Performance of non-recessed hole-entry hybrid journal bearing operating under turbulent regime

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HIGHLIGHTS

- Fluid film thickness of bearing increases with increasing Reynolds number at constant applied load.
- However, at higher value of Reynolds number, coefficient of friction increases significantly and remains lower when bearing operates under laminar regime.
- Under varying Reynolds number, the rotor dynamic coefficients of bearing are found higher than laminar regime.

ABSTRACT

The effect of turbulent flow on non-recessed hole-entry hybrid journal bearing system has been investigated numerically. For turbulent flow of lubricant in compensated hole-entry hybrid journal bearing by CFV, the Reynolds equation has been modified and solved using finite element method. The performance characteristics parameters of bearing have been presented for different values of Reynolds numbers. The bearing provides the higher values of minimum fluid film thickness and fluid film stiffness coefficients for constant restrictor design parameter when bearing operates under turbulent regime than laminar regime.

Keywords:
| Hybrid journal bearing | Turbulent flow | Restrictors | Finite element method |

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1.0 INTRODUCTION

The latest industrial developments have made the performance of machine to be more demanding. Thus, the bearings have to be designed more accurately for higher productivity of machines. The behavior of compensated hole-entry hybrid journal bearings using different restrictors operating in laminar regime was investigated by several authors (Shout and Rowe, 1974; Rowe and Koshal, 19980; Cheng and Rowe, 1995). Shout and Rowe presented the design and manufacture of externally pressurized journal bearings. They also compared the performance of recessed and non-recessed bearings. Later on, Rowe and Koshal proposed a procedure for optimizing the hybrid bearings for maximum load and minimum power dissipation. They reported an increase in load support with significant reduction in total power dissipation. Cheng and Rowe proposed a selection approach of bearing configurations and flow devices and material of bearing. In 1962, Constantinescu started to work on the turbulent flow in bearing taking into consideration Prandtl's mixing length theory. Kosasih and Tieu presented computational technique for determining velocity field for hydrodynamic lubrication under turbulent and laminar regime. They presented the effect of Reynolds number and pressure gradients on long bearing. Velescu presented new relations of coefficients for turbulent lubrication regime. The performance of centrally-pivoted tilting pad thrust bearings under turbulent lubrication was analyzed using finite difference equations by Capitao. They reported an increase in power loss and load carrying capacity of bearing under turbulent regime. The effect of turbulence on worn journal bearings was analyzed by Hashimoto et al. Later on, Hashimoto et al. evaluated dynamic characteristics of worn journal bearing under turbulent regime. The influence of turbulence on load carrying capacity, friction coefficient and oil flow for finite hydrodynamic porous journal bearings was presented by Kumar and Rao. A study by Kumar and Mishra investigated the non-circular worn journal bearings under turbulent flow. Recently, Nicodemus and Sharma analyzed the effect of geometric shapes of recess for capillary-compensated 4-pocket hybrid worn journal bearing operate under turbulent regime. More recently, the effect of wear on non-recessed journal bearing under turbulent regime was analyzed by Nathi and Sharma.

Available literature does not indicate the effect of turbulence on non-recessed CFV compensated hole-entry hybrid journal bearing. Therefore, the present study deals with the effect of turbulence on characteristics of non-recessed (hole-entry) hybrid journal bearing. The Reynolds equation considering Constantinescu's lubrication theory has been solved by using finite element analysis. For the chosen values of Reynolds numbers, the characteristics of bearing have been presented against restrictor design parameters.
2.0 ANALYSIS

For the flow of lubricant in hole-entry hybrid journal bearing system as shown in Fig. 1, the modified Reynolds equation is given as [4,11,12-13]

\[
\frac{\partial}{\partial \alpha} \left[ \frac{h^3}{G_\alpha} \frac{\partial \beta}{\partial \alpha} \right] + \frac{\partial}{\partial \beta} \left[ \frac{h^3}{G_\beta} \frac{\partial \beta}{\partial \beta} \right] = \frac{\Omega}{2} \frac{\partial h}{\partial \alpha} + \frac{\partial h}{\partial \tau} \tag{1}
\]

where, 

\[ G_\alpha = 12 + 0.026(R_e)^{0.8265} \tag{2} \]

\[ G_\beta = 12 + 0.0198(R_e)^{0.741} \tag{3} \]

The values of \( G_\alpha \) and \( G_\beta \) are given by Constantinescu [4] and for the laminar flow, \( G_\alpha = G_\beta = 12 \).
2.1 Fluid-Film Thickness

The value of fluid film thickness $\tilde{h}$ for symmetric journal bearing system is given as

$$\tilde{h} = 1 - \bar{X}_j \cos \alpha - \bar{Z}_j \sin \alpha$$  \hspace{1cm} (4)

2.2 Finite Element Formulation

The fluid flow is discretized using 4-noded quadrilateral isoparametric elements. System equations are obtained in matrix form after applying Galerkin’s technique and orthogonality condition for solving the Reynolds Eqn. (1) as:

$$[F]^e [p]^e = [Q]^e + \Omega \left( [R_H]^e + \bar{X}_j [R_{xj}]^e + \bar{Z}_j [R_{zj}]^e \right)$$  \hspace{1cm} (5)

where,

$$\bar{F}_{ij}^e = \int_{-1}^{+1} \int_{-1}^{+1} \frac{1}{\mu} \left[ \frac{\partial N_i}{\partial \alpha} \frac{\partial N_j}{\partial \alpha} + \frac{1}{c_{\alpha}} \frac{\partial N_i}{\partial \beta} \frac{\partial N_j}{\partial \beta} \right] \left[ \bar{V}_L \right] d\xi \ d\eta$$

$$\bar{R}_{Hj}^e = \int_{-1}^{+1} \int_{-1}^{+1} \frac{\partial N_i}{\partial \alpha} \left[ \bar{V}_L \right] d\xi \ d\eta$$

$$\bar{R}_{xij}^e = \int_{-1}^{+1} \int_{-1}^{+1} N_i \cos \alpha \left[ \bar{V}_L \right] d\xi \ d\eta$$

$$\bar{R}_{zij}^e = \int_{-1}^{+1} \int_{-1}^{+1} N_i \sin \alpha \left[ \bar{V}_L \right] d\xi \ d\eta$$

$$\bar{Q}_{ij}^e = -\frac{\Omega}{2} \int_{-1}^{+1} \bar{h} N_i \left| \bar{V}_{L2} \right| d\eta + \int_{-1}^{+1} \frac{\bar{h}^2}{\mu} \frac{1}{c_{\alpha}} \frac{\partial \bar{p}}{\partial \alpha} N_i \left| \bar{V}_{L2} \right| d\eta + \int_{-1}^{+1} \frac{\bar{h}^2}{\mu} \frac{1}{c_{\beta}} \frac{\partial \bar{p}}{\partial \beta} N_i \left| \bar{V}_{L1} \right| d\xi$$

where

$$\left| \bar{V}_L \right| = \begin{vmatrix} \frac{\partial \alpha}{\partial \xi} & \frac{\partial \beta}{\partial \xi} \\ \frac{\partial \alpha}{\partial \eta} & \frac{\partial \beta}{\partial \eta} \end{vmatrix}, \quad \left| \bar{V}_{L1} \right| = \begin{vmatrix} \frac{\partial \alpha}{\partial \xi} \\ \frac{\partial \alpha}{\partial \eta} \end{vmatrix}, \quad \left| \bar{V}_{L2} \right| = \begin{vmatrix} \frac{\partial \beta}{\partial \xi} \\ \frac{\partial \beta}{\partial \eta} \end{vmatrix}, \quad (i,j = 1,2,3,4)$$
2.3 Boundary Conditions

The boundary conditions (Nathi and Sharma, 2013) applied for solving the Reynolds equations (5) are:

a) The nodes have zero pressure on external boundary of bearing, i.e., \( \bar{p} \bigg|_{x=\pm 10} = 0.0 \).

b) Nodes have equal pressure on holes.

c) Flow through the CFV restrictor is equivalent to input flow of bearing at hole.

d) Reynolds boundary condition at trailing edge of positive region has been applied, \( \bar{p} = \frac{\partial \bar{p}}{\partial \alpha} = 0.0 \).

2.4 Restrictor Flow Equation

The lubricant flow in the bearing via constant flow valve restrictor is given as

\[
\bar{Q}_R = \bar{Q}_{sp}
\]  
(6)

2.5 Total Frictional Torque \( (\bar{T}_{fric}) \)

The total frictional torque \( (\bar{T}_{fric}) \) exerted on journal due to shearing action at surface is given as [4]

\[
\bar{T}_{fric} = \int_0^{2\pi} \int_{-1}^1 \left( \frac{h}{2} \frac{\partial p}{\partial x} + \frac{\mu \omega}{h} \frac{\bar{e} R^2}{h} \right) R f dX \, dY
\]  
(7)

Where \( \bar{e} \) is called normalized turbulent coquette shearing stress and is given by,

\[
\bar{e} = 1 + 0.0023(R_e^{0.855})
\]  
(7a)

For laminar flow \( \bar{e} \) becomes unity. Expressing the above equation in the non-dimensional form,

\[
\bar{T}_{fric} = \int_0^{2\pi} \int_{-1}^1 \left( \frac{\bar{R}}{2} \frac{\partial \bar{p}}{\partial \alpha} + \frac{\bar{e} \alpha}{\bar{R}} \right) d\alpha \, d\beta
\]  
(8)
2.6 Fluid Film Stiffness Damping Coefficients

The computation of fluid-film stiffness coefficients and fluid-film damping coefficients requires the pressure derivatives w. r. t journal center displacement \( (\ddot{a}_j = \ddot{X}_j, \ddot{Z}_j) \) and journal center velocity \( (\dot{a}_j = \dot{X}_j, \dot{Z}_j) \) respectively. The fluid film stiffness coefficients in matrix form are given as:

\[
\begin{bmatrix}
S_{11} & S_{12} \\
S_{21} & S_{22}
\end{bmatrix}
= - \begin{bmatrix}
\frac{\partial F_x}{\partial X_j} & \frac{\partial F_x}{\partial Z_j} \\
\frac{\partial F_z}{\partial X_j} & \frac{\partial F_z}{\partial Z_j}
\end{bmatrix}
\]  

(9)

The fluid film damping coefficients in matrix form are given as:

\[
\begin{bmatrix}
C_{11} & C_{12} \\
C_{21} & C_{22}
\end{bmatrix}
= - \begin{bmatrix}
\frac{\partial F_x}{\partial X_j} & \frac{\partial F_x}{\partial Z_j} \\
\frac{\partial F_z}{\partial X_j} & \frac{\partial F_z}{\partial Z_j}
\end{bmatrix}
\]  

(10)

2.7 Threshold Speed

Threshold speed is given as:

\[
\bar{\omega}_{th} = \left[ \frac{\bar{M}_c}{\bar{F}_0} \right]^{1/2}
\]

(11)

Where, \( \bar{M}_c \) is the critical mass. The non-dimensional critical mass \( \bar{M}_c \) of the journal is expressed as:

\[
\bar{M}_c = \frac{\bar{G}_1}{\bar{G}_2 - \bar{G}_3}
\]

\[
\bar{G}_1 = \left[ C_{11} \bar{C}_{22} - C_{21} \bar{C}_{12} \right]
\]

\[
\bar{G}_2 = \frac{\left[ S_{11} \bar{S}_{22} - \bar{S}_{12} \bar{S}_{21} \right] \left[ C_{11} + \bar{C}_{22} \right]}{\left[ S_{11} \bar{C}_{22} + \bar{S}_{22} \bar{C}_{11} - \bar{S}_{12} \bar{C}_{21} - \bar{S}_{21} \bar{C}_{12} \right]}
\]

\[
\bar{G}_3 = \frac{\left[ S_{11} \bar{C}_{11} + \bar{S}_{12} \bar{C}_{12} + \bar{S}_{21} \bar{C}_{21} + \bar{S}_{22} \bar{C}_{22} \right]}{\left[ C_{11} + \bar{C}_{12} \right]}
\]

A journal bearing system is asymptotically stable when the operating speed of the journal is less than the threshold speed (i.e. when \( \Omega < \bar{\omega}_{th} \)).
3.0 SOLUTION PROCEDURE

For obtaining the solution, the Eqn. (5) together with restrictor Eqn. (6) and boundary conditions has been solved assuming initial values of journal centre positions based on analysis. This requires an iterative technique to obtain the matched solutions. Additional iterations are also needed for the establishment of equilibrium journal centre position for a particular vertical load. When journal centre equilibrium is attained then the performance of the bearing is computed.

The solution schemes employed for the solution of bearing has been shown in Fig. 3.2. The overall solution consists of following stages:

a) Initially, the unit LDATA takes input data and 2-D mesh of lubricant domain.

b) Then, FLMTH determines film thickness at nodes by equation (4) taking tentative values of $x_J$ and $z_J$.

c) Then, the unit TRBULENT calculates the values of turbulent coefficients ($G_\alpha$ and $G_\beta$) given by equations (2) and (3).

d) By using values of film thickness, the matrices of equation (5) are developed and assembled in unit FLUD.

e) Then, unit BODRY modifies system equation for particular boundary conditions.

f) The modified system equations are then solved for nodal pressure in unit SOLVG by Gaussian elimination technique.

g) The pressure field for the lubrication flow field obtained, from BLOC PRS as shown in Fig.2.

h) Then, STATCH & DYNACH computes the performance of journal bearing.

i) Then the unit LPRINT gives the output data and then unit STOP terminates the program.

4.0 RESULTS AND DISCUSSION

A program has been developed using finite element method for investigating the effect of turbulence on symmetric hole-entry hybrid journal bearing. The simulated results have been compared with the published results for hydrodynamic bearing under turbulent regime [11] in Figure 3. The relevant expressions for computing characteristics of bearing have been used as given in Ref. [13]. The static performance characteristics such as maximum fluid film pressure, minimum fluid film thickness, bearing flow and frictional torque and dynamic characteristics have been evaluated for the following parameters as:

Aspect ratio $\lambda = L/D = 1.0$;

Land width ratio $(\bar{\alpha}_b) = 0.25$;
No. of rows of holes = 2;
No. of holes per row = 12;
Speed parameter ($\Omega$) = 1;
Reynolds numbers, $Re = 5000$, 10000, 15000 and 20000 for turbulent flow;
Reynolds number, $Re = 0$ for laminar flow;
External load ($\bar{W}_o$) = 1.25;
Concentric design pressure ratio $\beta^* = 0.5$;
Restrictor design parameter, $\bar{C}_{s2} = 0.05 - 0.25$

The numerically simulated results have been presented in Figs.4-10 for hole-entry hybrid journal bearing.

Figure 2: Overall solution scheme for bearing
4.1 Influence On Maximum Fluid Film Pressure ($\overline{p}_{\text{max}}$)

Figure 4 shows the increasing trend for the value of maximum fluid film pressure ($\overline{p}_{\text{max}}$) with $\overline{c}_{s_2}$. The maximum fluid film pressure $\overline{p}_{\text{max}}$ increases with increasing value of $\overline{c}_{s_2}$ when bearing is operating in laminar as well as turbulent regimes. However, the value of $\overline{p}_{\text{max}}$ is higher at constant value of $\overline{c}_{s_2}$ corresponding to increasing value of Reynolds number as compared to bearing operates in laminar regime. The increase in value of $\overline{p}_{\text{max}}$ is observed to be of the order of 76.26% at $\overline{c}_{s_2} = 0.15$ for the value of Reynolds number of $R_e = 5000$ as compared to bearing operates in laminar regime.
4.2 Influence On Minimum Fluid Film Thickness ($\bar{h}_{min}$)

Figure 5 shows the value for minimum fluid film thickness $\bar{h}_{min}$. The minimum fluid film thickness $\bar{h}_{min}$ increases as Reynolds number increases at constant value of $\bar{C}_{s2}$ than bearing operates under laminar regime. Further, minimum fluid film thickness is greater for the Reynolds number $R_e = 20000$ than the bearing operates in laminar regime. The increase in film thickness $\bar{h}_{min}$ is observed to be of the order of 10.54% at $\bar{C}_{s2} = 0.05$ and 7.27% at $\bar{C}_{s2} = 0.1$ respectively for Reynolds number of $R_e = 20000$ as compared to bearing operates in laminar regime.

![Figure 5: Variation of $\bar{h}_{min}$ with $\bar{C}_{s2}$](image)

4.3 Influence On Frictional Torque ($\bar{T}_{fri}$)

Figure 6 shows the variation for the frictional torque ($\bar{T}_{fri}$). The lower value of $\bar{T}_{fri}$ has been observed when bearing operating in laminar regime. However, the effect of turbulence increases the value of $\bar{T}_{fri}$ for constant $\bar{C}_{s2}$ than the bearing operates under laminar regime.
### 4.4 Influence On Direct Fluid Film Stiffness Coefficient ($\bar{\mathcal{S}}_{11}, \bar{\mathcal{S}}_{22}$)

Figure 7 and Figure 8 depict the variation of direct fluid film stiffness coefficient ($\bar{\mathcal{S}}_{11}, \bar{\mathcal{S}}_{22}$). The values of $\bar{\mathcal{S}}_{11}$ and $\bar{\mathcal{S}}_{22}$ increase as the value of restrictor design parameter ($\bar{c}_{s2}$) increases for the bearings operating under both turbulent and laminar regimes. For the increased value of Reynolds number, the values of $\bar{\mathcal{S}}_{11}$ and $\bar{\mathcal{S}}_{22}$ increases at constant $\bar{c}_{s2}$ than bearing operates under laminar regime. It has also been noticed that the values of fluid film stiffness coefficients are higher for all values of Reynolds number for restrictor design parameter at $\bar{c}_{s2} = 0.25$ than bearing operates in laminar regime.
4.5 Influence On Direct Fluid Film Damping Coefficient ($\bar{C}_{11}, \bar{C}_{22}$)

It has been observed from Figure 9 and Figure 10 that the values of direct fluid film damping coefficient ($\bar{C}_{11}, \bar{C}_{22}$) are higher at constant value of $\bar{C}_{s2}$ due to influence of turbulence. The values of $\bar{C}_{11}$ and $\bar{C}_{22}$ get increased by 72.27% and 72.15% for $Re = 20000$ at $\bar{C}_{s2} = 0.05$ than bearing operates in laminar regime. Further due to influence of turbulence the values of $\bar{C}_{11}$ & $\bar{C}_{22}$ gets increased significantly for the values of Reynolds number $Re = 20000$ at constant value of $\bar{C}_{s2}$ than bearing operates under laminar regime.

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

The variation of stability threshold speed margin ($\bar{\omega}_{th}$) against $\bar{C}_{s2}$ for the bearing is shown in Figure 11. When the bearing operates in turbulent regime, the stability threshold speed margin gets enhanced at constant $\bar{C}_{s2}$ than bearing operates under laminar regime. For a bearing operating at $\bar{C}_{s2} = 0.15$ and $Re = 20000$, stability threshold speed margin $\bar{\omega}_{th}$ increases by 104.05% than similar bearing under laminar regime.
Figure 9: Variation of $\ddot{c}_{11}$ with $\ddot{c}_{s2}$

Figure 10: Variation of $\ddot{c}_{22}$ with $\ddot{c}_{s2}$
5.0 CONCLUSIONS

A numerical study for CFV compensated non-recessed hole-entry journal bearing considering the influence of turbulence has been carried out. From the presented results, following conclusions have been derived.

1. With an increase in the value of Reynolds number, the minimum fluid film thickness enhances when bearing is operating at constant value of $\lambda_{s2}$ under turbulent.

2. When bearing operates under increased value of Reynolds number, the stiffness and damping coefficients is observed higher than similar bearing under laminar regime.

3. Bearing threshold speed increases significantly at $\lambda_{s2} = 0.25$ for a bearing operates at $Re = 20000$ than similar bearing operates under laminar regime.

REFERENCES


