EFFECT OF WEAR ON SYMMETRIC HOLE-ENTRY HYBRID JOURNAL BEARING COMPENSATED BY ORIFICE RESTRICTOR UNDER TURBULENT REGIME

Reference NO. IJME 1356, DOI: 10.5750/ijme.v1i1.1356

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KEY DATES: Submission date: 25.11.2023 / Final acceptance date: 20.02.2024 / Published date: 12.07.2024

SUMMARY

This study examines the impact of wear and turbulent flow on symmetric hole-entry hybrid journal bearings with orifice restrictors. Dufranes's abrasive model for wear effect and Constantinescu's lubrication model for turbulent flow have been used. A modification has been made to the Reynolds equationand utilized the finite element method to solvealong withflow equation of an orifice restrictor. For selected wear depth parameter values and Reynolds numbers,computed results have been acquired. The minimum fluid film thickness increases for worn bearings when operating under a turbulent regime rather than a laminar regime. Further, the stiffness coefficient decreases for constant external load when worn/unworn bearings function in a turbulent regime.

KEYWORDS

Fluid film journal bearings, Turbulent flow theories, Control flow devices, Wear, Finite element method

NOMENCLATURE

D	Bearing diameter, mm
F_o	Reaction of Fluid-film, N
L	Length of bearing, mm
Q	Flow of bearing, mm^{-3}s^{-1}
R_J, R_b	Journal radius, bearing radius, mm
W_o	External load, N
a_b	Land width of bearing, mm
a_o	Orifice diameter, mm
c	Radial clearance, mm
p	Pressure, Nmm^{-2}
t	Time, s
$\underline{C}_{ij}$	Damping coefficients, $C_{ij}\left(\dfrac{c^3}{p_s R_j^4}\right)$
$\underline{C}_{s2}$	$\left[\dfrac{1}{12}\left(\dfrac{3\pi a_o^2 \mu_r \psi_d}{C^3}\right)\right]\left(\dfrac{2}{\rho\rho_s}\right)^{\frac{1}{2}}$, orifice restrictor design parameter
$\underline{h}$	h/c
$\underline{Q}$	$Q\left(\mu / c^3 p_s\right)$
$\underline{S}_{ij}$	Stiffness coefficients, $S_{ij}\left(\dfrac{c}{p_s R_j^2}\right)$
$\underline{W}_o$	$W_o / p_s R_j^2$
α, β	$\left(X, Y\right)/R_J$, Circumferential and axial coordinates
$\underline{\delta}_w$	δ_w / c, wear depth parameter
$\underline{\omega}_{th}$	$\dfrac{\omega_{th}}{\omega_I}$
ω_j, ω_{th}	Journal rotational, threshold speed, rad per second
μ	Dynamic viscosity of lubricant, Ns per m^2
λ	Aspect ratio, $\dfrac{L}{D}$
ψ_d	Discharge Coefficient of orifice
β^*	Concentric designed pressure ratio, $\dfrac{p^*}{p_s}$
ω_I	$\left(\dfrac{g}{c}\right)^{\frac{1}{2}}$
δ_w	Wear depth, mm

1. INTRODUCTION

The frequent start and stop operations of machines is responsible for the impact of journal bearing wear. Forrester [1] examined the effect of wear on bearings due to operations of starting and stopping of machines. Later on, Duckworth and Forrester [2] analyzed the worn journal bearings' functionality. They analyzed the several damages in the bearings to investigate the failure in bearings caused by wear. The effect of wear on bearing surface, examined for use in steam turbine generators by Dufrane et al. [3]. He proposed model for worn geometry of the bearing for the analysis of journal bearings. Constantinescu [4] provided the computations for thrust and journal bearings that are lubricated turbulently. Afterwards, Constantinescu and Galetuse [5] proposed a technique for determining the friction forces in turbulent lubrication of bearings. Taylor and Dowson [6] presented a comprehensive review analysis of the already-in-place theories of turbulent lubrication and the limitations of the alternative approaches. Hashimoto et al. [7] presented the impact of turbulence on worn journal bearings' static performance parameters. They reported that the bearing offers a higher fluid film pressure value when operating in a turbulent regime for higher eccentricity ratios. Kumar and Mishra [8] presented the wear effect on non-circular worn bearings under turbulent lubrication. Later on, Kumar and Rao [9] examined the effect of turbulent lubrication in finite hydrodynamic porous journal bearings. Gertzos et al. [10] presented wear depth parameter's impact on performance of rotor-bearing systems. They also used the same for online wear identification.

The hybrid bearings, due to their activities that are both hydrostatic and hydrodynamic, are very useful in various industrial applications. Many studies about the effectiveness of hole-entry bearings were reported by several investigators [11-13]. Rowe et al. [12] proposed an analysis for the functionality of journal bearing designs with and without recessed spaces. They concluded that hole-entry bearings exhibits better performance as contrasted withother bearing configurations. Later on, Stout and Rowe [13] studied various bearing configurations, compensating devices, materials for gas and liquid fed bearings. Recently, Nicodemus and Sharma [14] analyzed the wear-causing factor of a four-pocket hybrid journal-bearing type system under turbulent lubrication for different recess geometric forms.

The existing studies point out that the turbulent flow's impact on hole-entry worn bearing using an orifice restrictor has not yet been reported. Hence, the current study investigates the impact of turbulent flow and wear on hole-entry hybrid symmetric type bearing. For obtaining the functionality of bearing, The Reynolds equation has been solved utilizing Newton Rapshon and Finite Element.

2. ANALYSIS

The laminar and turbulent flow in the symmetric journal bearing as demonstrated in Fig.1 is dictated by modifications in Reynolds equation is provided as [4,8,14]

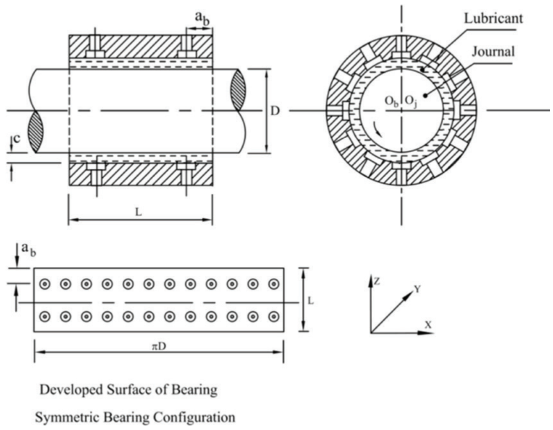

Figure 1. System for symmetric hole-entry journal bearing

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

where,

$$G_\alpha = 12 + 0.026\left(R_e\right)^{0.8265} \tag{2}$$

$$G_\beta = 12 + 0.0198\left(R_e\right)^{0.741} \tag{3}$$

The values of G_α and G_β are 12 for laminar flow and local Reynolds number R_e is zero.

Four noded quadrilateral isoparametric elements have been taken into consideration in order to discretize the lubricant flow domain. The system equation in matrix form is obtained by employing method of Galerkin on Reynolds equation (1) as:

$$\left[\bar{F}\right]^e\left\{\bar{p}\right\}^e = \left\{\bar{Q}\right\}^e + \Omega\left\{\bar{R}_H\right\}^e + \dot{\bar{X}}_J\left\{\bar{R}_{X_J}\right\}^e + \dot{\bar{Z}}_J\left\{\bar{R}_{Z_J}\right\}^e \tag{4}$$

Fluid-Film Thickness

For worn bearing's design and geometry as represented in Fig.2, the thickness of fluid $\bar{h}$ for worn geometry of bearing is given as [3,8,14]

$$\bar{h} = 1 - \bar{X}_j cos\alpha - \bar{Z}_j sin\alpha + \partial\bar{h} \tag{5}$$

where, $\bar{X}_j$ and $\bar{Z}_j$ represent the co-ordinates of equilibrium of journal's centre,

$$\partial\bar{h} = \bar{\delta}_w - 1 - \sin\alpha \;\text{ for }\; \alpha_b \le \alpha \le \alpha_e \tag{6}$$

$$\partial\bar{h} = 0 \;\text{ for }\; \alpha\langle\alpha_b or \alpha\rangle\alpha_e \tag{7}$$

The angles α_b and α_e are determined through Equation (7) with Equation (8) as

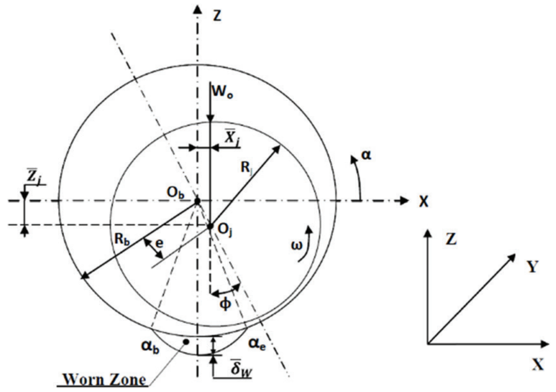

Figure 2. Worn bearing geometry

$$\sin\alpha = \bar{\delta}_w - 1 \quad (8)$$

Flow Equation of Restrictor

The lubricant flow equation through an orifice restrictor in the bearing can be given as [12]

$$\bar{Q}_R = \bar{C}_{s2}\left(1 - \bar{p}_c\right)^{1/2} \quad (9)$$

3. SOLUTION METHODOLOGY

An iterative numerical approach has been used to solve the hole-entry hybrid worn journal-type bearing functioning in the turbulent regime. The system Eqn. (4) along with restrictor flow Eqn. (9) and boundary conditions given in Ref. [14] have been solved for achieving pressure field. The modifications in system equations become non-linear for the field of lubricant flow for a bearing compensated by an orifice restrictor. Therefore, these non-linear type equations have been rectified by Newton Raphson Method. After converged solution, performance characteristics of bearing are calculated with the appropriate expressions [14].

4. RESULTS AND DISCUSSION

A computer program has been used to calculate the effects of wear and turbulence on symmetric hole-entry hybrid journal type bearings, formulated in FORTRAN77. The outcomes of the numerical simulation are very close to the results given in Ref. [8] in case of hydrodynamic journal bearing which are shown in Fig.3. Simulations of the bearing's performance defining characteristics have been conducted for an external load range $(\bar{W}_o)$ = 0.25-1.5 using parameters such as: parameter of speed (Ω) = 0.25; number of rows of holes = 2; number of holes/row = 12; land width ratio $(\bar{a}_b)$ = 0.25; pressure ratio of concentric design (β^*) = 0. 5; Reynolds numbers 0 (for laminar flow) and 10000

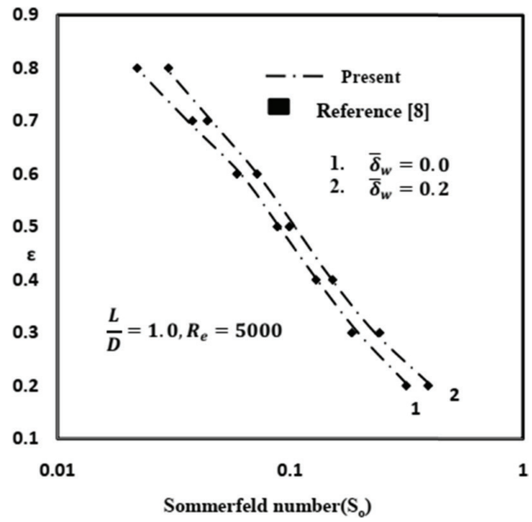

Figure 3. ε with S_o variation

(for turbulent flow), wear depth parameter, $\bar{\delta}_w$ =0, 0.25, 0.5; aspect ratio (λ) = 1.0.

The fluid film thickness at minimum $\bar{h}_{min}$ decreases for external load range of $(\bar{W}_o)$ = 0.25-1.5 for worn/unworn bearing under laminar/turbulent regimes as shown in Fig.4. For the fixed value of $\beta^* = 0.5$ and $\Omega = 0.25$, a decrease of 11.44% in value of $\bar{h}_{min}$ is observed at constant load of $(\bar{W}_o) = 1.0$ according to the wear depth parameter $\bar{\delta}_w = 0.5$ for a bearing under a laminar regime in comparison to an unworn bearing. However, when operating under a turbulent regime, $\bar{h}_{min}$ value is compensated for worn or unworn bearing. A reduction in minimum fluid film thickness $\bar{h}_{min}$ is 6.92% at $\bar{W}_o$ =1.0 for worn bearing under turbulent regime and operates at $\bar{\delta}_w = 0.5$ than unworn bearing in laminar regime.

As shown in Fig.5, when bearing functions under the laminar regime, the flow of bearing $\bar{Q}$ rises as the parameter of wear depth $\bar{\delta}_w$ rises. The rise in bearing flow value $\bar{Q}$ is 8.67% at $\bar{\delta}_w = 0.5$ corresponding to load at $\bar{W}_o$ =1.0 for bearing than unworn bearing under laminar regime. However, the flow of bearing $\bar{Q}$ is observed to become lower when unworn bearing operates under turbulent regime. Due to effect of turbulence, the worn and unworn bearing requires less amount of bearing flow than worn/unworn bearing under laminar regime. It is observed that the bearing flow has decreased by approximately 39.96% at $\bar{W}_o$ =1.0 for unworn bearing and 26.64% for worn bearing operates at $\bar{\delta}_w = 0.25$ respectively operates under turbulent lubrication than unworn bearing in laminar regime.

The film stiffness coefficient ($\bar{S}_{11}$) decreases for constant load $\bar{W}_o$ when bearing operates under turbulent regime as shown as in Fig.6. The reduction of 17.69% in the value

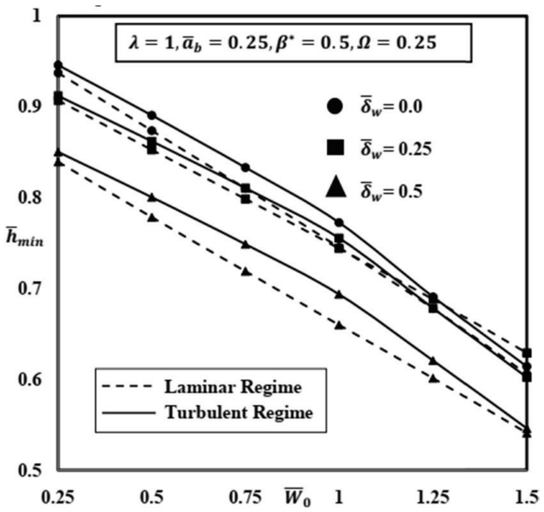

Figure 4. Variation of $\bar{h}_{min}$ with $(\bar{W}_o)$

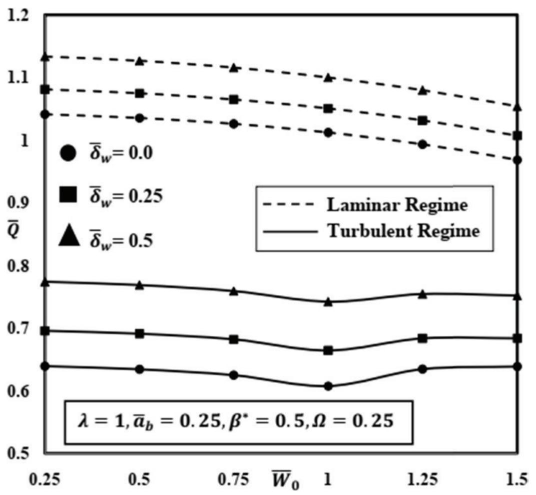

Figure 5. Variation of $\bar{Q}$ with $(\bar{W}_o)$

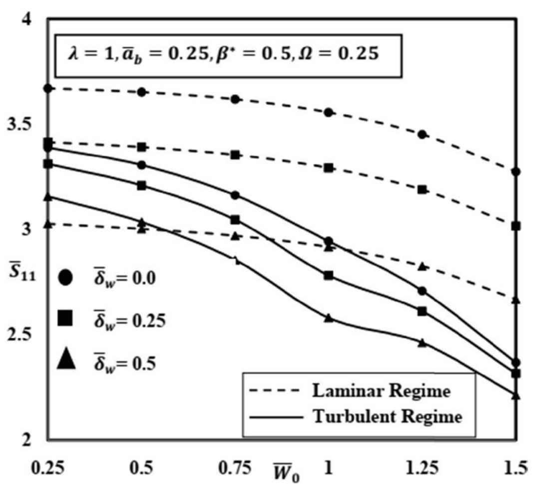

Figure 6. Variation $\bar{S}_{11}$ with $(\bar{W}_o)$

of $\bar{S}_{11}$ is noted at $\bar{W}_o$=1.0 corresponding to wear depth parameter $\bar{\delta}_w = 0.5$ for worn bearing under laminar regime than unworn bearing. Further, the film stiffness coefficient for a bearing gets reduced with increasing value of wear depth parameter operating under turbulent regime. The value of $\bar{S}_{11}$ reduces by 27.47% at wear depth parameter $\bar{\delta}_w = 0.5$ for bearing operating at $\bar{W}_o$=1.0 under turbulent regime than unworn bearing in laminar regime.

Coefficient of direct fluid film damping $\bar{C}_{11}$ values in order for a bearing to function at load rang of $\bar{W}_o = 0.25$ to 1.5 is shown in Fig.7. For constant load $\bar{W}_o$, the value of $\bar{C}_{11}$ reduces in the case of worn bearings as opposed to unworn bearings operates under laminar regime. The decrease of 13.54% at $\bar{\delta}_w = 0.5$ for coefficient of damping at constant load $\bar{W}_o$=1.0 is noted when a bearing operates under laminar regime. The unworn bearing runs under a turbulent regime, however, and this value of the coefficient of damping are larger. The $\bar{C}_{11}$ values increases by 57.62% at $\bar{\delta}_w = 0.25$ and 50.31% at $\bar{\delta}_w = 0.5$ respectively for a bearing under turbulent regime, operates at $\bar{W}_o$=1.0 than unworn bearing under laminar regime.

Fig. 8 illustrates that bearing stability threshold speed value $\bar{\omega}_{th}$ gets decreased by 5.56% at $\bar{W}_o = 1.0$ for worn bearing having the value of $\bar{\delta}_w = 0.5$ operates under laminar regime than unworn bearing. The value of $\bar{\omega}_{th}$ reduces as wear depth parameter increases in laminar regime while a bearing is in operation. In a turbulent regime while the bearing is in operation, the threshold speed $\bar{\omega}_{th}$ further decreases in comparison of unworn bearing in laminar regime. The threshold speed $\bar{\omega}_{th}$ is lower in the instance of unworn bearing operates in turbulent regime and a decrease of 11.47% in the value of $\bar{\omega}_{th}$ at $\bar{W}_o = 1.0$ is noticed contrasted with unworn bearing within the laminar regime. But still, threshold speed $\bar{\omega}_{th}$ is discovered of the sequence of 10.95% at $\bar{\delta}_w$=0.25 and $\bar{W}_o$=1.0 for a bearing operates under turbulent regime.

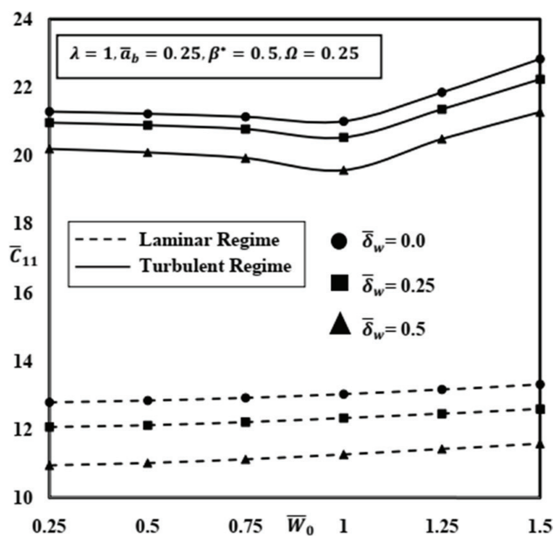

Figure 7. Variation $\bar{S}_{11}$ with $(\bar{W}_o)$

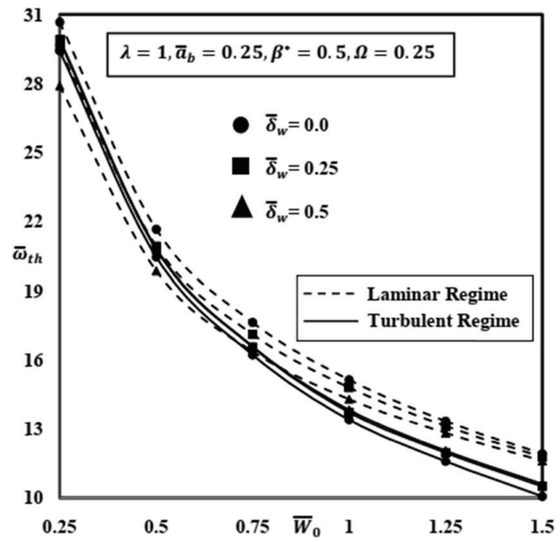

Figure 8. Variation of $\bar{\omega}_{th}$ with $(\bar{W}_o)$

5. CONCLUSIONS

The impact of wear and turbulence on hole-entry hybrid journal bearings has been assessed conceptually and following conclusions have been depicted.

1. Notedly, the minimal thickness of the fluid film ($\bar{h}_{min}$) decreases for constant external load in case of worn bearing operates under laminar regime than unworn bearing. However, the unworn/worn bearing functioning in a turbulent regime improves the value of $\bar{h}_{min}$.
2. In worn bearings, the value of lubrication flow increases for constant load when functions within a laminar regime than unworn bearing. Observations indicate that the lesser quantity of the lubricant flow is required for worn/unworn bearings operating under turbulent regime.

In case of worn bearing operating at $\Omega = 0.25$ in turbulent regime, the coefficient of stiffness ($\bar{S}_{11}$) reduces as wear depth parameter rises under constant load compared to a laminar regime unworn bearing.

In the laminar regime, worn bearings experience a lower damping coefficient at constant load than unworn bearings. However, as a worn or unworn bearing runs in a turbulent regime, it's damping coefficient value increases significantly than unworn bearing operating under laminar regime.

When the pressure ratio of concentric design $\beta^* = 0.5$ remains constant and operating speed parameter at $\Omega = 0.25$, stability threshold speed ($\bar{\omega}_{th}$) gets decreased for worn/unworn bearing in turbulent regime than unworn bearing under laminar regime.

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