An effect on the pressure and fluid film thickness of the worn journal bearing system using JFO cavity model

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Abstract
The present study is basically concerned with the wear which occurs due to frequent start / stop operation as well as due to running-in operation as these operations are very usual and the wear from these operations is commonly observed in fluid-film journal bearing system. In order to model the worn journal bearing geometry, the current study considers the geometry of the worn smooth surface based on the Dufrane model, which assumes that the footprint created by the journal on the bearing is symmetrical at the bottom of the bearing and the wear pattern is uniform along the axial length of the bearing. In order to compute the fluid film pressure for fluid film journal bearing system the two dimensional Reynolds is solved numerically using FDM. The code developed is validated for its accuracy by comparing the results with already published results. Effect of wear on performance parameters (maximum pressure and minimum fluid thickness) of journal bearing are calculated.

1. Introduction

Bearing failures are one of the chief causes in the shutdown of the turbines generators and they are caused by wear in journal bearings due to myriad reasons. Wear is a paramount factor that govern the quality of bearing material. Wear due to transient (starts and stops) conditions and running-in wear in bearings is inevitable. Under transient periods, the bearing bush progressively worn out due to abrasive method The wear caused by these operations may influence the bearing performance and in turn may affect the bearing life [1-4]. In the past since 1957 many investigators have examined psychological measurement of several destruct bearings. To measure wear, Durfane [5] was first to propose mathematical model. Lubricant flow rate increases while load carrying capacity reduces with increase in wear [6,7].

Vaidyanathan and Keith [8] studied the worn journal problem and concluded that as wear depth increases, indicating a greater worn bearing surface, the same load carrying capacity can be achieved by requiring the bearing to operate at greater eccentricity. Ilie and Cicone [9] develop a simplified solution for Reynolds equation for full film journal bearings affected by wear. The results were obtained numerically, based on simple integration of a 2ndorder differential equation and it was shown that the presence of the worn region extends the circumferential pressure distribution together with an increase of the film thickness. Experimental data and results obtained from numerically simulated were compared on a journal lobed bearing subjected to numerous starts and stops by Bouyer et al. [10]. They concluded that wear modifies bearing characteristics, with an increase in maximum pressure and an overall decrease in temperature.

A study of bearing dynamics is important for enhancing smooth bearing life. There appears a very little work in the published literature, dealing with the time transient analysis of worn hydrodynamic journal bearing.The main objective of the current work is to make numerical analysis of a worn out fluid film journal bearing system for predicting static parameters at various eccentricities and at various wear depths by incorporating ADI technique In order to account the wear, a numerical model for wear depth used by Dufrane [5] has been considered in the analysis. The JFO based model of Fluid film journal bearing is selected for this purpose as this model leads to more accurate prediction of pressure vis-a-vis to Reynolds boundary condition based model. The details have been given in section 2.

Figure 1 shows a typical example of a worn out bearing of a turbine with symmetrical wear pattren in bottom of bearing.

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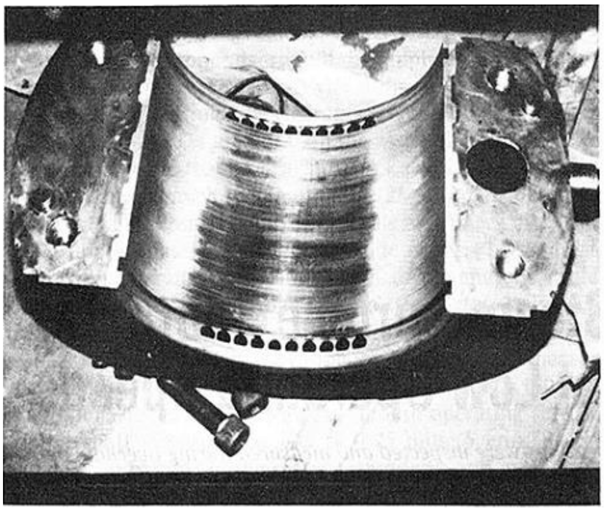

Fig. 1 Wear pattern on turbine bearing after approximately 15 years of service [5]

2. Analysis

In order to compute the fluid film pressure for fluid film journal bearing system the two dimensional Reynolds equation as described below is solved numerically using FDM.

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\eta}\frac{\partial P}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\eta}\frac{\partial P}{\partial z}\right)=6h\frac{\partial}{\partial x}(u+u_2)h+6(u_1-u_2)\frac{dh}{dx}+12\frac{\partial h}{\partial t} \quad (1)$$

For the current study FDM is used to solve Reynolds equation which is described below.

2.1 Nominal Fluid-Film Thickness

Based on the observations of Dufrane et al [5], a wear model was formulated in which the wear depth was written as:

$$d = d_o - c\,[1 + \cos(\alpha + \varnothing)] \quad (2)$$

Where d is the depth of wear, d_o is maximum depth of wear, and c the radial clearance**.**

The film thickness in the non-worn regions of the bearings is given by [5]:

$$h = c + e\cos(\alpha) \quad (3)$$

The film thickness in the worn region is obtained by the superimposing the wear model onto the above expression thus:

$$h = do + e\cos(\alpha) - c\cos(\alpha + \varnothing) \quad (4)$$

The extent of the wear region is found by solving equation (4) for the starting (α_s) and ending points (α_f) points. This can be accomplished by allowing the depth of wear in equation (1) to vanish and solving the resulting expression

$$\cos(\alpha + \varnothing) = \delta_o - 1 \quad (5)$$

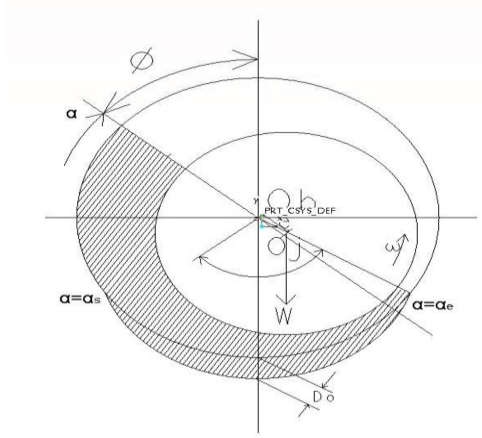

Fig. 2 A worn circular journal bearing

3. Results and Discussion

3.1Validation

The influence of wear depth on the performance characteristics of worn journal bearing system have been computed by developed computer code in FORTRAN.

In order to validate the accuracy of the developed code, the simulated results were compared with the published results. The computational grid selected for this purpose is 72 * 10 (with 73 nodes in circumferential direction and 11 nodes in axial direction). This computational grid is selected as a compromise between accuracy and computational time. This software computes the value of pressure and film thickness at all nodal points. The simulated results for a specified value of β = 1.0 has been plotted and compared with the published results. In order to validate the code computed results for eccentricity ratio versus Sommerfeld number are compared with already published results [8] at wear depths of δ_w=0.0 and δ_w= 0.25.

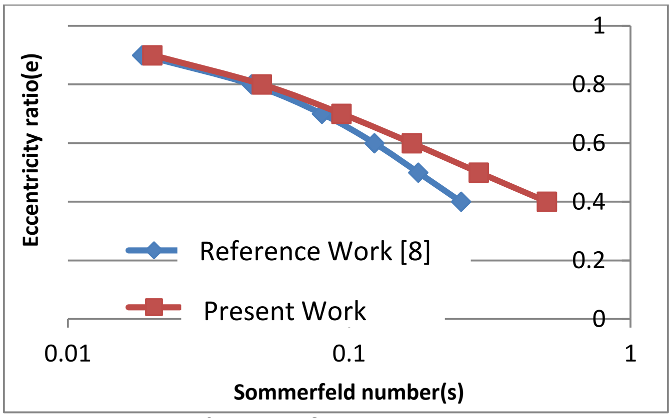

Fig. 3 Comparison of results at δ_w=0.0

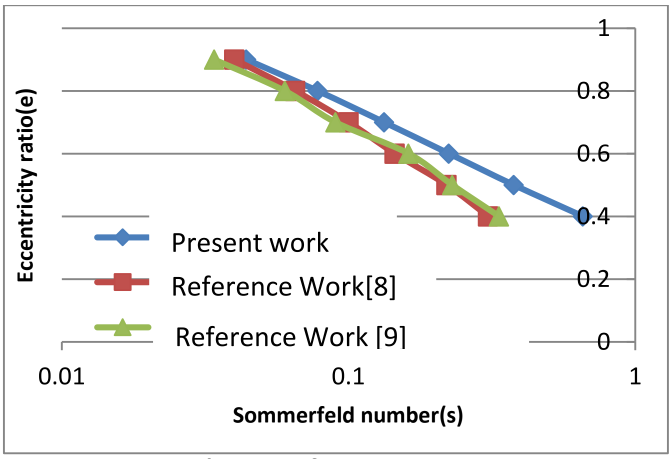

Fig. 4 Comparison of results at δ_w=0.25

It is clear from Figure 3 and 4 that with the increase in eccentricity ratio, Sommerfeld number is decreasing, which results in increase in load carrying capacity (LCC) and hence pressure also increases (as pressure increases with eccentricity ratio is requirement for published results [8].

4.Effects of wear depth

4.1 Circumferential pressure distribution

The distribution of fluid film pressure impacts bearings static and dynamic performance parameters, and thus has an important significance to predict the bearing behavior. Therefore, the pressure distribution in the fluid domain has been described in terms of axial mid plane and corresponding to the lower most axial direction.

The fluid-film pressure distribution in the circumferential direction at the axial mid plane is shown in Figure 5. The fluid-film pressure field is altered with the wear defect which represents the pressure fields with δ_w = 0.0, 0.25 and 0.5 for the three different cases.

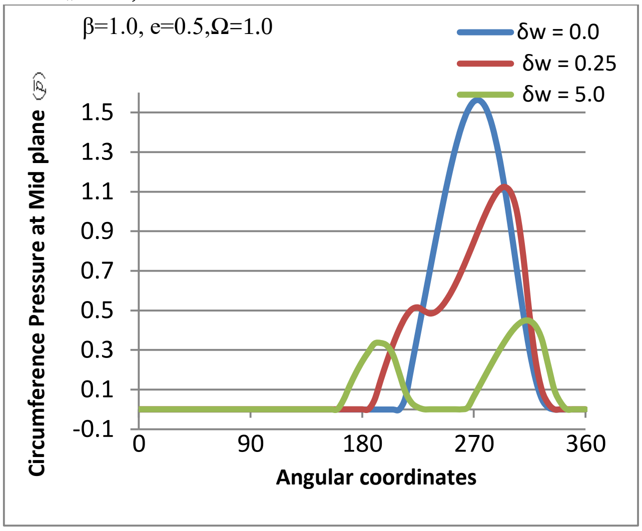

Fig. 5 Circumference Pressure

Figure 5 shows typical pressure distributions for the same eccentricity in the case of unworn bearing i.e δ_w=0.0and on the same bearing at two different stages of wear i.e δ_w= 0.25 and 0.5. The value of maximum pressure at the damaged portion i.e at δ_w= 0.25 and 0.5, at the midplane is reduced due to less resistance over the large clearance. As the $\delta_w$increases the peak pressure shifts towards the divergent zone. The defect also creates a second convergent zone which makes the bearing to behave like a two lobe bearing. The two peaks pressure in worn bearing is mainly due to increased film thickness in the worn out region which causes to lower the pressure. The change in film thickness can be seen in next figure.

4.2 Film thickness

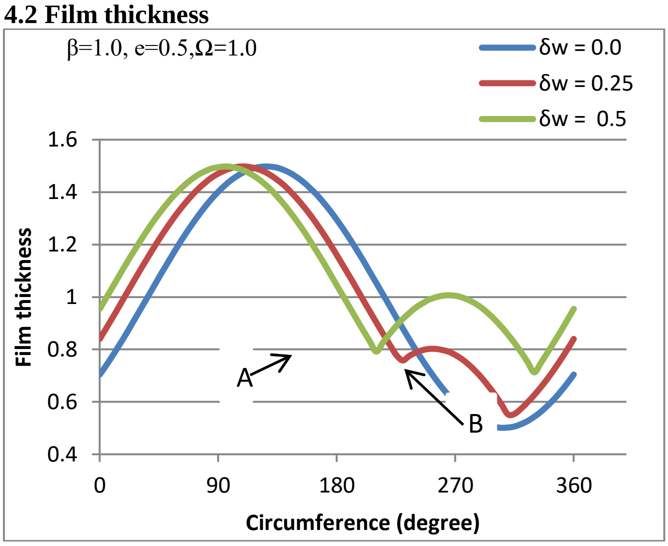

Fig. 6 Fluid Film Thickness

Figure 6 shows the development of the film thickness in the fluid film zone of the bearing for differentvalue of the wear defect. As it is observed that film geometry is significantly modified by the presence of a wear defect, with abrupt changes near the footprint. Aincrease in film thickness and a break of slope can be observed at points A and B i.e. at start and stop of wear region. This configuration creates two pressure field in worn out bearing. At a fixed angular speed of Ω=1.0 the film thickness increases in the wear zone as there is more space in wear zone to accommodate the lubricant which is clearly shown between points A and B. From the above discussion it can be said that increased film thickness in worn region causes less chances of shaft contact with bearing

4.3 Axial Pressure

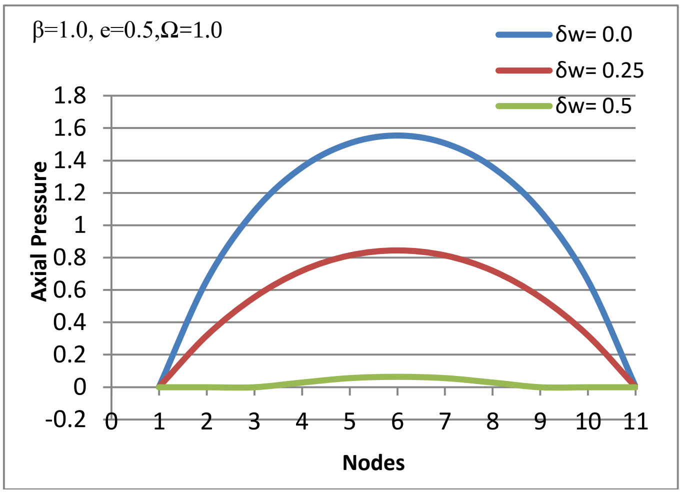

Fig. 7 Axial pressure at lower most portions

It is observed from Figure 7 that the pressure formed at the center node 6 is maximum and the pressure distribution is symmetrical about the center. At same node 6, the axial pressure drops due to wear defect δ_w=0.25 and δ_w = 0.50 due to increased film thickness due to worn region as clearly shown in Figure 7. The pressure at node 1 and node 11 is equal due to boundary conditions which are pressure at bearing ends is at atmospheric pressure and also equal in magnitude at ends.

5. Conclusions

The work reported in the article presents a theoretical study concerning the effect of wear on the performance characteristic parameters of fluid-film journal bearing systems. The wear study due to transient (e.g. start / stop) operations has been carried on fluid-film journal bearing system.

1. The code developed in Fortran 90 is accurate and can be used for analysis of worn journal bearing system.
2. With the increase of wear depth the peak pressure shifts towards divergent zone and creates second divergent zone which causes bearing to behave like a two lobe journal bearing.
3. The change in values of bulk modulus (β) has no effect on performance parameters which is in agreement with reference [5].
4. Increased film thickness in worn region causes less chances of shaft contact with bearing which is positive thing for worn out journal bearings.

It can be concluded from this work is that worn out bearings are not always unsafe and unusable. If we are using bearings in applications where performance is secondary priority then wear in bearing can be advantageous since contact between shaft and bearing is reduced.

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Nomenclature

i	: Index for Axial direction
j	: Index for circumferential direction
x	: Axial coordinate (direction)
z	: Circumferential coordinate (direction)
Δz	: Grid spacing in the circumferential direction
Δx	: Grid spacing in the axial direction
β	: Bulk Modulus
ε	: Eccentricity ratio, (e/c)
ϕ	: Attitude angle, (radians)
e	: Eccentricity
c	: Radial clearance
R	: Shaft radius, m
S	: Somerfield number
$\frac{L}{D}$	: Aspect ratio
m	: Mass, (kg)
h	: Film thickness, m
g	: Switch function (cavitation index)
F_x	: Horizontal force component, N
F_y	: Vertical force component, N
Ω	: Angular velocity of the journal, (radians / second)