Research Paper

BEHAVIOUR AND PREDICTION OF FLUID FILM BEARING PRESSURE

Er.Sunil Tuli

Address for Correspondence

Department of Mechanical engineering, BCET, PTU, PUNJAB, India

ABSTRACT—
The cylindrical (plain) hydrodynamic journal bearing is the most basic hydrodynamic bearing with cylindrical bore plays a vital role in design and stable operation of rotating machinery. For stable operation of machinery a proper understanding of the behaviour of hydrodynamic journal bearing is needed. The main objective of the current work is to make numerical analysis of a fluid film journal bearing system for predicting fluid film pressure distribution at various eccentricities by incorporating ADI technique and developing a computer code in Fortran 90 for pressure prediction. In order to compute the fluid film pressure for fluid film journal bearing system the two dimensional Reynolds equation is solved numerically using FDM. In order to accomplish the numerical study, a computer code developed in Fortran 90 using Salford compiler has been used. The code developed using various algorithms is required to be debugged for accuracy at several steps. This optimum grid size is selected for the present study for computation of pressure distribution and fluid film thickness at various eccentricity ratios.

KEYWORDS—Journal bearings, FDM, JFO model, ADI method.

I.INTRODUCTION

During operation of fluid film hydrodynamic journal bearings two wedges (zones) convergent and divergent. In convergent region of fluid film hydrodynamic journal bearing, the hydrodynamic pressure rises to a peak, decreasing to ambient pressure at the end sides and trailing edge of the thin film. While in the divergent portion pressure is equivalent to ambient pressure. The fluid pressure may drop to ambient, thus releasing the dissolved gases within the lubricant and hence gaseous cavitation, or below ambient to its vapor pressure causing lubricant vaporization and hence rupture of continuous fluid film or vapor cavitation [1]. Hydrodynamic fluid film bearings generally counter balances (supports) the radially applied load with the pressure generated due to rotation of shaft inside the bearing. Therefore this generated pressure due to rotation of shaft has great importance in studying the other performance parameters of the fluid film bearing. This paper mainly focus on the prediction of fluid film bearing pressure distribution for a finite bearing at various eccentricities. Cavitation is a common phenomenon that occurs in fluid film bearing. It can be vaporous or gaseous, and causes damage to fluid film bearings and hence failure of machinery [1]. For obtaining the pressure distribution JFO theory is considered, as it provides boundary conditions which conserve the mass at the interface between the full film and the cavitation zones at rupture and reformation boundary. Due to consistency of this condition with mass conservation principle, a better understanding of the phenomenon of cavitation is available [2]. Therefore performance of fluid film bearing can be predicted more precisely using JFO theory but the main disadvantage of this of this condition is difficulty to accommodate in computer codes [2]. Elrod developed a finite difference algorithm (Elrod algorithm) that makes the applicability of JFO theory in a very simple manner, and this algorithm automatically handles cavitation regions and operates satisfactorily with and without cavitation being present [2]. Therefore this paper focuses on numerical analysis of a fluid film journal bearing system for predicting fluid film pressure distribution at various eccentricities by incorporating ADI technique and developing a computer code in Fortran 90 for pressure prediction. The JFO based model of fluid film journal bearing is selected for this purpose.

II. LITERATURE REVIEW

N.P. Petroff, B. Tower and Osborne Reynolds were three persons who worked in the area hydrodynamic lubrication separately. Both N.P. Petroff and B. Tower provided their concepts via experimentation, which were (concepts) necessary for formulating mathematical modelling.

The essence of hydrodynamic lubrication was first clarified experimentally by British railroad engineer Beauchamp Tower (1845 – 1904) in 1883 [3,4].

Beauchamp Tower has given evidence through experimentation that there was development of pressure in the lubricant film.

Osborne Reynolds in 1886 studied the Tower's experiments and gives theoretical results and it was true that no one before Osborne Reynolds has performed theoretical work [6]. Today the Reynolds' theoretical work has been the basis of journal bearing analysis for over a century. Reynolds in his classical study, while identify the mechanism of hydrodynamic pressure generation in lubricant films, clearly recognized the possible influence of cavitation on bearing behaviour [7]. These conditions do not account for mass conservation in the flow domain [8].

Later in 1930's Reynolds or Swift-Stieber has given condition which calls for a null pressure gradient and ambient pressure at the onset of cavitation [9].

Later Jackobsson and Floberg [10] and Olsson [11] developed a theory named JFO theory.

In order to make extensive study on the fluid film journal bearing system, the current study using JFO cavitated based model of fluid film journal bearing has been adopted.

III. ANALYSIS

This part provides details of analytical formulation and numerical solution procedure. The geometry of a circular journal bearing considered is shown in figure 1 and the control volume is shown in figure 2.

A. Analytical formulation

Elrod algorithm is a mass conservation algorithm and therefore we begin our work by applying principle of mass conservation to control volume shown in Figure 2.

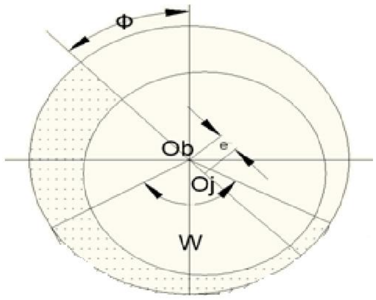

Fig.2 Circular Journal bearing

$$\frac{\partial(h\rho)}{\partial t} + \nabla . \vec{m} = 0 \qquad (1)$$

where $\vec{m} = \dot{m}_x + \dot{m}_z$

$$\dot{m}_X = -\frac{\rho h^3}{12\mu}\frac{\partial P}{\partial x} + \frac{\rho h U}{2}$$

and

$$\dot{m}_Z = -\frac{\rho h^3}{12\mu}\frac{\partial P}{\partial z} \qquad (2)$$

Inserting (2) in (1) will allow us to write [9]

$$\frac{\partial(\rho h)}{\partial t} + \frac{\partial}{\partial x}\left(-\frac{h^3 \rho}{12\mu}\frac{\partial P}{\partial x} + \frac{U\rho h}{2}\right) + \frac{\partial}{\partial z}\left(-\frac{\rho h^3}{12\mu}\frac{\partial P}{\partial z}\right) = 0 \qquad (3)$$

Elrod introduced a variable α, which is known as fractional film content and it is defined as the ratio of density of lubricant to the density of lubricant at the cavitation pressue i.e.

$$\alpha = \rho / \rho_c \qquad (4)$$

Elrod related the density, fractional film content, to fluid film pressure by the relation given below [5]

$$\beta = \rho \frac{\partial P}{\partial \rho} \qquad (5)$$

Integrating (5) with pressure limits

$p \rightarrow p_c$

$$p = p_c + \beta(In\alpha) \qquad (6)$$

For moderate to heavily loaded bearings, the pressure-induced flow within the cavitated regions is small and the flow existing in the cavitation regions is primarily due to shear component, and the pressure is nearly constant [18]. Hence to finish its role (pressure induced term) in cavitation region we use a switch function (g) which automatically removes the pressure effects in cavitated regions. Therefore introduction of switch function (g) into (5) will give

$$g\beta = \rho / \rho_c \left(\frac{\partial P}{\partial \rho / \rho_c}\right) = \alpha \frac{\partial P}{\partial \alpha} \qquad (7)$$

Using (7) in steady state version of (3) will give us a single expression which is valid in both full film and cavitation regions.

$$\frac{\partial}{\partial x}\left(\frac{\rho_c g \beta h^3}{12\mu}\frac{\partial \alpha}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{\rho_c g \beta h^3}{12\mu}\frac{\partial \alpha}{\partial z}\right) = \frac{U}{2}\frac{\partial(\rho_c \alpha h)}{\partial x} \qquad (8)$$

This expression is valid in full film region for g = 1 and in cavitation region for g = 0.

B. NUMERICAL PROCEDURE

Applying principle of mass conservation to control volume will permit us to write [5]

$$\frac{\Delta(\rho h)}{\Delta t} - \left(\frac{\Delta \dot{m}_X}{\Delta x} + \frac{\Delta \dot{m}_Z}{\Delta z}\right) = 0 \qquad (9)$$

As the flow in the x-direction (circumferential direction) is due to both shear and pressure, and in z-direction (axial direction) flow is entirely due to pressure, so we can write

$$\Delta \dot{m}_X = (\Delta \dot{m}_X)_{shear} + (\Delta \dot{m}_X)_{pressure}$$

$$\Delta \dot{m}_Z = (\Delta \dot{m}_Z)_{shear}$$

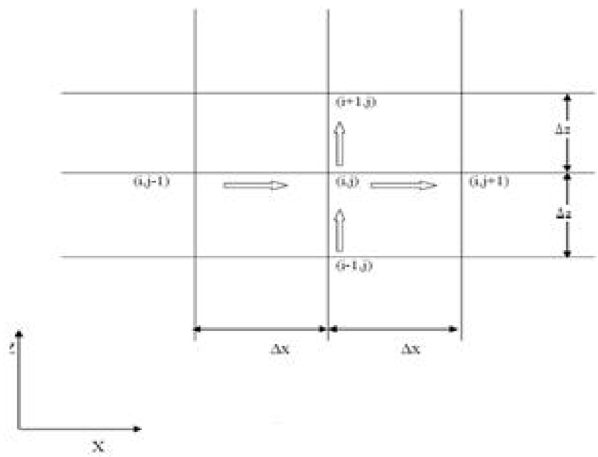

Fig. 2 Control Volume

Elrod proposed the following flow rates [5]

$$\left(\Delta\dot{m}_x\right)_{SHEAR} = \frac{U\ \rho_c}{2}\left[\, h_{i,j-1}\,\alpha_{j-1}\,(1-g_{i,j-1}) - (1-g_{i,j})\,h_{i,j}\,\alpha_{i,j} + \frac{h_{i,j-1}\,g_{i,j-1}\,(2-g_{i,j})}{2} + \frac{g_{i,j}\,h_{i,j}}{2}\,(g_{i,j-1}-2+g_{i,j+1}) - \frac{g_{i,j+1}\,h_{i,j+1}\,g_{i,j}}{2}\right]$$

$$\left(\Delta\dot{m}_x\right)_{PR>} = \left(\frac{\beta\,\rho_c}{12\,\mu\,\Delta x}\left[\begin{array}{l} h^3_{i,j+1/2}\ g_{i,j+1}\ (\alpha_{i,j+1}-1) - (h^3_{i,j+1/2} + h^3_{i,j-1/2}) \\ g_{i,j}(\alpha_{i,j}-1) + h^3_{i,j-1/2}\ g_{i,j-1}\ (\alpha_{i,j-1}-1) \end{array}\right]\right)$$

$$\left(\Delta\dot{m}_z\right)_{PR.} = \left(\frac{\beta\rho_c}{12\mu\Delta z}\left[\begin{array}{l} h^3_{i+1/2,j}g_{i+1,j}(\alpha_{i+1,j}-1)-(h^3_{i+1/2,j}+h^3_{i-1/2,j}) \\ g_{i,j}(\alpha_{i,j}-1)+h^3_{i-1/2,j}\ g_{i-1,j}(\alpha_{i-1,j}-1) \end{array}\right]\right)$$

Inserting above expressions given by Elrod in (9) will permit us to write [5]

$$D_{i,j}\alpha_{i,j} = A_{i,j}\alpha_{i,j+1} + B_{i,j}\alpha_{i,j-1} + E_{i,j}\alpha_{i+1,j} + F_{i,j}\alpha_{i-1,j} + C_{i,j} \quad (10)$$

Where $A_{i,j}, B_{i,j}, E_{i,j}, F_{i,j}, D_{i,j}, C_{i,j}$ are given below [5]

$$\boldsymbol{A_{i,j}} = a\ \left[(\bar{h}_{i,j+1} + \bar{h}_{i,j})^3 g_{i,j+1}\right]$$

$$\boldsymbol{B_{i,j}} = b\left[\bar{h}_{i,j-1}\ (1-g_{i,j-1})\right] + a\left[(\bar{h}_{i,j} + \bar{h}_{i,j-1})^3 g_{i,j-1}\right]$$

$$\boldsymbol{D_{i,j}} = f\left[\left(\bar{h}^3_{i,j} + \bar{h}^3_{i,j}\right)g_{i,j}\right] + b\left[\bar{h}_{i,j}\ (1-g_{i,j})\right] + a\left[(\bar{h}_{i,j} + \bar{h}_{i,j-1})^3 g_{i,j}\ (\bar{h}_{i,j} + \bar{h}_{i,j+1})^3 g_{i,j}\right]$$

$$\boldsymbol{C_{i,j}} = b\left\{\left[\bar{h}^3_{i,j-1}\ g_{i,j-1} + \frac{g_{i,j}\,g_{i,j-1}}{2}\left(\bar{h}_{i,j} - \bar{h}_{i,j-1}\right)\right] - \left[\bar{h}_{i,j}\ g_{i,j} + \frac{g_{i,j}\,g_{i,j+1}}{2}\left(\bar{h}_{i,j+1} - \bar{h}_{i,j}\right)\right]\right\} + a\left[\left(\bar{h}_{i,j} + \bar{h}^3_{i,j}\right)(g_{i,j} - g_{i,j-1}) + \left(\bar{h}_{i,j} + \bar{h}_{i,j+1}\right)^3(g_{i,j} - g_{i,j+1})\right] - f\ \bar{h}^3_{i,j}\left(g_{i+1,j} - 2\,g_{i,j} + g_{i-1,j}\right)$$

$$\boldsymbol{E_{i,j}} = f\ \left(\bar{h}^3_{i,j}\ g_{i+1,j}\right)$$

$$\boldsymbol{F_{i,j}} = f\ \left(\bar{h}^3_{i,j}\ g_{i-1,j}\right)$$

$$\bar{X} = \frac{X}{R}$$

$$\bar{Y} = \frac{Y}{R}$$

$$b = \frac{\Delta\bar{Y}}{4(L/D)}$$

$$a = \frac{\bar{\beta}}{192\pi}\left(\frac{\Delta\bar{Y}}{\Delta\bar{X}}\right)\left(\frac{1}{L/D}\right)$$

$$\bar{h} = \frac{h}{c}$$

$$f = \frac{\bar{\beta}}{24}\left(\frac{\Delta\bar{X}}{\Delta\bar{Y}}\right)$$

$$\bar{P} = \frac{P}{\omega\mu\ \left(\frac{R}{C}\right)^2}$$

$$\bar{W} = \frac{W}{\omega\mu(R^2)\left(\frac{R}{C}\right)^2}$$

Equation (10) was solved using alternating direction implicit (ADI) scheme as it reduces the complexity of the problem by reducing this two-dimensional equation (10) to two separate one-dimensional equations (11) and (12). When these equations are written in matrix form for each grid point they reduces to tridiagonal form and these tridiagonal type of equations can be easily solved by Thomas algorithm [5]. In this scheme the whole iteration cycle consists a sweep in circumferential direction and then sweep in axial direction.

STEP 1:- **Circumferential sweep:-** In this step values of fractional film content at half time level (n+1/2) is calculated at all grid points along the length of the bearing making use of values for fractional film content α at time level (n). Mathematically circumferential sweep gives [5]

$$\left(\frac{\Delta\dot{m}_X}{\Delta X}\right)^{n+1/2} = -\left(\frac{\Delta\dot{m}_z}{\Delta Z}\right)^{n} \quad (11)$$

In expanded form (11) reduces to [5]

$$-A_{i,j}\alpha^{n+1/2}_{i,j+1} + D_{i,j}\alpha^{n+1/2}_{i,j} - B_{i,j}\alpha^{n+1/2}_{i,j-1} = E_{i,j}\alpha^{n}_{i+1,j} + F_{i,j-1}\alpha^{n}_{i-1,j} + C_{i,j} \quad (12)$$

STEP 2:- **Axial Sweep**:-

$$\left(\frac{\Delta\dot{m}_z}{\Delta z}\right)^{n+1}_{PRESSURE} = -\left(\frac{\Delta\dot{m}_x}{\Delta x}\right)^{n+1/2}_{PRESSURE} \quad (13)$$

Here the updated values of fractional film content (α) of level (n+1/2) are used in (13) and in expanded form (13) can be written as [5]

$$-F_{i,j}\alpha^{n+1}_{i+1,j} + D_{i,j}\alpha^{n+1}_{i,j} - E_{i,j}\alpha^{n+1}_{i1,j} = A_{i,j}\alpha^{n+1/2}_{i,j+1} + B_{i,j}\alpha^{n+1/2}_{i,j-1} + C_{i,j} \quad (14)$$

Equation (12) and (14) are solved using Thomas algorithm [5,19].

IV. RESULTS AND DISCUSSION

The steady state 2-D universal mass conservation equation is solved numerically (FDM approach) using ADI technique. Firstly, an optimum computational grid is selected. After trying various grid sizes a 72×10 size is chosen as optimal grid. All results for pressure distribution and fluid film thickness are obtained at different eccentricity ratios corresponding to grid size 72×10.

V. CONCLUSION

- The code developed in Fortran 90 is accurate and can be used for analysis of finite journal bearing system.
- Optimal grid size selected after trying different grid sizes on the basis of accuracy in the value of load carrying capacity and minimum computational time is 72×10.
- Circumferential pressure distribution increases with increase in the value of eccentricity ratio or applied load.
- The maximum pressure value increases with the increase in eccentricity ratio (i.e. radially applied load) for a journal rotating at fixed angular speed thus increasing bearing load carrying capacity at the cost of decrease in minimum film thickness.
- The minimum film thickness decreases with the increase in eccentricity ratio (i.e. radially applied load) for a journal rotating at fixed angular speed.
- Maximum axial pressure at θ = 270 degrees rise with the rise in eccentricity ratio. For a particular value of eccentricity ratio the axial pressure distribution is parabolic with only one maximum value of pressure and the rate of pressure rise is same as that for pressure fall.

NOMENCLATURE

ε : Eccentricity ratio, *(e/c)*
ϕ : Attitude angle, (*radians)*
e : Eccentricity
c : Radial clearance
ρ : Density of lubricant, $(\frac{kg}{m^3})$
μ : Dynamic viscosity, $\frac{N}{s}m^2$
p : Fluid film pressure, $\frac{N}{m^2}$
p_c : Cavitation pressure, $\frac{N}{m^2}$
h : Film thickness, *m*
g : Switch function
F_x : Horizontal force component, *N*
F_y : Vertical force component, *N*
Ω : Angular velocity of the journal
θ : Fractional film content $(\frac{\rho}{\rho_c})$
W : Load carrying capacity, *N*
δ_0 : Maximum Wear depth
α_s : Wear start point
α_f : Wear end point
o_b : Bearing centre
o_j : Journal centre
δ_w : wear depth, mm

REFRENCES

[1] Hirani H, Athre K, Biswas S ,"A Simplified Mass Conserving Algorithm for Journal Bearing under Large Dynamic Loads", International Journal of Rotating Machinery , Vol. 7, No. l, ,2001, pp. 41-51.

[2] Tower, B. (1883), "First Report on Friction-Experiments (Friction of Lubricated Bearings)",Proc. Institution of Mechanical Engineers, November 1883, pp. 632 – 659.

[3] Tower, B. (1885), "Second Report on Friction-Experiments (Experiments on the Oil Pressure in a Bearing)", Proc. Institution of Mechanical Engineers, January 1885, pp. 58 - 70.

[4] Vaidyanathan K. and Keith T.G., "Numerical Prediction of Cavitation in Noncircular Journal Bearings," Tribol. Trans., vol. 32, 1989, pp 215-224Fillon M. and Bouyer J., "Thermo Hydrodynamic Analysis of a Worn Plain Journal Bearing," Tribol. Int., vol. 37, 2004, pp 129-136

[5] Reynolds, O. (1886), On the Theory of Lubrication and its Application to Mr. Beauchamp Tower's Experiments, Including an Experimental Determination of the Viscosity of Oiive Oil. PMos. Trans. R. 5oc. London, vol.177, pp. 157-234.

[6] Hori, Y. (2002), "Hydrodynamic Lubrication" Published by Yokendo Ltd., japan.

[7] Stachowiak, G.W. and Batchelor, A.W (1993), "Engineering Tribology" Tribology series 24, Elsevier Science Publishers B.V., Amsterdam, The Netherlands.

[8] Muthusubramaniyan, S. (2006), "Simulation of Cavitation in Engine Bearings" UMI Number: 1430440.

[9] Jakobsson, B. and Floberg, Leif. (1957), "The Finite Journal Bearing Considering Vaporization," Trans. Chalmers Univ. Tech., Goteborg, No. 190, 1957.

[10] Olsson, K.O. (1965), "Cavitation in Dynamically Loaded Bearings," Chalmers University of Technology, Goteborg.