Analysis of Hydrodynamic Journal Bearing Using Fluid Structure Interaction Approach

Dinesh Dhande[1], Dr D W Pande[2], Vikas Chatarkar[3]

1. Assistant Professor, AISSMS College of Engineering, Pune -1
2. Professor and Dean R&D, College of Engineering, Pune-5
3. Assistant Professor, RMD Sinhgad College of Engineering,Pune

Abstract— **Hydrodynamic journal bearings are analyzed by using Computational fluid dynamics (CFD) and fluid structure interaction (FSI) approach in order to find deformation of the bearing.**
Journal bearing models are developed for different speeds and eccentricity ratios to study the interaction between the fluid and elastic behaviour of the bearing. The nodal fluid forces computed by CFD are used in order to find deflection of the bearing. Cavitations in the bearing are neglected by setting all negative pressures to ambient pressures. The CFD results were compared in order to validate the model with the experimental work carried by Ferron et al [8] and a good agreement were found. It is observed that CFD-FSI method provides a useful platform to study the combined effect of hydrodynamics and elastic behaviour of the bearing. It is observed that the deformations of the bearing are significant and should be considered in order to predict accurate performance of the hydrodynamic journal bearings.

Keywords— **Fluid Structure Interaction, Deformation, Static P Pressure**

I. INTRODUCTION

Now-a-days the research is focused on increasing the output of the internal combustion engine and to reduce their weights. Hence bearing housing of the connecting rod big end bearings and main bearings are subjected to severe operating conditions. The increase in bearing loads and desire to reduce the dimensions and component masses in modern combustion engines leads to substantial elastic deformations in the connecting rod and main bearings which alternatively affects the properties of the lubricating fluid and henceforth the performance. In such applications, the conventional assumption of rigid bushing fails to predict the accurate performance of the bearing and hence the combination of hydrodynamic lubrication with the structural analysis i.e. an elasto-hydrodynamic analysis must be considered.

Huiping Liu *et al*.[1] studied hydrodynamic journal bearings with elastic insert and found that the elastic deformation of the bearing had a significant influence on the rotor–bearing system, particularly for the polymeric-based materials. Kuznetsov *et al*.[2] experimentally analyzed compliant journal bearings considering liner deformations . A thin layer of PTFE coating was applied in bearing performance in order to study steady state characteristics. It was observed that a slight increase in radial clearance in the PTFE lined bearing results in improved operating characteristics (higher load carrying capacity and lower power loss) compared to the white metal bearing. Qiang Li *et al*.[3]

applied CFD method based on the fluid-structure interaction (FSI) technique to predict the transient flow field of a misaligned journal bearing in a rotor-bearing system and concluded that journal misalignment in the bearing hole should be included in calculating the performance characteristics of a journal bearing. ATTIA HILI Molka *et.al*.[9] used finite element analysis to predict the effect of the bearing elastic deformation on the performance characteristics of a cylindrical journal bearing. It was noted that flexibility of bearing liner has a significant influence on the performance of a cylindrical journal bearing. In the present work, the efforts are taken to analyze cylindrical journal bearing using CFD-FSI approach.

II. ANALYSIS

The geometry and the co-ordinate system of the journal bearing is as shown in fig 1. The journal rotates with a angular velocity, ω. The journal remains in equilibrium position under the action of external load, W and developed hydrodynamic pressure. The journal centre O is eccentric to the bearing centre O'. The film thickness h(θ) varies from its maximum value hmax at bearing angle θ = 0 to its minimum value, hmin at θ = π.
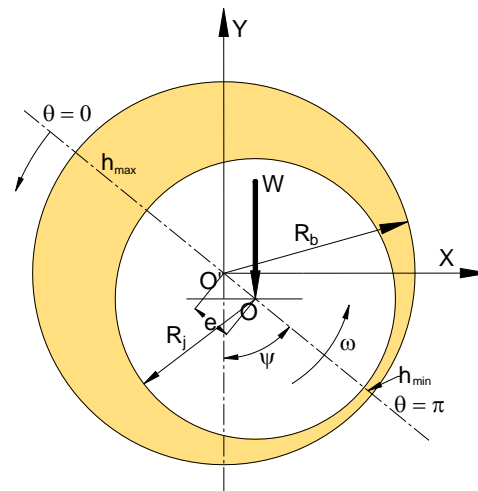

Fig 1. Schematic of Journal bearing geometry

III. ASSUMPTIONS

Initially a journal bearing with rigid bushing as shown in fig.1 is considered in order to find hydrodynamic forces with steady state condition. The flow is laminar and isothermal and a constant vertical load W is applied on the journal.

IV. EQUATIONS TO BE SOLVED

FLUENT is based on solution of conservation equations for mass and momentum. For thermal considerations energy conservation equation is solved. Additional transport equations are solved when for turbulent flows. In this work, results for laminar flow are presented. In Fluid structure interaction fluid flow exerts a pressure on solid structure causing it to deform. The multiphysics capability of ANSYS is used to find deformation in the bearing.

The governing finite element equations are:

$$[M_s]\{\ddot{U}\} + [K_s]\{U\} = [F_s] + [R]\{P\}$$

$$\begin{bmatrix} M_s & 0 \\ \rho R^T & M_f \end{bmatrix} \begin{Bmatrix} \ddot{U} \\ \ddot{P} \end{Bmatrix} + \begin{bmatrix} K_s & R \\ 0 & K_f \end{bmatrix} \begin{Bmatrix} U \\ P \end{Bmatrix} = \frac{F_s}{F_f}$$

Where $[M_s]$ is the structural mass matrix; $[M_f]$ is the fluid mass matrix; $[F_s]$ and $[F_f]$ is the structural and fluid force matrix; $[R]$ is a coupling matrix that represents the effective surface area associated with each node in fluid structure interface.

V. GEOMETRICAL MODEL

The bearing dimensions used in the present work are as given below:

TABLE I

JOURNAL BEARING PROPERTIES

Journal Radius, R_b	50 mm
Bearing Length, L	80 mm
Radial Clearance, C	145 μm
Rotational Speed Range, N	4000- 10000 RPM
Lubricant viscosity, μ	0.0277 Pa-sec
Lubricant density, ρ	860 kg/m3
Lubricant Specific Heat, Cp	2000 J/kg °C
Bearing thickness, t	5 mm
Young's modulus, E	200 GPa
Poison's ratio	0.3

The model is constituted as one cylinder with a radius of R_b 50 mm and another one with a radius of 50.145 mm, tra999nslated by the value of eccentricity. The geometry and meshing is done with the help of pre-processor GAMBIT and the CFD- FSI analysis is done with ANSYS Fluent and multiphysics modules.

For meshing, the fluid ring is divided into two connected volumes. Then all thickness edges are meshed with 360 intervals, and then volumes are meshed with the "Hex/Wedge e cooper" method. The mesh is carried out using ANSYS GAMBIT. A hexahedral structure mesh is used. (Fig. 2) 100 divisions were taken along the length. So the total number of elements is 75840. As the load is assumed to be constant, the eccentricity ratio depends on the pressure equilibrium over the journal surface. The mesh was generated for different values of eccentricity ratios from 0.2 mm to 0.8. The mesh quality is always around 0.5 for all generated elements.
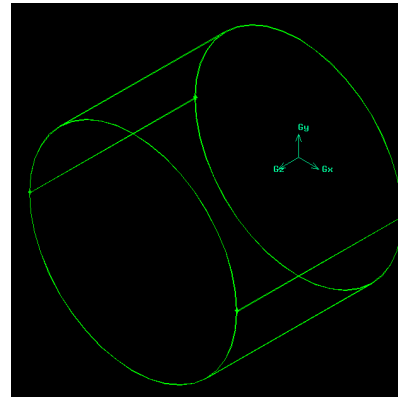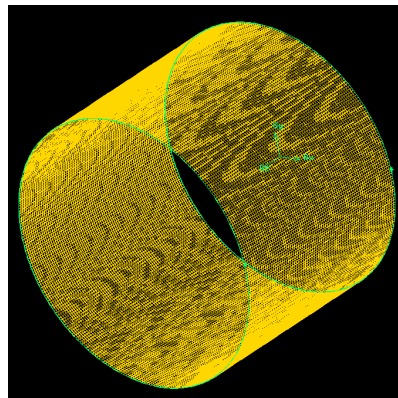

Fig 2. Geometry before meshing

Fig 3. Meshed Model

VI. BOUNDARY CONDITIONS

The bearing wall is considered as stationary and journal is modelled as moving wall. The sides of the lubricant volume have been assigned with a zero pressure condition, meaning that the lubricant is free to flow there. The cavitation within the lubricant was modelled using the half Sommerfeld boundary condition.

$$p - p_a \geq 0, at \ \pi \leq \theta \leq 2\pi, z = 0, z = L$$

The half Sommerfeld condition, utilized in the present work, neglects all negative pressures in the diverging part of the

fluid film, which are physically unrealistic. The half Sommerfeld condition, offers sufficient accuracy, fast convergence, and is selected in this work to accelerate the solution of the CFD problem. The use of zero pressure boundary condition at the sides of the bearing implies the leakage of the lubricant at the sides. The boundary condition for entry of the lubricant would simply be unnecessary because of the half Sommerfeld boundary condition. In other words, since the lubricant enters the bearing space at atmospheric pressure, a separate boundary condition for the lubricant inlet would be overplayed to half Sommerfeld and thus would be redundant. The Reynolds boundary condition, not utilized here, assumes that the positive pressure curve terminates with a zero gradient in the divergent part of the film; it gives in some cases more accurate results than the half Sommerfeld boundary condition. Nonetheless, it is still an approximation to the transition from single-phase flow to multi-phase flow, and is computationally more demanding. The external surface of the bearing was fixed. The lubricant film was created in the CFD module of the software. Fluid–structure interface pairs were used to connect the fluid and solid domains, and a non-slip boundary condition was applied. As with the Fluid models, the desired rotational speed was applied to the inner surface of the oil film for all CFD–FSI models.

VII. RESULTS

Fig. 4 shows the static pressure contour plot for $\varepsilon = 0.2$ and L/D ratio = 0.8 by considering negative pressures and fig 5 shows static pressure contour plot neglecting negative pressures.
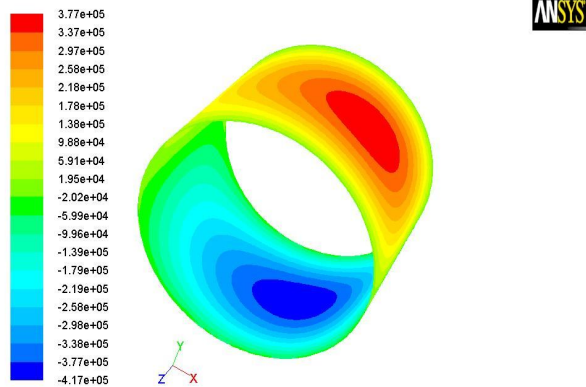

Fig 4. Static Pressure Contour plot for $\varepsilon = 0.2$ and L/D = 0.8 with negative pressures
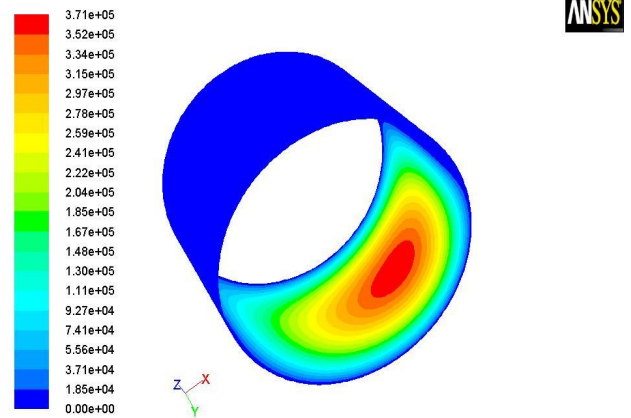

Fig 5. Static Pressure Contour plot for $\varepsilon = 0.2$ and L/D = 0.8 without negative pressures

Fig 6 shows the stress distribution and Fig 7 shows displacement vector for $\varepsilon = 0.2$ and L/D ratio = 0.8.
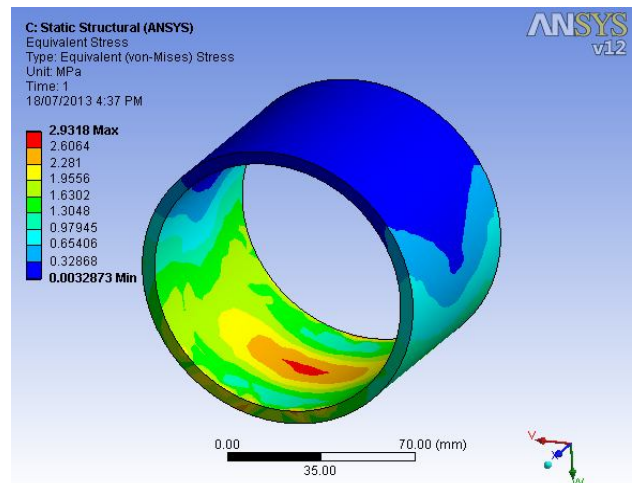

Fig 6. Stress distribution for $\varepsilon = 0.2$ and L/D = 0.8
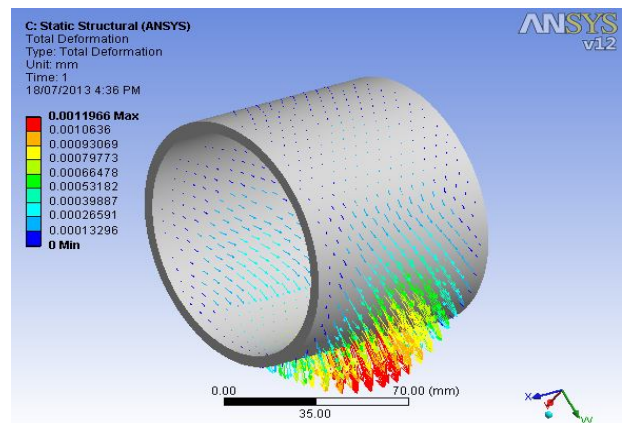

Fig 7. Deformation vector for $\varepsilon = 0.2$ and L/D = 0.8

Fig.8 shows pressure distribution in circumferential direction at mid plane for various eccentricity ratios. Fig 9. Shows variation of maximum displacement with eccentricity ratios.
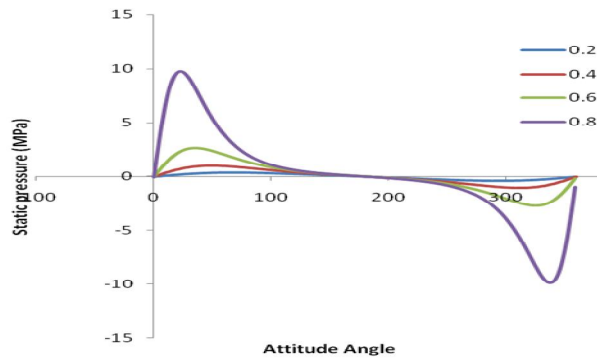

Fig 8. Pressure distribution in circumferential direction at mid plane for various eccentricity ratios.
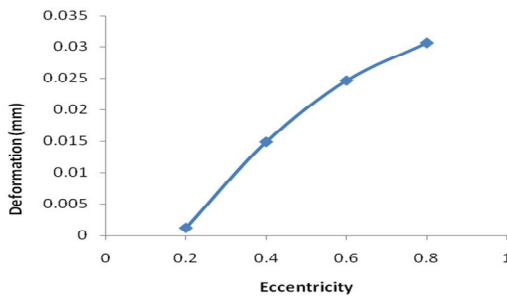

Fig 9. Variation of maximum deformation for various eccentricity ratios.

Fig.10 shows pressure distribution in circumferential direction at mid plane for various rotational speeds.
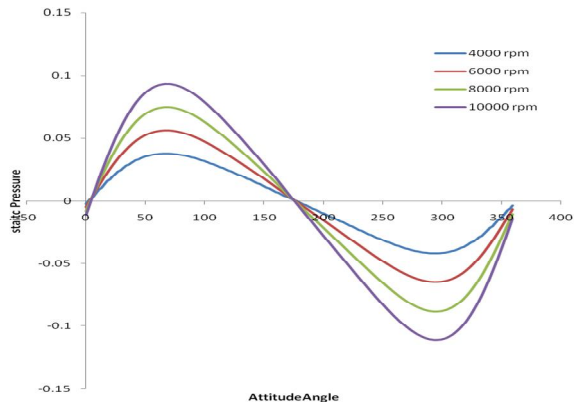

Fig 10. Pressure distribution in circumferential direction at mid plane for various RPMs

VIII. CONCLUSION

The elastohydrodynamic study of the hydrodynamic journal bearing using computational fluid dynamics (CFD) and fluid structure interaction (FSI) approach. The simulation result of pure fluid model has a good agreement with previous experimental work done. This technique gives the deformation of the bearing due to action of hydrodynamic forces developed which is important for accurate performance of the bearings operation under severe conditions It is observed that there is substantial amount of deformation of the bearing.

REFERENCES

[1] Huiping Liu, Hua Xu , Peter J. Ellison ,Zhongmin Jin , "Application of Computational Fluid Dynamics and Fluid–Structure Interaction Method to the Lubrication Study of a Rotor–Bearing System ", *Tribol Lett* (2010) 38:325–336, DOI 10.1007/s11249-010-9612-6

[2] Evgeny Kuznetsov a, Sergei Glavatskih a, b, n, Michel Fillon, "THD analysis of compliant journal bearings considering liner deformation", *Tribol Lett* (2011) , doi: 10. 1016/ j. triboint. 2011 .05 .013

[3] Qiang LI, Shu-lian LIU, Xiao-hong PAN, Shui-ying ZHENG," A new method for studying the 3D transient flow of misaligned journal bearings in flexible rotor-bearing systems, *Journal of Zhejiang University-SCIENCE A (Applied Physics & Engineering), 2012 13(4):293-310.*

[4] ADRIAN SESCU,1 CARMEN SESCU,1 FLORIN DIMOFTE, SORIN CIOC, ABDOLLAH A. AFJEH, and ROBERT HANDSCHUH, "A Study of the Steady-State Performance of a Pressurized Air Wave Bearing at Concentric Position", , *Tribology Transactions, 52:* 544-552, 2009

[5] M. Deligant, P. Podevin, G. Descombes ,"CFD model for turbocharger journal bearing performances" , *Applied Thermal Engineering 31 (2011)* ,
doi: 10.1016 /j. applthermaleng. 2010. 10.030.

[6] Dimitrios A. Bompos, Pantelis G. Nikolakopoulos , " CFD simulation of magneto rheological fluid journal bearings " , *Simulation Modeling Practice and Theory 19 (2011),* 1035–1060

[7] K.P. Gertzos, P.G. Nikolakopoulos, C.A. Papadopoulos ,"CFD analysis of journal bearing hydrodynamic lubrication by Bingham lubricant" , *Tribology International* 41 (2008) 1190–1204

[8] J. Ferron, J. Frene, R. Boncompain ,"A Study of the Thermohydrodynamic Performance of a Plain Journal Bearing Comparison Between Theory and Experiments" **,** *Transactions of the ASME*, Vol. 105, JULY 1983, pg.422-428.

[9] Attia Hili Molka, Bouaziz Slim , Mattar Mohamed , Fakhfakh Tahar , Haddar Mohamed, " Hydrodynamic and ElastoHdroDynamic Studies of Cylendrical Journal Bearing " , Journal of hydrodynamics, 2010,22(2):155-163,
DOI: 10.1016/S1001-6058(09)60041-X.

[10] P G Tucker, and P S Keogh," A generalized computational fluid dynamics approach for journal bearing performance prediction" , 1995, *Instn Mech Engrs, Part J: Journal of Engineering Tribology* Vol.209,pg.99-108

[11] J. Padhye, V. Firoiu, and D. Towsley, "A stochastic model of TCP Reno congestion avoidance and control," Univ. of Massachusetts, Amherst, MA, CMPSCI Tech. Rep. 99-02, 1999.