Dhande et al. 2016. *Int. J. Vehicle Structures & Systems*, 8(4), 224-228
ISSN: 0975-3060 (Print), 0975-3540 (Online)
doi: 10.4273/ijvss.8.4.08

International Journal of Vehicle Structures & Systems
Available online at www.maftree.org/eja

Numerical Analysis of Hydrodynamic Journal Bearing Lubrication using Computational Fluid Dynamics and Fluid Structure Interaction Approach

Dinesh Y. Dhande[a] and D W Pande[b]

[a]*Dept. of Mech. Engg., AISSMS College of Engg., Pune, Maharashtra, India*
Corresponding Author, Email: dydhande@aissmscoe.com
[b]*Dept. of Mech. Engg., College of Engg., Pune, Maharashtra, India*
Email: dean@aissmscoe.com

ABSTRACT:

Now-a-days, journal bearings are subjected to severe loads and higher operating speeds causing generation of high hydrodynamic pressures which in turn deform the bearing shell thus modifying the lubricating film in the operating region. Hence, there is need for optimized bearing performance parameter estimation considering the realistic change in lubricating film along with less computational time. In this paper, response surface optimization module coupled with static structural and fluent, available in ANSYS workbench is used for analysing the performance of the bearing. The optimization is based on Response Surface evaluations. It has been observed that the computation time is considerably reduced. The bearing is analysed for various rotational speeds and eccentricity ratios to obtain load carrying capacity and pressure distribution. It is observed that the results are following the expected trend i.e. as speed increases the load carrying capacity as well as maximum pressure is increasing.

KEYWORDS:

Hydrodynamic journal bearings; Fluid structure interaction; Response surface optimization

CITATION:

D.Y. Dhande and D.W. Pande. 2016. Numerical analysis of hydrodynamic journal bearing lubrication using computational fluid dynamics and fluid structure interaction approach, *Int. J. Vehicle Structures & Systems*, 8(4), 224-228. doi:10.4273/ijvss.8.4.08.

ACRONYMS AND NOMENCLATURE:

C	Radial clearance (mm)
e	Eccentricity between shaft and bearing (mm)
h_{min}	Minimum film thickness (mm)
h_{max}	Maximum film thickness (mm)
h	Fluid film thickness at an angle θ, (mm)
L	Bearing length, (mm)
N	Shaft rotation speed, RPM
O	Shaft centre
O'	Bearing centre
W	Load, N
ψ	Attitude angle (°)
ε	Eccentricity ratio = (e/C)
μ	Viscosity of the Lubricant, Pa-sec

1. Introduction

Hydrodynamic journal bearings are widely used due to their simplicity and better damping characteristics in high load, high speed and high precision applications such as gas turbines, electric generators, marine propellers, hydro turbines, IC Engines, hard disk drives and turbo generators. In hydrodynamic journal bearing analysis two simplified approaches are used: One is 2D analysis with cavitation approximation and constant viscosity neglecting viscous heating and second is approximation of the films thermal interaction with solid components such as shaft or bearing and neglect elastic deformation in the components. Even though these approaches give solutions they are not accurate enough to the designs which demand accurate and detailed hydrodynamic bearing performance including elastic deformations of the components. Previously used approaches include steady state simulations with free floating shafts and transient simulation for CFD and structural analysis. The first approach doesn't work as any imbalance in the shaft allowed the contact between shaft and bearing and hence perfect force balance is never achieved.

Second approach requires long simulation time to reach to the equilibrium position and hence is too slow. The transient structural coupled to steady state CFD approach works but requires expertise. Also it requires artificial dampers to stabilize the structural motion. Also there is need to consider bearing deformations due to developed hydrodynamic pressure forces developed and remodel the geometry accordingly for realistic modelling of hydrodynamic journal bearing. In this work, optimization approach is used to study the effect of rotational speed on pressure distribution and load carrying capacity of the bearing by taking into considerations the bearing deformations and dynamically remodelling of bearing to get optimized solution of the problem.

2. Problem formulation

The outer part is modelled as bearing and inner rotating part is modelled as journal. The gap between journal and bearing is considered as a lubricant volume. Initially the shaft position is assumed and fluid reaction forces are estimated using fluent. These fluid reaction forces are coupled with static structural in order to find deformation in the bearing. These deformations are again fed into the geometry and the geometry and mesh is modified accordingly. The solution is optimized for the position of the shaft for the specified speed and load. The results are verified with the developed experimental setup. In the present work the geometry considered is as shown in Fig. 1. O' is the bearing centre and O is the journal or shaft centre, 'e' is the eccentricity of the bearing, ψ is the attitude angle h_{min} and h_{max} are the minimum and maximum film thickness respectively and L is the bearing length. The external load W is assumed as acting vertically along Y axis and is constant. With the geometrical considerations, the fluid film thickness is given as:

$$h = C + (1 + \varepsilon \cos\theta) \quad (1)$$

where C is radial clearance and ε = e/C is the eccentricity ratio. Fig. 2 shows CAD model used in simulation.

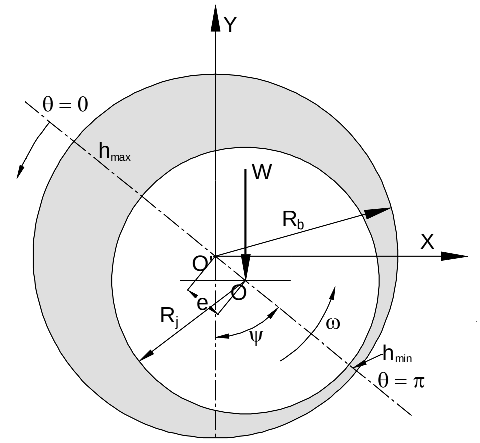

Fig. 1: Bearing geometry and parameters

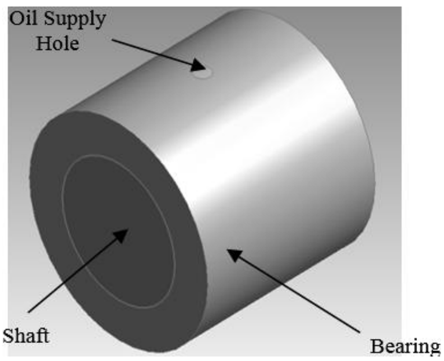

Fig. 2: CAD model

Since the load applied is constant, the shaft eccentricity and attitude angle depends on fluid reaction forces on the shaft. Hence to model the eccentricity, the dynamic meshing technique is used. As the clearance is very small as compared to journal and bearing dimensions, hexahedral cells are used for the meshing with a meshing size of 5E-04 m is used. The meshing used is as shown in Fig. 3.

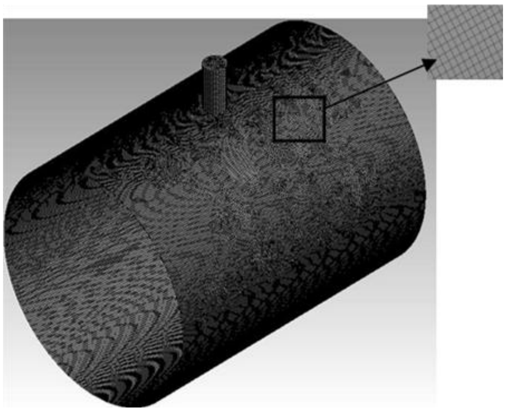

Fig. 3: Meshing

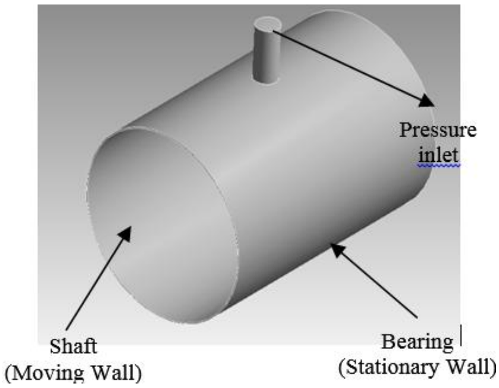

Fig. 4: CFD fluent boundary conditions

The Nervier stokes equations are solved and 3D double precision pressure based steady state analysis is carried out. As the Reynolds number *Re* is very low, laminar flow conditions are used. The lubricant supply hole is specifies as "pressure inlet" and the sides of the lubricant are specified as "pressure outlet" with gauge pressure as zero. The bearing is modelled as "stationary wall" and the shaft is modelled as "moving wall" with an absolute rotation speed. Initially the shaft axis position is defined by arbitrary value of eccentricity and attitude angle and these values are given as input to shaft rotation axis origin. To model the change in thickness of fluid domain, dynamic mesh technique in FLUENT is used. The mesh is transferred to fluent for flow analysis. The smoothing mesh method is used with a convergence tolerance of 10e-6 and number of iterations equal to 50. The dynamic mesh zones defined are: (a) bearing as ''stationary'', (b) the two sides and the lubricant volume set to ''deforming'' with ''spring smoothing'' method, and (c) the journal as system coupled. The boundary conditions used are shown in Fig.4. The bearing details are given in Table 1.

Table 1: Bearing details

Parameters	Values
Shaft diameter, d	45 mm
Clearance, C	300 μm
Length of the bearing, L	67.5 mm
Viscosity of the lubricant, μ	0.0227 Pa-sec
Density of the lubricant, ρ	860 kgm^{-3}
Steel shaft	E = 210 GPa, ρ_s = 7850 kgm^{-3} ν = 0.3
Aluminium bearing	E = 70 GPa, ρ_A = 2700 kgm^{-3} ν = 0.334

In solution, mass flow at inlet and outlet along with pressure force on shaft in X and Y direction is monitored. For optimization, the fluent and static structural are coupled and response surface optimization module is attached to both. In response surface optimization, initially various design points are solved. The design of experiments is the initial step building a response surface over the design space. This section describes the selected input parameters and their variation range, the chosen design of experiments type, and the generated matrix of experiments. The explored design space is defined by the range of variation of the input parameters. The response surface is a meta-model built from the design of experiments for an efficient exploration of the design space. This section describes the selected type of meta-model, including its properties, the obtained quality, and the generated response points and charts. The minimum and maximum section reports the minimum and maximum values for each output parameter. These values are approximations found by the min-max search on the response surface. The optimization is based on response surface evaluations. this section describes the chosen optimization type and the generated candidates and charts. The explored design space is defined by the range of variation of the input parameters eccentricity and attitude angle. Then optimization is set for maximizing the vertical load and minimizing the x-imbalance.

3. Results and discussions

The bearing is analysed for four speeds viz. 1000 rpm, 2000 rpm, 3000 rpm and 4000 rpm. Fig. 5 shows the contour plot of circumferential pressure distribution over the journal. Similarly pressure distribution is plotted against circumferential length for constant eccentricities and varying speed of the journal as shown in Fig. 6. It has been observed that as speed increases, the maximum pressure increases.

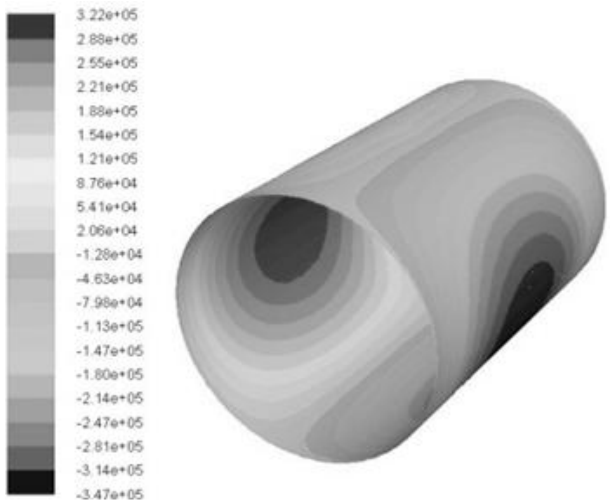

Fig. 5: Circumferential pressure distribution at ε = 0.6 and N = 4000 rpm

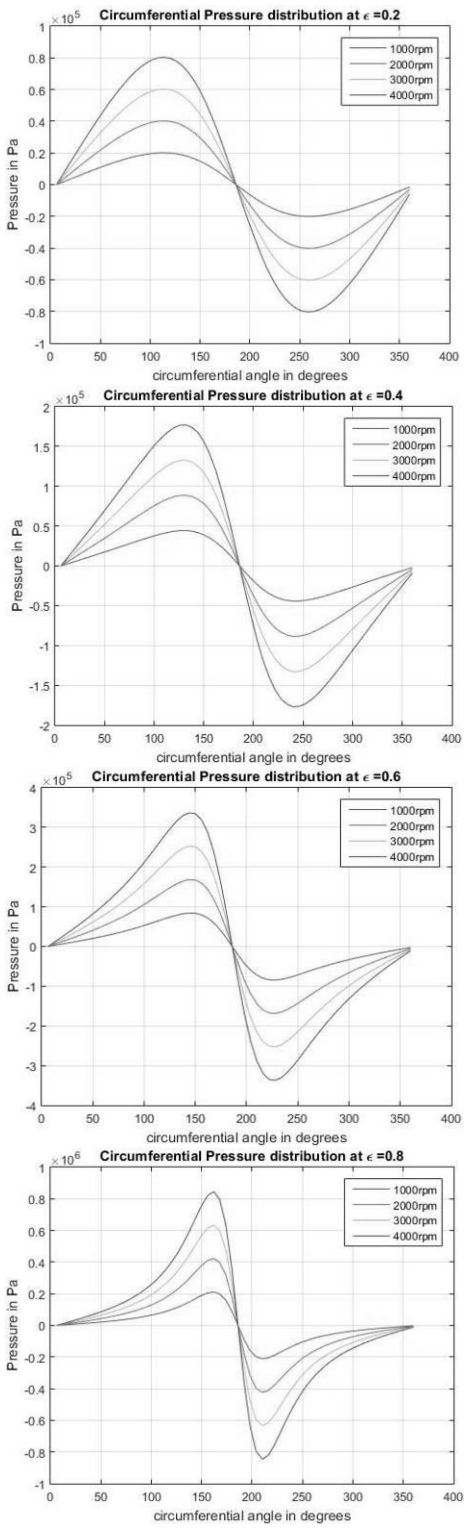

Fig. 6: Circumferential pressure distribution for constant eccentricity ratio and various speeds

Fig. 7 shows the circumferential pressure distribution for constant speed with various eccentricity ratios. It is observed that as the eccentricity ratio increases, the maximum pressure also increases. Fig. 8 shows the variation of the load carrying capacity with shaft rotational speed of the bearing for various eccentricity ratios. The increase in load capacity is less at lower eccentricity ratios (0.2 to 0.6) but more at 0.8 and 0.9. Fig. 9 shows the variation of load capacity with various speeds.

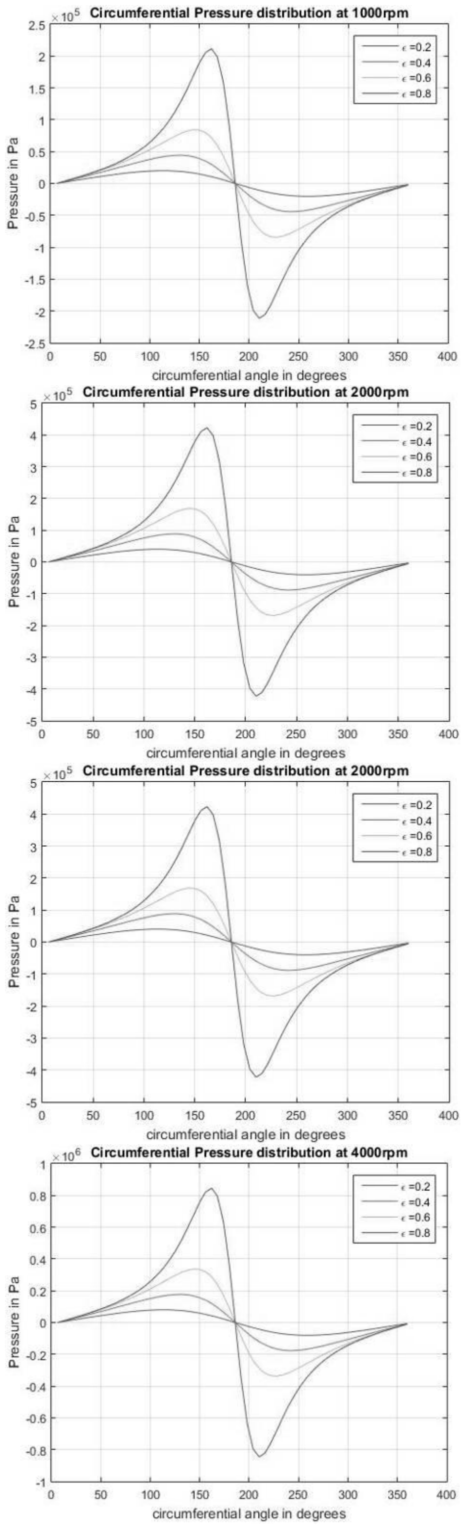

Fig. 7: Circumferential pressure distribution for constant speed and various eccentricity ratios

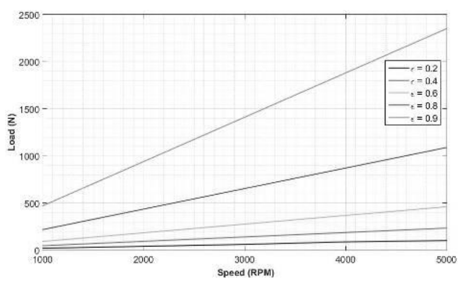

Fig. 8: Load carrying capacity of the journal bearing at various eccentricity ratios

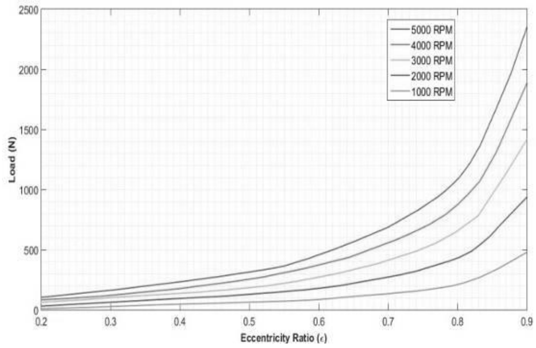

Fig. 9: Load carrying capacity of the journal bearing at various speeds

4. Conclusions

In this paper, response surface optimization module coupled with static structural and fluent, is successfully used for analysing the performance of the bearing. It has been observed from the pressure plots that the peak value of the pressure shifts towards the mid plane of the bearing for constant eccentricity ratio with reduction in the range of the peak value. On the other hand, for constant speeds, the peak value of the pressure remains at constant position with sudden increase in peak from eccentricity ratio value 0.6 to 0.8. At lower values of eccentricities the pressure is not sufficient to separate the shaft and the bearing surface. The load capacity increases with increase with eccentricity ratio as well as speed. The more increase is observed at eccentricity ratio 0.8. So based on the study carried out eccentricity ratio 0.8 and speed 4000 is recommended for the operation of the bearing. Load carrying capacity of the bearing also found increasing with increase in speed. It is found that this method reduces the computation time than alternative approaches and helps in predicting realistic solutions of performance of the bearing by considering the modified deformed geometry of the bearing due to action of hydrodynamic forces.

REFERENCES:

[1] H. Liu, H. Xu, P.J. Ellison and Z. Jin. 2010. Application of computational fluid dynamics and fluid-structure interaction method to the lubrication study of a rotor–bearing system, *Tribol. Letters*, 38(3), 325-336. http://dx.doi.org/10.1007/s11249-010-9612-6.

[2] E. Kuznetsov, S. Glavatskih and M. Fillon. 2011. THD analysis of compliant journal bearings considering liner deformation, *Tribology Int.*, 44(12), 1629-1641. http://dx.doi.org/10.1016/j.triboint.2011.05.013.

[3] Q. Li, S. Liu, X. Pan and S. Zheng. 2012. A new method for studying the 3D transient flow of misaligned journal bearings in flexible rotor-bearing systems, *J. Zhejiang University-Science A (Applied Physics & Engg.)*, 13(4), 293-310.

[4] A. Sescu, C. Sescu, F. Dimofte, S. Cioc, A.A. Afjeh and R. Handschuh. 2009. A study of the steady-state performance of a pressurized air wave bearing at concentric position, *Tribology Trans.*, 52(4), 544-552. http://dx.doi.org/10.1080/10402000902774275.

[5] M. Deligant, P. Podevin and G. Descombes. 2011. CFD model for turbocharger journal bearing performances, *Applied Thermal Engg.*, 31(5), 811-819. http://dx.doi.org/10.1016/j.applthermaleng.2010.10.030.

[6] D.A. Bompos and P.G. Nikolakopoulos. 2011. CFD simulation of magneto rheological fluid journal bearings, *Simulation Modeling Practice and Theory*, 19(4), 1035-1060. http://dx.doi.org/10.1016/j.simpat.2011.01.001.

[7] K.P. Gertzos, P.G. Nikolakopoulos and C.A. Papadopoulos. 2008. CFD analysis of journal bearing hydrodynamic lubrication by Bingham lubricant, *Tribology Int.*, 41(12), 1190-1204. http://dx.doi.org/10.1016/j.triboint.2008.03.002.

[8] J. Ferron, J. Frene and R. Boncompain. 1983. A study of the thermohydrodynamic performance of a plain journal bearing comparison between theory and experiments, *Trans. of the ASME*, 105(3), 422-428. http://dx.doi.org/10.1115/1.3254632.

[9] A.H. Molka, B. Slim, M. Mohamed, F. Tahar and H. Mohamed. 2010. Hydrodynamic and elasto hydrodynamic studies of cylendrical journal bearing, *J. Hydrodynamics*, 22(2), 155-163. http://dx.doi.org/10.1016/S1001-6058(09)60041-X.

[10] P.G. Tucker and P.S. Keogh. 1995. A generalized computational fluid dynamics approach for journal bearing performance prediction, *Int. J. Multiphase Flow*, 22(1001), 99-108. https://doi.org/10.1243/pime_proc_1995_209_412_02.