Thermoelastohydrodynamic Behavior of a Journal Bearing with Turbulent Flow

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Abstract: Turbulence is an irregular fluid motion in which the various flow properties such as velocity and pressure show random variations with time and position. A number of authors proposed different solutions e.g. for pressure distribution, temperature prediction and Thermo-Hydrodynamic (THD) analyses. In a fluid film bearing, the pressure in the oil film satisfies the Reynolds equation with a variation in the thickness of the lubricating film. In the presented cases of fluid-structure interaction analyses, all important phenomena accompanying bearing operation are considered, e.g. lubricant flow, structure movements and their deformations as well as heat transfer in case of thrust bearing. In this paper, the authors have developed an empirical relationship to determine the effect of lubrication when considering thermoelastohydrodynamic (TEHD) lubrication with turbulent flow. The critical point of this work is to import the matrix data (the pressure and temperature fields...) from the fluid domain to the internal surface of the bearing with a precision of the mesh especially in the contact surface. The results are presented in the median plane as a function of the bearing angle. A parametric study deals with the influence of rotation speed and the type of turbulence model on the pressure, temperature, deformation and stress intensity fields.

Key words: TEHD, journal bearing, computational fluid dynamics (CFD).

1. Introduction

In this study, a fully three-dimensional CFD analysis has been successfully implemented for simulation of hydrodynamic journal bearing considering the realistic deformations of the bearing with fluid structure interactions (FSI). The first research on lubrication problems using the finite element method (FEM) was published by Reddi in [1] 1969. Over 40 years of development of the numerical methods and access to high-performance workstations have given researchers new possibilities of carrying out theoretical investigations. Commercial computer programs, based mainly on FEM or finite volume method, offer many advantages when compared with “in-house” developed codes, e.g., easily accessible meshes built for very complicated geometries and many types of analyses and boundary conditions. FEM codes were usually used separately to calculate bearing deformations [2, 3]. On the other

hand, CFD codes were applied to investigate flow phenomena in the lubricating oil and water or in the regions surrounding the bearing, both in journal [4, 5] and thrust bearings [6]. Reliable theoretical prediction of bearing properties requires simultaneous consideration of interacting phenomena such as lubricant flow, heat transfer and the distortion of bearing elements. That is why, fluid-structure interaction technique (FSI) seems to offer promising possibilities for bearing analysis. It allows for coupling of different physical domains in one computational task, taking into account the interaction between them. The FSI technique, based on CFD and FEM packages, allows for the analysis of the whole bearing system, including lubricant flow, thermal effects, bearing pad and shaft equilibrium or bearing components deformation. Thus far, the FSI technique has been applied only in a few research studies that have focused on bearing performance prediction. Liu et al. [7] used FSI to investigate journal-bearing behavior under steady-state and unbalanced loading conditions assuming an isothermal

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model. Shenoy et al. [8] used FSI model to calculate parameters of oil film in full journal bearing and took into account bush deformation (cavitation and all negative pressures were neglected). Cabrera et al. [9] used fluid mechanics and FSI to simulate rubber bearing. Ricci et al. [10] carried out FSI analysis.

2. Model Description

The journal bearing geometry used in the present work is shown in Fig. 1 [12]. The bearing centre is represented by O' and O is the journal or shaft centre, e is the eccentricity between shaft and bearing centres and L is the bearing length. The external load W is assumed as acting vertically along y-axis and is constant.

The hydrodynamic pressure developed in the convergent region, separates the bearing from shaft with a fluid film and balances the external force acting on the shaft. The fluid properties in this region remain constant. As the fluid enters the divergent region, the fluid pressure falls and reaches saturation pressure. In this region, liquid is converted into vapour and as the fluid advances, oil vapour expands and more vapour bubbles are released. This phenomenon is assumed as isothermal expansion and the energy required for the phase change is neglected as the amount of oil vapour formed is small. As the fluid further advances to convergent region, the high pressures react with fluid vapour by diffusion and the vapour is dissolved in the fluid again. This reduces the positive pressure built up in the convergent region.

Fig. 1 Journal bearing geometry [12].

3. Theory

3.1 Governing Equations

The pressure distribution in hydrodynamic journal bearing is governed by Reynolds equation which is derived from Navier Stokes continuity and momentum equations. In CFD-FLUENT [15], these equations are solved for mass and momentum that are valid for all types of flows:

$$\begin{cases} \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{v}) = 0 \\ \frac{\partial (\rho \vec{v})}{\partial t} + \nabla \cdot (\rho \vec{v} \otimes \vec{v}) = -\nabla P + \nabla \cdot \bar{\bar{\tau}} + \rho \vec{g} + \vec{F} \\ C_p \left[\frac{\partial \rho T}{\partial t} + (\vec{v} \cdot \nabla) T \right] = k_h \nabla^2 T + \phi \end{cases} \quad (1)$$

where ρ is fluid density, $\vec{v}$ is fluid velocity vector and P is the pressure. $\bar{\bar{\tau}}$ is the stress tensor, $\vec{g}$ and $\vec{F}$ are the gravitational body force and external body forces, C_p is the specific heat, T is the temperature and ϕ is the dissipation function, k_h the thermal conductivity of the fluid.

3.2 Cavitation Model

In hydrodynamic bearings, shaft rotates eccentrically with bearing forming a convergent and divergent region. In the convergent zone, shaft rotation forces the fluid to pass into an ever decreasing cross-sectional area resulting in an increase in pressures. Conversely, in the divergent region the pressures will decrease to an equal yet negative value as that in the convergent zone.

These negative pressures are non-physical and can not occur in a real fluid. Instead as the pressure begins to drop below atmospheric, the fluid will begin to cavitate and a gaseous phase will begin to fill the divergent region [11]. The mixture model solves the continuity and the momentum equation of the mixture and the volume fraction equation of the vapour-phase. The mass transfer between the phases is needed for these equations. This mass transfer will be solved with a cavitation model using vapour transport equation

given below:

$$\frac{\partial(a\rho)_{vap}}{\partial t} + \nabla \cdot (a\rho v)_{vap} = C_e - C_c \quad (2)$$

where C_e and C_c account for the mass transfer between liquid and vapour phases in cavitation.

3.3 Fluid Structure Interaction (FSI)

The fluid and the bearing affect each other. Fluid flow exerts a pressure on the bearing causing it to deform thus modifying the flow domain. The static structural capability of ANSYS is used to find deformations in the bearing. The governing equations are:

$$\begin{aligned} [M_s]\{\ddot{U}\} + [K_s]\{U\} &= [F_s] + [R]\{P\} \\ \begin{bmatrix} M_s & 0 \\ \rho R^T M_f \end{bmatrix} \begin{Bmatrix} \ddot{U} \\ \ddot{P} \end{Bmatrix} + \begin{bmatrix} K_s R \\ 0 \quad K_f \end{bmatrix} \begin{Bmatrix} U \\ \bar{P} \end{Bmatrix} &= \begin{Bmatrix} F_s \\ \bar{F}_f \end{Bmatrix} \end{aligned} \quad (3)$$

where $[M_s]$ is the structural mass matrix; $[M_f]$ is the fluid mass matrix; $[F_s]$ and $[F_f]$ are 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.

3.4 Lubricant Film Thickness

The relative displacement of the fluid-structure interfaces in the fluid domain is consistent with that of the solid domain. The film thickness is defined as the distance between the rotor-lubricant interface and the bearing-lubricant interface, including both rigid and elastic deformations between the two bearing surfaces as:

$$h = C + \Delta h + \delta \quad (4)$$

where h is film thickness, C is radial clearance of the bearing system, Δh is relative rigid displacement of the two bearing surfaces and δ is total elastic deformation of the two bearing surfaces.

4. Results and Discussion

4.1 Assumption and Boundary Conditions

The Navier Stokes equations are solved using 3D

double precision pressure based transient state analysis. Turbulent flow conditions are assumed with k- ϵ model realizable with non-equilibrium wall functions. The fluid domain, shown in Fig. 2 is meshed using hexahedral elements in CFD with 8261 Nods/32219 Elements. The solid domain shown in Fig. 3 is meshed using tetrahedral meshing with 38667 Nods/21270 Elements. The inlet pressure is taken as 101,325 Pa. The bearing is modelled as “stationary wall” and the shaft is modeled as “moving wall” with absolute rotation speed. Initially the shaft axis position is defined by an arbitrary value of eccentricity and the 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 geometric parameters and the operating conditions of the bearing are presented in Table.1.

The mesh is then transferred to fluent for flow analysis. The smoothing mesh method is used with a

Fig. 2 Fluid domain.

Fig. 3 Solid domain.

Table 1 Parameters used in analysis.

Parameter	Value	Parameter	Value
Shaft diameter, D	50 mm	Thermal conductivity	0.136 wmk
Clearance, C	50 μ m	Elastic modulus shaft	210 GPa
Bearing length, L	25 mm	Density ρ_s	7,850 kg/m ³
Speed, N	4,000 RPM	Poisson ratio	0.3
Eccentricity, e	0.25 mm	Elastic modulus bearing	210 GPa
Lubricant viscosity, μ	0.0125 Pa·s ⁻¹	Density ρ_b	2,700 kg/m ³
Lubricant density, ρ	850 kg/m ³	Poisson ratio	0.334

convergence tolerance of 10^{-6} . The fluid forces are computed in computational fluid dynamics (CFD) domain and structural deformations are computed in structural domain. These two systems are system coupled to perform thermoelastohydrodynamics (TEHD). The fluid forces and the temperature body developed in CFD are imported to structural domain and the results of the fluid part will be presented in the median plane of the bearing.

For an eccentricity of 0.25 mm and with a shaft rotation speed of 4,000 RPM, Fig. 4 ((a), (b)) shows respectively the three-dimensional distribution of the fields of static pressure and static temperature of the lubricant film. It is observed that the maximum value of the pressure is reached in the zone where the thickness of the film is minimal. The temperature distributed in the circumferential direction reached the value of 38 °C.

The velocity magnitude and Turbulent Kinetic Energy (TKE) are present in Fig. 4 ((c), (d)) respectively. It is observed that the velocity magnitude increases in the divergent zone, i.e. in the ends of the cavitation zone, the maximum values of TKE are located in a significant way in the zone where the thickness of the film is with the maximum value $50.2 \text{ m}^2\cdot\text{s}^{-2}$.

The total deformation field U and the stress intensity S within the pad are shown in 3D in Fig. 5. It is observed that the pad deforms more significantly in the median plane where the thickness of the lubricant film is minimal. The deformation field has reached its

maximum, of order $U_{\max} = 37.18 \text{ e}^{-5}\cdot\text{m}$. The intensity stress accesses its maximum value $S_{\max} = 2.75 \text{ e}^2\cdot\text{MPa}$ in the ($z = \pm L/2$) ends.

Fig. 4 The distribution of the pressure, temperature, velocity and TKE in the median plane.

Fig. 5 The distribution of the total deformation and the stress intensity.

Fig. 6 Influence of the rotational speed on the fields of pressure, deformation and stress.

Fig. 7 Influence of flow type on fields of pressure, deformation and stress.

This is due to the fact that, at these ends, embedding (a boundary condition) generates significant constraints. Fig. 6 ((a), (b) and (c)) respectively shows the influence of the shaft rotation speed on the circumferential distribution of the pressure and therefore on the deformation and stress of the solid. It can be seen that the pressure increases with the increase of the speed in a large way and the deformation increases with a high rate, concerning the stress one notices that there exists IS a shift between the two internal and external surfaces of the solid. The influence of the flow model is shown in Fig. 7 ((a), (b) and (c)). It has been noticed after a special numerical treatment on the three types that are laminar, $k-\epsilon$ and $k-\omega$. The $k-\omega$ model gives important results with respect to $k-\epsilon$ and laminar and there is a small difference between laminar and $k-\epsilon$.

5. Conclusion & Perspective

This paper has studied the nonlinear analysis of the lubrication TEHD of a hydrodynamic bearing. Using on the one hand the CFD, we are allowed to determine the pressure and temperature field in the lubricant film. On the other hand, the fields of pressure and temperature are imported into the “coupling model” FSI contact zone to calculate the total deformation and the stress intensity. It should be noted, however, that the results given in this paper are valid only for the specific cases we have studied, and that they are not independent of bearing and lubricant characteristics. Our work also deals with the influence of the velocity and different types of the flow on different variables of the problem. In the next works we will treat the bearings with grooves for different locations and see the influence of the eccentricity and take into account the vibration of the shaft.

References

- [1] Reddi, M. M. 1969. “Finite-Element Solution of the Incompressible Lubrication Problem.” *J Lubr Technol* 91 (3): 524-33.
- [2] Brown, A. L., Medley, J. B., and Ferguson, J. H. “Spring-Supported Thrust Bearings Used in Hydroelectric Generators: Finite Element Analysis of Pad Delection.” In *Proceedings of 27th Leeds Lyon Symposium on Tribology*, 99-110.
- [3] De Kraker, A., Van Ostayen, R., and Rixen, D. J. 2007. “Calculation of Stribeck Curves for (Water) Lubricated Journal Bearings.” *Tribol Int* 40 (3): 459-69.
- [4] Edney, S. L., Heitland, G. B., and De Camillo, S. M. 1998. “Testing, Analysis, and CFD Modeling of a Profiled Leading Edge Groove Tilting Pad Journal Bearing.” In *Proceedings of the International Gas Turbine & Aeroengine Congress & Exhibition Stockholm*, Paper No. 98-GT-409.
- [5] Pai, R., Hargreaves, D. J., and Brown, R. 2001. “Modelling of Fluid Flow in a 3-Axial Groove Water Bearing Using Computational Fluid Dynamics.” In *14th Australasian Fluid Mechanics Conference*, Adelaide, Australia.
- [6] Maneshian, B., and Gandjalikhan Nassab, S. A. 2009. “Thermodynamic Characteristics of Journal Bearings Running under Turbulent Condition.” *IJE Trans* 22 (2): 181-94.
- [7] Liu, H., Xu, H., and Ellison, P. J. 2010. “Application of Computational Fluid Dynamics and Fluid-Structure Interaction Method to the Lubrication Study of a Rotor-Bearing System.” *Tribol Lett* 38: 325-36.
- [8] Shenoy, B. S., Pai, R. S., Rao, D. S., et al. 2009. “Elasto-hydrodynamic Lubrication Analysis of Full 360 Journal Bearing Using CFD and FSI Techniques.” *World J Modell Simul* 5 (4): 315-20.
- [9] Cabrera D L Woolley N H Allanson David Roy Tridimas YD. 2005. “Film Pressure Distribution in Water-Lubricated Rubber Journal Bearings.” In *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*.
- [10] Ricci, R., Chatterton, S., Pennacchi, P., et al. 2011. “Multiphysics Modeling of a Tilting Pad Thrust Bearing with Polymeric Layered Pads.” In *Proceedings of 10th EDF/Pprime Workshop*, 995-1002.
- [11] Dhande, D. Y., and Pande, D. W. 2018. “Multiphase Flow Analysis of Hydrodynamic Journal Bearing Using CFD Coupled Fluid Structure Interaction Considering Cavitation.” *Journal of King Saud University Engineering Sciences* 30 (4): 345-54. <http://dx.doi.org/10.1016/j.jksues.2016.09.001>.
- [12] Braun, M. J., and Hannon, W. M. 2010. “Cavitation Formation and Modelling for Fluid Film Bearings: A Review.” *Proc. Inst. Mech. Eng. Part J J. Eng. Tribol.* 224: 839-63.
- [13] Gandjalikhan Nassab, S. A., and Moayeri, M. S. 2002. “Three-Dimensional Thermodynamic Analysis of Axially Grooved Journal Bearings.” *J Engineering Tribology* 216: 35-47.
- [14] ANSYS 2017. *ANSYS Documentation*. Houston, PA: Swanson Analysis Inc.