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Multiscale Modelling and Analysis on the Heavy-duty Hydrostatic Journal Bearing for a Precision Press Machine

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Abstract. The present study aims to predict the performance of hydrostatic journal bearings during steady-state operation through multiscale modelling and analysis. An innovative multiscale modelling approach is presented for numerical modelling and analysis of the hydrostatic journal bearing as configured and designed, particularly for the precision press machine. With the challenging environment for journal bearings under hydrostatic pressures and surface features, , elastic modulus of oil and viscosity-temperature characteristics are often neglected while against heavy-loading and precision engineering requirement. This research takes into account of oil elastic modulus and viscosity-temperature characteristics to investigate the intrinsic relationship between the pressures within the hydrostatic bearing and its surface texture, feature size and locations being considered. This paper concludes with a further discussion on the potential and application of the approach in precision engineering.

Keywords: Hydrostatic Journal Bearing, Multiscale Modelling, Textured Surface, Computational Fluid Dynamics (CFD) Simulation, Precision Press Machine

1. Introduction

Hydrostatic journal bearings have been investigated through a number of simulation cases. In particular, the effects of surface roughness on Elastohydrostatic Lubrication (EHL) line contact are investigated against the smooth surface, because experimental measurement is difficult. This explains why the experiment on mixed lubrication only provides general trends of contact performance, while the detailed in-depth study on surface roughness and relevant scale is through numerical methods.

Patir and Cheng [1] presented an average flow model to solve the effects of roughness. In order to intensively study the flow factors of stochastic model of Patir and Cheng, several researchers proposed modified average Reynolds equations. Wang et al. [2,3], Epstein and Yu et al. [4], introduce flow factors in thermal elastohydrodynamic analysis of journal bearings. Based on those studies, de Kraker et al. [5, 6], presented the development of a micro–macro multiscale method. The micro flow effects for a single surface pocket was analyzed by using the Navier–Stokes equation and compared to the Reynolds solution for a similar smooth piece of surface. Gao et al. [7], presented Heterogeneous Multiscale Method (HMM) with cavitation effects to solve the EHL problem. Almqvist and Larsson [8], presented the commercial CFD software to simulate the EHL line contact issue based on the continuity equation and the energy equation. Hartinger et al. [9] used this approach to compare with the solution of Reynolds equation, the results of both approaches have good agreement. Recently Bruyere and Fillot [10] by adding the fluid thickness dimension to the EHL problem and by modelling solids with finite dimension and the thermal effects on friction have been investigated, Meng et al. [11] used Ansys Fluent software applied to isothermal cases with surface roughness effects under varied loads. Gao and Cheng et al. [12] used Ansys Fluent to investigate the influence of orifice chamber shapes of thrust bearings and optimized the shapes of the orifice.

In this paper, the hydrodynamic effect of hydrostatic bearings with surface texture and microstructure are investigated. Through the mass-conserving approach at macro-micro scales, it investigates the surface texture and the associated hydrodynamic performance of the journal bearing. ANSYS Fluent is used to simulation the bearing model and investigate the optimal location of the micro structures.

2. Development of the multiscale modelling approach

The performance of hydrostatic journal bearing depends on both local and global film height variation. Due to the oil film thickness compared with the overall bearing size is very small, the surface texture becomes important in macro-scale, and in micro-scale we should concern about the variations of oil viscosity coefficient and elasticity modulus, which in different rotating speed and oil supply pressure.

2.1. Macroscale model

The macroscale simulation describes the fluid–structure interaction in the global lubrication domain, In consideration of global surface roughness, the deterministic method still not solve 3D EHL problem. Hence, Patir and Cheng [1] presented an average rough Reynolds equation.

$$\frac{\partial}{\partial x}\left(-\phi_{px}\frac{h^3}{12\eta}\frac{\partial p}{\partial x}+\frac{\left(U_1+U_2\right)h'}{2}+\frac{\phi_s\left(U_1-U_2\right)\sigma}{2}\right)+\frac{\partial}{\partial z}\left(-\phi_{pz}\frac{h^3}{12\eta}\frac{\partial p}{\partial z}\right)=0 \tag{1}$$

With h' the mean oil film thickness, in this paper the surface roughness is Gaussian distributed, assume the surfaces to be stationary (U_2=0) in equation (1), leaving roughness deformation out of consideration, h' is equal to h, ϕ_{px} and ϕ_{pz} are pressure flow correction factors and ϕ_s is the shear flow factor and σ the surface rms roughness.

$$h\left(x,z\right)=C\left[1-\varepsilon\cos\left(\theta_{inlet}+\frac{x}{r}-\varphi\right)\right]+u\left(x,z\right)+h_{\sigma}\left(x,z\right) \tag{2}$$

Consider the effect of elastic deformation, the fluid film thickness h in function of the initial geometry, equation (2), by calculating the bearing on a large scale, various parameters of the bearing during the movement can be calculated accurately, especially the iterative calculation of the offset angle. This provides a reliable basis for geometric modelling for small scale calculation by CFD method.

2.2. Microscale model

Considering the influence of texture on elastohydrostatic lubrication bearing, differ with global smooth bearing, the pressure is incorrect and modified.

$$p=p_c+p_f+\overline{p} \tag{3}$$

In macroscale, with P_c the contact pressure equation (3) and P_f the fluid pressure. Two mechanisms have been pointed out contributing to the pressure generating effect of surface texture in full film lubrication. $\overline{p}$ the additional pressure between texture geometry in microscale and average pressure at macro level P_f equation (3).

$$P_c=H\overline{a}_c=\frac{H}{2}\left[1-\mathrm{erf}\left(\frac{1}{\sqrt{2}}\frac{h}{\sigma}\right)\right] \tag{4}$$

With H bearing material hardness, based on the plastic asperity model of Bowden and Tabor.

$$\overline{p} = \frac{1}{L_x L_y} \int_{\Omega} \max\left(p_f + p_{tex}, 0\right) d\Omega - p_f \tag{5}$$

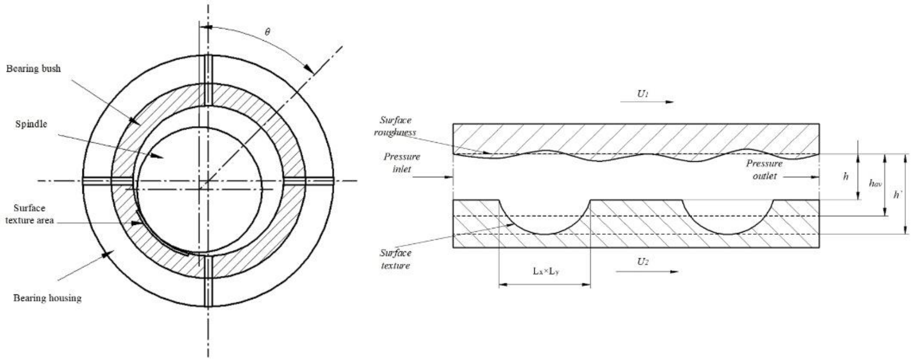


Figure 1. Schematic diagram of the journal bearing and surface microstructure

2.3. Density and Viscosity equation

The density and viscosity of oil fluid are concerned to be a constant parameter in previous studies. In reality, they are depended on the interior pressure. The linear variation of density can be written as:

$$\rho = \rho_0 \left(\frac{0.6 \times 10^{-9} p}{1.7 \times 10^{-9} p + 1} + 1 \right) \tag{6}$$

With ρ is the density of oil at any time, ρ_0 is the initial density of oil. And the relationship between viscosity and pressure can be expressed as:

$$\eta = \eta_0 \exp\left\{ \left(\ln \eta_0 + 9.61 \right) \left[\left(1 + 5.1 \times 10^{-9} p \right)^k - 1 \right] \right\} \tag{7}$$

With η is the viscosity of oil at any time. η_0 is the initial viscosity of oil, *k* is the coefficient of viscosity (usually take 0.5~0.7), in this paper we take 0.6.

2.4. Cavitation and Elastic modulus of oil

To solve cavitation phenomenon, we usually assumed all negative pressures are equal zero, because the Reynolds function based on mass conservation. In this paper the macro-cavitation we use full cavitation model to interpretation in the CFD method:

$$\frac{\partial(\rho\zeta)}{\partial t} + \nabla\left(\rho v \zeta\right) = \nabla\left(\gamma \nabla \zeta\right) + \beta \tag{8}$$

With ζ is Volume fraction, β is Volume coefficient of vapour-oil variation. In the micro-cavitation, the cavitated area is found due to dissolved gasses that appear when the pressure drops below the vapour pressure. For more accurate calculation we introduce the elastic modulus of oil.

$$E_{oil} = 700 + 800 \log p \tag{9}$$

3. Geometry and simulation

The geometry domain is subdivided into a finite number of micro-scale, as shown in Figure 1. In order to start the iteration process, the velocity and pressure fields are approximated for the first iteration. Then these parameters are used to calculate the momentum conservation equation and the pressure of journal bearing. These values will be corrected in each iteration, until acceptable convergence of pressure and velocity is achieved as shown in Figure 2. In this research, representative hydrodynamic journal bearing that is used in the simulations.as discussed previously, the design and material parameters are shown in Table 1. The values that were used for the Young`s modulus, hardness and Poisson ratio are given.

Table 1. Main shaft bearing characteristics.

Geometry		
Bearing diameter	D	140 mm
Bearing width	B	140 mm
Radial clearance	c	30 μm
Relative eccentricity	ε	0.8-1.2
Surface rms roughness	σ	0.6 μm
Rotational speed	ω	300-1000 rpm
Viscosity at T=40.0℃	μ	0.035 $Pa \cdot s$
Operating temperature	T	40.0℃
Young`s modulus (pad)	E	0.063 MPa
Poisson`s ratio (pad)	v	0.33

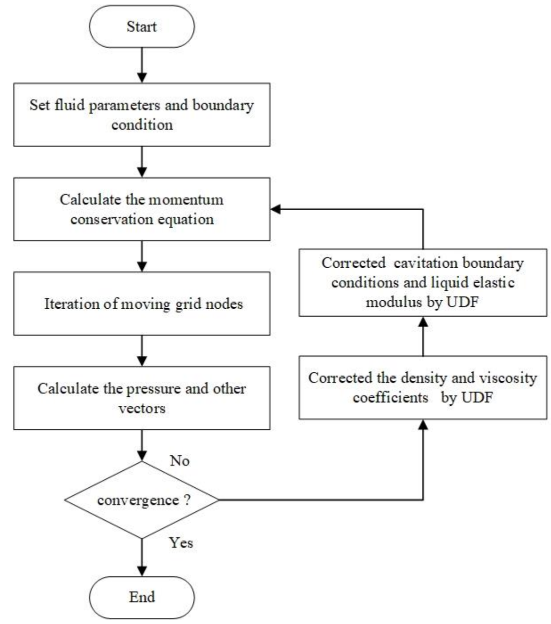
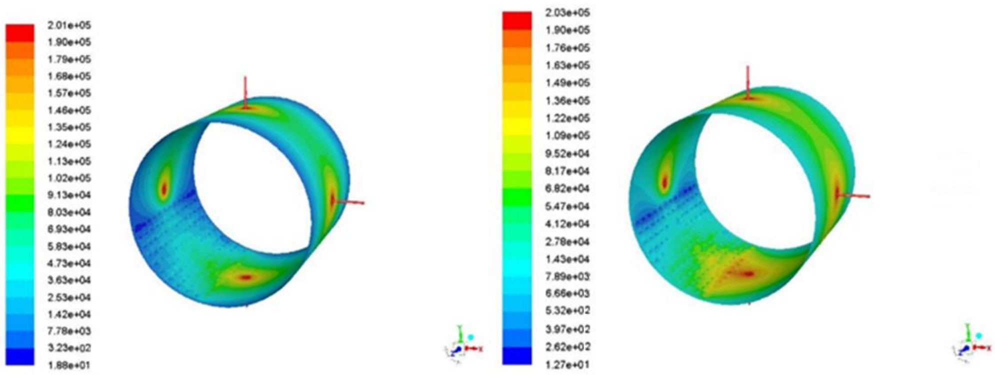


Figure 2. Schematic diagram of calculation flow

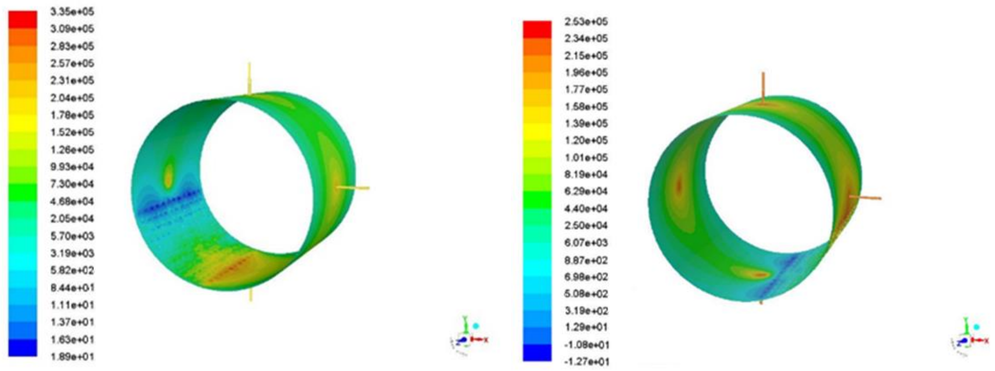
4. Results, analysis and discussion

4.1. Simulation on the oil film with surface textures

The surface microstructure of bearing is analyzed and studied by CFD method. It can be clearly seen from the pressure distribution contours of journal bearing with the rotate speed at 300 and 1,000 rpm, oil supply is 2MPa.



(a) (b)



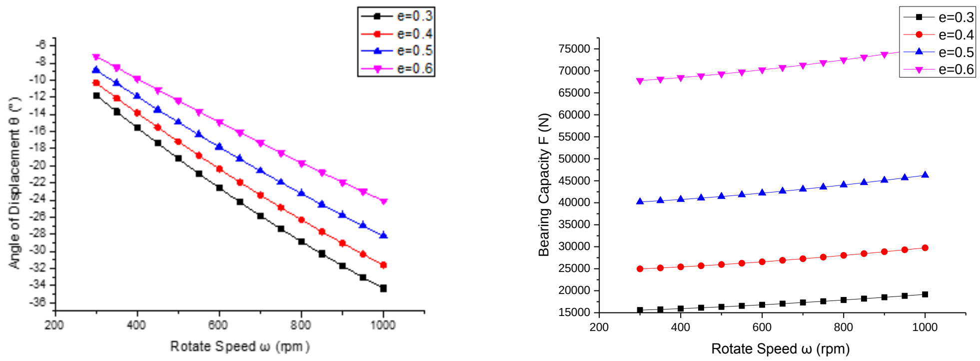
(c) (d)

Figure 3. Pressure distribution contours with surface microstructure at different rotate speed
(a) Microstructure in loaded area at 300 rpm (b) Microstructure in loaded area at 1000 rpm
(c) Microstructure in loaded area at 1000 rpm (d) Microstructure in unloaded area at 1000 rpm

In figure 3(a) and (b). the pressure increased at approximate contact area with multiscale method. Because the density and viscosity coefficient increased with pressure and rotate speed, and the cavation area decreased due to the elastic of oil gradually increasing. The surface deformation increase with the rotating speed, at low speed, the load capacity varies with the eccentricity. But in high rotating speed the effect is very clear, (c) and (d) at 1000 rpm.

4.2. Influence of EHL on static characteristics of the bearing

The change trend of attitude angle (left) and bearing capacity (right) are shown in Figure. 4. From top to bottom, eccentricity (e), oil supply pressure (P_0), average oil film thickness (h_0) is varied. From the first set of figures it is clear that for smaller eccentricity, a lower coefficient of friction and larger minimum film height is obtained for the same operating conditions in EHL. Reduction of the radial clearance by a factor of eccentricity the capacity of load, when the eccentricity e<0.3, the film thickness less than average surface roughness h_{av} the bearing wear and tear will be occurs. With the pressure increasing the trend of attitude angle increased obviously, especially at rotate speed 600 to 700 rpm (P_0=2 MPa).



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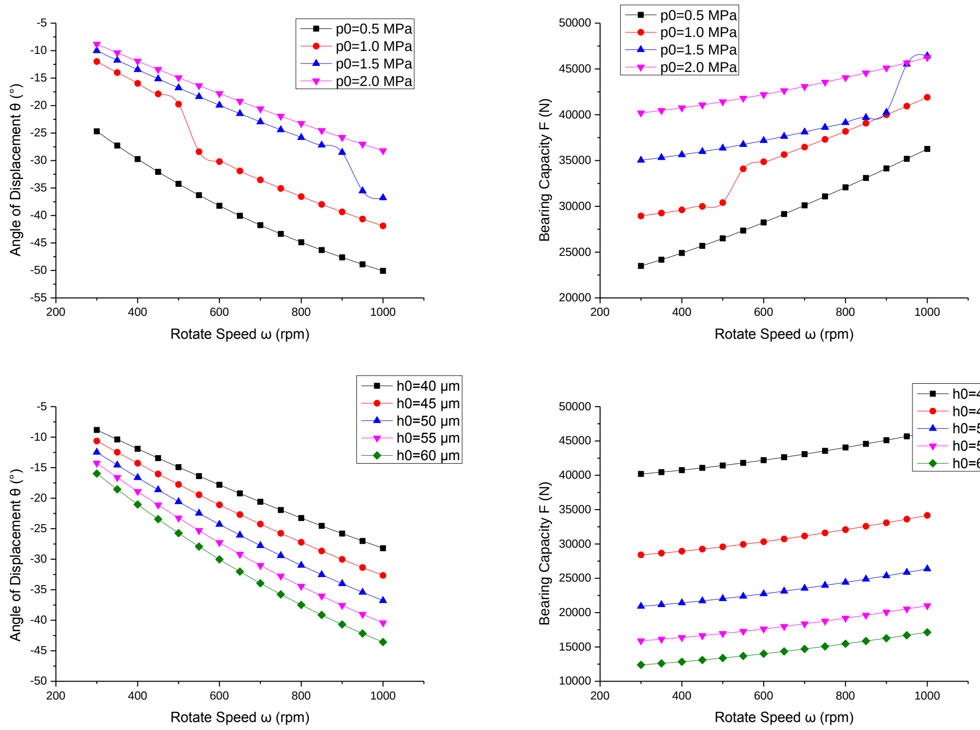

Figure 4. The change trend of attitude angle and bearing capacity

This is due to the formation of micro-cavitation inside oil at this speed, the growth of carrying capacity is also evident. In the third set of figures, the load is varied with the average oil film thickness h_0. With the deformation increasing, the contact pressure is distributed in larger area. it can be seen that the thinner of minimum film thickness the larger carrying capacity in load area.

5. Conclusions

The research presented is to study a multiscale modelling and analysis approach for hydrostatic bearings, The following conclusions are obtained through the simulation-based investigation:

- In the actual working condition of the bearing, the surface roughness of the bearing surface, elastic deformation and other factors must be taken into account.
- The bearing loading capacity is related to its speed and viscosity, so choosing the appropriate operation parameters has a great influence on the load carrying capacity of the bearing.
- The influence of the surface microstructure at the bearing surface includes the layout of the microstructure, location and the ratio of the bearing oil film thickness. The oil film thickness is smaller, the greater the influence of microstructure on the bearing capacity.

References

[1] Patir N and Cheng H S 1978 Average flow model for determining effects of 3- dimensional roughness on partial hydrodynamic lubrication *ASME J Lubri Tech* **100** 12–17

[2] Shi F and Wang J Q 1998 A Mixed-TEHD Model for Journal-Bearing Conformal Contacts—Part I: Model Formulation and Approximation of Heat Transfer Considering Asperity Contact *ASME J. Tribol.* **120** 198–205

[3] Gao S and S Srirattayawong 2012 CFD Prediction of the Effects of Surface Roughness on Elastohydrodynamic Lubrication under Rolling/Sliding Conditions *Applied Mechanics and Materials* **184-185** 86-89

IOP Conf. Series: Materials Science and Engineering **825** (2020) 012010 doi:10.1088/1757-899X/825/1/012010

[4] Epstein D, Yu T, Wang J, Keer L M, Cheng H S and Liu S 2003 An Efficient Method of Analyzing the Effect of Roughness on Fatigue Life in Mixed-EHL Contact *STLE Tribol. Trans.*, **46** 273–281
[5] A de Kraker, R A J van Ostayen and D J Rixen 2010 Development of a texture averaged Reynolds equation. *Tribol. Int* **43(11)** 2100-09
[6] A de Kraker, R A J Ostayen and D J Rixen 2007 A multiscale method modelling surface texture effects *ASME J Tribol* **129** 221–30
[7] Gao L M, de Boer GN, Hewson R W 2015 The role of micro-cavitation on EHL: A study using a multiscale mass conserving approach. *Tribol Int* **324** 331–90
[8] Almqvist T and Larsson R 2008 Thermal transient rough EHL line contact simulations by aid of computational fluid dynamics *Tribol. Int.* **41** 683-93
[9] Hartinger M, Dumont M L, Ioannides S, Gosman D and Spikes H 2008 CFD modelling of a thermal and shear-thinning elastohydrodynamic line contact *ASME J. Tribol.* **130(4)** 041503
[10] Bruyere V, et al 2012 Computational fluid dynamics and full elasticity model for sliding line thermal elastohydrodynamic contacts *Tribol Int* **46(1)** 3-13
[11] Meng F M and T Yang 2012 Preliminary study on mechanism of cavitation in lubricant of textured sliding bearing *Part J: Journal of Engineering Tribology*, **227(7)** 695-708
[12] S Y Gao, K Cheng, et al 2015 CFD based investigation on influence of orifice chamber shapes for the design of aerostatic thrust bearings at ultra-high speed spindles *Tribol Int* **92** 211–21