INFLUENCE OF SURFACE TEXTURING ON THE PERFORMANCE OF TILTING PAD THRUST BEARINGS

Session 2H - Las Vegas 1 - Fluid Film Bearings II

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INTRODUCTION

Surface texturing has been shown to be a promising method of enhancing the tribological performance of various applications and has attracted considerable attention in recent years. Recently, the authors have summarized the key findings and revealed the challenges still encountered in this field [1]. The present numerical study is focused on a less explored application of surface texturing, namely tilting pad thrust bearings. Zouzoulas and Papadopoulos have recently shown that partial texturing of the bearing pads can result in enhanced hydrodynamic performance concerning friction torque and minimum film thickness [2]. One of the major drawbacks is the dependency of optimal texturing parameters (depth, size, distribution) for best performance enhancement on the operating conditions. If designed wrong, texturing may even become detrimental. This emphasizes the need for robust and fast theoretical models that allow the evaluation of textured bearings under a wide range of conditions.

The objective of this study was the development of such a model. The proposed model is (1) based on a Finite Volume discretization of a modified Reynolds equation on an adaptive non-uniform polar coordinate grid; (2) includes an iterative mass-conserving cavitation algorithm; (3) incorporates a special treatment of film discontinuities to limit discretization errors; (4) takes advantage of multicore processing; (5) uses the results (equilibrium position, pressure distribution, temperature) of the untextured pad as first approximation for the textured pad to improve computational performance; (6) selects the most suitable algorithm to find the pad's equilibrium position (Newton-Raphson, Broyden, and Continuation Methods have been implemented) and (7) considers thermal effects through an effective temperature method, while taking into account the hot-oil-carry-over effect.

In the following sections details of the developed numerical model are given, the model's accuracy and speed is investigated and a parametric study is run to evaluate the influence of surface texturing on a typical line-pivoted tilting pad thrust bearing as shown in Fig. 1. All simulations were run in *MATLAB 2016a* on a workstation with 16GB RAM and *Intel Core i7-3770* @ 3.40GHz CPU with four physical/eight logical cores.

The results show that the proposed model exhibits good accuracy and remarkable computational speed. The chosen texture design may decrease friction torque by up to 4.5% and increase minimum film thickness by up to 7% as compared to the untextured bearing. More details are given on our poster in the poster session.

MATHEMATICAL MODEL

Bearing & texture geometry

The thrust bearing geometry, chosen texture design and operating conditions considered in this study are given in Fig. 1 and Table 1.

Fluid flow

For the Reynolds numbers ($Re_{max} \approx 50$) and texture aspect ratios ($\lambda_{max} = dimple\ depth/length_{min} \approx 0.0263$) encountered, the Reynolds equation can safely be applied as long as concentrated inertia effects are taken into account [3].

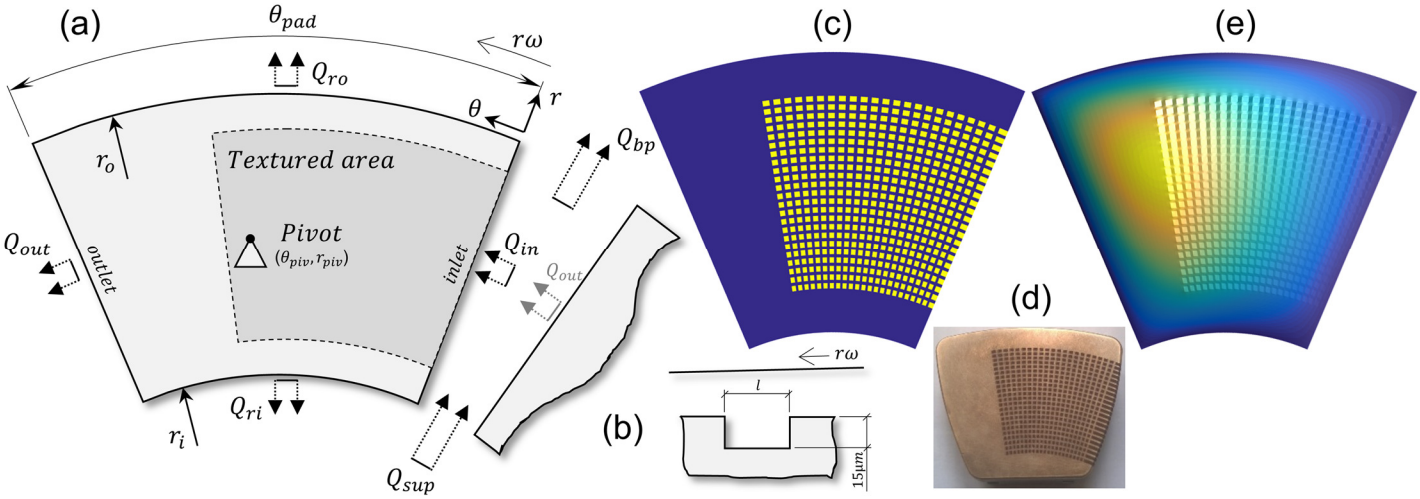

Fig. 1 (a) Thrust pad schematic, (b) Dimple detail, (c) Texture design, (d) Picture of the laser textured pad and (e) Exemplary pressure distribution.

In polar coordinates and considering mass-conserving cavitation, which may occur inside dimples close to the pad inlet under certain operating conditions, the applied Reynolds equation reads:

$$\frac{\partial}{\partial r}\left(r\frac{\rho_{liq}h^3}{\eta_{liq}}\frac{\partial p}{\partial r}\right)+\frac{1}{r}\frac{\partial}{\partial\theta}\left(\frac{\rho_{liq}h^3}{\eta_{liq}}\frac{\partial p}{\partial\theta}\right)=6\omega r\frac{\partial(\Theta\rho_{liq}h)}{\partial\theta}\quad with\quad p>p_{cav}\quad for\quad \Theta=1\quad and\quad p=p_{cav}\quad for\quad 0\le\Theta<1 \tag{1}$$

Eq. (1) is discretized using a Finite Volume Method on a non-uniform mesh and solved iteratively and simultaneously for p and $\Theta=V_{liq}/(V_{liq}+V_{gas})=\rho/\rho_{liq}=\eta/\eta_{liq}$ using a Gauss-Seidel method with successive over-relaxation as described in detail in reference [4]. As the film thickness function is discontinuous at all dimple edges, special Finite Volume schemes as proposed by Arghir *et al.* [5] are applied. In this method film thickness values are defined not only at the cell centers, but also at all cell boundaries, resulting in discontinuous first-order pressure derivatives. Also, concentrated inertia effects can be considered by solving a generalized Bernoulli equation, making pressure and its first-order derivatives discontinuous. The solution of Eq. (1) is computationally expensive and required multiple times during the overall solution procedure. For this reason it is solved by a *MATLAB* sub-function that runs in *C* to decrease computation time. For untextured pads uniform meshes and standard Finite Volume discretization schemes are used.

Mesh details

To limit discretization errors and improve computational performance an adaptive and non-uniform polar coordinate grid is used to discretize the complex pad and texture geometry. Control volume faces are aligned with the dimple's edges and additional control volumes can be placed at either side of all discontinuities. Also, the mesh density can be separately controlled for the area inside dimples, between dimples and the untextured pad area. This methodology ensures an efficient solution and retards discretization errors.

Film thickness and equilibrium position

The film thickness distribution can be fully described by the pad's pitch angle (α_r), roll angle (α_θ) and film thickness at pivot location (h_p) and is expressed as follows:

$$h(\theta,r)=h_p+r\sin(\theta_{piv}-\theta)\sin\alpha_r+\left[r_{piv}-r\cos(\theta_{piv}-\theta)\right]\sin\alpha_\theta+\Delta h_{texture}\qquad [\alpha_\theta=0\ for\ line\ pivoted\ pads] \tag{2}$$

The pad's equilibrium position is the specific film geometry that balances the applied load and results in zero resultant moments around the pivot. This leads to a multidimensional nonlinear system that is typically solved by a Newton-Raphson method with quadratic convergence. However, this method requires to determine the Jacobian matrix at each iteration, which results in high computation times for the fine meshes encountered in the simulation of textured surfaces. For this reason the authors propose using Broyden's algorithm [6], which only requires the evaluation of the Jacobian matrix once. Although the convergence of this method is only superlinear, the algorithm is considerably faster in the present study.

One of the drawbacks of Broyden's method is the need for a good initial approximation. Two ways to address this issue have been implemented: (1) for untextured pads a Newton-Raphson or Continuation method [6] is run for a limited number of iterations. This is usually enough to obtain a sufficiently accurate solution to start Broyden's algorithm; (2) for textured pads the results from an equivalent untextured pad, which only takes a fraction of the computing time as compared to the textured pad, are used as first approximation for Broyden's method. This could be considered a kind of a Multigrid technique and highly decreases computation time.

The elements of the Jacobian matrix are approximated by Finite Difference formulae and calculated simultaneously on multiple processor cores to further improve computational performance.

Thermal effects

Thermal effects are taken into account by applying an effective temperature method [7]. The average temperature rise due to viscous shear heating and the resulting effective temperature are calculated using the following expressions:

$$\Delta T=\frac{\Pi}{Q_{in}\rho c_p}\quad and\quad T_{eff}=T_{inlet}+k\Delta T \tag{3}$$

Where Π is the total frictional power loss and k is the amount of the heat evacuated by convection (in this study: $k=0.7$). The hot-oil-carry-over effect is considered with the following expression and updated iteratively [2].

$$T_{inlet}=\frac{T_{out}Q_{out}+T_{sup}Q_{sup}+T_{ir}Q_{ir}}{Q_{sup}+Q_{out}+Q_{ir}} \tag{4}$$

The viscosity variation with temperature is modelled using McCoull and Walther's relation [8].

$$log_{10}[log_{10}(\nu_{cSt}+a)]=n-mlog_{10}(T_{\circ K}) \tag{5}$$

Numerical procedure

After generating the required meshes for the textured and equivalent untextured pad, the simulation starts by calculating the pressure distribution for the untextured pad. Load capacity and center of pressure are evaluated and the film thickness is iteratively updated by a Newton-Raphson method until the results are sufficiently accurate to start Broyden's

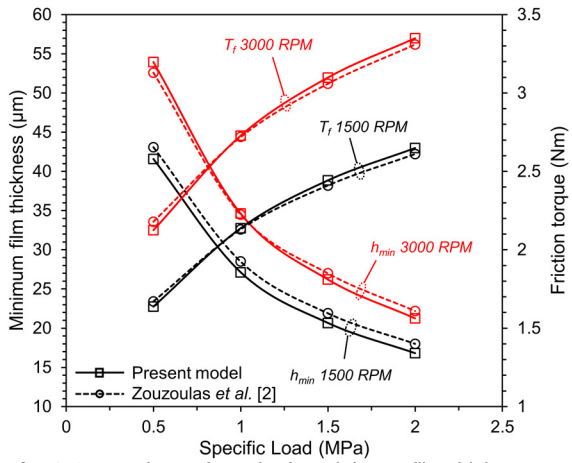

Fig. 2 Comparison of results for minimum film thickness and friction torque as obtained by the present model and the CFD study conducted by Zouzoulas and Papadopoulos [2].

Table 1 *Bearing characteristics, operating conditions and texturing parameters.*

Description	Quantity	Description	Quantity
Inner pad radius (mm)	30.25	Viscosity @ 40°C (cSt)	42.65
Outer pad radius (mm)	70.25	Viscosity @ 100°C (cSt)	6.5
Pad angle (°)	46.05	Supply oil temp. (°C)	40
Pivot type	Line pivot	Heat capacity (J/kg/K)	2035
Relative pivot location	0.6	Texture depth (µm)	15
Rotational speed (RPM)	1000…4000	Rel. texture extend in θ	0.70
Specific Load (MPa)	0.5…4	Rel. texture extend in r	0.70
Lubricant	ISO VG 46	Texture density	0.40
Supply flow rate per pad (l/min)	2.2	Number of textures in θ	23
Lubricant density (kg/m^3)	855	Number of textures in r	23

algorithm. Once the results are converged, the procedure is repeated with updated effective temperature, inlet temperature and effective viscosity until temperature convergence is reached. The same procedure is then repeated for the textured pad, however, all obtained results (equilibrium position, temperature, pressure distribution) are used as initial approximations and only Broyden's method is applied. Running numerous simulations, this procedure has proven to be highly efficient.

VALIDATION

To validate the developed model simulations were run for the thrust bearing with point-pivoted pads investigated through CFD by Zouzoulas and Papadopoulos [2]. Eight load/speed scenarios were simulated simultaneously on the eight logical processor cores available, resulting in a total calculation time of just 101 seconds. A uniform mesh with $101x101$ control volumes and a relaxation parameter of 1.94 were used. Results are presented in Fig. 2 and show good agreement between the two models with maximum errors of 6.6% and 2.5% regarding minimum film thickness and friction torque, respectively.

RESULTS

Treatment of discontinuities

To investigate the effect different treatments of discontinuities have on computation time and accuracy, the model was run for the textured pad at 2000rpm/70°C and a predefined film geometry ($h_p = 25.17\mu m$ & $\alpha_r = 471.5\mu rad$). The cases considered and meshes used are given in Fig. 3. As each case leads to a slightly different system of equations and thus numerical behavior, each case was run with the corresponding optimum relaxation parameter (ω_{opt}). Texturing results in a clear distortion of the pressure distribution and increased maximum pressure. The highest load carrying capacity is predicted when discontinuities are not treated at all (case 1). Both, placing additional points around discontinuities (case 2) and using special discretization schemes (case 3 & 4) result in lower load carrying capacities. While case 3 and 4 are calculated significantly faster than case 2, predicted load carrying capacities are similar. Although more complicated to implement, special discretization schemes seem to be the most efficient treatment of discontinuities. The method used in case 4 was thus applied for the following parametric study.

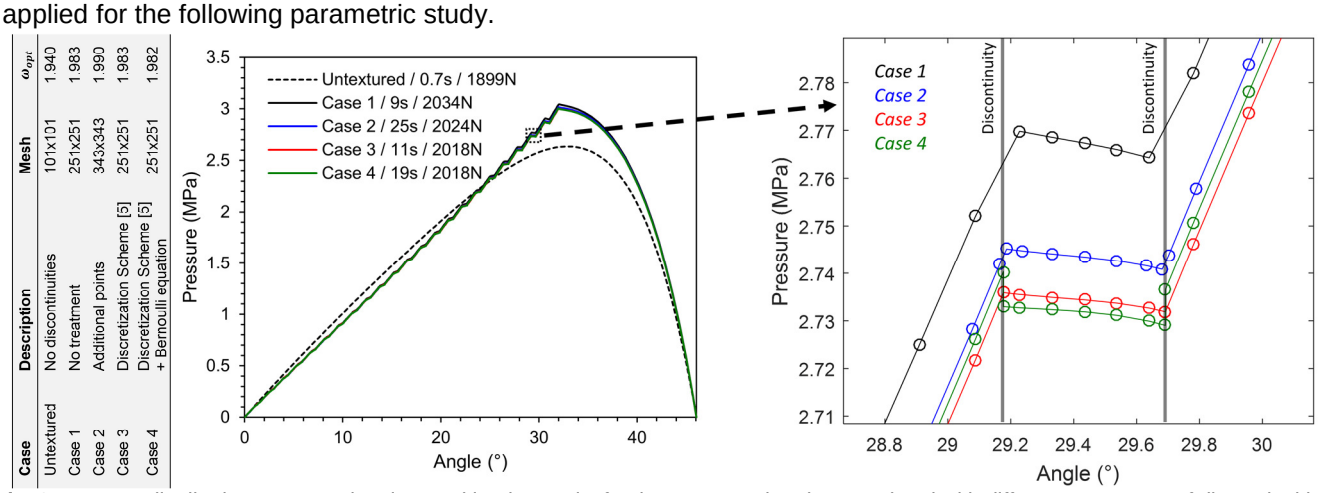

Fig. 3 *Pressure distribution, computation time and load capacity for the untextured and textured pad with different treatments of discontinuities.*

Parametric study

To investigate the influence of surface texturing on the performance of tilting pad thrust bearings, a parametric study was run. As can be seen in Fig. 4 (a) and (b), the chosen texture design enhances bearing performance in terms of friction torque and minimum film thickness. For nearly all operating conditions, friction torque is reduced and minimum film thickness increased. Plotting the relative change of these parameters due to texturing allows an overall evaluation of the texture design (see Fig. 4 (c) and (d)). It can clearly be seen that the impact of texturing depends on the operating conditions. Whereas the highest increase in minimum film thickness ($max[\Delta h_{min}] \approx 7\%$) occurs at medium loads and higher speeds, the highest reduction in friction torque ($max[\Delta T_f] \approx -4.5\%$) occurs at low speeds and high loads only. For highest loads and lowest speeds minimum film thickness is decreased by nearly 1.5%. The results show that although some general guidelines exist (e.g. texturing the pad inlet [1]), textures need to be carefully designed for a particular bearing application. Knowing the expected range of operating conditions and applying fast and robust theoretical models are thus crucial.

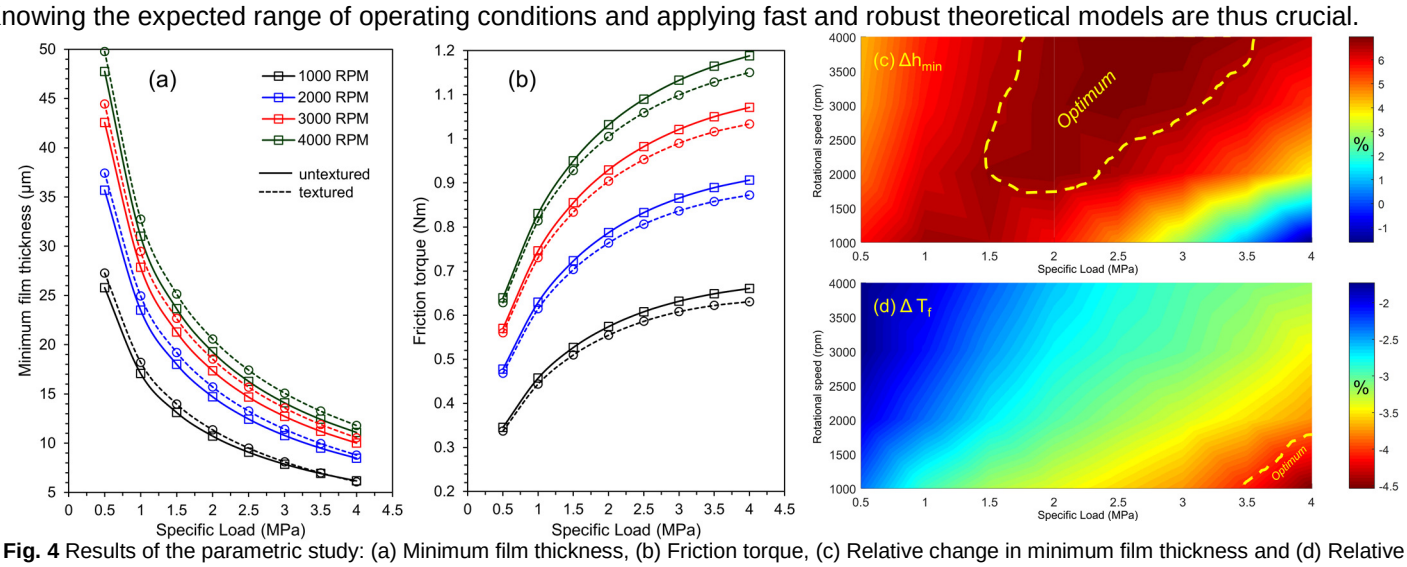

Fig. 4 Results of the parametric study: (a) Minimum film thickness, (b) Friction torque, (c) Relative change in minimum film thickness and (d) Relative change in friction torque.

CONCLUSIONS AND FUTURE WORK

Surface texturing can notably enhance bearing performance in terms of friction torque and minimum film thickness. However, the magnitude of performance improvements highly depends on the operating conditions, hence, selecting a texture design for a given bearing geometry and operating conditions is a challenging but key task. Applying special discretization schemes to handle discontinuities, taking advantage of multicore processing and strategically using Broyden's Method to find the pad's equilibrium position are ways to numerically study textured bearings most efficiently, allowing for arbitrary parametric studies necessary in the field of surface texturing.

To better approximate thermal effects and improve the accuracy of the developed model, a temperature model based on the Energy equation will be implemented in future work. A developed bearing test rig will help with further validation.

ACKNOLEDGEMENTS

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REFERENCES

[1] Gropper D, Wang L, Harvey TJ. Hydrodynamic lubrication of textured surfaces: A review of modeling techniques and key findings. Tribology International. 2016;94:509-29.

[2] Zouzoulas V, Papadopoulos CI. 3-D Thermohydrodynamic Analysis of Textured, Grooved, Pocketed and Hydrophobic Pivoted-Pad Thrust Bearings. 14th EDF/Pprime Workshop. Poitiers - Futuroscope, France2015.

[3] Dobrica MB, Fillon M. About the validity of Reynolds equation and inertia effects in textured sliders of infinite width. Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology. 2009;223:69-78.

[4] Ausas RF, Jai M, Buscaglia GC. A Mass-Conserving Algorithm for Dynamical Lubrication Problems With Cavitation. Journal of Tribology. 2009;131:031702-1.

[5] Arghir M, Alsayed A, Nicolas D. The finite volume solution of the Reynolds equation of lubrication with film discontinuities. International Journal of Mechanical Sciences. 2002;44:2119-32.

[6] Burden RL, Faires JD. Numerical Analysis: Brooks/Cole, Cengage Learning; 2011.

[7] Stachowiak GW, Batchelor AW. Engineering Tribology. 4th ed. Amsterdam; London: Elsevier/Butterworth-Heinemann; 2014.

[8] Frene J, Nicolas D, Degueurce B, Berthe D, Godet M. Hydrodynamic Lubrication: Bearings and Thrust Bearings: Elsevier Science; 1997.

KEYWORDS

Hydrodynamics: Tilting-Pad Bearings, Surfaces: Surface Roughness, Hydrodynamics: Cavitation in Hydrodynamics.