Modelling and Analysis of Elastic and Thermal Deformations of a Hybrid Journal Bearing

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Abstract

The continuous demand for higher power density leads to a very challenging operational environment for sliding bearings regarding pressures and deformations. Understanding of the deformation behavior of heavily loaded bearings becomes even more pronounced when modern hybrid multilayer designs are considered. The aim of this study is to develop a numerical, multi-physical model for the evaluation of journal bearing performance. Hydrodynamics were based on the Reynolds equation and deformations were calculated using the integrated finite element method. Elastic and thermal deformations have a significant effect on bearing performance and those deformations can be adjusted with properties of polymer layer. The design of hybrid bearings is delicate and their properties must be tailored according to the operating conditions.

Keywords: Hydrodynamic lubrication, Deformation, Journal bearing, Polymer hybrid bearing

1. INTRODUCTION

A hydrodynamic sliding bearing is a very commonly used type of bearing in heavy industry. It is based on a hydrodynamic lubrication film which forms as a result of the relative movement and shape of the gap between the sliding surfaces. In case of a journal bearing, a narrowing gap forms between the bearing and the shaft as they are eccentrically located relative to each other due to external forces. An analytical solution to hydrodynamic lubrication of journal bearings can be found by assuming simple steady state operational conditions and bearing geometry conditions. For more complex cases, a numerical solution of Reynolds equation is needed in order to take into account, for example, dynamic operation conditions, thermal effects, shaft misalignment and out-of-round bearing geometries. In modern heavy machinery with continuously increasing demand of higher output (power density), elastic and thermal deformation of bearing surfaces and housings need also to be considered.

The interest in compliant liners has increased in recent years and several studies have shown that more uniform pressure distribution with reduced maximum pressure can be achieved with bearings utilizing compliant polymer layer, where the thickness of the elastic layer is in the range of 0.5-2mm and commonly used materials with different strengthening materials are Polytetrafluoroethylene (PTFE), Polyether ether ketone (PEEK) or Polyamide-imide (PAI), depending on the operational conditions. In this case it is obvious that a relatively high elastic deformation takes place in the compliant lining with relatively low hydrodynamic pressure. For example: Kuznetsov et al. [1] developed a model to investigate the effect of a polymer liner compliance on the bearing characteristics. They used a simple plane strain hypothesis for the calculation of elastic deformations. Thermal deformation was based on the solution of the energy equation and the hydrodynamic pressure on the solution of Reynolds equation. They concluded that compared to a standard bronze bearing, polymer hybrid bearings with a 2mm PTFE liner has higher load carrying capacity, more even oil film pressure distribution with up to 40% lower maximum pressure, similar or slightly higher maximum oil film temperature, more favorable (pocket shape) oil film distribution, increased minimum oil film thickness at the bearing midplane and slightly lower at the bearing edges and similar or higher power loss depending on operating conditions. Thomsen and Klit [2] studied the effect of polymer layer on large scale dynamically loaded bearings. Their hydrodynamics model was also based on Reynolds equation. Thermal response is calculated by a constant viscosity method and the equivalent temperature rise. Elastic deformation of the polymer liner is considered using the Winkler/column compliance method and thermal deformation is not considered. They concluded that the eccentricity increases slightly with the compliant layers but the maximum pressure decreases significantly. Thomsen and Klit [3] later improved their model by introducing finite element model for the calculation of deformations and two dimensional energy equation was used to calculate the lubricant temperature over film thickness without considering thermal deformations This study also covers shaft misalignment and flexure bearings with and without polymer liner compared with standard bearing with and without polymer liner. They concluded that in the misalignment case a significant reduction of maximum pressure using polymer liner with standard bearing can be achieved. With the flexure bearing the reduction is not so clear. Generally, standard journal bearings benefited much more from compliant layers than the flexure bearing did. Aksoy and Aksit [4] presented a 3D thermoelastohydrodynamic (TEHD) model for bump-type compliant layer journal bearings. Their model is largely based on finite element method, except for film temperature iteration which is performed using finite difference method.

Other models with deformation but without a separate compliant layer can also be found in literature. These include, for example, Fatu et al. [5] that developed a full TEHD-model for journal bearings with dynamic loading conditions. They used a finite element method for the calculation of both elastic and thermal deformations while the hydrodynamic solution was calculated using the Reynolds equation. Bendaoud et al. [6] studied the effect of journal bearing elastic deformation in extreme loading conditions. They focused only on elastic deformation neglecting the thermal characteristics. In this case also the elastic deformation was calculated using the finite element method and the hydrodynamics were based on Reynolds equation. Piffeteau et al.[7] developed a full TEHD model for engine connecting rod bearings. The model included a finite element method for discretization and solution of Reynolds equation, energy equation and heat transfer equation. This means that all of the governing equations are solved in each time step of the dynamic model using the Garlekins method resulting in less iterations. They assumed heat transfer to be constant in the axial direction and used a 2D mesh for heat transfer. Zhao et al.[8] upgraded an existing hydrodynamic journal bearing model to a full TEHD model by integrating a commercial finite element - software to the old model based on Reynolds equation. They used a basic beam theory to calculate the shaft bending and resulting misalignment and used a finite element - model only to calculate the local deformations of bearing surfaces.

In conclusion most of the developed models used a finite element method to calculate the deformations and a numerical model based on Reynolds equation for solving the hydrodynamic lubrication. This method has also been adopted by Bouer and Fillon [9] in their study considering journal bearings in extreme operating conditions and by Kim and Kim [10] in their dynamic, connecting rod bearing, analysis. Also Kucinschi et al. [11] used this approach for their TEHD study of steadily loaded journal bearings. This kind of effective approach was chosen also for this work.

The main purpose of this work is to develop a parameterized calculation model for hydrodynamic radial hybrid journal bearings incorporating a compliant polymer liner. The model takes into account elastic and thermal deformations of the bearing surfaces and especially those of the polymer liner. Hybrid bearing performance is evaluated in a variety of operating conditions including shaft misalignment.

2. MODEL DESCRIPTION

A basic journal bearing consist of a rotating shaft within a stationary bearing (bushing) as shown in Fig. 1. The figure shows also the coordinate system, main parameters and unwrapped bearing layout, which is used for results handling. The coordinate x1 refers to the line of centers where minimum and maximum film thicknesses are located. A polymer liner is added to the bearing surface, while maintaining the same bearing inner diameter. [12]

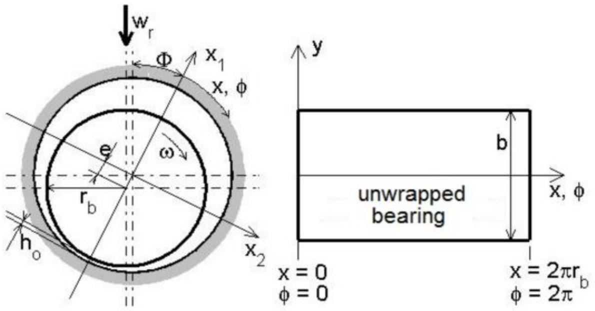

Figure 1. *Illustration of hydrodynamic journal bearing*

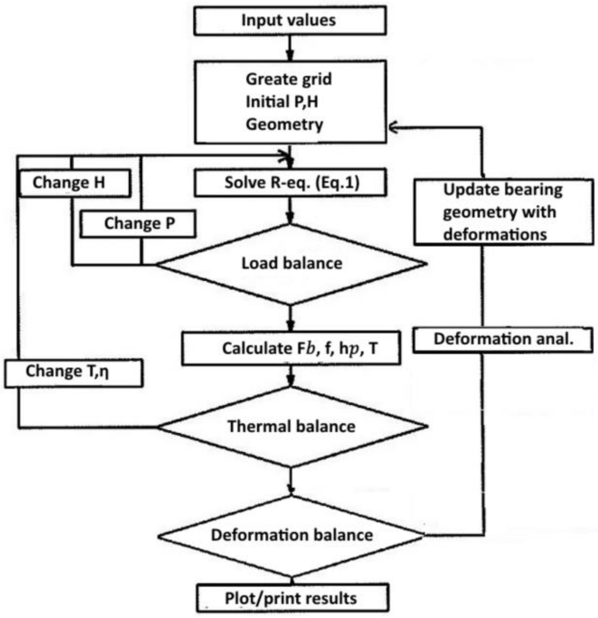

Figure 2. *Flow chart for the calculation procedure*

The solution of hydrodynamic lubrication between the shaft and bearing is based on the numerical solution of Reynolds equation. A more extensive description of the hydrodynamic model can be found in [13] but the governing equations are as follows. Allowing a side leakage, assuming Newtonian lubricant and neglecting fluid inertia effects, the transient Reynolds equation for an incompressible lubricant can be formulated as (Eq. 1). The bush is assumed to be fixed ($u_a = 0$) and the shaft sliding is pure rotational sliding without axial movement ($u_b = \omega r_b$).

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(h^3\frac{\partial p}{\partial y}\right)=6\eta_e u_b\frac{\partial h}{\partial x}+12\eta_e\frac{\partial h}{\partial t} \qquad \text{(Eq. 1)}$$

where h is the lubrication film thickness, p is the fluid film pressure, u_b is the shaft sliding speed in x direction and t is time. η_e is the effective viscosity of the lubricant according to (Eq.2).
The viscosity-temperature relationship is governed by: [14]

$$\eta_e=\eta_x e^{\left(\frac{159.56}{T_e+95°C}-0.181913\right)*ln\frac{\rho VG}{10^6*\eta_x}} \qquad \text{(Eq. 2)}$$

In (Eq. 2) η_x is a constant, T_e is the effective lubricant temperature shown in (Eq. 7), ρ is the density of the lubricant and VG is the lubricants ISO-VG class.

From (Eq. 1) the film thickness h and the pressure distribution p will be solved numerically. Assuming a steady state condition and taking into account elastic and thermal deformations of bearing surfaces the film thickness h can be written as follows:

$$h(x,y,t)=h_{co}(t)+h_{geom}(x,y)+h_{def}(x,y) \qquad \text{(Eq. 3)}$$

where $h_{co}(t)$ represents minimum film thickness at $y = 0$ at time t, $h_{geom}(x,y)$ is film thickness induced only by the bearing geometry and $h_{def}(x,y)$ is the film thickness caused be deformation of the bearing surfaces including the liner, bearing backing material, housing and shaft surface. The shaft misalignment and axially variable geometry are allowed in the steady state case by modifying the $h_{geom}(x,y)$ accordingly.

Bearing friction is calculated using equation (Eq. 4) by integrating the lubricant shear stress over the bearing surface. In the parts of the bearing where cavitation occurs, a zero value for the shear stress is applied.

$$F_b=\int_{-b/2}^{b/2}\int_{0}^{2\pi r_b}\left(\frac{h}{2}\frac{dp}{dx}+\frac{\eta_e u_b}{h}\right)dxdy \qquad \text{(Eq.4)}$$

Friction force is then converted into power loss by multiplying with sliding speed according to equation (Eq. 5).

$$h_p = F_b u_b \qquad \text{(Eq.5)}$$

Power loss h_p is then used to calculate the temperature rise in the lubricant according to equation (Eq. 6)

$$\Delta T = \frac{2Kh_p}{\rho c_p q_{in}} \qquad \text{(Eq. 6)}$$

where K is the ratio of frictional heat going to the lubricant, ρ is the density of the lubricant, c_p is the heat capacity of the lubricant and q_{in} is the lubricant inlet flow rate. K is a variable that is dependent on the loading conditions as well as thermal design of the bearing.

For the lubricant, an effective mean temperature is used

$$T_e = T_{in} + \Delta T/2 \qquad \text{(Eq. 7)}$$

The effective temperature is used to calculate the effective viscosity from (Eq. 2). This kind of isothermal lubricant model is valid and tested for bearings rotating relatively slowly. The effective lubricant temperature concept may underestimate the actual contact temperature at high sliding speeds, resulting in overestimated viscosity, film thickness and friction.

2.1 Bearing data

The bearing having inner diameter of 60 mm, width of 30 mm was modelled to study the bearing performance. The reference bearing was a 5 mm thick bronze bearing with an elastic modulus of 96 GPa. A polymer hybrid bearing had 4 mm of steel as a backing material with elastic modulus of 208 GPa and a 1 mm thick layer of bearing grade Torlon® 4301 (PAI) polymer as a liner material. Torlon® 4301 has an elastic modulus of 5.3 GPa, thermal conductivity of 0.54 W/mK and thermal expansion coefficient of 25.2E-6 1/K [15]. A relatively low thermal expansion coefficient was one of the reasons of choosing this liner material along with excellent mechanical properties. The thermal expansion coefficient is still however roughly 1.4 times that of the bronze, resulting in more thermal deformation, which is undesirable for the operation of the bearing.

A cylindrical bearing geometry was assumed with a radial clearance of 2 ‰. Steady state and full film conditions were also assumed with a radial loading of 10.8 kN and 21.6 kN, which resulted in a specific pressures of 6 MPa and 12 MPa accordingly, sliding speeds of 4 m/s and 8 m/s were used and oil inlet temperature was 50 ºC. ISO-VG150 lubricant was used in the analysis. A shaft misalignment β of 0.0229º, to the worst possible direction (along minimum film thickness), was used. The misalignment β refers to angle between tilted shaft vs shaft aligned parallel to the bearing axis. Every misalignment case has a comparable reference case (without misalignment) to study the effect that this has to key operational parameters of the bearing.

2.2 Calculation procedure

The effect of heat distribution between bearing surfaces was modelled in an approximate way. Heat distribution is a challenge to predict since it depends on the operational conditions. In this study the frictional heat carried away by the lubricant is assumed to be constant having the value of 80 % (K = 0.8). The value was estimated from the TUT test rig [13][16] energy balance based on the measured total friction torque, shaft velocity, oil flow rate and inlet and outlet temperatures in the corresponding operating conditions. That value was used for all of the investigated cases in this study. The remaining 20 % of frictional heat is divided between shaft and bearing surfaces. A variable T was created to describe the ratio of frictional heat going to the bearing. First assumption was that since the polymer layer is a good insulator, all of the structural heat goes to the shaft i.e. the T-ratio of heat division to bearing and shaft is zero. This assumption seems to be the mostly used in literature. It is very likely that with a polymer bearing a bigger portion of the heat will go to the oil and to the journal due to insulating properties of the polymer. The other extreme is to assume that the polymer layer has the same *T*-ratio (50 %) that is often used for the standard bronze bearing in the calculations without energy equation. The correct value is almost certainly something between these two, hence third case is taken from the middle of these two (25 %). The estimation of heat division based on the thermal conductivity of the materials and the sliding speed profile was ignored at this time. The exact solution of frictional heat division would need the numerical solution of energy equation, which was left for future studies. The presented approach was considered to be accurate enough for a conceptual study of polymer layer, where exact numbers are not the primary interest.

A flow char of the calculation procedure is presented in (Fig. 2). At first step the Reynolds equation is iterated for load balance using initial undeformed geometry. After that deformations are calculated, using the pressure and frictional heat distributions from the hydrodynamic model, before moving to the next iteration step. Deformation results, i.e. displacement of the calculation nodes over the shaft and bearing surfaces, are then used to modify the (initial) bearing geometry in the hydrodynamic calculations by altering the local gaps node by node between a shaft and a bearing according to (Eq. 2.) A few iterations are needed to obtain a satisfactory balance between the state of the hydrodynamic calculations and the state of deformation calculations.

Deformations are calculated using commercial finite element software, Abaqus [17], which is fully integrated as a part of the Matlab-based hydrodynamic calculation model. Both mechanical and thermal loading for the FE model are taken from the outcome of the hydrodynamic calculations in the form of pressure and friction force (power loss) distributions in each deformation iteration step. Bearing and shaft are calculated separately. The FE-mesh is shown in Figs. 3 and 4. Polymer liner is highlighted with darker colour on the inner side of the bearing in Fig 3. Section of the bearing is also enlarged.

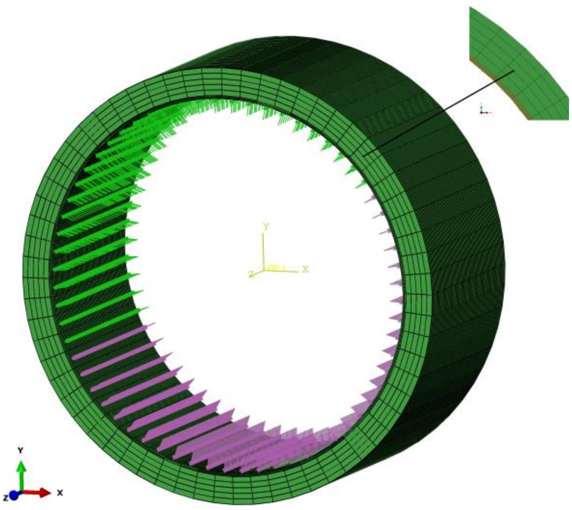

Figure 3. *FE-mesh for bearing*

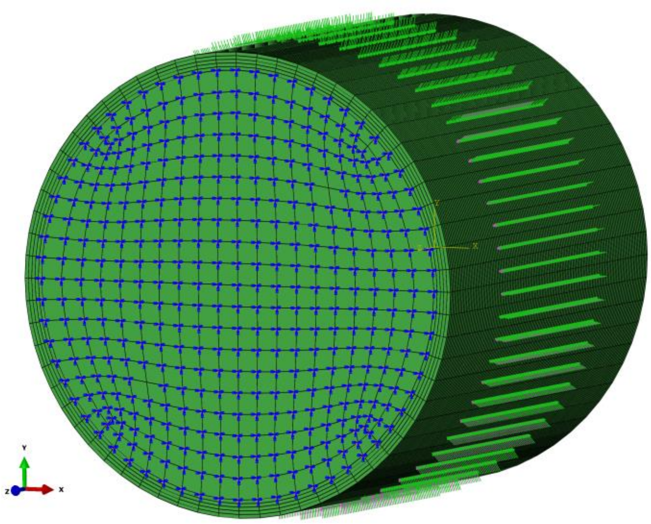

Figure 4. *FE-mesh for shaft*

Meshing of the bearing and shaft outer layer was done manually in Matlab in order to achieve full control over mesh parameters. The bearing assembly used in the model corresponds to the TUT sliding bearing test rig. This was done so that the future comparison of results to the test data and model verification is possible. A description of the test rig can be found in [16].

The bearing housing was modelled using tetrahedral combined thermal-stress elements and was supported similarly to the test rig construction. The test bearing is fixed to bearing housing and modelled using hexahedral thermal-stress elements. A section of the test shaft twice the length of bearing was modelled using hexahedral thermal-stress elements. The shaft was supported at both ends with boundary conditions simulating again the test rig construction. Bearing and shaft surfaces are modelled with 65*65 nodes and the FE-model loading can be seen in figures Figs.3 and 4 purple arrows being the pressure loading and green arrows being the thermal loading. A convectional heat transfer coefficients were estimated based on the individual characteristics of the surfaces. Also a radiative heat transfer boundary condition was used. Heat transfer calculations in the finite element software are based on the Green-Naghdi energy balance. [17]

3. RESULTS

The developed model was utilized to evaluate the potential of using a polymer liner in the bearing surface to improve bearing performance and especially its robustness against shaft misalignment. The main idea of compliant polymer layer is to increase the amount of elastic deformation and therefore reduce maximum pressure and increase minimum film thickness. It was concluded earlier by the author [12] that thermal deformation cancels out some of the beneficial elastic deformation since they have an opposite effect on bearing geometry at the pressurized area of the bearing. Consequently, a low thermal expansion coefficient of the polymer material is crucial in order to obtain full performance benefits from the elastic deformation of the polymer layer such as decreased maximum pressure. A bronze bearing, which is commonly used by the industry, was used as a reference case where the hybrid bearing performance was compared.

The calculated key performance parameters for bronze and polymer hybrid bearings are shown in Fig. 5. Generally, the variation of heat division between bearing and shaft affected through temperatures and thermal deformation as can be expected. As the *T*-ratio decreases, i.e. insulation increases, bearing clearance decreases. Increase in insulation causes lubricant mean temperature to increase and friction and film thicknesses to decrease.

Figure 5a shows that a polymer layer has very little, if any, effect on friction on its own compared to standard bronze bearing. However, the variation of *T*-ratio has a slight effect on friction through increased lubricant temperatures. Sliding speed makes a clear difference in friction which was expected since in hydrodynamic journal bearings, friction comes from viscous shear of the lubricant and higher sliding speed means higher film thickness and more viscous shear. Also increased specific pressure had an effect on friction due to high pressures, which reduces film thickness according to (Eq. 3).

Figure 5b shows that in all comparable cases, the maximum hydrodynamic pressure decreases when the hybrid bearing is used instead of bronze bearing. This pressure decrease is largest when the polymer layer has high insulation and when the specific pressure is high. The pressure profile is slightly wider and lower in the axial direction as can be seen in Fig 6. This is anticipated and desired behavior caused by elastic deformation of the compliant layer creating more space for the lubricant at the loaded side of the bearing. Along the length of the bearing, the profile is fairly similar, as can be seen from Fig 7. In the most extreme case of 12 MPa specific pressure with misalignment the maximum pressure was considerably lower than that of the reference bearing. The hybrid bearing effectively reduced the maximum pressure in the shaft misalignment case when the specific pressure is high, but this effect is fairly small with lower specific pressure. This indicates that hybrid bearing properties should be tailored carefully according to given operating conditions and that the steel-polymer hybrid used for this study was a bit too stiff for the lower specific pressure, since the reduction in maximum pressure was quite low.

Figures 5c and 5d show that in almost all of the comparable cases polymer layer increases the minimum film thickness in the center of the bearing ($B = 0$), creating a beneficial pocket shape for the lubricant in the center of the bearing. Also the overall minimum film thickness, which for some cases is at the edge of the bearing, generally increases with the liner. Film thickness at the center of the bearing however increases in all of the cases with polymer liner as can also be seen from a film thickness profile in Fig 8. The bearing center zone is the more critical area due to pressure distribution presented in Fig 6. Again under the most severe operating conditions the difference in minimum film thickness is clearest indicating that the polymer hybrid was too stiff for the lower specific pressure. The *T*-ratio has an effect to film thickness due to reduced clearance of the bearing, which leads to decreased fluid flow rate and hence elevated lubricant mean temperature as seen in (Fig. 5e). Higher lubricant temperature reduces film thickness and friction and is typical for a journal bearing operating at low clearance. Attitude angle Φ shown in Fig. 5f is typically affected by sliding speed and loading. The polymer layer seems to have no effect on that, but the *T*-ratio has a small effect in increasing the attitude angle as *T* decreases.

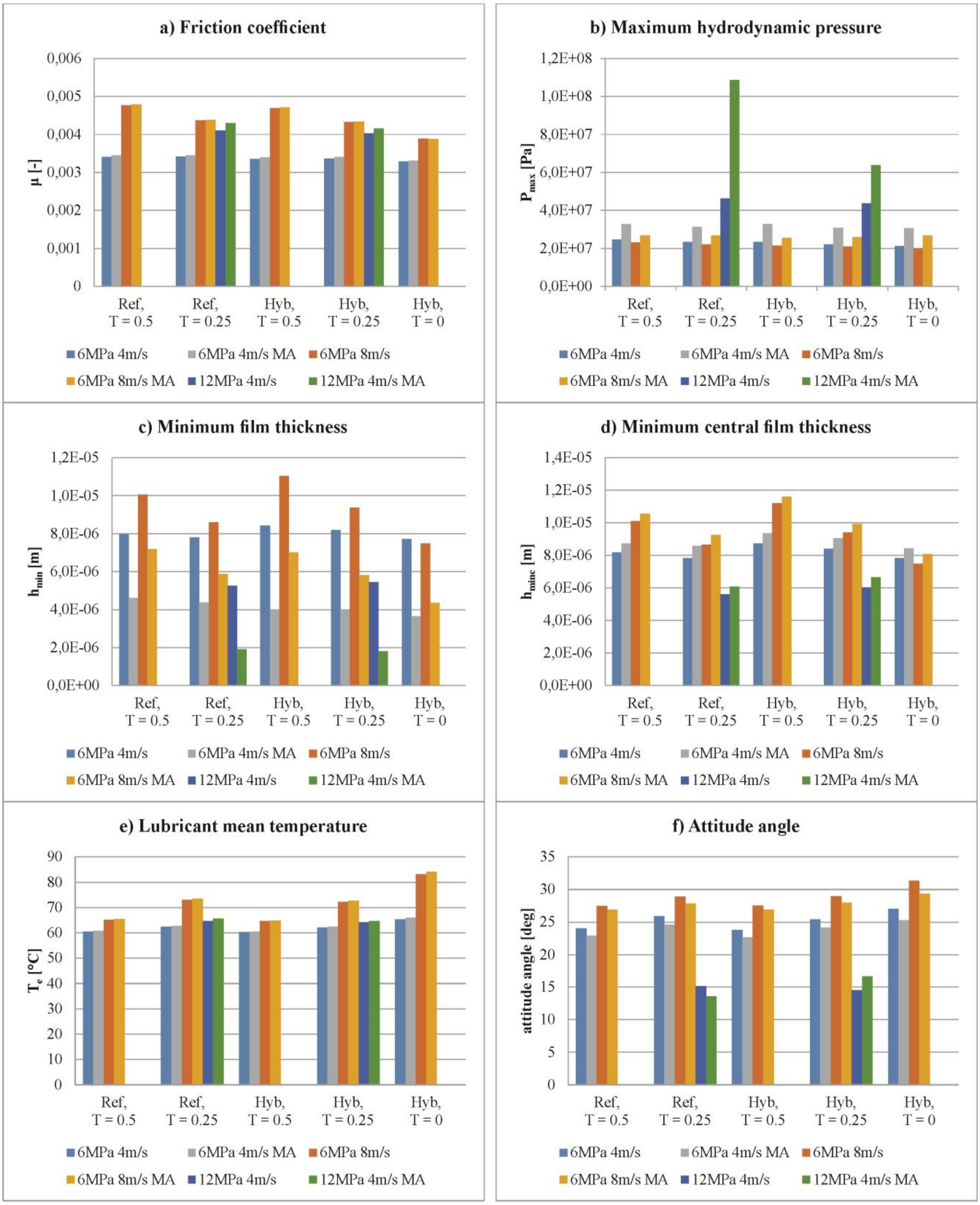

Figure 5. *Key bearing performance parameters: MA = shaft misalignment, T = Portion of the frictional heat going to the bearing, Ref = Standard bronze bearing, Hyb = Polymer hybrid bearing*

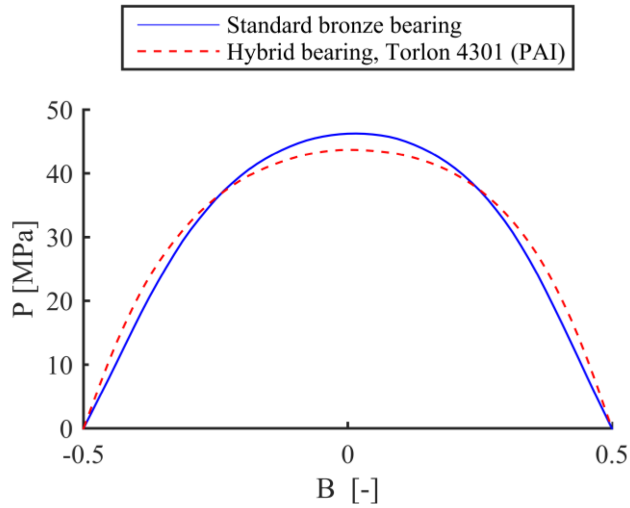

Figure 6. *Axial pressure profile at the location of maximum pressure, L = 0.5, 12MPa, 4m/s*

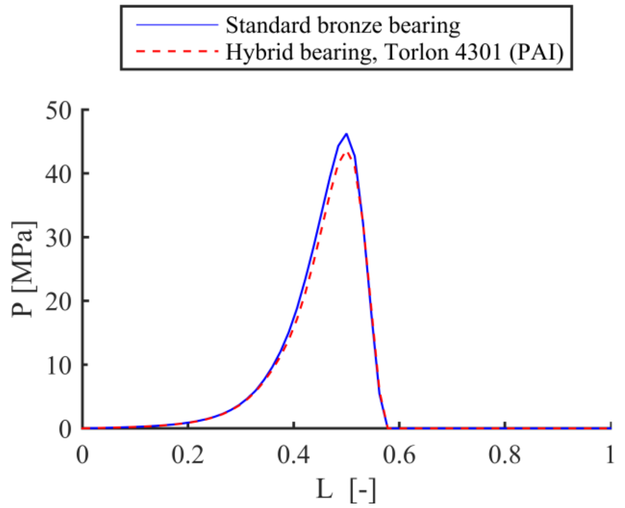

Figure 7. *Pressure profile along the length of the bearing at the location of maximum pressure, B = 0, 12MPa, 4m/s,*

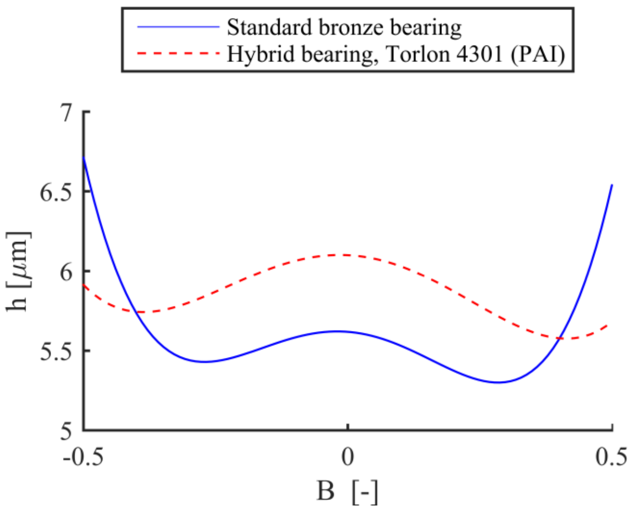

Figure 8. *Axial film thickness profile at the location of minimum film thickness, L = 0.5, 12MPa, 4m/s*

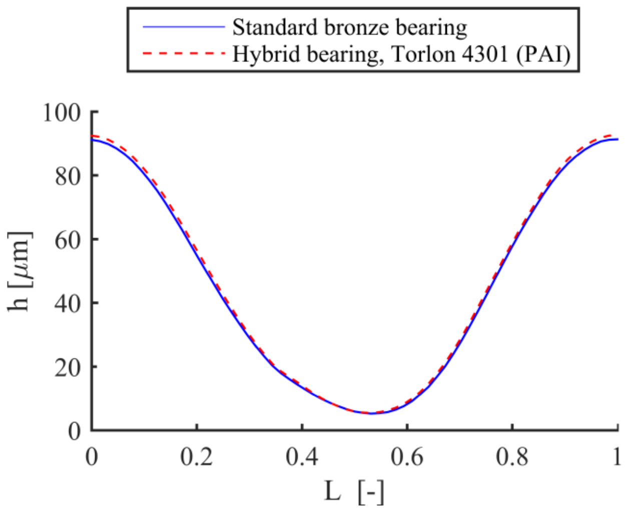

Figure 9. *Film thickness profile along the length of the bearing at the location of minimum film thickness, B =0.4 for hybrid bearing and B = 0.3 for standard bearing, 12MPa, 4m/s*

The relative effects of elastic and thermal deformations to the performance of the bearing increase when the bearing is heavily loaded and a bigger advantage due to compliant layer can be achieved with higher loads. With higher hydrodynamic pressures the elastic effect becomes relatively more significant compared to thermal effect due to increased specific pressure while maintaining almost the same amount of frictional heat. The highest hydrodynamic pressures occur especially at misalignment cases. In figures Fig. 10–13 are pressure profiles and film thicknesses of the most severe loading case where polymer liner had the biggest effect. A clear reduction of maximum pressure can be seen along with the increase in minimum film thickness. Compliance of the polymer layer causes the pressure to be more evenly distributed in the hybrid bearing. This results a pressure profile closer to that of the perfectly aligned bearing i.e. the resultant of the pressure is closer to the center of the bearing. It must be noted that the load carrying capacity cannot be determined from these 2D profiles. A small kink in the film thickness curves of polymer bearing in Fig 13 at *L=0.4* is most likely due to the relatively high Poisson's ratio of the polymer. This causes the layer to chance shape rather than compress. Stresses and temperatures in the polymer layer were also calculated to make sure they are within limits of the selected polymer material.

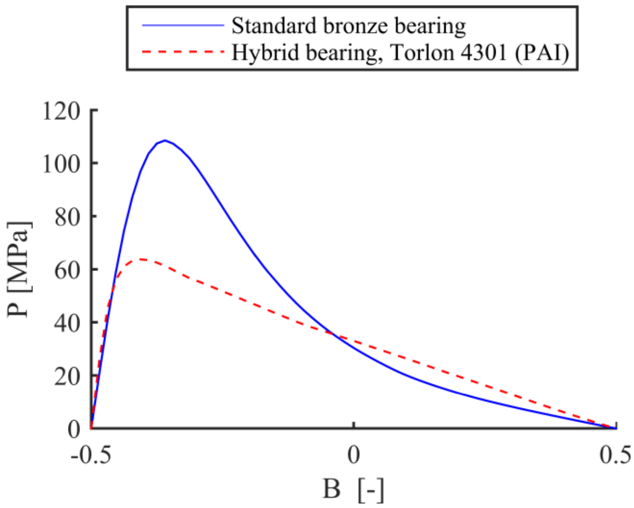

Figure 10. *Axial pressure profile at the location of maximum pressure, L = 0.55 for polymer bearing and L = 0.5 for standard bearing, 12MPa, 4m/s, β = 0.0229º*

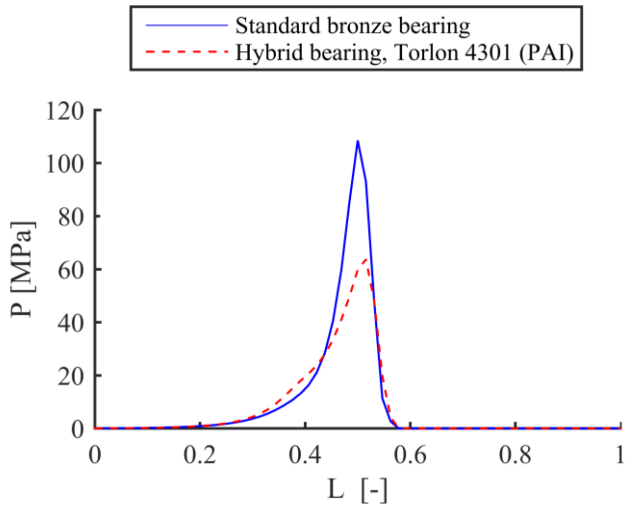

Figure 11. *Pressure profile along the length of the bearing at the location of maximum pressure, B = -0.4 for polymer bearing and B = -0.35 for standard bearing, 12MPa, 4m/s, β = 0.0229º*

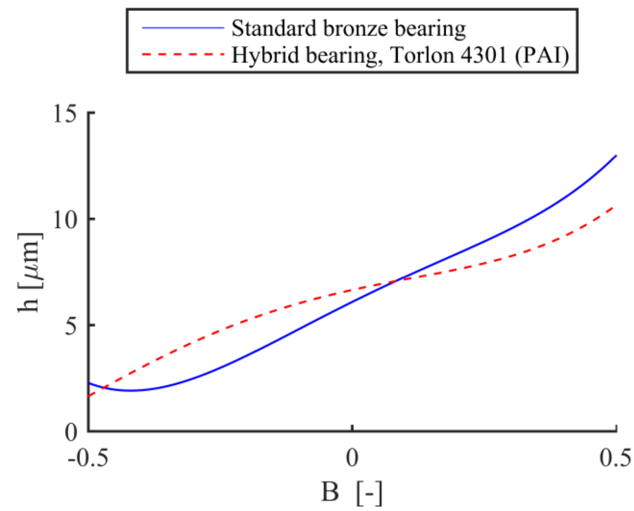

Figure 12*. Axial film thickness profile at the location of minimum film thickness, L = 0.5, 12MPa, 4m/s, β = 0.0229º*

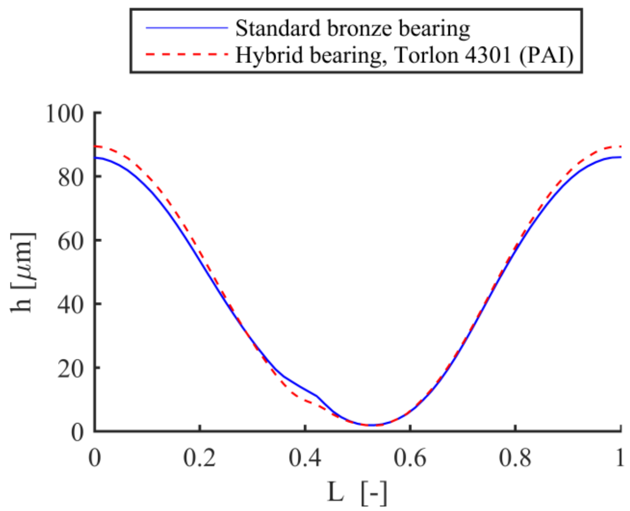

Figure 13. *Film thickness profile along the length of the bearing at the location of minimum film thickness, B = -0.5 for hybrid bearing and B = -0.4 for standard bearing, 12MPa, 4m/s, β = 0.0229º*

From the results it can be seen that the model is not extremely sensitive for the *T*-ratio alone. However, thermal behavior of the bearing is very complex and in reality *T*-ratio is dependent of the portion of heat carried away by the lubricant, *K*-ratio, which is not constant but a function of loading conditions and thermal design of the bearing. As a whole, thermal behavior clearly has a high importance on bearing performance. Since the model currently has no energy equation, the partitioning of frictional heat proved to be a problem that needs further attention. Naturally the structural temperatures and therefore thermal deformation of bearing and shaft are greatly dependent of this ratio also and for that reason the model needs to be upgraded in the future.

However, it is possible to draw some conclusions on the effect of polymer liner based on this study. Firstly, it is reasonable to assume that some minor benefits in friction coefficient can be achieved since polymer as an insulator in bearing surface will cause a larger portion of heat to transfer to oil and shaft, bigger *K*-, smaller *T*-ratio, in both cases reducing the friction. This aspect needs further studying with improved thermal model. This kind of behavior would actually be beneficial since the same polymer layer would provide reduced maximum hydrodynamic pressure at low speed and high radial loading situations. It would also provide reduced friction at high speed and low radial loading situations expanding the usable operational speed range of the bearings as far as the limiting design temperature of the bearing has not been reached.

Reduced maximum pressure, higher film thickness and good behavior in extreme loading conditions also indicates increased load carrying capacity which would allow smaller bearings to be used and thus further reducing the friction.

Careful planning and choice of materials is needed in the design process of a hybrid journal bearing. A liner should have just right amount of elasticity: enough to allow for elastic deformation, not too much to cause liner to collapse under load. Low thermal expansion coefficient of the liner is beneficial, preventing a formation of thermal deformation bubble, which is caused by thermal expansion of a local bearing hotspot. The bubble at least partly cancels out the advantages of elastic deformation. A good thermal conductivity could be a positive property even though it was seen that a low L-ratio leads to low friction but it would bring the temperature of the polymer down, preventing it from melting. If the bearing is operating near its limits with high temperature and low clearance, bad thermal conductivity might lead to a seizure.

4. CONCLUSIONS

The main purpose of this work was to develop a parameterized calculation model for hydrodynamic radial hybrid journal bearings incorporating a compliant polymer liner. The model takes into account elastic and thermal deformations of the bearing surfaces and especially those of the polymer liner. Hybrid bearings key performance parameters were evaluated in variety of operating conditions including shaft misalignment. Following main conclusions can be drawn:

- Hybrid bearings with a compliant polymer layer have good potential and key performance parameters such as maximum pressure and minimum film thickness indicate improvement compared to standard bronze bearing.
- Hybrid bearings have a potential to reduce local maximum edge pressure involved in shaft misalignment cases.
- Due to material properties of typical polymers the deformation behavior of the hybrid bearing is considerably different to that of the standard bearing. More beneficial elastic deformation exists but also more unwanted thermal deformation
- Hybrid bearing properties should be tailored carefully according to given operating conditions i.e. it needs to be designed correctly in order to get the benefits
  Some of the effects of polymer layer are indirect: like reduced friction caused by reduced clearance, which can be tracked to low thermal conductivity of the layer.

5. ACKNOWLEDGEMENTS

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