Simulation of Microstructured Rolling-Sliding Contacts

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Abstract: To reduce friction in lubricated tribological contacts, the surfaces of the contacting bodies can be microstructured to improve lubricating conditions. For lower loaded contacts this approach has already reached industrial applications, e.g. the piston-liner contact. For higher loaded contacts the effects are currently in basic research. Elastic deformation in the contact area plays an important role in those cases. This paper presents an approach to compute microstructured elastohydrodynamic contacts using Comsol Multiphysics and compares results with those attained on a test bench.

Keywords: Tribology, Lubrication, Microstructures, Elastohydrodynamics

1. Introduction

In this presentation the impact of microstructures on highly loaded rolling-sliding contacts – like cam-tappet or gear tooth contacts – is investigated. In such contacts elastohydrodynamic lubrication theory applies, i.e. lubricating film and elastic deformation are of similar size [1]. For lower loaded, sliding contacts – such as piston-liner contacts – several authors showed that microstructures can improve lubricant film formation and reduce friction, e.g. [2]. For highly loaded contacts this question is currently in basic research. Due to high pressure in the contact area elastic deformation of contacting bodies plays an important role. Moreover, microstructures influence this deformation resulting in a local change of lubricating film.

Friction reduction itself is a main research promoter with the goal of increased energy efficiency. The valve train for example is a major contributor to combustion engine losses at low crankshaft rotation speeds. Therefore, the cam-tappet line-contact, see Figure 1, dominating these losses, is chosen as a demonstrator. The presentation will focus on the simulation of microstructured contacts using Comsol Multiphysics based on a “Full-System Approach” from [3] for stationary contacts. An outlook on experimental results for the tribotesting of a cam-tappet contact will complete the paper.

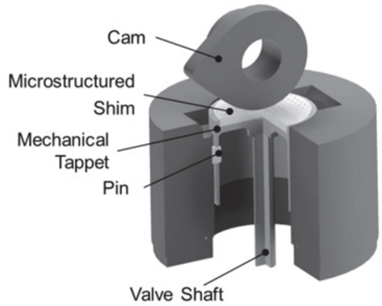


Figure 1. Cam-tappet line-contact with microstructured shim as used in experimental investigations.

2. Governing Equations

To compute the fluid film pressure and gap distribution in an elastohydrodynamic contact, a coupled solution of the hydrodynamic problem, expressed by the REYNOLDS equation, and the elastic deformation is necessary. In the following, the hydrodynamic problem shall be described a bit more in detail for a two dimensional (2D) case. For a three dimensional (3D) case, additional terms for gradients in the second dimension of the lubricant gap have to be added. The equations for the elastic problem shall be excluded, as the standard structural mechanics module is used.

The numerical solution of a lubricated cam-tappet contact can be reduced to a cylinder on plane problem, as shown in Figure 2. The radius, the normal force and the boundary velocities would then change with time, which is neglected in this study.

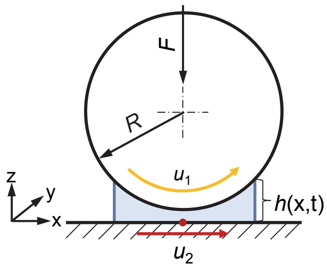


Figure 2. Equivalent model for a cam-tappet contact.

The elastic deformation of both bodies is calculated on one body only (the flat one) using an equivalent Young's modulus and an equivalent Poisson's ratio. For identical materials the Young's modulus is half of the nominal one and Poisson's ratio remains constant.

The REYNOLDS equation for a 2D case is expressed by

$$\frac{\partial(\rho\cdot h)}{\partial t}+\frac{\partial}{\partial x}\left(-\frac{\rho\cdot h^3}{12\cdot\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial x}(\rho\cdot h\cdot\bar{u})=0.$$

The first term is only present in time dependent cases and reflects squeeze effects. The second term describes the influence of pressure gradients and the third, the couette term, accounts for boundary velocities of the contacting bodies and the wedge shape of lubricant gap. Viscosity η and density ρ are dependent on pressure. To describe this behavior viscosity is modelled with an equation proposed by ROELANDS [4]

$$\eta(p)=\eta_{p=0}\cdot\exp\left[\left(\ln(\eta_{p=0})+9.67\right)\left(-1+\left(1+\frac{p}{P_R}\right)^{z_R}\right)\right]$$

and density with an equation by DOWSON/ HIGGINSONS [5]

$$\rho(p)=\rho_{p=0}\cdot\frac{0.59\cdot10^9+1.34\cdot p}{0.59\cdot10^9+p}.$$

In the couette term, the mean entrainment velocity $\bar{u}$ is the arithmetic mean of the surface velocities of the two contacting bodies. The last remaining variable is the height distribution h, namely

$$h(x,t)=h_0(t)+\frac{x^2}{2\cdot R}+\delta_{\mathrm{elastic}}(x,t)+h_{\mathrm{structure}}(x,t).$$

The first term represents the rigid body motion of the upper cylinder, the second a simplification of the radius by a quadratic approximation, the third the elastic deformation in the contact and the last one the microstructure shape and also the positon of the microstructure.

3. Use of COMSOL Multiphysics

The implementation in Comsol Multiphysics is similar to that described in [3], but the used geometry differs as well as the fact that a time dependent study is necessary.

As stated before, for the elastic deformation, the structure mechanics module is used and an Equivalent Young's modulus and Poisson's ratio are employed. The lubricant film builds up a pressure that is applied as a boundary load to the elastic body. In this study the meshed body is a square for the two 2D case and a quad in the 3D case, as shown in Figure 3. The dimensions are related to those proposed in [3]. The quad in the 3D case resembles a cutout of the central area of a finite line contact as it exists in the cam-tappet contact. In this case zero displacement boundary conditions in the y-direction for the elastic problem and zero flux boundary conditions for the hydrodynamic problem are used.

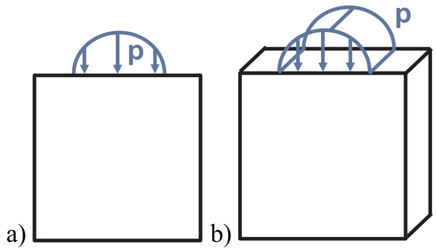


Figure 3. Geometry for the simulations used in a) 2D case and b) 3D case.

For the hydrodynamic problem, the REYNOLDS equation is solved in a weak form on boundary of the upper surface of the elastic bodies. The integral of the pressure on the boundary must balance the contact load F. To achieve this balance, the value of the rigid body motion h_0 has to be adjusted. The effect of cavitation at the diverging outlet of the contact is dealt with a penalty method, where negative pressures are multiplied by a large positive number and therefore the negative pressures are forced to become zero, see [6].

As the elastohydrodynamic pressure distribution and the elastic deformation are similar to that of a dry HERTZian contact and to facilitate the numerical solution, the hydrodynamic and the elastic problem are non-dimensionalized using the HERTZian parameters contact width and pressure. For more details concerning the dimensionless parameters see for example [7].

4. Simulation Results

First, a comparison to results from [8] shall be undertaken. Those results were achieved by using the traditional half space approach for elastohydrodynamic computations together with multigrid techniques [7]. In this case a cos-shaped microstructure moves through a line contact with a pressure of 2 GPa. The microstructured surface moves faster than the unstructured one, i.e. the contact is in condition of rolling and sliding.

It can be seen (Figure 4) that the pressure and lubricant gap distribution differ to some extent to the solution of [8]. It is assumed that the differences are due to the automatic timestepping methods in Comsol Multiphysics, as in [8] a constant timestep is used. The timesteps computed here were smaller for the condition shown in Figure 4. Thus, automatic timestepping allows the resolution of smaller gradients in the solution and therefore an additional constriction can be resolved before the microstructure in that case. For other microstructure positions almost no difference to the solution in [8] can be recognized, see Figure 5.

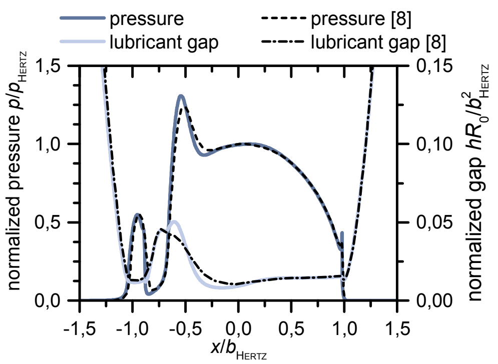


Figure 4. Comparison to traditional half-space approach by Venner et. al. [8] (digitalized) for a cos-shaped microstructure, microstructure at a dimensionless position of microstructure x_{Struct} = -0.75.

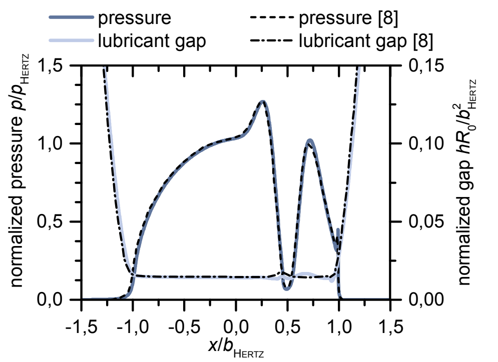


Figure 5. Comparison to traditional half-space approach by Venner et. al. [8] (digitalized) for a cos-shaped microstructure, microstructure at a dimensionless position of microstructure x_{Struct} = 0.5.

Second, the influence of slide-to-roll-ratio (SRR) on the lubricant gap of a microstructured line contact (pressure 0.6 GPa) shall be investigated. Therefore a line contact and a nearly rectangular microstructure shape are used. Using a rectangular shape for the microstructure increases effects at the transition from unstructured areas to the microstructure. This is done to enhance them, even this does not need to be an optimal microstructure shape for tribological purposes.

Figure 6 shows the pressure and gap distribution for the microstructure in the center of the contact area. It can be seen that an additional constriction occurs at the right hand side of the microstructure and therefore an additional pressure peak is present.

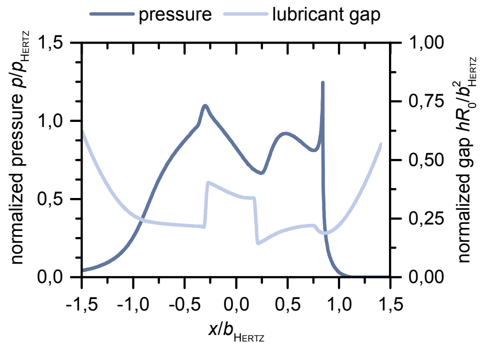


Figure 6. Pressure and gap distribution for the case of SRR = 0, i.e. pure rolling; microstructure in the middle of the contact.

One passage of this rectangular microstructure through the contact is shown in Figure 7. The central gap height (that is at a position of $x = 0$ in Figure 6) is plotted over the dimensionless position of the microstructure.

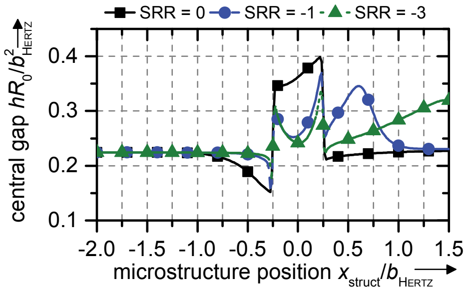


Figure 7. Microstructured two-dimensional line contact and influence of slide-to-roll ratio (SRR) on central film thickness.

It can be seen, that for the highest value of SRR (negative SRR indicates that the structured surface moves faster than the smooth one) the reduction of the central film thickness is the smallest and an increased thickness results after the microstructure passed the central position. This is due to an entrapped lubricant following the microstructure.

Last, a three-dimensional line contact (pressure 0.6 GPa) in the case of SRR = -1 was studied to account for lateral effects. The rotation-symmetric microstructure has a Gaussian-like cross section. Figure 8 shows a plot of the fluid film thickness of the contact area where the microstructure moves from left to right. Also in this case a trailing effect and an increase in lubricant film (yellow) can be observed following the microstructure (red circle). Nevertheless, it has to be noted that a film reduction occurs (light blue) at the side boundaries and in front of the microstructure.

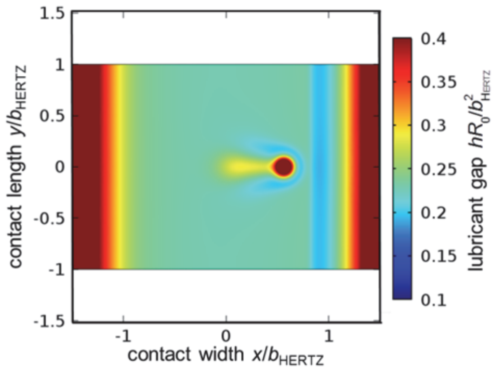


Figure 8. Microstructured 3D line-contact in case of sliding, microstructures moves from left to right.

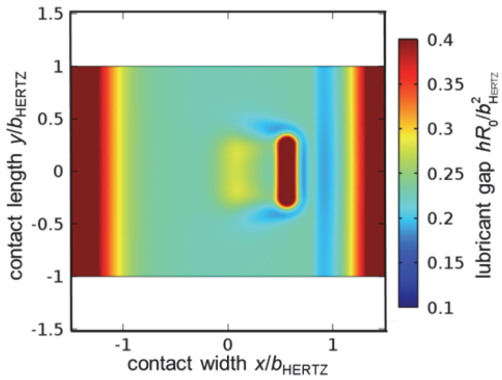


Figure 9. Microstructured 3D line-contact in case of sliding: linear extruded microstructure shape.

When the structure is extruded to a more linear shape perpendicular to the direction of motion, the lubricant gap around the microstructure changes compared to the rotation symmetric variant from Figure 8. The area with an increased lubricant gap is enlarged but also at the sides and in front of the microstructure the film reduction is increased (Figure 9).

5. Experimental Setup and Results

The simulation results presented before focused on SRR and lateral microstructure shape but neglected the lateral arrangement and distance between individual microstructures on a real tribological surface. Nevertheless, for a qualitative comparison friction measurements on a single cam-tappet component test rig shall be discussed here. For testing standard engine parts for cam, spring and valve are employed. Mechanical tappets with microstructured top shims, see Figure 1, are used. Rotation of the mechanical tappet within its guidance is restricted. The contact is lubricated with pure mineral oil FVA 3 (viscosity 25 mm^2/s at 70 °C). For comparison with simulation results two different microstructure shapes with similar arrangement shall be discussed here, see Table 1. The microstructures were manufactured using a femtosecond pulsed laser. The depth of the microstructures is about 1 µm.

Table 1: Microstructures for tribotesting

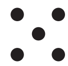
	V1	V2
Structure shape and arrangement relative to direction of motion 		

Figure 10 shows the change in friction force compared to a polished specimen for different cam rotation speeds. It can be observed, that linearly formed microstructures can significantly reduce friction compared to a polished specimen, whereas the effect for rotation symmetric microstructures is not that distinctive.

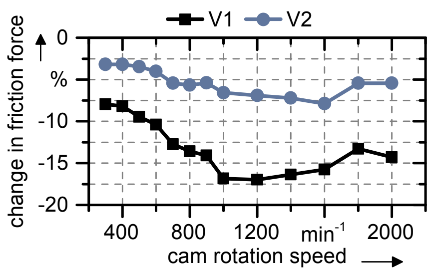


Figure 10. Change of friction force compared to polished shims for different microstructure arrangements (samples n = 3).

6. Conclusions

Simulation results underline positive effects of microstructures in the case of rolling-sliding. A friction reduction effect, measured on a test rig, is possibly due to a reduced amount of areas with mixed friction due to the cavity of the microstructure itself and the sliding induced trailing effect. But further investigation is necessary to verify that.

8. References

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