

Numerical investigations in mixed friction systems

Albert Albers, Benoit Lorentz and Lukas Nowicki

IPEK – Institute of Product Engineering, Karlsruhe Institute of Technology, South Campus, Kaiserstr. 10, 76131 Karlsruhe, Germany
e-mail: lorentz@ipek.uni-karlsruhe.de

ABSTRACT

A numerical approach is here selected to investigate mixed friction phenomena where testing rigs cannot be used or need complex adaptations to deliver reliable measurements. The following work focuses on the numerical investigations of mixed friction systems combining fluid-solid and solid-solid interactions at the micro scale. Goal is to improve the accuracy of future macro models by applying them more precise boundary conditions derived from micro models. A three dimensional model is built in a Finite Elements (FE) software composed of one fluid lubricating two sliding rough surfaces. Both surfaces are generated according to a statistical method making use of measured technical surfaces. To model the interactions between the fluid and the solid structure, the Arbitrary-Lagrangian-Eulerian remeshing process is used. A model is built, based on an axial floating bearing on which the adhesion effects are the most present. Global friction coefficient between both lubricated solids is established using the FE solver and solid-solid friction can be separated from the fluid-solid friction with further post-processing operations.

Key words: Mixed friction, tribology, technical surfaces, FSI, FEM

1. INTRODUCTION

In a current context of global warming, improvements in energy saving are highly demanded. Friction effects occurring in mechanical systems are responsible for up to 10% loss of the overall worldwide produced energy [1] and therefore need to be reduced. To make that possible it becomes necessary to increase the knowledge of different friction phenomena.

Present in almost any technical system, friction contacts are of vital importance for the reliability of industrial products. The functional adequate design of contact surfaces requires developing methods that enable studying friction mechanisms of tribological systems. Two approaches make that possible. On the one hand, the experimental way is the nearest to the reality because measures can directly be taken on prototypes of the existing products. On the other hand, the numerical approach is principally based on simplified mathematical models issued from experimental way. Due to the continuous development of hardware technology, the interest in numerical simulations has been put forward thanks to the high level of details reached by software to represent the reality. The main advantage of such an approach is a better modularity when systems become complex. Indeed, numerical simulations only need the CAD data of the studied system whereas the other requires building a prototype what is much expensive. Another advantage concerns the analysis scale.

The purpose of this paper is the study of mixed friction systems which necessitates taking into account the surface roughness. This implies that investigations have to be done at the microscopic scale where expected phenomena occur. That is the reason why the simulation

way is chosen in this work because such conditions make physical measurements quite more difficult. However, only few works have been done in numerical modelling of mixed friction and without considering real surface roughness, which is due to the complexity of coupling fluid mechanics with structural mechanics. One-way Fluid-Structure-Interactions (FSI) modelling still exists since the Computational Fluid Dynamics (CFD) programs are available. Such a coupling implies the following limitation that only the fluid is able to apply pressure on the structure and no contact topology changes are possible during the simulation. That is why a two way coupling is necessary as well as the concept of Arbitrary-Lagrangian-Eulerian (ALE) remeshing technique of C.W. Hirt [2], particularly interesting to model deformations of fluid domains. Without using these techniques L. Nowicki has modelled FSI in his work [3, 4] in order to calculate the friction coefficient of mixed lubricated systems composed of rough surfaces also called technical surfaces.

As exposed in the present paper, methods initiated by L. Nowicki are still developed at the Institute of Product Engineering (Karlsruhe) that enables calculation of lubricated friction problems. A mixed lubricated model is investigated by means of a three dimensional fluid flow simulation which also takes into account the technical surfaces characteristics. In addition to the global aim consisting in calculating friction coefficients, two secondary goals were defined as follows:

- Complete automatic process enabling technical surfaces generation
- Integration of standard software to built a micro-mixed-friction model

Transcending academic examples, this method will guaranty the calculation of complex lubricated friction problems. Figure 1 illustrates the complete simulation process needed to

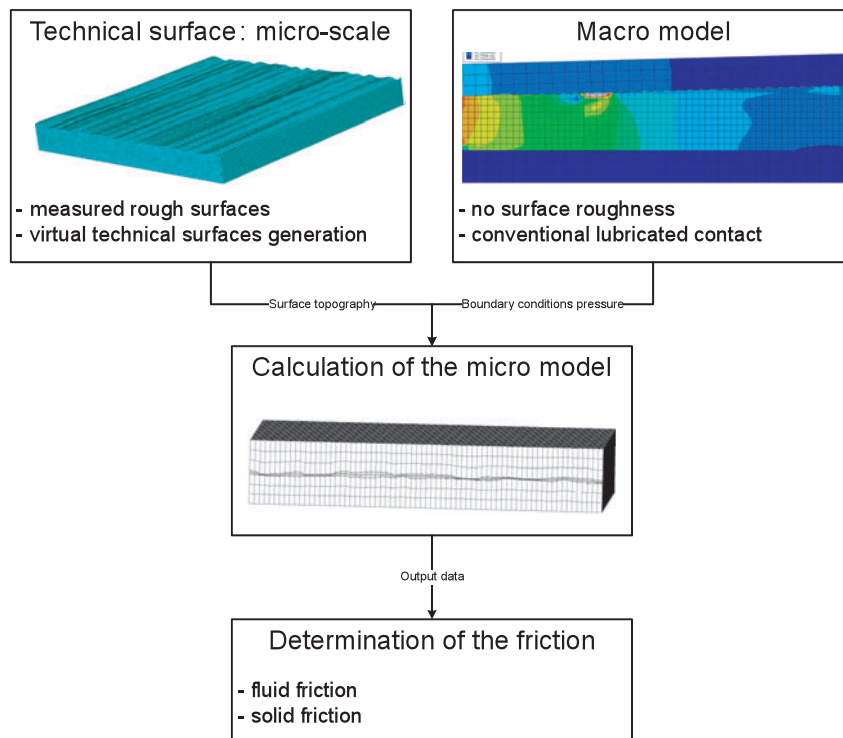


Figure 1. Flow chart of the overall process.

reach specified goals. In order to set up the micro model, two tasks must be achieved. It is first necessary to generate rough surfaces and to import them for the building of the geometrical micro model. Secondly, in order to get realistic boundary conditions for the micro model, a macro model is created and calculated. In this model the influence of the surface roughness is not considered, i.e., it is assumed that both bodies are perfectly smooth. Therefore, the pressure distribution in the fluid resulting from the macro model is used as boundary conditions. Then, the micro models enable calculating partial friction coefficients with post-processing operations that issue on the calculation of the global friction coefficient.

The following section presents the method used to generate virtual rough surfaces coming from experimental measures. Then, section 3 focuses on the simulation models whose first model is composed of two flat surfaces submitted to conventional hydrodynamic lubrication regime. The second model refers to the mixed lubrication model composed of two rough surfaces and one lubricant. Issuing from this model, the post processing operations separate the fluid friction from the solid friction by using a data treatment method. The last part synthesizes the presented method and also gives an outlook of the future experimentations.

2. GENERATION OF VIRTUAL TECHNICAL SURFACES

One challenge as regards the description of the surface characteristic is the allocation of a numerical value to the surface, which describes its substantial characteristics [5]. For a complete two dimensional description of the surface properties the shape of the profile must be taken into account exceeding the well-known arithmetic (R_a) and square (R_q) roughness factors or the averaged depth of roughness ($R_{z(DIN)}$). As it is shown in Figure 2 although both surfaces can have the same arithmetic mean value R_a and the same profile height, the profile characteristics such as material proportion may be different. As a result, it becomes necessary to use in addition to these values other characteristics to compare two surfaces. This happens by dint of the linear material ratio curve (Abbott curve) illustrating the increase of the material load share (M) as a function of the increasing depth of profile (R) of the rough surface. The function is divided into three sections by means of three parameters: reduced peak height (R_{pk}), reduced valley depth (R_{vk}) and core roughness depth (R_k) [6]. The core roughness depth describes the profile area which shows the highest increase of the material load share. The reduced peak height gives the shape and the number of the roughness amplitudes. Analogously, the R_{vk} parameter describes the valley section. The sum of these three parameters constitutes the maximum height of the profile (R_t). This approach gives a good idea of the contact area of the friction system but has also its limitation because it does only give it in a two dimensional sight.

For contact examination this means that the multitude of peaks only guaranties a small nominal contact area, which finally results in high contact friction and heat. The lower surface in Figure 2 has the same mean roughness but a larger valley depth section. Therefore, the nominal contact area is bigger. Also, in lubricated contacts grease pockets can occur in the valley depth section, which, for instance, allows a better drawing off heat.

To complete these investigations information on the three dimensional disposition of the asperities are necessary. L. Nowicki [7] made use of the Fourier method in order to get the asperities spectrum of the technical surface.

2.1 MATHEMATICAL DESCRIPTION

A particular demand on the mathematical description method of technical surfaces is its applicability to manufacturing processes. This requirement excludes the use of polynomials of a constant order since it changes in function of the manufacturing processes. For a given

surface area, describing lapped surfaces through polynomial method will need a higher degree than for lathe surfaces because more points are needed to characterize the first surface. To solve this task, harmonic functions are used instead of polynomials such as Discrete Fourier Transformation (DFT) that can be used in the most common manufacturing processes.

For the analysis and synthesis of technical surfaces, a two-dimensional transformation based on the DFT equations [8] is used. Two different manufactured examined surfaces have a width and a length of $500\text{ }\mu\text{m}$ (sample rate $2\text{ }\mu\text{m}$). In addition to the Abbott curve and arithmetic values, each surface is characterised through its frequency spectrum as shown on Figure 3. The examination focuses on the role played by the high quantity number of amplitudes.

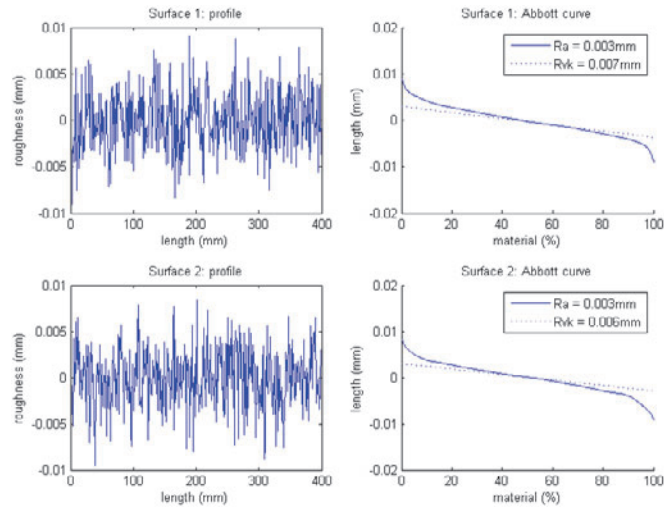


Figure 2. Stochastically generated surfaces

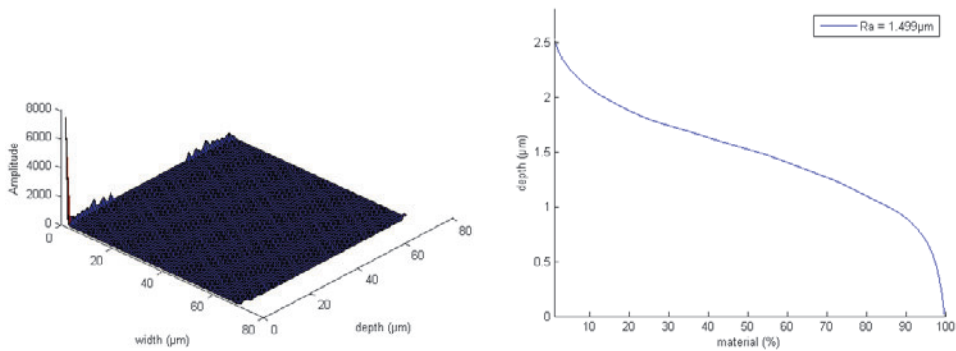


Figure 3. Frequency spectrum (left) and Abbott curve of measured turned surface (right), number of dominant amplitudes: 18.

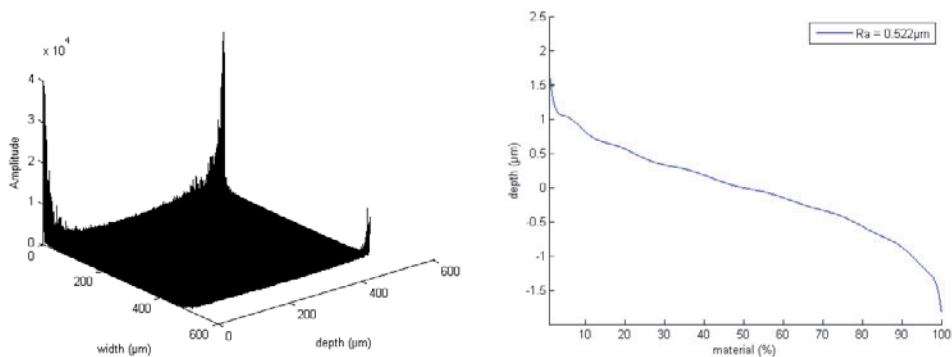


Figure 4. Frequency spectrum (left) and Abbott curve of measured grinded surface (right), number of dominant amplitudes: 36

Figure 3 shows the analysis results of a turned surface ($R_a = 1.49 \mu\text{m}$). This example reflects very well the trend of the whole examination. All surfaces can be described by a manageable amount of frequencies. All significant amplitudes were in the first five columns and rows of the Fourier matrix. Manufacturing processes with geometric definite cutting edge showed less dominant amplitudes than manufacturing processes with geometric indefinite cutting edge from Figure 4. The number of amplitudes depended a lot on the roughness value. With decreasing R_a -values, more high frequency components occurred.

2.2 PROCESS OF THE VIRTUAL ROUGH SURFACES GENERATION

To generating the technical surface with a given characteristic, an optimization algorithm based on the biological process of evolution was chosen. This algorithm was developed in the early 70's [9, 10, 11] and is nowadays established as a standard tool for optimization tasks. In comparison with the gradient that is efficient but not always able to find a global optimum, this optimization algorithm is used for problems with a huge area of design and is highly nonlinear with lots of local extrema.

As presented in Figure 5, the process needs a measured surface whose roughness is transformed into frequency signals. On the basis of these signals the frequencies are filtered and their amplitude A_i , resp. phase ϕ_i are used to create the vectors **S** representing individuals. Using the vector **A** composed of the normalized material load share M , a fitness function is calculated. Combined with the constraint function, the objective function is developed and needs to be optimized.

3. MODEL USED FOR THE NUMERICAL ANALYSIS

3.1 MACRO MODEL WITH TWO SMOOTH SURFACES

A first macro model was developed in order to check the pressure to be used in the micro model. Composed of two ideal flat surfaces, this model exemplifies an axial friction bearing (see Figure 6) in which the velocities and pressure load of conventional hydrodynamic lubrication conditions are calculated. The lower body is fixed on the ground whereas the upper is sliding on the lower with a velocity U of 4 m/s. Lubrication is assured by a mineral oil film minimum thickness of 5 μm . An artificial pressure is applied on both extremities in order to ensure an initial velocity to the fluid so that it does not leave the system.

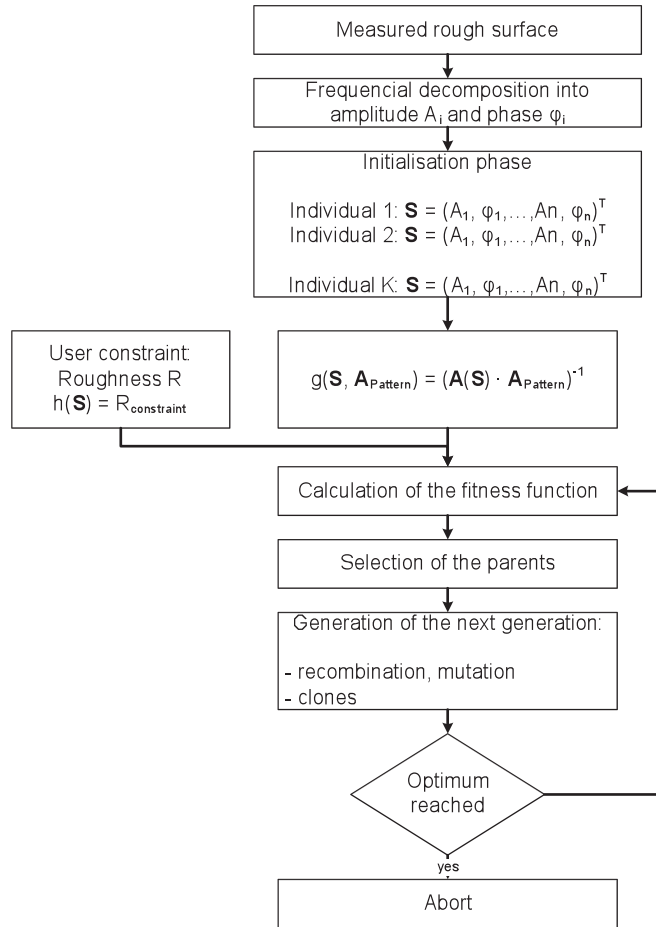


Figure 5. Virtual surface generation process

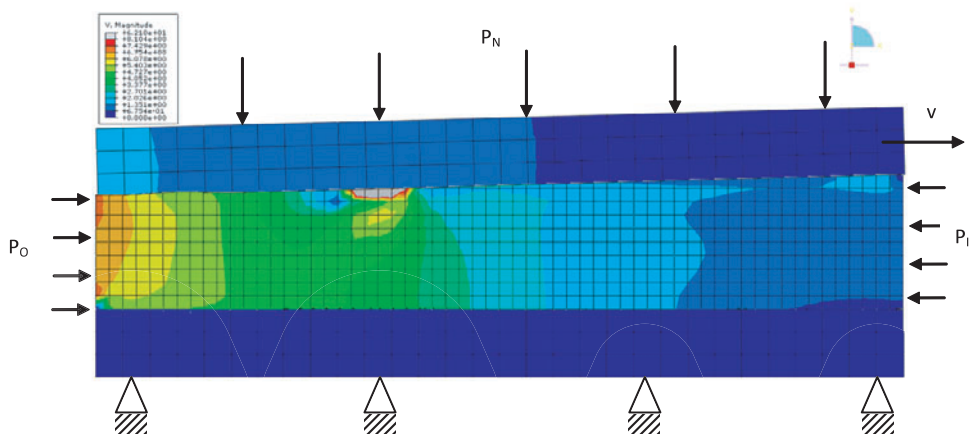


Figure 6. Axial friction bearing with velocity profile in m/s

Due to the sliding movement of the shoe the oil is pressed into the bevelled crack. This causes a rise of pressure, which prevents mechanical contact of the shoe and the sliding surface. Inside of the crack the pressure distribution is a typical hydrodynamic profile calculated with using the formula

$$p(x) = 6\eta U \int \frac{h(x) - H}{h(x)^3} dx + C \quad (1)$$

where $p(x)$ is the pressure, η the oil viscosity, $h(x)$ the oil film thickness, H and C representing constant that can be calculated using two known pressure values. The maximum reached pressure to be used in the next micro model is of 100 MPa.

3.2. MICRO MODEL CONFIGURATION

The micro model shown in Figure 7 is made of two three-dimensional rough virtual generated solid bodies that are submitted to lubricated contact conditions. Two methods are used to model the occurring phenomena. The finite element method was used to model the structural mechanics side and the finite volume method in order to model the fluid mechanics. Both methods are using a discrete mesh of the bodies.

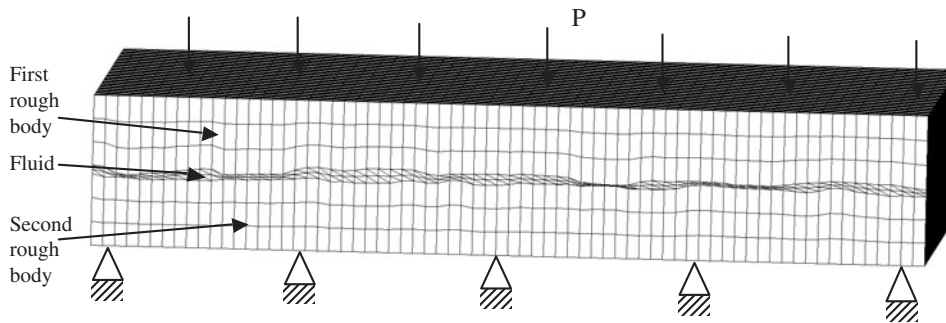


Figure 7. Micro model configuration

First step is to build the finite element model which will be solved through pre-processing phase. That concerns the importation of the geometries, the application of boundary conditions, contact definitions and material properties.

– Importation of the geometry, mesh creation

For both solids, importation is done using a virtual generation process exposed in the second part of this paper. Measured or virtual generated points are collected and sorted on both surfaces, according to their coordinates. These surfaces will also be called contact surfaces and each of them is generated according following equation

$$z(x, y) = Ax + by + Cxy + D \quad (2)$$

where A, B, C, D are constants. To achieve the body construction, it is necessary to generate five plane surfaces to generate a closed volume for both solids. Afterwards a two dimensional mesh is generated based on built surfaces and imported points that are converted into nodes supposed to be equidistant from each other. The same process is used to generate a fluid mesh whose nodes are directly coupled with both solid mesh contact nodes. Four

surrounding surfaces also need to be created to close the volume.

Once each body has a two dimensional mesh, a solid mesh with tetrahedrons is generated. Node- and element-face groups are built up in order to define all necessary boundary conditions. Concerning the material affectation, basic structural steel is used for the solid with a maximal shear stress of 900 MPa. The fluid is a conventional mineral oil with a dynamic viscosity of 0.088 Pa.s.

– Boundary conditions

The bottom nodes of the lower body (see Figure 7) are fixed on the ground in order to guarantee static determinations. On the upper body a normal load is applied in form of a displacement whereas both tangential displacements are forbidden. Figure 8 exposes the constraints applied to the fluid side.

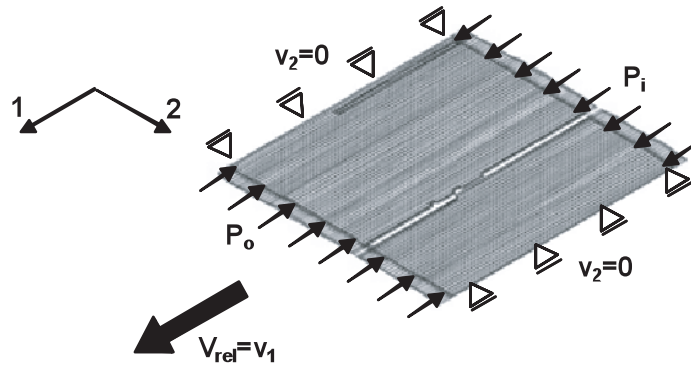


Figure 8. Preprocessing of the fluid model.

On the top of the fluid mesh a velocity v_1 of 4m/s is applied to simulate a translation of the upper body. Parallel to velocity vector two pressure loads are applied respectively 100 MPa at the inlet (P_i) and 90 MPa at the outlet (P_o). Another limitation is required in order to stay a laminar flow is the wall condition ($v_2 = 0$ m/s) on both extremity of the fluid mesh to keep the fluid inside it. In order to describe the ratios in the non-Newtonian fluid the Navier-Stokes equation is set up and solved numerically. The dependence of the viscosity on pressure and temperature is not considered in this work.

– Contact definition

Contact definition is a conventional one and concerns both rough surfaces and also the fluid mesh. Concerning the fluid solid contact, the coupling is a two way coupling which means that fluid can transmit a pressure load that can generate deformations in the solid. On the other side, the solid is also able to apply a pressure on the fluid. As a consequence, the fluid mesh that is directly fixed on the solid mesh can be deformed. Because of the large deformations due to the displacement of the upper solid an adaptive remeshing method (ALE) is used to keep an acceptable mesh quality. When the elements get a too large distortion this method remeshes the intern mesh meaning that the boundary nodes are kept. Nevertheless this method is not able to take into account mesh separation for example when both solids are very close so that no fluid can stay between them anymore. If there are two points on both solid surfaces closer than a given limiting value, the fluid mesh elements are

removed. This has to be done manually by building a second model with a new fluid mesh configuration.

For the solid-solid contact the upper body contact surface is selected as master surface meaning that it is able to penetrate into the bottom surface defined as slave surface (also called target surface). The contact definition is a frictionless contact where only the normal loads are transmitted through the penalty algorithm using an exponential increasing stiffness and based on Hertz pressure theory.

3.3. SOLVING PROCEDURE AND RESULTS

Due to the high nonlinearities the micro model is solved with an explicit solver and during the transient analysis the upper body displacement causes too many mesh distortion so that the ALE remeshing is used. On the places where solid-solid contact is initiated in state of fluid-solid contact, the mesh is manually remeshed.

Outgoing results after solving the model are illustrated in Figure 9 where three domains can be established. First of all the biggest one is the domain of elastic deformation occurred by the fluid pressure. Then, come the elastic domain of solid-solid contact and the plastic domain where irreversible deformations occur.

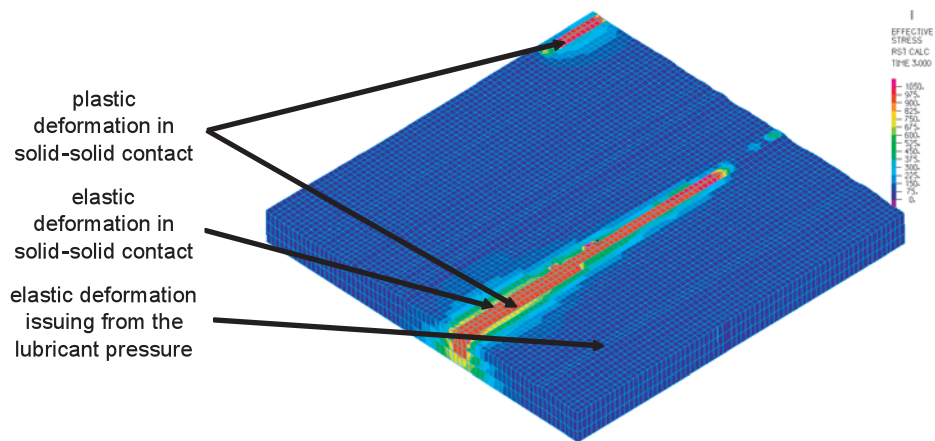


Figure 9. Pressure distribution on the lower body

Figure 10 shows the respective fluid pressure of the fluid mesh laying between both rough surfaces.

The example shows very well that the surface roughness may cause local pressure fluctuations. The mean distance between the midplanes of both solids is set to $1.3\mu\text{m}$ by the displacement constraint (see Figure 7). Pressure repartition gives a good overview on how the mixed lubricated system is behaving when the oil film thickness decreases. The pressure load is much bigger on the side where the fluid space is the smallest (right side in Figure 10). The explanation for this is a limiting volume which increases the speed as well as the pressure in case of film thickness decrease. Another important point is a significant peak in the region where both fluid regions are communicating an effect due to laminar whirlpool present at this place.

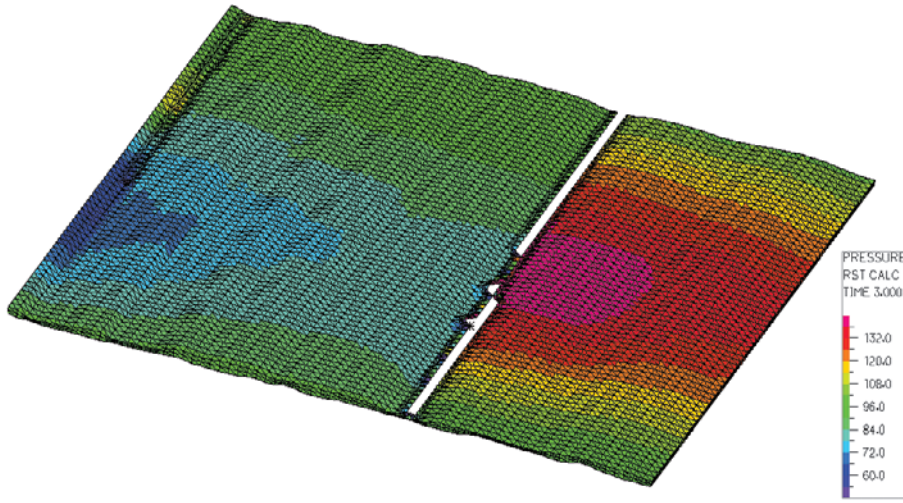


Figure 10. Pressure field in the lubricant film (MPa)

In addition to the ability to model systems where two contact types are occurring, the goal of this work is to predict and calculate the friction coefficient of such systems. Basically, the friction coefficient can be separated into two parts

$$\mu = \mu_f + \mu_s \quad (3)$$

where μ_s friction resulting from solid contact and μ_f represents the fluid friction part coming from different forces. Regarding the solid friction, predominant forces are a number of three

$$F_{pl} = \frac{\sum_i F_{N,i} \cdot s_{pl,i}}{\sum_i F_{N,i}} \quad (4)$$

for plastic friction force,

$$F_{el} = \frac{\sum_i F_{N,i} \cdot s_{el,i}}{\sum_i F_{N,i}} \quad (5)$$

for elastic friction force and

$$F_{Ad,BoT} = \tau_{crit} \cdot A_{ad,BoTa} \quad (6)$$

for Bowden and Tabor [12] adhesion forces.

The fluid friction is also issued from two forces: the adhesion forces of Hamaker

$$F_{Ad,Ham} = \frac{A_{Ad,Ham} \cdot H}{6 \cdot \pi \cdot c^3} \quad (7)$$

where $A_{Adh, Ham}$ is the Hamaker adhesion area, c a constant and

$$H = \left(\sqrt{H_{solid1}} - \sqrt{H_{Fluid}} \right) \cdot \left(\sqrt{H_{solid2}} - \sqrt{H_{Fluid}} \right) \quad (9)$$

a Hamaker constant and the hydrodynamic forces

$$F_d = \sum_i \tau_i A_i \quad (9)$$

which is the sum of each cell element of shear stress multiplied by the corresponding element area. τ is the shear stress of the fluid and A the surface of the elements present in Figure 10. The friction values are resumed into the Table 1.

Table 1. Friction coefficients

| Friction type | Friction coefficient |
|-----------------------------|----------------------|
| Adhesion (Bowden and Tabor) | 0.07748 |
| Adhesion (Hamaker) | 0.00886 |
| Hydrodynamic | 0.00063 |
| Plastic deformation | 0.00036 |
| Elastic deformation | 0.00001 |

One can conclude from Table 1 that the biggest contribution to friction comes from the adhesion effects of Hamaker and Bowden and Tabor. Hydrodynamic forces as well as elastic and plastic deformation forces are not relevant in the global friction coefficient.

4. CONCLUSION AND OUTLOOK

The presented approach gives an overview of the difficulties encountered in modelling mixed lubricated systems. Overriding the difficulty of mesh distortion is possible through the ALE method but is also limited because very large contact topology changes cannot be simulated with it, meaning that a tangential displacement is not possible to be simulated with this technique.

Also, the method tells how the decomposition of the global friction coefficient into fluid and solid friction components is established through post-processing treatments. This study allows announcing that the elastic, plastic and hydrodynamic forces are not set as prior forces to take into account for a future implementation of friction forces in models. The most relevant one are the adhesion forces of Bowden and Tabor and then the Hamaker.

Another point is the importance of the high CPU resources necessary for these investigations. Due to high nonlinearities, explicit solving is needed. In addition to the small mesh dimension of a micro scale model, the CPU time increases significantly because of the time step that are limited by the element global size.

Concerning the future work, one objective is to use the coupled Eulerian-Lagrangian method in order to model mixed lubricated systems with large contact topology changes and translational displacements as well. This method is expected to overcome the problem of mesh distortion but can be limited by the necessary CPU resources. Apart from the measure of friction forces, one other important objective consists in implementing such forces according to the Bowden and Tabor model to the used Software packages.

5. REFERENCES

- [1] ASME, Strategy for energy conservation through tribology, 1977
- [2] Hirt, C. W., An arbitrary Lagrangian-Eulerian computing technique, *Proceedings of the Second International Conference on Numerical Methods in Fluid Dynamics*, Springer, Berlin, 1971
- [3] Albers, A., Nowicki, L., Enkler, H-G., Development of a method for the analysis of mixed friction problems, *International Journal of Applied Mechanics and Engineering*, vol. 11, No. 3, pp. 479-490, 2006

- [4] Albers, A., Nowicki, L., Enkler, H-G., Methode zur Berechnung von geschmierten Friktionsproblemen in Mischreibungsbereichen, *Gesellschaft Für Tribologie*, 2006
- [5] Rometsch R. and Letzner R-D., *Rauheitsmessung, Theorie und Praxis*, Hommelwerke GmbH, Schnurr Druck, 1993
- [6] DIN EN ISO 13565, *Oberflächenbeschaffenheit: Tastschnittverfahren*, Part 1 till Part 3. Hrsg, Deutsches, Institut für Normung. Berlin, Köln, Beuth-Verlag, 1997
- [7] Nowicki, L., *Raue Oberflächen in geschmierten Tribokontakten*, PhD Thesis, Band 30, University of Karlsruhe, 2008
- [8] Sundararajan D., *The Discrete Fourier Transform*, World Scientific Publishing Co. Pte. Ltd., Singapore, New Jersey, London, Hong Kong, 2001
- [9] Kursawe F. and Schwefel H-P, Optimierung mit Evolutionären Algorithmen, *Automatisierungstechnische Praxis*, vol.39 No.9, pp.10-17, 1997
- [10] Rechenberg I., Evolutionsstrategie: Optimierung technischer Systeme nach Prinzipien der biologischen Evolution, *Frommann-Holzboog*, Stuttgart, 1973
- [11] Albers A., Nowicki L. and Minx J., A method for the consideration of influences of the surface roughness on the behaviour and design of global systems, *15th International Conference on Engineering Design ICED 05*, Melbourne, Australia, August 15th -18th, 2005 [12] Bowden, F.P., Tabor, D., Reibung und Schmierung feste Körper, *Springer*, Berlin, Göttingen, Heidelberg, 1959