Wear Modeling of non-conformal Rolling Contacts subjected to Boundary and Mixed Lubrication

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1. Introduction

The striving for friction- and wear-optimized machine elements and the associated increasing use of low-viscosity lubricants leads to a shift of operating conditions from full film lubrication to the mixed lubrication or even the boundary lubrication regime. Therefore, detailed wear simulations offer great potential for the design of machine elements: On the one hand, operating conditions with an undesirably high wear rate can be systematically avoided. On the other hand, it enables the optimization of running-in processes, which have a decisive influence on the service life of machine elements subjected to mixed lubrication or boundary lubrication.

2. Numerical Wear Modeling

Within this contribution, a general method for numerical wear modeling of machine elements operated under mixed and boundary lubrication is briefly described. The entire wear-modeling scheme is implemented using a commercial FEM software.

2.1. Mixed Lubrication Model

Wear simulation of the mixed lubrication regime, as depicted in Figure 1, is implemented by the application of an FEM-based EHL-Model according to HABCHI [1] to solve for the REYNOLDS equation:

$$\frac{\partial}{\partial x}\left(\frac{\rho(p_\mathrm{h})\cdot h^3}{12\cdot\eta(p_\mathrm{h})}\frac{\partial p_\mathrm{h}}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\rho(p_\mathrm{h})\cdot h^3}{12\cdot\eta(p_\mathrm{h})}\frac{\partial p_\mathrm{h}}{\partial y}\right)$$
$$=\frac{\partial}{\partial x}\left(\rho(p_\mathrm{h})\cdot h\cdot\frac{u_1+u_2}{2}\right)+\frac{\partial}{\partial y}\left(\rho(p_\mathrm{h})\cdot h\cdot\frac{v_1+v_2}{2}\right)$$

A statistical contact model of rough surfaces (e.g. GREENWOOD/WILLIAMSON-model [2]) is used to calculate the asperity contact pressure. Moreover, the surface topography model of SUGIMURA and KIMURA [3] is used to consider the time-dependent change of the surface height distribution function, which is in turn required as an input for the applied statistical asperity contact model. EHL simulation and the asperity contact model are coupled in order to fulfil the equilibrium of the load balance equation:

$$F=\int_{\Omega_\mathrm{c}} p_\mathrm{total}\,\mathrm{d}\Omega_\mathrm{c}=\int_{\Omega_\mathrm{c}}\left[p_\mathrm{h}+p_\mathrm{a}\right]\mathrm{d}\Omega_\mathrm{c}$$

The Profile variation is calculated by means of ARCHARD's wear model [4].

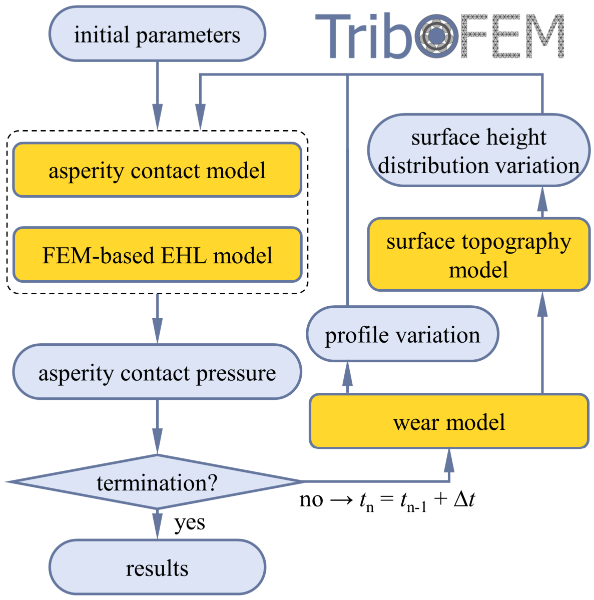

Figure 1: Wear Modeling (Mixed Lubrication)

2.2. Boundary Lubrication Model

In contrast to the first mentioned approach, wear simulation of the boundary lubrication regime relies on a FEM-based contact pressure calculation, see Figure 2. This modification was implemented since EHL simulations tends to become numerically unstable in the boundary lubrication regime.

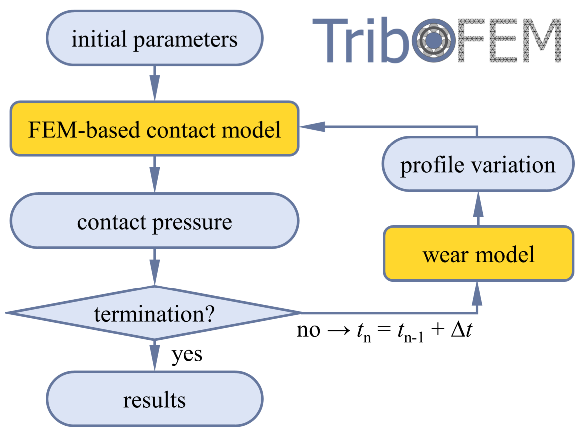

Figure 2: Wear Modeling (Boundary Lubrication)

In analogy to the FEM-based EHL model according to HABCHI [1], a substitute body is defined which possesses equivalent mechanical properties of the base and counter body. This substitute body is contacted with a rigid surface, which in turn possesses the equivalent geometry of the base body and the counter body, see Figure 3.

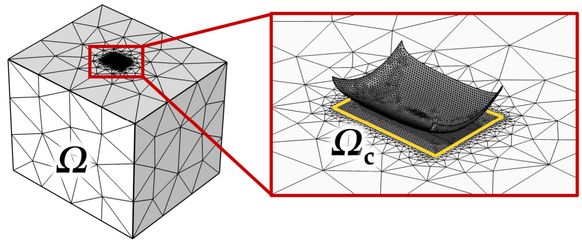

Figure 3: Contact Model (Boundary Lubrication)

The remaining simulation procedure is based on the mixed lubrication model as described in section 2.1.

3. Experimental Determination of the wear coefficient

The aim of this wear modeling approach is to utilize a universal wear coefficient that is valid for both the boundary lubrication and the mixed lubrication wear simulations. Therefore, the wear coefficient needs to be determined in the boundary lubrication regime. But since in the mixed lubrication regime only a part of the total load is carried by the asperities, the asperity contact pressure – not the total contact pressure – is used to calculate the wear volume by means of ARCHARD's wear law:

$$h_{\mathrm{wear}}(x,y) = k_{\mathrm{boundary}} \cdot s(x,y) \cdot p_{\mathrm{a}}(x,y)$$

A two-disc tribometer was chosen as the experimental setup for the determination the boundary lubricated wear coefficient, see Figure 4.

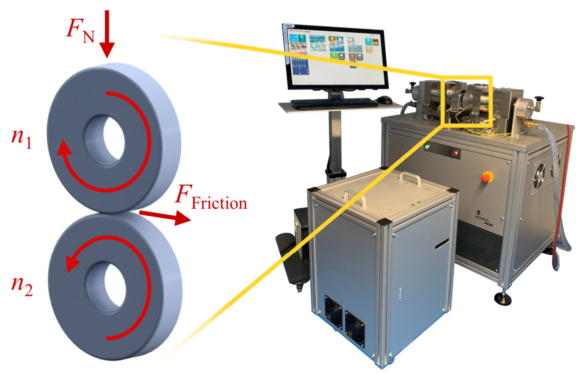

Figure 4: Two-Disc Tribometer

The material of the discs as well as their surface roughness and the lubricant ought to match the conditions of the application to be investigated. However, the geometry of the discs, the kinematics and the lubricant film thickness should be selected so as to ensure that the two-disc contact operates within the boundary lubrication regime.

On the one hand, the wear volume can be determined gravimetrically and converted via the density of the disc material:

$$V_{\mathrm{wear}} = \frac{\Delta m}{\rho}$$

On the other hand, the wear volume can also be determined by profile measurement of the worn disc. In this case, the worn cross-sectional area A_{wear} must be determined:

$$V_{\mathrm{wear}} = A_{\mathrm{wear}} \cdot \pi \cdot D_{\mathrm{disc}}$$

Finally, the wear coefficient can be calculated:

$$k_{\mathrm{boundary}} = \frac{V_{\mathrm{wear}}}{F_{\mathrm{N}} \cdot s}$$

4. Conclusion and Outlook

The presented wear modeling approach offers the possibility to calculate the surface profile evolution as well as the time-dependent change of the surface height distribution in any lubrication regime. Commercial FEM software is used to calculate contact pressures. Moreover, an experimental setup for the determination of the wear coefficient for ARCHARD's wear law was presented.

Since lubricant additives and the chemical processes at the interfaces can strongly influence the wear behavior, future research should focus to a greater extent on the influence of interface chemistry on the wear of tribological systems.

5. References

[1] HABCHI, W., "Finite Element Modeling of Elastohydrodynamic Lubrication Problems", John Wiley & Sons Incorporated, 2018.

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