KTH Industrial Engineering
and Management

Interface modeling: friction and wear

Anders Söderberg

Doctoral thesis
Department of Machine Design
Royal Institute of Technology
SE-100 44 Stockholm

TRITA – MMK 2009:03
ISSN 1400-1179
ISRN/KTH/MMK/R-09/03-SE
ISBN 978-91-7415-215-9

TRITA – MMK 2009:03
ISSN 1400-1179
ISRN/KTH/MMK/R-09/03-SE
ISBN 978-91-7415-215-9

Interface modeling: friction and wear

Anders Söderberg

Doctoral thesis

Academic thesis, which with the approval of Kungliga Tekniska Högskolan, will be presented for public review in fulfillment of the requirements for a Doctorate of Engineering in Machine Design. Public review: Kungliga Tekniska Högskolan, Flodissalen (F3), Lindstedtsvägen 26, Stockholm, on February 9, 2009, at 13.00

Abstract

The general trend toward increased use of computer models and simulations during product development calls for accurate and reliable product models. The function of many products relies on contact interfaces between interacting components. Simulating the behavior of such products requires accurate models of both components and interfaces. Depending on the purpose of the simulation, interface models of different degrees of detail are needed. In simulating very large systems with many interfaces, it might be computationally expensive to integrate detailed models of each individual interface. Condensed models, or abstractions, that describe the interface properties with the fewest degrees of freedom are therefore required.

This thesis deals with the modeling and simulation of mechanical interfaces in a systems context. The five appended papers discuss the issue from both the simulation and tribological points of view. The aim is to study how friction and wear can be modeled in the behavioral simulation of technical systems and to discuss the convenience and applicability of using different types of models as building blocks of a system model in simulations.

Paper A reviews existing friction models of sliding contacts under different running conditions.

Paper B uses a simplified contact model, the elastic foundation model, to model friction in a boundary-lubricated rolling and sliding contact. The model is integrated into a dynamic rigid body model of a mechanical system, and the system behavior is simulated.

Paper C discusses the application of the elastic foundation model to rough surface contact problems and investigates how the error in its results depends on surface roughness.

Papers D and E address how the wear of the contact surfaces at the pad-to-rotor interface in a passenger car disc brake can be simulated using finite element analysis (FEA).

Keywords: Interfaces, Models, Simulation, Friction, Wear

Preface

The research presented here was carried out between April 2004 and February 2009 in the Department of Machine Design at the Royal Institute of Technology, Stockholm, Sweden. Standing at the end of this long journey I would like to express my gratitude to all the people that have helped me and made it possible to reach this far.

First of all, I wish to thank my supervisors, Sören Andersson, Ulf Sellgren, and Stefan Björklund for giving me the chance to write this thesis and their excellent guidance throughout the research.

I would also like to thank all present and former colleagues at the department for creating a stimulating working environment. Special thanks go to my ex-colleague and co-writer Christer Spiegelberg for helping me out during my first years. I also dedicate a special thought to Jon Sundh, who has been with me for the most part of this journey, and the "newcomer" Jens Wahlström for much appreciated help and friendship. Riding this rollercoaster would have been much scarier without you by my side.

My family has always believed in me and is an important part of my life. I take this chance to thank you for taking good care of me both in Stockholm and back home. Your support has been much appreciated, especially during the latter part of writing this thesis.

My last thoughts go to Lucia. Without your endless support and patience this thesis could never have been written. Thanks for putting up with me during these five years. Sometimes it feels like you have put more effort into this than I have. I love you!

Stockholm, January 2009

Anders Söderberg

List of appended papers

This thesis consists of a summary and the following five appended papers:

Paper A

Sören Andersson, Anders Söderberg and Stefan Björklund, “Friction models for dry, boundary and mixed lubricated contacts,” *Tribology International*, Volume 40, Issue 4, April 2007, Pages 580–587

Paper B

Anders Söderberg and Christer Spiegelberg, “Modeling transient behavior of a mechanical system including a rolling and sliding contact,” Proceedings of IMECE05 2005 ASME International Mechanical Engineering Congress and Exposition, 5-11 November 2005, Orlando, Florida, USA

Paper C

Anders Söderberg and Stefan Björklund, “Validation of a simplified numerical contact model,” *Tribology International*, Volume 41, Issues 9-10, September–October 2008, Pages 926–933

Paper D

Anders Söderberg and Sören Andersson, “Simulation of wear and contact pressure distribution at the pad-to-rotor interface in a disc brake using general purpose finite element analysis software,” Proceedings of 13th Nordic Symposium on Tribology NORDTRIB 2008, 10–14 June 2008, Tampere, Finland

This paper has been selected for publication in the Nordtrib 2008 special issue of *Wear*.

Paper E

Anders Söderberg, Ulf Sellgren, and Sören Andersson, “Using finite element analysis to predict the brake pressure needed for effective rotor cleaning in disc brakes,” Proceedings of SAE 26th Annual Brake Colloquium & Exhibition, 12–15 October 2008, Grand Hyatt, San Antonio, Texas, USA

Division of work between authors

The work presented in this thesis was initiated and supervised by Professor Sören Andersson, Ulf Sellgren, and Stefan Björklund.

Paper A

Simulations were performed by Söderberg. Most of the writing was done by Andersson, but Söderberg also contributed to writing and editing the text.

Paper B

System modeling and simulations were performed by Söderberg. Spiegelberg provided the contact model that was integrated in the simulations. Experimental work and writing were equally divided between the authors.

Paper C

Work and writing were mainly performed by Söderberg. Björklund provided the reference model and supervised.

Paper D

Work and writing were mainly performed by Söderberg. Andersson supervised.

Paper E

Work and writing were mainly performed by Söderberg. Andersson and Sellgren supervised.

List of published papers not included in this thesis

Anders Söderberg and Ulf Sellgren, "Modelling strain hardening and strain rate hardening of dual phase steels in finite element analysis of energy absorbing components," Proceedings of NAFEMS World Congress 2005, 17-20 May 2005, St Julian, Malta

Anders Söderberg and Christer Spiegelberg, "Simulation models of test equipment for measuring transient friction in a rolling and sliding contact", Proceedings of OST Symposium of Mechanical Design 2005, 20-21 October 2005, Stockholm, Sweden

Sören Andersson, Anders Söderberg and Ulf Olofsson, "A random wear model for the interaction between a rough and a smooth surface", *Wear*, Volume 264, Issues 9-10, 10 April 2008, Pages 763-769

Sören Andersson and Anders Söderberg, "Modelling and simulation of an oscillating journal bearing of a rocker arm in a cam mechanism," Proceedings of 13^{th} Nordic Symposium on Tribology NORDTRIB 2008, 10–14 June 2008, Tampere, Finland

Jens Wahlström , Anders Söderberg , Lars Olander and Ulf Olofsson, "A disc brake test stand for measurement of airborne wear particles," Proceedings of 13^{th} Nordic Symposium on Tribology NORDTRIB 2008, 10–14 June 2008, Tampere, Finland

Contents

Appended papers

1 Introduction

The general trend toward increased use of computer models and simulations during product development calls for accurate and reliable product models. In many respects, modern products can be viewed as technical systems, i.e., sets of subsystems or machine elements interrelated with each other and with the whole so as to achieve a common goal. Most subsystems are assemblages of elementary machine elements, so both subsystem and machine elements will here be called "components." The function of a component, and ultimately of the whole system, often relies on physical interactions at mechanical interfaces within a component or between the component and the surrounding components or environment.

With today's computer-based modeling and simulation techniques, the behavior of single-part components can be simulated fairly accurately, but technical systems in which the components are connected by mechanical interfaces still cannot be directly modeled and simulated with reliable results. Another way to view this problem is to note that component properties, such as structural strength, can be predicted fairly well, but system responses at the interfaces, such as friction and wear, cannot. Consequently, one way to obtain more accurate and reliable simulations of complex technical systems is by developing models that describe the characteristics of mechanical interfaces under different conditions. These models should address a number of different aspects, such as contact stiffness, damping, friction, and wear.

This thesis deals with the modeling and simulation of mechanical interfaces in a systems context. It consists of this summary and five appended papers that discuss the issue from both the simulation and tribological points of view. Their common aim is to study how friction and wear can be modeled in the behavioral simulation of technical systems and to discuss the convenience and applicability of using different types of models as building blocks of a system model in simulations.

Paper **A** reviews existing friction models of sliding contacts under different running conditions.

Paper **B** uses a simplified contact model, the elastic foundation model, to model friction in a boundary-lubricated rolling and sliding contact. The model is integrated into a dynamic rigid body model of a mechanical system, and the system behavior is simulated.

Paper **C** discusses the application of the elastic foundation model to rough surface contact problems and investigates how the error in its results depends on surface roughness.

Papers **D** and **E** address how the wear of the contact surfaces at the pad-to-rotor interface in a passenger car disc brake can be simulated using finite element analysis (FEA).

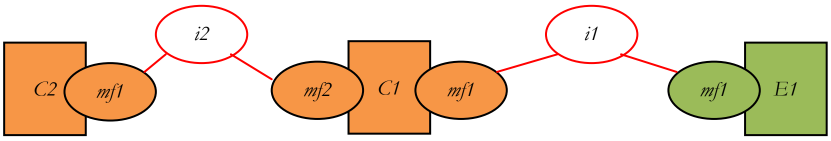
2 Mechanical interfaces in a systems context

Modern products are often technical systems, consisting of a number of different components that work together to fulfill given functions. The main functions are accompanied by undesired side effects, such as vibration, wear, friction, heat, fatigue, and crack growths. Engineering design is therefore largely concerned with maximizing intended behavior, minimizing unintended behavior, and avoiding accidental behavior [1].

The components of a technical system interact at their geometrical boundaries, referred to as mating faces. The relations between the mating faces are given by interfaces. To model the behavior of such a system, it is important to model both the components and their interactions with each other and their surroundings. A model is a simplified representation of reality that serves a well-defined purpose. From an engineering point of view, the model should always be as simple as possible but as complex as necessary. Depending on the design question addressed by the simulation, behavior models of different degrees of detail may be needed. It is therefore convenient to apply a modular model architecture dividing the system model into component models, which describe the behavior of the individual components or subsystems, and interface models, which define their interactions. The work in this thesis is inspired by the modular modeling approach proposed by Sellgren and Drogou [2] and further developed by Sellgren [3][4]. This approach divides the system model into component models with mating features that are connected by interface models. Surrounding components or subsystems that interact with the modeled system are defined as environment models and must also be taken into consideration. The structure of a modular system model can be seen in overview using an architecting tool called the model structure matrix (MSM) [4]. The MSM representation of a system model can either be condensed, showing only the components and the interfaces, or fully expanded, showing the relations between the individual mating features (Figure 1).

The modeling principle is illustrated by the link mechanism of the lifting system shown in Figure 2. The different components of the mechanism are represented by CAD models, and the interfaces are represented by symbols *i1* to *i12*. An interface can be either a mechanical contact or an interface element, as when it is a bearing or a system component such as a coupling, depending on the level of abstraction.

This thesis considers interfaces in the sense of mechanical contacts and focuses on models describing the friction and wear in these contacts. Friction and wear in mechanical contacts are influenced by many parameters, including contact surface geometry and their properties, the running conditions, and any lubricants used.



(a)

	#	1	2	3	4	5	6	7	8
Environment1	1	E							
Environment1.mf1	2		mf		i1				
Component1	3			C					
Component1.mf1	4		i1		mf				
Component1.mf2	5					Mf		i2	
Component2	6						C		
Component2.mf1	7					i2		mf	
Component2.mf2	8								Mf

	#	1	3	6
Environment1	1	E	i1	
Component1	3	i1	C	i2
Component2	6		i2	C

(b) (c)

Figure 1. A graphic representation (a), full MSM representation (b), and condensed MSM representation (c) of a model of a system consisting of two components (C1, C2) interrelated with each other and with a surrounding environment model (E1) at mating features (mf) through two interface models (i1, i2).

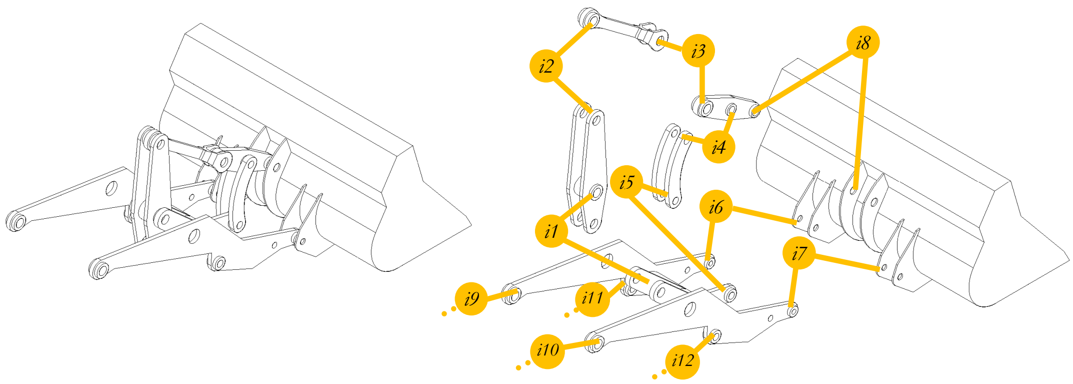


Figure 2. Mechanical components and interfaces in a model of a wheel loader lifting unit. The interfaces are indicated by circles labeled i1 to i12.

3 Physics of mechanical interfaces

To discuss models of mechanical interfaces, a basic understanding of the physics of interacting surfaces is needed. This knowledge can be found in the field of tribology: the science and technology of interacting surfaces in relative motion. It includes the study and application of the principles of friction, lubrication, and wear.

3.1 Real engineering surfaces in contact

It is hard to understand how friction and wear arises without knowing how surfaces interact when brought into contact. It is well known that real engineering surfaces are not perfectly smooth. Even highly polished surfaces possess some degree of roughness. Surface roughness has a significant effect on how loads are transmitted at the contact interface between solid bodies. Because of roughness, only parts of the surfaces are actually in contact, and the real contact area is only a fraction of the apparent contact area (Figure 3). The actual surface topography can significantly influence the global behavior of a technical system. There is, for example, a direct relation between the thermal and electromagnetic resistivity and the real contact area between the contact surfaces [5].

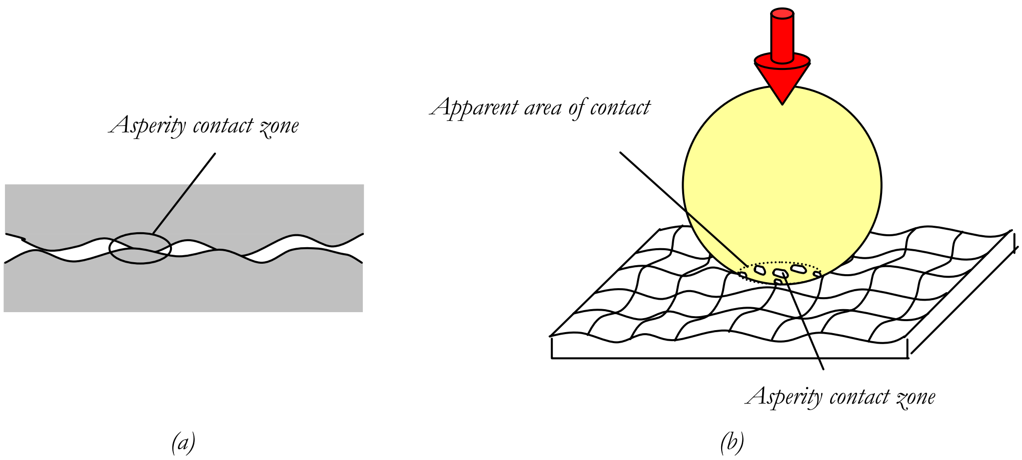


Figure 3. Contact between engineering surfaces: (a) schematic of local contact between surface asperities, and (b) the concepts of apparent and real contact area exemplified by the contact between a smooth ball and a rough plane.

3.2 Nature of friction

When two solid bodies in contact are subjected to forces that tend to produce relative sliding motion, stresses develop on the interfaces that tend to oppose that motion. The phenomenon is called friction and is often discussed in terms of the resultant of the stresses, i.e., friction force. Friction phenomena have long been of interest. The classic friction laws discussed in most textbooks on the subject can be traced back to the fifteenth-century scientist Leonardo da Vinci, although they are often attributed to Amontons and Coulomb, who published their work in the seventeenth and eighteenth centuries, respectively [6]. The classic friction laws apply to nonlubricated contacts between metallic bodies and can be summarized as follows:

- the friction force is independent of the apparent area of contact;
- the friction force is proportional to the normal contact force; and
- the friction force is independent of the sliding velocity.

The first two laws are generally observed to hold for gross motions, but the third law is known to be invalid [7]. However, for many purposes in which only limited velocity ranges are of interest, the friction can be assumed to be constant in the studied range.

The most widely accepted theory of the mechanisms of friction between metallic surfaces is that of Bowden and Tabor [8]. Their theory states that the friction in contacts between metallic bodies is mainly due to two causes: shearing of metallic joints between the surface asperities (adhesion) and the plastic deformation of the softer surface by harder asperities (abrasion).

The usual measure of sliding friction is the coefficient of friction, i.e., the ratio between the friction force and the normal contact force. Often two values of the coefficient of friction are quoted: the coefficient of static friction, which applies to the onset of sliding, and the coefficient of kinetic friction, which applies during sliding motion. In most cases, the static coefficient is known to be higher than the kinetic coefficient. Because the friction between sliding contact surfaces is a result of stochastic interactions between rubbing asperities, the friction force in nearly all friction tests is highly stochastic in nature and the coefficient of kinetic friction is often taken as a mean value.

Although not universally applicable, the classic laws of friction give a basic understanding of nonlubricated contacts. However, many mechanical interfaces in engineering applications are lubricated, and in lubricated contacts the friction is known to display velocity dependence. This behavior is often described by the curve shown in Figure 4. This curve is named after Stribeck, who first studied such phenomena in the early twentieth century [9]. The form of the Stribeck curve is often explained by defining different lubrication regimes. At low velocities, the hydrodynamic pressure buildup in the contact is negligible and the contact load is assumed to be transmitted mainly by mechanical contact between the asperities. This lubrication regime is referred to as boundary lubrication. In the mixed-lubrication regime, the contact load is divided between asperity contacts and the lubricant. Consequently, the friction in the contact is partly due to asperity interactions and partly due to the shear stresses in the lubricant. In the full-film regime, the surfaces are completely separated by the pressure buildup in the lubricant and the friction corresponds to the shear stresses in the lubricant.

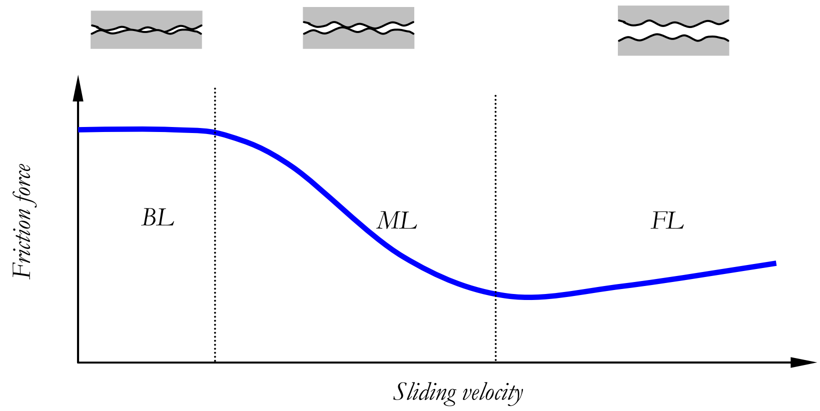


Figure 4. Schematic of the Stribeck curve and the different lubrication regimes; BL = boundary lubrication, ML = mixed lubrication, and FL = full-film lubrication.

The classic friction laws only consider bodies sliding relative to each other, but deformations and local sliding will occur in the contact interface before global sliding can be observed [10]. This phenomenon, called microslip, is believed to be caused by elastic and plastic deformation of the surfaces combined with local sliding in the contact. The contact interface can be considered to be divided into zones in which the bodies stick together and slip zones in which they slide against each other. The amount of slip in the contact increases with the tangential load until gross slip is obtained and the whole contact interface is sliding (Figure 5). Microslip has a strong influence on the contact stiffness and damping in apparently static contact interfaces, such as screw joints. Both the tangential contact stiffness and the length of the microslip zone are affected by the surface roughness [11]. Microslip can be found in both sliding and rolling contacts. Rolling contacts with microslip are often referred to as rolling and sliding contacts.

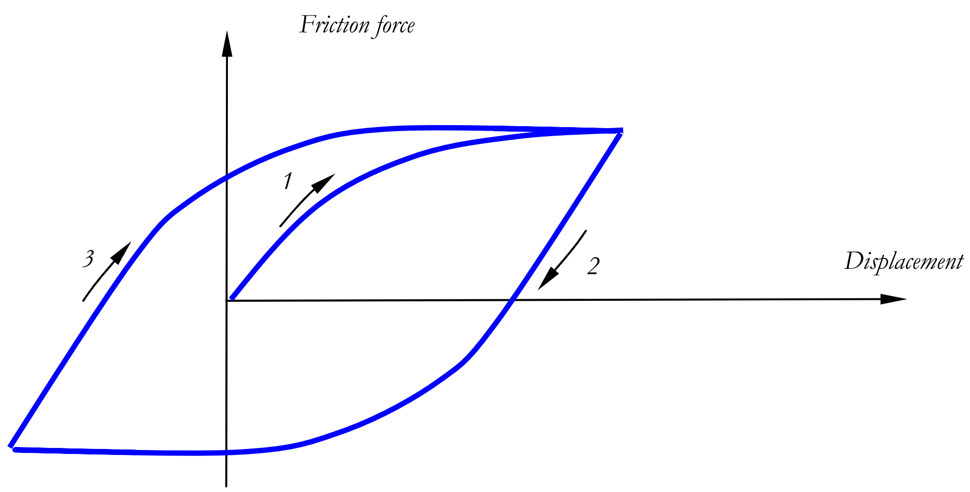


Figure 5. Schematic tangential load–displacement curve for an oscillating sliding contact with microslip.

3.3 Wear

Wear can be defined as the loss or displacement of material from a solid surface as a result of mechanical action. Material can be lost in the form of debris, whereas material can be displaced by the transfer of material from one surface to another. Wear is almost inevitable when two solid surfaces in contact move relative to each other and can appear in many ways depending on the material properties of the contact surfaces, the environment, and the running conditions. The wear rate of a surface is conventionally defined as the volume or mass lost from the surface per unit of distance slid. From an engineering point of view, wear is often classified as either mild with a low wear rate or severe with a high wear rate [12] (Figure 6). Mild wear refers to processes that produce smooth surfaces, which are often smoother than the original surfaces and display minimal plastic deformation. Mild wear is what engineers strive for, as it causes a smooth running-in of the surfaces. In the running-in process, more or less severe wear changes to mild wear by removing the surface peaks and in this way flattening the surfaces. However, sometimes severe wear with a higher wear rate may occur, resulting in rough and scored surfaces with extensive plastic deformation. Severe wear is often associated with seizure and a severe wear situation is usually unacceptable.

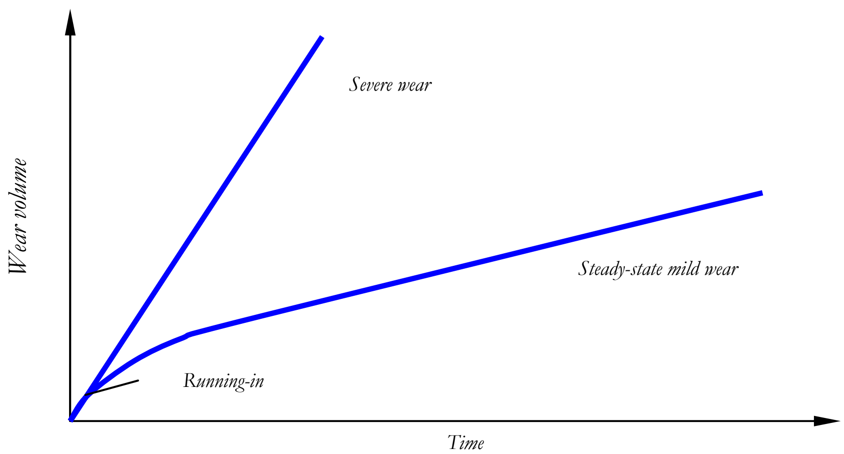


Figure 6. Wear volume versus time for a sliding contact for mild wear with running-in and severe wear.

The classification into mild and severe wear does not say much about the active wear mechanism. In many situations, there are several mechanisms operating simultaneously. Researchers therefore often tend to classify wear based on the dominant active mechanism. Some such mechanisms often mentioned in the literature are adhesive, abrasive, oxidative, delamination, fatigue, and fretting wear.

Wear involves a number of strongly interacting mechanical and thermal processes [13]. The heat produced by friction and plastic flow may lead to thermal softening of the surfaces and of the bulk material, or to changes in the oxidation rate of the surface affecting the dominant wear mechanism and the wear rate. Hence, changes in contact conditions, such as contact pressure or sliding velocity, affect the frictional heating and may lead to transition between different wear mechanisms. The transition from one dominant wear mechanism to another can be studied by constructing diagrams showing the rate and dominance regime of a number of wear mechanisms [14]. Such diagrams are often referred to as wear maps (see Figure 7).

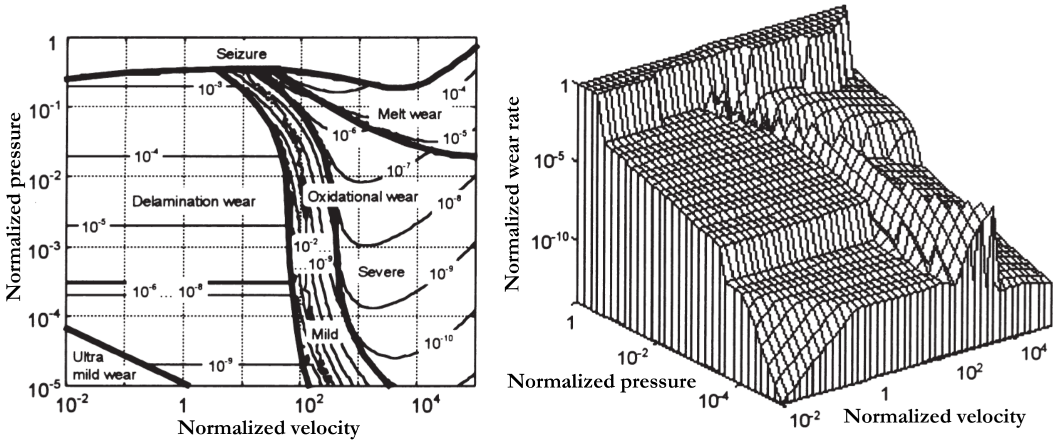


Figure 7. Wear map for the unlubricated sliding steel on steel contact from Lim and Ashby [14] and its three-dimensional plot from Podra and Andersson [39].

4 Modeling mechanical interfaces

4.1 Contact stresses and deformations

Both friction and wear are intimately related to the local stresses and deformations of the interacting surfaces. Stresses and deformations in nonlubricated and boundary-lubricated contacts can be modeled using knowledge and methods from the field of contact mechanics. There exist analytical solutions to only a very limited class of contact problems [15]. For a general contact problem, stresses and deformations in the contact interface need to be computed numerically.

In the last two decades, several surface-based numerical methods have been proposed for calculating the contact stresses and deformations in contacts between three-dimensional surfaces [16]-[20]. These methods work by replacing the continuous traction distributions on the surfaces with a discrete number of traction elements ("traction" is here used to refer to all forces acting on the surfaces, i.e., both normal pressure and tangential shear stresses). This means that only the surfaces of the interacting bodies are meshed with elements, not the entire bodies. The geometries of the bodies and the applied global displacements provide the boundary conditions. Knowing the relations between tractions and deformations for the discrete elements, the unknown tractions can be solved. In a discrete element, the tractions are approximated by a function, the deformations of which are known. This function can be either a concentrated force at the element centre or a uniform traction over the element. Some proposed methods assume frictionless contacts [16]-[17] and thus ignore tangential tractions, while others offer a full solution to the contact problem [18]-[20].

The above-mentioned, surface-based contact models are, however, limited to cases in which the half-plane approximation is valid. The finite element (FE) method, in contrast, can describe the state of objects with arbitrary shape. Today, many general-purpose FE software packages are commercially available, most of which can be used to solve general contact problems. In an FE-based contact model the interacting bodies in their entirety, not just their surfaces, are meshed. The problem is then solved for the deformations and stresses of the whole body. A general contact algorithm in FE software consists of two distinct parts. The first part is a contact search algorithm that identifies the penetration between the different bodies. Examples of various common types of contact search algorithms are master and slave algorithms [21][22] and pinball algorithms [23]. The second part of a general contact algorithm satisfies the kinematic contact condition at the interface and calculates the pressure and the tangential traction on the contact surfaces. The impenetrability constraints on the displacements are usually enforced either approximately using the penalty method [24] or more exactly using the Lagrange multiplier method [25]. Augmented Lagrangian methods, which combine the two methods, are also common [26]. For accurate results, nonlinear elastic and plastic material behavior and finite friction should be included in the model. These issues can be taken into account in the numerical solution, but the price paid is that it becomes more computationally demanding.

Regardless of the method used, a problem encountered when solving problems including contact between rough surfaces is that the elements must be small enough to allow the effects of individual surface irregularities to be included in the solution. This means that when the nominal contact area is large compared with the surface features of interest, the necessary number of elements becomes too large for a reasonable computational effort.

Most numerical contact models are computationally demanding. Using these models in wear and friction simulations where repeated contact simulations are needed results in very time-consuming simulations. Therefore, computationally efficient contact models are required, and, depending on the purpose of the simulation, simplified and less accurate models may be preferable to more accurate, but more complex, models.

One of the things that makes most numerical contact models time consuming is the fact that the deformation at one point in the contact will be influenced by the stresses and deformation at all other points. A fast, but approximate, method for calculating the pressure distribution, referred to as the *Winkler elastic foundation* model, is to ignore this mutual influence between different points in the contact. A similar approach, often called the *brush model*, can also be used for tangentially loaded contacts. The fundamentals of both methods are described by Johnson [15]. A contact model based on the above methods will be referred to in this thesis as the elastic foundation model. Because the mutual influence between surrounding points is ignored, the elastic foundation model gives only an approximate solution to the contact problem. To be considered valid for a specific contact, the parameters of the elastic foundation model must be inversely modeled, i.e., calibrated. For many nonconformal contacts between smooth surfaces, this can be done by comparing the results of the elastic foundation model with analytical solutions for known standard cases [27]. For other more complex contacts, detailed numerical contact simulations could be used as independent references. Because of its inherent nature, the results of the elastic foundation model cannot correspond to all aspects of the reference solution at the same time. Which aspect of the contact is most relevant, for example, area of contact, stiffness, or pressure, depends on the purpose of each specific simulation.

4.2 Friction modeling

Friction models are often expressed as equations describing the friction force. Most models are empirically based and must be calibrated to experimental data to be considered valid for a specific contact. While the friction force in nearly all friction tests is highly stochastic in nature, varying significantly in both amplitude and frequency, most friction models do not take this variation into account and instead represent the friction forces by a smooth mean value.

The most commonly used friction model is the Coulomb friction model, which is based on the classic laws of friction. Although it is known that a Coulomb friction model does not always properly represent the friction behavior in a contact, such models are often used to describe the friction in mechanical contacts. The model is used for nonlubricated contacts as well as boundary- and mixed-lubricated contacts. An improved model of lubricated contacts can be obtained by replacing the constant Coulomb friction force with a velocity-dependent function describing the Stribeck curve. Here this type of model is referred to as the Stribeck friction model.

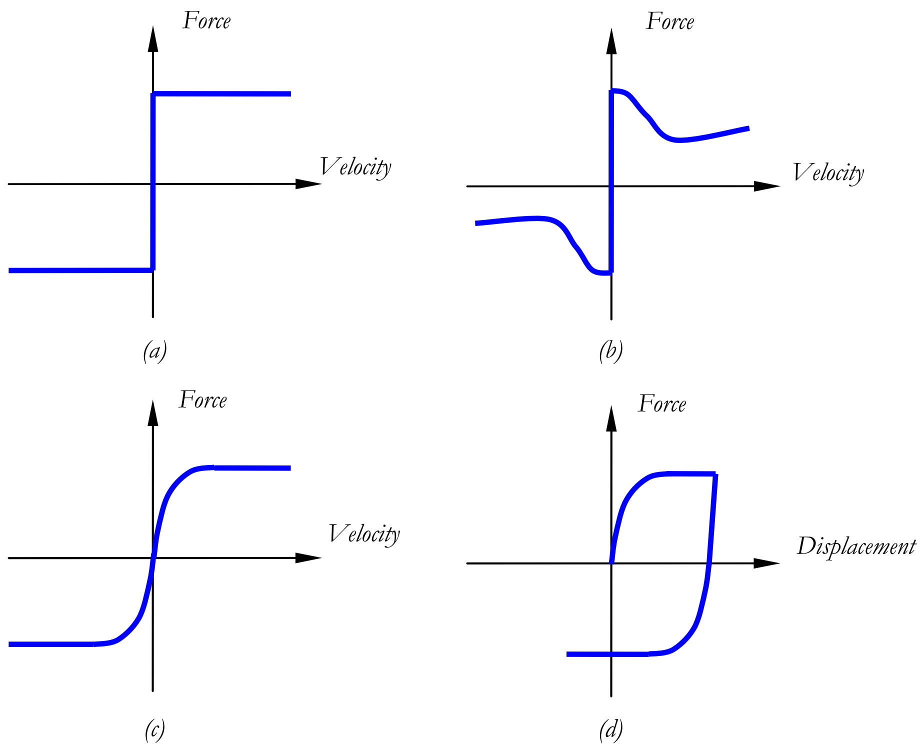


Figure 8. Examples of different friction models: (a) Coulomb friction model, (b) Stribeck friction model, (c) Coloumb frction model where a tanh function is used to model the transition from negative to positive silding velocity, and (c) Dankowicz friction model.

From a simulation point of view, the main disadvantage of the Coulomb and Stribeck friction models is the undetermined friction force at zero sliding speed. These friction models run into problems both at the start of a motion and at the point where the motion reverses. Although the behavior at these points can be modeled by a *sign* function, which represents the behavior fairly well, this representation complicates simulations and necessitates extra condition checks of the system states or interruptions of the simulations. Various tricks can be used to overcome the problem, such as replacing the *sign* function with a more appropriate continuous function, for example, the *tanh* function. These modifications improve the ability of friction models to simulate the behavior of systems, but they do not represent small displacements very well. Small displacements, or microslips, are important in many high-precision and high-control applications, so models that can handle microslip are needed.

Attempts to construct friction models that incorporate microslip are often based on the assumption that friction is a function of the contact surfaces' displacements. More than 50 years ago, Mindlin presented a theoretical model of microslip in elastic contacts between two spheres [28]. More recently, theoretical models of microslip in contacts between nominally flat surfaces have been presented by Olofsson and Hagman [29] and Björklund [30]. To incorporate the energy dissipation in oscillating contacts, most microslip models use different relations between friction and displacement for the loading and unloading of the contact. This requires extra condition checks of the system state to determine which relation to use, so these models are unsuited for implementation in dynamic simulations. Both Dahl [31] and Dankowicz [32] have proposed models that deal with the problem more conveniently. In their models, the level of microslip in the contact interfaces is modeled by a first-order differential equation. Both these models behave as linear springs in the case of small displacements, which can lead to undamped high-frequency oscillations in the friction force. This behavior suggests that there is no sliding whatsoever in the contact, just elastic deformation. Theoretically, there would exist local sliding under any tangential load, no matter how small, leading to energy dissipation and damping. Canudas de Wit et al. [33] propose a model similar to the Dahl and Dankowicz models that include a damping term coupled with the time rate of the microslip to give energy dissipation even at very small displacements. Furthermore, the model proposed by Canudas de Wit et al. incorporates Stribeck effects and thus gives the possibility of modeling contacts running under different conditions.

4.3 Wear modeling and simulation

Wear of the contact surfaces changes their topography and can therefore drastically change the contact conditions and the behavior of the interface. Modeling and predicting the type and amount of wear in an interface is generally considered difficult, although many wear models can be found in the literature [34][35]. Most models are empirical and only valid for a specific application under certain running conditions. Even though wear prediction is generally considered difficult, much effort has gone into trying to simulate how wear develops over a surface with time. The basic modeling principle often followed considers wear on a local scale by following particular points on the contact surfaces and finding a relation between the wear depth h at a particular point and the distance s that the point slides against the interacting surface. Wear is analyzed as a dynamic process, and the prediction of the process is an initial value problem. In such cases, a wear model of the form

$$\frac{dh}{dt} = f(\text{Load, Velocity, Temp., Material}, \ldots) \tag{1}$$

has been shown to be appropriate [36]-[43]. The most commonly used model for predicting wear is the linear wear law proposed by Archard [44]. A common generalization of Archard's wear law is based on the assumption that the wear rate at any point on the contact surface is proportional to the local contact pressure p and the relative sliding velocity v according to

$$\frac{dh}{dt} = k \quad p \quad v \tag{2}$$

where k is contact-state dependent wear coefficient. Although it is possible to implement a more complex wear model, most wear simulations presented in the literature assume a wear law based on equation (2) and different types of wear mechanism are simulated using different values of the wear coefficient. Transitions between different wear mechanisms could be simulated by evaluating wear maps and state variables such as local sliding velocity, contact pressure, and temperature to determine which wear model to use in each time step. The wear process is simulated by the numerical integration of equation (2) using a finite difference method. This means that knowledge of the local contact pressure and sliding velocity is needed to compute the wear rate at each integration step. The sliding velocity can be derived from the motions of the interacting surfaces, but the contact pressure needs some more attention.

Because there exist analytical solutions to only a very limited class of contact problems, and because the wear process alters the contact surface, the numerical contact model is often used to find the contact pressure distribution. This makes wear simulations computationally demanding and time consuming, since repeated contact solutions are needed. Depending on the type of interface studied, researchers tend to use different types of numerical contact models. For example, Spiegelberg and Andersson [36] used a three-dimensional elastic foundation model to simulate the wear in the rolling and sliding contact between the valve bridge and rocker arm pad in the cam mechanism in a diesel engine, whereas Hugnell et al. [37] simulated wear in cam-follower contact using the numerical contact model developed by Björklund and Andersson [38].

Today, many general-purpose FE software packages are commercially available, most of which can be used to solve general contact problems. With this in mind, Podra and Andersson [39] proposed a wear simulation procedure in which general-purpose finite element software is used to obtain the contact solution. In this procedure, wear is considered a quasi-static process and treated in a post-processing step of the finite element analysis. The proposed procure loops through a series of static simulation steps with the FE model, each with an updated contact surface geometry. The procedure is fairly straightforward as long as load history-dependent behavior is excluded from the FE model and the model can be modified between the simulation steps. The use of general-purpose finite element software gives the possibility of using coupled thermo–mechanical analysis to include the effects of frictional heating. This will, however, lead to a load history-dependent solution, since the temperature field will develop during the simulation, and at least the temperature profile must be mapped onto the new geometries after the surface geometries have been updated [40]. Sui et al. [41] also included a re-meshing scheme in which the stress–strain state of the old mesh is mapped onto the new mesh to incorporate load history-dependent material behavior in the wear simulation. Most FE-based wear simulations reported in the literature use two-dimensional geometries and only consider the wear of one of the surfaces in the contact pair, but both Hegadekatte et al. [42] and Kim et al. [43] have included a three-dimensional geometry and simulated the wear of both surfaces in the contact pair.

5 Summary of appended papers

This thesis comprises five appended papers (Appendices **A**-**E**) that discuss different aspects of interface modeling. The focus of all papers is on models and simulation techniques and three main topics are discussed as summarized below.

5.1 Friction models of sliding contacts

Paper **A** reviews existing friction models of sliding contacts running under different contact conditions. The purpose is not to give a complete review of all existing friction models, there are simply too many, but rather to give insight into the possibilities available. The models are discussed from both the simulation and tribological points of view.

5.2 Elastic foundation models

Papers **B** and **C** treat three-dimensional elastic foundation models, giving a deeper understanding of how these models can be used and of their limitations.

Paper **B** gives an example of how an elastic foundation model can be integrated into a rigid body model of the dynamic behavior of a mechanical system to improve the accuracy of behavioral simulation. The mechanical system that is modeled and simulated is a piece of testing equipment for studying a rolling and sliding contact between two discs. The results of the simulations are compared with experimental observations. Simulations are made assuming smooth as well as rough contact surfaces. If the contact surfaces are modeled as smooth, the tangential stiffness of the contact model must be set to a lower value than that suggested in the literature to obtain model output correlated with experimental results. The simulations also indicate that the roughness of the contact surface may disturb the system behavior.

Paper **C** discusses the application of elastic foundation models to rough surface contact problems. The objective is to establish the consequences of the simplifications made in formulating the models. This is done by comparing the results of the simplified models with those of a more accurate contact model. The contact between a smooth sphere and a rough plane is investigated, in particular, how surface roughness influences the errors in the elastic foundation solution in terms of predicted pressure distribution, real contact area, and normal and tangential contact stiffness. The results presented can be used to estimate the extent of error in the elastic foundation model, depending on the degree of surface roughness. The main conclusion is that the elastic foundation model is not accurate enough to give a correct prediction of the actual contact stresses and contact areas, though it might be good enough for simulations where contact stiffness are of interest.

5.3 Wear simulations using general-purpose finite element software

The two final papers address how general-purpose finite element software can be used to simulate wear at the pad-to-rotor interface in passenger car disc brakes. Both papers focus on the modeling principles and the wear simulation algorithm.

In paper **D**, a three-dimensional finite element model of the piston, brake pad, and rotor is developed to calculate the pressure distribution in the pad-to-rotor contact. The model is structured following the modular modeling approach discussed earlier. A wear simulation procedure based on a generalized form of Archard's wear law and the Euler integration scheme is adopted to simulate wear of the brake pad. The usefulness of the numerical model is demonstrated by simulating running-in wear under steady-state drag conditions. It is also shown that the frictional traction at the interface affects the pressure distribution and, consequently, the wear process as well. In paper **E**, the model is further developed to include the wear of both surfaces in the contact pair.

6 Conclusions and discussion

The function of many technical systems relies on contact interfaces between interacting components. Simulating the behavior of such systems requires accurate models of both components and interfaces. This thesis deals with the modeling and simulation of friction and wear in mechanical interfaces. The focus has been on models and simulation techniques, and the five appended papers treat friction models of sliding contacts, elastic foundation models, and FE-based wear simulations.

A review of existing friction models that can be implemented in behavioral simulations of technical systems has been presented (paper **A**). The review leads to the conclusion that, although the Coulomb friction model is usually used in simulation, it does not always properly represent the friction behavior in the contact. Other, more appropriate models have been proposed for sliding contacts under different running conditions. The friction in lubricated contacts running in different lubrication regimes can be modeled by a Stribeck model, and microslip can be modeled by formulating the friction model as a differential equation.

Detailed contact models have the advantage of giving more information about contact conditions, but are often complex and computationally demanding. Integrating detailed contact models in dynamic system simulation leads to time-consuming simulations, so fast but less accurate models may sometimes be preferred to accurate and complex models. The elastic foundation model is an example of a simplified contact model that can be used for this purpose. The elastic foundation model incorporates many major simplifications, but it can be evaluated quickly and therefore has the potential to be used in simulations where repeated contact evaluations are needed. An example of this is given in paper **B**, where the elastic foundation model is integrated into a rigid body model of a technical system, used to simulate the dynamic behavior of the system. Because of the simplifications made in the elastic foundation model, the applicability of the model to the numerical analysis of rough surfaces can be questioned. The results presented in paper **C** indicate that the model is useful for predicting the contact stiffness but not for predicting the detailed contact state. This means that the elastic foundation model might not be accurate enough to give a correct prediction of the actual contact stresses and deformations, though it might be good enough for system dynamics simulations where contact stiffness and damping are crucial.

The last two papers, **D** and **E**, address FE-based wear simulation. The developed simulation routines allow the simulation of wear on both contact surfaces at the pad–rotor interface. An FE model with three-dimensional geometry is used in the simulation, and the model could easily be modified and used to simulate wear in other technical systems with similar geometries, such as couplings or pin-on-disc machines.

The friction and wear models discussed in this thesis indicate that all models are empirical and need to be calibrated to experimental data before they can be considered valid for a specific contact. It has also been pointed out that both friction and wear are system phenomena that depend on many different parameters, such as the geometry of the interacting surfaces, material parameters, environmental conditions, and running conditions. With this in mind, it can be argued that it is impossible to predict the effects of friction and wear in a technical system a priori, and that such simulations serve no purpose in the design process. However, one must not forget that a major task in the design process is to design robust products the performance of which is insensitive to unpredictable phenomena, such as friction and wear, so such models and simulation techniques may be valuable tools in this process.

7 References

[1] Whitney, D.E., Why mechanical design cannot be like VLSI design, Research in Engineering Design (1996) 8 125-128

[2] Sellgren, U. and Drogou, R., Behavior modeling in mechanical engineering: a modular approach, Engineering with Computers 14 (1998) 185–196

[3] Sellgren, U., Simulation driven design: motives, means and opportunities, Doctoral thesis, Dept. of Machine Design, Royal Institute of Technology, Stockholm, 1999

[4] Sellgren, U., Architecting models of technical systems for non-routine simulations, Proceedings of ICED 03 – 14th International Conference on Engineering Design, 19–21 August 2003, Stockholm

[5] Thomas, T.R., Rough surfaces, second edition, Imperial College Press, London, 1999

[6] van Beek, A., Machine lifetime performance and reliability, TU Delft, Delft, 2004

[7] Oden J.T. and Martins J.A.C., Models and computational methods for dynamic friction phenomena, Computer methods in applied mechanics and engineering 52 (1985) 527–634

[8] Bowden F.P. and Tabor D., The friction and lubrication of solids, Clarendon Press, Oxford, 1950

[9] Stribeck R., Die Wesentlichen eigenschaften der gleit- und rollenlager, Zeitschrift des Vereins Deutscher Ingenieure 46 (1902)1341-1348 1432-1438 1463-1470

[10] Hagman L.A., Measurement and modeling of microslip for engineering surfaces in contact, Doctoral thesis, Dept. of Machine Design, KTH, Stockholm, 1997

[11] Olsson H., Control systems with friction. Doctoral thesis; Department. of Automatic Control, Lund Institute of Technology, Lund, 1996

[12] Archard J.F. and Hirst W, The wear of metals under unlubricated conditions, Proceedings of the royal society of London Series A 236 (1956) 397–410

[13] Bhushan, B., Introduction to tribology, John Wiley & Sons, New York, 2002

[14] Lim S.C, and Ashby M.F., Wear-mechanism maps, Acta metal 35 (1987) 1–24

[15] Johnson K.L., Contact Mechanics, Cambridge University Press, Cambridge, 1985

[16] Liang, X. and Linqing, Z., A numerical model for the elastic contact of three-dimensional real rough surfaces, Wear 148 (1991) 91–100

[17] Ren, N. and Lee, S.C., Contact simulation of three-dimensional rough surfaces using moving grid method, Journal of Tribology 115 (1993) 597–601

[18] Kalker, J.J., Three-dimensional elastic bodies in rolling contact, Kluwer Academic Publishers, Dordrecht, 1990

[19] Jaeger, J., Elastic impact with friction, Doctoral Thesis, University of Delft, Delft, The Netherlands, 1992

[20] Björklund, S. and Andersson, S., A numerical method for real elastic contacts subjected to normal and tangential loading, Wear 179 (1994) 117–122.

[21] Hallquist,, J.O. Goudreau, G.L and Benson, D.J., Sliding interfaces with contact-impact in large-scale Lagrangian computations, Computer Methods in Applied Mechanics and Engineering 51 (1985) 107-137

[22] Hallquist, J.O. and Benson D.J., A single surface contact algorithm for the post-buckling analysis of shell structures, Computer Methods in Applied Mechanics and Engineering 78 (1990) 141-163

[23] Belytschko, T. and Neal, M.O., Contact-impact by the pinball algorithm with penalty and Lagrangian methods, Int. J. Numer. Methods Engrg, 31 (1991) 547–572

[24] Oden, J.T. and Pires, E.B., Non local and non linear friction laws and variational principles for contact problems in elasticity, J. Appl. Mech. 50 (1983) 67–76

[25] Brathe, K.J. and Chaudhary, A. A solution for planar and axisymmetric contact problems, Int. J. Numer. Methods Engrg. 21 (1985) 65–88

[26] Simon, J.C and Laursen, T.A., An augmented Lagrangian treatment of contact problems involving friction, Computers & Structures 42 (1992) 97-116

[27] Põdra P. and Andersson S., Wear simulations with the Winkler surface model, Wear, 207 (1997) 79–85

[28] Mindlin R.D., Compliance of elastic bodies in contact, Transactions of the ASME, Journal of Applied Mechanics, 16 (1949) 259–268

[29] Olofsson U. and Hagman L.A., A model for micro-slip between flat surfaces based on deformation of elliptical elastic bodies, Tribology International 30 (1997) 599–603

[30] Björklund S., A random model for micro-slip between nominally flat surfaces, Transactions of the ASME, Journal of Tribology 119 (1997) 726–732

[31] Dahl P.R., Measurement of solid friction parameters of ball bearings, Proceedings of the 6th Annual Symposium on Incremental Motion, Control System and Devices, 24–27 May 1977, University of Illinois, Urban-Champaign, Illinois, USA.

[32] Dankowicz H., Modelling of dynamic friction phenomena, Zeitschrift für angewandte Mathematik und Mechanik, 79 (1999) 399–409

[33] Canudas de Wit C., Olsson H., Åström K.J., and Lischinsky P., A new model for control of systems with friction, IEE Transactions on Automatic Control 40 (1995) 419–425

[34] Meng H.C. and Ludema K.C., Wear models and predictive equations: their form and content, Wear 181–183 (1995) 443–457

[35] Hsu, S.M., Shen, M.C, and Ruff, A.W., Wear prediction for metals, Tribology International 30 (1997) 377–383

[36] Spiegelberg, C. and Andersson, S., Simulation of friction and wear in the contact between the valve bridge and rocker arm pad in a cam mechanism, Wear 261 (2006) 58–67

[37] Hugnell, A., Björklund, S. and Andersson S., Simulation of mild wear in a cam follower contact with rotation, Wear 199 (1996) 202–210

[38] Björklund, S. and Andersson, S., A numerical method for real elastic contacts subjected to normal and tangential loading, Wear 197 (1994) 117–112

[39] Podra, P. and Andersson S., Simulating sliding wear with finite element method, Tribology International 32 (1999) 71–81

[40] Gonzales, C., Martín, A., Llorca, J., Garrido, M.A., Gómez, M.T. Rico, A. and Rodríguez, Numerical analysis of pin on disc tests on Al–Li/SiC composites, Wear, 259 (2005) 609–612

[41] Sui, H., Pohl, H., Schomburg, U., Upper, G. and Heine, S., Wear and friction of PTFE seals, Wear 224 (1999) 175–182

[42] Hagedekatte, V., Huber, N., and Kraft, O., Finite element based simulation of dry sliding wear, Modelling Simul. Mater. Sci. Eng. 13 (2005) 57–75

[43] Kim, N.H., Won, D., Burris, D., Holtkamp, B. Gessel, G.R, Swanson, P. and Sawyer, W.G., Finite element analysis of metal/metal wear in oscillatory contacts, Wear 258 (2005) 1787–1793

[44] Archard, J.F, Contact and rubbing of flat surfaces, J. Appl. Phys. 24 (1953) 981–988