Linköping Studies in Science and Technology.
Licentiate Thesis No. 1603

Simulation of Thermal Stresses in a Brake Disc

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LIU–TEK–LIC–2013:37
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Linköping, May 2013
Preface

First of all I would like to express my profound gratitude to my supervisor, Niclas Strömberg, for his support and guidance. I am thankful to all the colleague at JTH for a nice working environment. A special thanks to my former colleague, Magnus Hofwing, for providing the relevant data to the project. I would also like to thank Martin Tapanov for his helpful suggestions for solving Latex related issues and recommending very useful softwares.

I am very grateful for the funding by Vinnova and Volvo 3P. I would also like to express my gratitude to the people at Volvo 3P especially Magnus Levinsson and Per Hasselberg for providing the relevant data and fruitful discussions. Another thanks to Erik Holmberg at Linköping University for providing the Latex template for this thesis.

Finally, I would like to thank my family for their support and patience.

Asim Rashid
Jönköping, 2013-05-20
Abstract

In this thesis thermal stresses in a brake disc during a braking operation are simulated. The simulations are performed by using a sequential approach where the temperature history generated during a frictional heat analysis is used as an input for the stress analysis. The frictional heat analysis is based on the Eulerian method, which requires significantly lower computational time as compared to the Lagrangian approach. The stress analysis is performed using a temperature dependent material model both with isotropic and kinematic hardening behaviors. The results predict the presence of residual tensile stresses in circumferential direction for both hardening behaviors. These residual stresses may cause initiation of radial cracks on the disc surface after a few braking cycles. For repeated braking an approximately stable stress-strain loop is obtained already after the first cycle for the linear kinematic hardening model. So, if the fatigue life data for the disc material is known, its fatigue life can be assessed. These results are in agreement with experimental observations available in the literature.

The simulation results predict one hot band in the middle of the disc for a pad with no wear history. It is also shown that convex bending of the pad is the major cause of the contact pressure concentration in middle of the pad which results in the appearance of a hot band on the disc surface. The results also show that due to wear of the pad, different distributions of temperature on the disc surface are obtained for each new brake cycle and after a few braking cycles, two hot bands appear on the disc surface.

This sequential approach has proved tremendously cheap in terms of computational time so it gives the freedom to perform multi-objective optimization studies. Preliminary results of such a study are also presented where the mass of the back plate, the brake energy and the maximum temperature generated on the disc surface during hard braking are optimized. The results indicate that a brake pad with lowest possible stiffness will result in an optimized solution with regards to all three objectives. Another interesting result is the trend of decrease in maximum temperature with an increase in back plate thickness.

Finally an overview of disc brakes and related phenomena is presented as a literature review.
List of Papers

This thesis is based on the following five papers:

I. An Efficient Sequential Approach for Simulation of Thermal Stresses in Disc Brakes

II. Sequential Simulation of Thermal Stresses in Disc Brakes for Repeated Braking

III. Thermomechanical Simulation of Wear and Hot Bands in a Disc Brake by adopting an Eulerian approach

IV. Multi-Objective Optimization of a Disc Brake System by using SPEA2 and RBFN

V. Overview of Disc Brakes and Related Phenomena - a literature review

Papers have been reformatted to fit the layout of the thesis.
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Disc brakes are an important component of a vehicle retardation system. They are used to stop or adjust the speed of a vehicle with changing road and traffic conditions. During braking, a set of pads is pressed against a rotating disc and, due to friction, heat is generated at the disc-pad interface, which causes the disc surface temperature to rise in a short period of time. This heat ultimately transfers to the vehicle and the environment, and the disc cools down.

As a result of higher temperatures, in addition to local changes of the contact surfaces, there are global deformations occurring in the disc and the pad. Due to different geometries of discs, each disc has different geometrical constraints to the thermal expansion. So the deformations can appear in different forms in different discs. Some of the most commonly observed thermal deformations are coning and buckling [1, 2, 3, 4, 5]. Such geometrical deviations could be avoided or reduced if thermal loading and disc geometry are symmetric about the midplane of the disc [6, 7], and the friction ring is decoupled from the mounting bell so that it has relatively more freedom of expansion in radial direction [8]. This is usually intended to achieve by using a so-called composite brake disc in which mounting bell and friction ring are separated from each other. Such a composite brake disc is shown in figure 1.

In addition to these deformations, macrocracks might also appear on a disc surface in the radial direction after some brake cycles, affecting the performance and life of a brake disc [10, 11]. It has been shown in many previous works, e.g. [4, 12, 13], that during hard braking, high compressive stresses are generated in the circumferential direction on the disc surface which cause plastic yielding. But when the disc cools down, these compressive stresses transform to tensile stresses. For repeated braking when this kind of stress-strain history is repeated, stress cycles with high amplitudes are developed which might generate low cycle fatigue cracks after a few braking cycles. Dufrénay and Weichert [4], confirmed the existence of residual tensile stresses on the disc surface by measuring with the hole drilling strain gage method. In the present work investigation of the stresses which cause these cracks on a disc surface, by using finite element simulations, is a major focus.
CHAPTER 1. INTRODUCTION

Figure 1: Simplified representation of a composite brake disc showing an integrally casted mounting bell with a friction ring [9].

Many researchers have used the finite element analysis (FEA) techniques to predict the thermomechanical behavior of disc brakes. To simplify the development of a FEA model for a solid disc (as compared to a ventilated disc) it is often assumed that pad is smeared over the entire $360^\circ$, implying that the disc-pad system can be considered axisymmetric, see e.g. [14, 15]. In this simplified model circumferential variation of temperature and contact pressure cannot be predicted. Another approach to simplify the model for a ventilated disc is to consider only a small sector of a disc by taking the rotational symmetry into account, see e.g. [3, 4]. Again the assumption has to be made about the smearing of the pad, implying that circumferential variations of temperature and contact pressure cannot be predicted satisfactorily. It has been shown in the previous works [16, 17] that a pad also undergoes thermal deformation, called convex bending, furthermore temperature distribution is not constant along the circumference of a disc [18]. So, it is clear that these approaches are not sufficient to model the real behavior, instead a FEA model with complete three dimensional (3D) geometries of a disc and pads is required. Some researchers, see e.g. [1, 12], have used complete 3D geometries to determine the thermomechanical behavior of disc brakes realistically.

Today, the prevalent way to simulate frictional heating of disc brakes in commercial softwares is to use the fully coupled Lagrangian approach in which the finite element mesh of a disc rotates relative to a brake pad and, thermal and mechanical analysis are performed simultaneously. Although this approach works well, it is not feasible due to extremely long computational times. Particularly, for simulating repeated braking, this approach is of little importance for practical use. As a brake disc could be considered a solid of revolution, partially or fully, which makes it possible to model it using an Eulerian approach, in which the finite element mesh of the
1.1. BACKGROUND

The simulations performed within this work are by using a sequential approach where temperature history from the frictional heat analysis is used as an input in a coupled stress analysis. The frictional heat analysis, based on the Eulerian method, is performed in an in-house software developed by Strömberg, which is described in his earlier works [19, 20]. In this Eulerian approach the contact pressure is not constant, but varies at each time step taking into account the thermomechanical deformations of the disc and the pad. This updated contact pressure information is used to compute heat generation and flow to the contacting bodies at each time step. In such manner, the nodal temperatures are updated accurately and their history is recorded at each time step. Later stress analysis is performed in the commercial software Abaqus, which uses thermal history from the frictional heat analysis as an input. Figure 3 shows the workflow of this sequential approach schematically. Stresses due to the applied normal brake force, centrifugal forces and deceleration forces are insignificant in comparison to the thermal stresses [21] so only thermal stresses are considered in this work.

The results show that during hard braking high compressive stresses are generated on the disc surface in the circumferential direction which cause yielding. But when the disc cools down, these compressive stresses transform to tensile residual stresses. For repeated braking an approximately stable stress-strain loop is obtained. So, if the fatigue life data for the disc material is known, its fatigue life can be assessed. It is also shown that convex bending of the pad due to thermal deformations is the major cause of contact pressure concentration and hence appearance of hot bands. The results show that when wear is considered, different distributions of temperature on the disc surface are obtained for each new brake cycle. After a few
braking cycles two hot bands appear on the disc surface instead of only one. These results are in agreement with experimental observations. The sequential approach requires significantly lower computational time as compared to the Lagrangian approach which makes it possible to perform multi-objective optimization studies. Preliminary results of such a study are also presented in this work.

1.2 Governing equations

The governing equations for the frictional heat analysis, and stress analysis while employing two different material hardening models will be described here.

1.2.1 Heat transfer analysis

Frictional heat power generated at the contact interface of a disc-pad system can be expressed as

\[ q_{\text{gen}} = \mu p_n \omega r, \]  

(1)

where \( \mu \) is the coefficient of friction, \( r \) is the distance of a contact pair from the center of the disc, \( p_n \) is the normal component of the contact traction vector, and \( \omega \) is the angular velocity of the disc. The frictional heat generated at the contact interface flows into the disc and the pad. Heat conduction for each body is governed by the classical heat equation

\[ \rho c \frac{\partial T}{\partial t} = k \sum_{i=1}^{3} \frac{\partial^2 T}{\partial x_i^2}, \]  

(2)
1.2. GOVERNING EQUATIONS

where $\rho$ is the density, $c$ is the specific heat capacity and $k$ is the thermal conductivity.

1.2.2 Stress analysis

Stress-strain relations used to describe deformation of a material are different for the elastic and plastic domain. Consequently, it is important to know if the stress state is in the elastic or plastic domain. For this purpose a yield criterion is used to suggest the limit of elasticity and the initiation of yielding in a material under any combination of stresses. There are several yield criterion used in practice. Some of these are: the maximum shear stress criterion, the maximum principal stress criterion and the von Mises stress criterion. These criteria could be expressed in terms of material constants obtained from different physical tests e.g. a shear or a uniaxial tensile test. In this work these material parameters are obtained by considering uniaxial tests at different temperatures.

According to the von Mises stress criterion, yielding depends on the deviatoric stress and not the hydrostatic stress. It is expressed as

\[ \sqrt{3J_2} - \sigma_y = 0 \]  
\[ \sqrt{3J_2} - \sigma_y < 0 \]

for plastic deformation

for elastic deformation

where $\sigma_y$ is the stress at yield in a uniaxial test and $J_2$ is the second invariant of the deviatoric stress, i.e.

\[ J_2 = \frac{1}{2} s : s , \]

where $s$ is the deviatoric stress, given by

\[ s = \sigma - \frac{\text{tr}(\sigma)}{3} I . \]

The von Mises yield criterion appears as a cylindrical surface in the principal stress space as shown in figure 4. A loading case where stresses lie inside this surface is said to be an elastic loading. The yield surface can be described as a boundary between elastic and plastic deformation regions.

During plastic deformations, subsequent yield surface can translate, expand or distort in the stress space [22]. Two models are frequently used to describe the hardening behavior of a material due to plastic deformations: isotropic hardening and kinematic hardening. Isotropic hardening assumes that the yield surface expands uniformly as shown in figure 5a. Kinematic hardening assumes that the yield surface translates in the stress space as shown in figure 5b. Pure isotropic hardening cannot predict the Bauschinger effect, as shown for a uniaxial loading
Given the temperature history, thermal strains are determined according to
\[ \epsilon^t = \alpha(T)(T - T_{ref}) - \alpha(T_i)(T_i - T_{ref}), \] (6)
where \( \alpha(T) \) is the thermal dilatation coefficient, \( T_{ref} \) is a reference temperature and \( T_i \) is the initial temperature. The infinitesimal strain \( \epsilon \) is split into elastic, plastic and thermal strains, expressed as
\[ \epsilon = \epsilon^e + \epsilon^p + \epsilon^t, \] (7)
where \( \epsilon^e \) and \( \epsilon^p \) represent the elastic and plastic strains, respectively. \( \epsilon^t \) is determined from this relation and then stresses can be computed by using Hooke’s law as
\[ \sigma = D\epsilon^e, \] (8)
where \( D \) is the elasticity tensor. The stresses satisfy the following equilibrium equation:
\[ \text{div}(\sigma) = 0. \] (9)
When the von Mises yield criterion with isotropic hardening model is used, the yield surface is defined as
\[ f(\sigma, \epsilon^p, T) = \sqrt{3J_2} - \sigma_y - K, \] (10)

Figure 4: Schematic of the von Mises yield surface in the principal stress space.
1.2. GOVERNING EQUATIONS

- **(a) Isotropic hardening**
- **(b) Kinematic hardening**

Figure 5: Evolution of the von Mises yield surface with isotropic and kinematic hardening. Solid line represents initial yield surface and dashed line represents subsequent yield surface.

Figure 6: Uniaxial stress-strain curves for isotropic and kinematic hardening.
where $\sigma_y = \sigma_y(T)$ is the uniaxial yield strength and $K = K(\epsilon^{\text{eff}}_{\text{pl}}, T)$ is the hardening parameter. The effective plastic strain $\epsilon^{\text{eff}}_{\text{pl}}$ is expressed as

$$
\epsilon^{\text{eff}}_{\text{pl}} = \int_0^t \sqrt{\frac{2\epsilon^p : \epsilon^p}{3}} \, dt.
$$

The plastic strain $\epsilon^p$ is governed by the following associative law

$$
\dot{\epsilon}^p = \dot{\lambda} \frac{3s}{2\sqrt{3J_2}},
$$

where $\lambda$ is the plastic multiplier, which is determined by the Karush-Kuhn-Tucker conditions:

$$
\dot{\lambda} \geq 0, \quad f \leq 0, \quad \dot{\lambda} f = 0.
$$

When the von Mises yield criterion with kinematic hardening model is used, the yield surface is defined as

$$
f(\eta, T) = \sqrt{\frac{3}{2}} \eta : \eta - \sigma_y,
$$

where

$$
\eta = s - \alpha
$$

and $\alpha$ is the back-stress tensor. The evolution of the back-stress is governed by Ziegler’s rule, which can be written as

$$
\dot{\alpha} = \frac{k}{\sigma_y}(s - \alpha) \dot{\epsilon}^{\text{eff}}_{\text{pl}},
$$

where $k = k(T)$ is the kinematic hardening modulus and the plastic strain $\epsilon^p$ is governed by the following associative law:

$$
\epsilon^p = \frac{3}{2} \frac{\eta}{\sqrt{\frac{3}{2} \eta : \eta}}.
$$

1.3 Material model

To predict the thermomechanical behavior of a component realistically, it is important to have a material model which represents its characteristics sufficiently accurately. During the frictional heat analysis, temperature independent material data has been used for all the components. For more realistic results, temperature dependent material data should be used. During stress analysis only the brake disc is considered. The brake disc is casted in a grey iron alloy. The material
model used in the present work was developed in an earlier work [23] in order to simulate residual stresses in castings from solidification and is now utilized for thermomechanical stress analysis.

Most of the material parameters required to develop this model were obtained from measurements. Young’s modulus, the yield strength and hardening behavior were obtained from tensile tests performed at 20°C, 200°C, 400°C, 600°C and 800°C. The data was assumed or collected from literature for temperatures above 800°C. This material data is used to build a temperature dependent material model with nonlinear hardening which is described in detail in Paper I. The same data is used to build a temperature dependent material model with linear hardening by connecting the first and last point of the hardening curve with a straight line. This linear hardening model is described in detail in Paper II.

The grey iron alloy shows different yield properties in tension and compression [7]. In the present work, it is assumed that the material has the same behavior both in tension and compression. Although this assumption is unrealistic, it is not the purpose of this work to develop a better material model. Moreover, in this work, the von Mises yield criterion is used both in tension and compression.

In [7] a material model which employs the maximum principal stress yield criterion in tension and von Mises yield criterion in compression was used. Another material model which considers different yield behaviors in tension and compression, and employs the von Mises yield criterion both in tension and compression, is reported in [7] and [3]. In the latter model, numerical results were much closer to the measured experimental data.

1.4 Residual stresses: a simple example

Sometimes, permanent stresses develop in a component even after the external cause, e.g. heat gradient or force, has been removed. In the case of thermal stresses, if they are sufficiently large to cause yielding in a component, then these stresses may develop to residual stresses when the component cools down. Residual stresses may affect the behavior and life of such a component. In order to describe this kind of phenomenon, finite element analysis of a bar subjected to a temperature load will be described and presence of residual stresses after the cooling of the bar will be shown.

In figure 7, a bar is shown with boundary conditions. This bar is subjected to a cyclic temperature load as shown in figure 8, where the peak temperature is $T_m = 600^\circ\text{C}$. The analysis is performed in Abaqus where the bar is meshed with 8-node biquadratic plane stress quadrilateral elements and reduced integration is used. The material models described in section 1.3 will be used to compute stresses in the bar.

In figure 9, a graph of different strain measures in the longitudinal direction is
shown for only one load cycle while employing the temperature independent material model with linear isotropic hardening. It can be seen that the thermal strain increases linearly with the temperature increase. As the bar is restrained in the longitudinal direction, consequently, expanding material causes compressive strain in the bar. In the beginning only compressive elastic strain appear but as the material reaches the elastic limit, compressive plastic yielding also starts. Both the elastic and the plastic strains keep on increasing as the thermal strain increases. After the thermal strain starts decreasing, the elastic strain first shows decreasing trend and later becomes tensile in nature. During this decrease and reversal of the elastic strain, plastic strain stays constant. With the further decrease in the thermal strain, the material reaches its elastic limit and later starts yielding in tension. This yielding in tension causes a reduction in the magnitude of plastic strain. At the end of the first load cycle as the thermal strain vanishes, residual elastic and plastic strains develop in the material. This causes residual stresses in the material even the external source of excitation has been removed. Figure 10a shows the evolution of longitudinal stress versus the longitudinal plastic strain for the three cycles of temperature load. Residual tensile stress can be seen at the end of loading cycles.

The computed stresses and strains strongly depend on the material model used. In figure 10 and 11, the stress-strain graphs for the bar are shown with temperature independent and temperature dependent material models, receptively. By comparing
1.5 Results

The assembly of the disc-pad system considered in this work is shown in figure 12 with one disc sectioned to reveal the ventilation vanes (patented [24]). This is the assembly of a disc brake system of a heavy Volvo truck. In this hybrid or composite design, mounting bell is not a part of the brake disc. The disc is geometrically symmetric about a plane normal to the z-axis. It is assumed that thermomechanical loads applied to the disc are symmetric. Due to these reasons it could be assumed that coning or buckling does not take place. Therefore only a half of this assembly seems sufficient to be considered for the simulation.

The splines at the inner periphery of the disc are used to mount the disc to the wheel hub by engaging corresponding splines. For the simulation of thermal stresses

Figure 9: Evolution of different strain measures with time while using the temperature independent linear isotropic hardening model.

In the results it can be seen that relatively, the stresses are lower for the temperature dependent models, as compared to the temperature independent material models. This reduction in the stresses is attributed to the reduction in hardening of the material at high temperatures. Furthermore with the temperature dependent material models, graphs show higher plastic strain which is attributed to larger thermal strain due to higher thermal expansion coefficient at higher temperatures.

The stress analysis results show that during a load cycle, with increasing temperatures high compressive stress is generated, but when the material cools down and thermal strain vanishes, the compressive stresses transform to tensile stresses. This can be observed for all material models but magnitude of the residual stress is relatively lower with the kinematic hardening as compared to the isotropic hardening models. Furthermore in the case of kinematic hardening model, it can be seen that after the first cycle the stress-strain behavior becomes approximately stable.
CHAPTER 1. INTRODUCTION

Figure 10: Evolution of the longitudinal stress versus the longitudinal plastic strain for the bar, with temperature independent material models, subjected to the cyclic temperature variations.

Figure 11: Evolution of the longitudinal stress versus the longitudinal plastic strain for the bar, with temperature dependent material models, subjected to the cyclic temperature variations.
Figure 12: The assembly of the disc-pad system with a disc shown sectioned.

these splines are not considered important so they have been removed to simplify the model. Similarly some geometry of the back plate has been removed to simplify the model. The assembly with simplified geometries of the disc and the back plate is shown in figure 13.

Simulation of thermal stresses has been performed with the sequential approach. The results show that during hard braking, high compressive stresses are generated on the disc surface in circumferential direction which cause plastic yielding. But when the disc cools down, the compressive stresses transform to tensile stresses. Such results for a single braking operation have been presented in Paper I where the plasticity model is taken to be the von Mises yield criterion with nonlinear isotropic hardening, and both the hardening and the yield limit are temperature dependent.

For repeated braking it is important to use the kinematic hardening model as the isotropic hardening model cannot represent the Bauschinger effect. It has been shown in [25] that in grey cast iron, for a cyclic loading resulting in plastic deformation in both tension and compression, the kinematic hardening model gives a somewhat better agreement with experimental data than isotropic hardening.

In Paper II results of an analysis for repeated braking are presented, where the plasticity model is taken to be the von Mises yield criterion with linear kinematic hardening and both the hardening and the yield limit are temperature dependent. Figure 14 shows the temperature distribution on the disc surface after a brake application during this analysis. A ring of high temperatures, called hot band, can be distinguished in the middle of the disc surface. Figure 15 shows a ring in the middle of disc surface, at the end of brake application, with relatively higher compressive circumferential stresses which roughly corresponds to the ring of high temperatures. The disc is cooled after this braking operation, completing one brake cycle. It is assumed that braking conditions are same for all the brake cycles so
they generate similar temperature history. Hence the temperature history generated during one brake cycle is merged three times in a sequence. In figure 16, graphs of circumferential stresses against different measures of strain in circumferential direction, for three brake cycles, are plotted. The node chosen for these plots is located on the disc surface at 180° from the middle of the pad and at a radius of 163.9 [mm]. It can be seen that residual tensile stresses in circumferential direction are predicted with both hardening models but with the kinematic hardening model these stresses are lower in magnitude as compared to the isotropic hardening model. After the first cycle an approximately stable stress-strain loop is obtained for the linear kinematic hardening model. So if the fatigue life data for the disc material is known, its fatigue life can be assessed. Furthermore results also show the appearance of tensile stresses in radial direction during braking and cooling of the disc. But the residual radial stresses are compressive as compared to the residual circumferential stresses which are tensile. This is indeed in agreement with the observation that radial microcracks on disc surfaces are more marked than circumferential ones, even when macroscopic cracks do not appear [4]. Figure 17 shows a ring in the middle of the disc surface, at the end of three brake cycles, where effective plastic strain is relatively higher. So the material in this area is most susceptible to fatigue cracks.

The simulation results presented in the first two papers predict one hot band in the middle of the disc. It has been explained by showing the contact pressure plots at different time steps. It is also shown (in Paper III) that convex bending of the pad due to thermal deformations is the major cause of contact pressure concentration and hence appearance of hot bands. In the first two papers wear of the pad is not considered as it does not show much influence on the temperature distribution during a single braking operation for a pad without wear history and hence on the stresses.
1.5. RESULTS

Figure 14: After the brake application, a ring of high temperatures develops on the disc surface.

Figure 15: Circumferential stresses at the end of brake application during first cycle with the linear kinematic hardening model.
Figure 16: Evolution of the circumferential stress versus the circumferential strain for the repeated braking.
1.5. RESULTS

The results show that when wear is considered, different distributions of temperature on the disc surface are obtained for each new brake cycle. After a few braking cycles two hot bands appear on the disc surface instead of only one, which is in agreement with experimental observation. The influence of wear on temperature distribution is discussed in Paper III.

This sequential approach has proved tremendously cheap in terms of computational time when compared to a fully coupled Lagrangian approach. Significantly lower computational resources required to simulate a disc brake by using the sequential approach gives the freedom to perform multi-objective optimization studies. Such a study is performed in Paper IV where the mass of the back plate, the brake energy and the maximum temperature generated on the disc surface during hard braking are optimized. The design variables are the applied load of braking, Young’s modulus of friction material and the thickness of back plate. The results indicate that a brake pad with lowest possible stiffness will result in an optimized solution with regards to all three objectives. The results also reveal a linear relation of applied braking load and brake energy. Another interesting result is the trend of a decrease in maximum temperature with an increase in back plate thickness.

Figure 17: Effective plastic strain at the end of third brake cycle with the linear kinematic hardening model.
Review of included papers

Paper I

In this paper results of a simulation of stresses in a brake disc for a single braking operation are presented. The plasticity model is taken to be the von Mises yield criterion with nonlinear isotropic hardening, where both the hardening and the yield limit are temperature dependent.

Paper II

In this paper results of a simulation of thermal stresses in a brake disc for repeated braking are presented. The plasticity model is taken to be the von Mises yield criterion with linear kinematic hardening, where both the hardening and the yield limit are temperature dependent.

Paper III

In this paper the influence of the wear history of a pad on the temperature distribution on a disc surface is presented. It is also shown that convex bending of a pad assembly as a result of thermal deformations is a significant factor towards the concentration of contact pressure in the middle of a pad.

Paper IV

In this paper results for a multi-objective optimization of a disc brake system are presented. The mass of the back plate of the brake pad, the brake energy and the maximum temperature generated in the disc during hard braking are optimized. The design variables are the applied load of braking, Young’s modulus of friction material and the thickness of the back plate.
CHAPTER 2. REVIEW OF INCLUDED PAPERS

Paper V

In this paper a literature review of disc brakes and related phenomena is presented. A detailed description of different geometries and materials for the components of a brake assembly is given. The evolution of tribological interface of disc-pad system is also covered in detail here. Different operational problems such as fade, geometrical deviations and noise are also discussed.
Bibliography


