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In-Vehicle Brake System Temperature Model

Master Thesis in the Master's programme in Automotive Engineering

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ABSTRACT

The brakes system is critical with respect to vehicle safety. One situation during which the brake system is put to the test is an Alpine descent. Such a descent causes very high brake system temperatures and may even induce brake fluid vaporization. In following report an In-Vehicle Brake System Temperature Model is developed and tested. This model makes use of the information that is available on the vehicle CAN-bus in order to estimate the temperature of the brake system and detect the risk of brake fluid vaporization. Implementing such a model in production vehicles would improve vehicle safety and in the long run allow downsizing of the brake system without giving in on any safety margins.

Firstly, possible approaches for estimating the amount of kinetic energy that is converted into heat by the brake system are investigated. A Feed-Forward model is developed that uses the pressure in the primary hydraulic circuit as input. The problem with such a Feed-Forward Model is that the friction coefficient is variable. A Feed-Back model that uses the vehicle deceleration as input gives better results. The challenge in this approach lies in the fact that not all required inputs for the Feed-Back Model are known.

Secondly, the Temperature Estimation Model is developed. This is composed of models of different parts of the brake system which are combined and matched to the measurements. The brake disc is modeled as a lumped mass. The brake pad is modeled as a Finite Difference Thermal Model. The fluid and surrounding caliper are modeled as one lumped mass which receives its heat through conduction from the brake pad and spreads the heat into the surroundings by means of convection.

Finally, for the Human Machine Interface, several active and passive intervention concepts are proposed. One important challenge to take into account is the uncertainty in the brake fluid boiling temperature, which decreases over the course of time due to brake hygroscopic effects.

Keywords: Brake System, Thermal Model, Brake Fluid Vaporization, Brake Pad, Alpine Descent

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List of Symbols

Acronym	Unit	Description
BCM	-	Brake Control Module
BSTM	-	Brake System Temperature Model
CAE	-	Computer Aided Engineering
FB	-	Feed-Back
FF	-	Feed Forward
TEM	-	Temperature Estimation Model
VCC	-	Volvo Car Corporation

Symbol	Unit	Description
A_s	m^2	Surface Area
C_d	-	Drag Coefficient
C_{fric}	-	Friction Coefficient
C_p	$J \cdot K^{-1}$	Thermal Capacity
f_r	-	Rolling Resistance
F	N	Force
$F_{i,j}$	-	View Factor
h	$W \cdot m^{-2} \cdot K^{-1}$	Convective Coefficient
J_w	$kg \cdot m^2$	Wheel Rotational Inertia
k	$Wm^{-1}K^{-1}$	Thermal Conductivity
m	kg	Mass
$n_{pistons}$	-	Number of Pistons
p	Pa	Pressure
Q	J	Amount of Heat Generated
$\dot{Q}$	W	Heat Generation Rate
R_{therm}	$m^2 \cdot K / W$	Thermal Resistance
r_{eff}	m	Brake System Effective Radius
$S_{d,p}$	m^2	Frictional Contact Surface (disc/pad)
t	s	Time

T, T_s	K	(Surface) Temperature
$T_{subscript}$	Nm	Torque
v	$m \cdot s^{-1}$	Vehicle Velocity
W_{drag}	J	Work Done by Drag Forces
z	mm	Pad Material Thickness
$\dot{z}$	$mm \cdot s^{-1}$	Pad Material Wear Rate
α	-	Absorbtivity
ε	-	Emissivity
μ_{fric}	-	Friction Coefficient
θ	rad	Road Gradient
ρ	$kg \cdot m^{-3}$	Density
σ	-	Heat Partition coefficient
σ_b	$W \cdot m^{-2} \cdot K^{-4}$	Stefan-Boltzmann Constant
ω	$rad \cdot s^{-1}$	Wheel Angular Velocity
$\dot{\omega}$	$rad \cdot s^{-2}$	Wheel Angular Acceleration
$\xi_{d,p}$	$W \cdot s^{1/2} \cdot K^{-1} \cdot m^{-2}$	Thermal Effusivity

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1. Introduction

1.1. Background

The continuously increasing demand for personal mobility has led automobile manufacturers to strive continuously for cheaper, more efficient and safer vehicles. These requirements are competing with respect to each other. However, the tough competition in today's automotive market forces vehicle manufacturers to strive for improvement in each of these fields.

To improve vehicle safety without the burden of increasing vehicle weight, more and more active safety features are introduced. However, the most critical part with respect to safety is, and has always been: the brake system. Due to its critical role, very high demands are imposed on the brake system with respect to reliability, durability and consistency in its behavior. These severe demands have forced the brake system development in a rather conservative direction. Introducing drastically new technology is never completely free of risk and in brake system development there is no room for mistakes. Therefore, the brakes on modern vehicles are an extreme optimization of a very old design.

Four-Wheel hydraulic brakes have been introduced in 1918 by Malcolm Loughead [1]. Both disc brakes and drum brakes have been developed in the beginning of the 20th century. Among these types, the disc brakes generally have better performance than drum brakes, though at a higher cost. From the 1950's onwards, hydraulic brake systems with disc brakes on the front axle and a vacuum operated brake booster have been the standard in the automotive market. This setup has proven to be very reliable, robust and well performing. Most modern vehicles have disc brakes at the front axle, where the toughest requirements are imposed on the brakes. On the rear axle either disc brakes or drum brakes are generally installed.

With modern technical standards, the rigorous demands that are imposed on a brake system can be met easily. This has led to a shift in focus in brake system development. The high standard with respect to durability, reliability, performance and comfort should be maintained, while at the same time reducing cost and weight of the brake system as much as possible. Currently this is done mainly by optimization of different parts of the brake system and the cooling airflow around it. In the near future however, additional help can come from vehicle dynamics and the sensors that are installed in the vehicle. Knowledge of vehicle dynamics in combination with on board computational capacity paves the way for safety systems that monitor/estimate the brake system condition.

1.2. Problem Definition

The rigorous demands for the brake system with respect to reliability are put to the test during situations in which the brake system is used extensively. Since brake system design is a tradeoff between costs, weight and performance, the brake system is designed to be able to cope with the most extreme situations it might encounter during its lifespan. One such situation in which brakes are put to the test is an Alpine descent. A standard test for European automobile manufacturers is descending Mount Glockner in Austria with a fully laden vehicle and without making use of engine braking. This Alpine descent is simulated at Volvo Car Corporation by means of a so called SimAlp test. During this test the Alpine descent is simulated by pushing the test vehicle with a second vehicle. During this test, the push force between both vehicles is measured since this corresponds to the brake force exerted by the front vehicle. During such an Alpine-descent the brake fluid temperature rises significantly. For well maintained vehicles this generally does not impose a large problem since pure brake fluid has a very high boiling temperature in the order of 200-300 degrees Celsius [2], [3].

The brake fluid is divided into DOT-categories, depending on the fluid boiling point [2], [3]. Over the course of time the brake fluid absorbs moisture, significantly reducing the fluid boiling temperature. For DOT 3 and DOT 4 brake fluids, which are used in a vast majority of today's vehicle park, the initial boiling point lies at 205 degrees Celsius and 230 degrees Celsius respectively. Due to hygroscopic effects, the brake fluid boiling temperature has already gone down to 140 degrees Celsius and 155 degrees Celsius respectively, after only a few years of operation. This gives a significant increase in the risk of brake fluid vaporization during an Alpine descent.

A vehicle brake system is designed to be able to perform an Alpine descent without the risk of brake fluid vaporization. However, during vehicle development it is assumed that the brake fluid is changed every second year. In reality many vehicle owners never change the brake fluid of their vehicle. This introduces the risk that some of the older vehicles could encounter brake fluid vaporization and strives towards the demand for a system that estimates the brake fluid temperature in order to take action in the case (the risk for) brake fluid vaporization occurs.

During braking, the kinetic energy of the vehicle is transformed by means of frictional contact into heat. For drum brakes 95% of this heat is dissipated through the brake drum. The remaining part goes into the brake lining [4]. Due to the poor conductivity between the brake lining and the wheel cylinder, the brake fluid is effectively insulated from the brake heat generation for drum brakes [2]. Therefore there is only a considerable risk for brake fluid vaporization disc brake systems. Next to this, the vast majority of vehicles are equipped with disc brakes on the front axle where most of the brake torque is exerted in order to maintain vehicle stability. These factors combined lead to the decision to focus the brake system temperature model on disc brakes and not on drum brakes.

1.3. Objective

The objective of this thesis is to develop preliminary version of a Brake System Temperature Model (BSTM) for vehicle implementation and perform the required testing to tune this model. This system should estimate the temperature of the different brake system components in order to detect dangerous situations that can arise due to over-heating of the brake system.

The BSTM(1) should fulfill following requirements:

- The input for the BSTM consists of the sensor signals that are readily available in the Brake Control Module (BCM) and on the vehicle CAN-bus. No additional sensors or other hardware should be installed for the BSTM.
- The amount of parameters that are to be tuned empirically should be as small as possible. In other words: it should be possible to install the BSTM on a new vehicle with a limited amount of additional testing/tuning required.
- The BSTM is compatible with the brake system model at VCC. To ensure this, regular meetings and discussions with the VCC brake system CAE Engineers are needed. This will ensure easy exchange of the information and models in both directions.
- The estimation of the fluid temperature should be sufficiently accurate. The brake fluid temperature should be estimated with a maximum deviation of plus or minus 30 K over a complete Alpine descent including heat soaking. On top of this, the time at which the fluid peak temperature occurs should be predicted within an accuracy of 5 minutes.
- The BSTM development will focus on an Alpine descent. Therefore, the BSTM should be able to detect the risk of brake fluid vaporization during this driving situation. An additional application could be to detect high brake system temperatures during sportive driving.
- During the development of the BSTM several tests will be performed to determine brake system characteristics. These tests should be set up in cooperation with the brake system CAE Engineers since some of the info coming out of some of these tests will be of use for them also.
- Future application and improvement of an on-board BSTM should be thought through and a strategy should be proposed.

(1): It should be noted that the BSTM as developed in this report is only a preliminary version. The requirements as stated are therefore the requirements for this report and not for the complete project.

2. Brake System Description

2.1. General Description

The function of the brake system is to allow the driver to reduce the vehicle kinetic energy according to a highly controllable rate. The brake pedal travel and force determines the brake fluid pressure that is present in the hydraulic circuit. In between the brake pedal and the hydraulic circuit, a brake booster is present to amplify the force exerted by the driver. This allows any driver to have complete control over the brakes with limited effort and a proper pedal feedback. The brake booster makes use of a vacuum which is either driven by the vacuum that occurs in the combustion engine at the intake manifold for most gasoline engines or by a separate vacuum pump for most diesel engines. The increased pressure in the brake fluid pushes one or more pistons outwards. In a disc brake setup, these pistons push the brake pad against the disc.

In modern disc brake systems, two types are mostly used:

- **Floating caliper**: A sliding contact in which there are only brake pistons on one side of the brake disc. These piston(s) push the inner pad against the brake disc. At the same time, they push a part of the caliper backwards. By means of a sliding contact this pulls the brake pad at the other side against the brake disc.
- **Fixed caliper**: Brake pistons which are located on both sides of the brake disc and push the brake pads directly against the brake disc.

A more elaborate and detailed description of different setups of disc brake systems can be found in [5]. In Figure 2.1 and Figure 2.2 below an illustrative sketch of a brake system is shown.

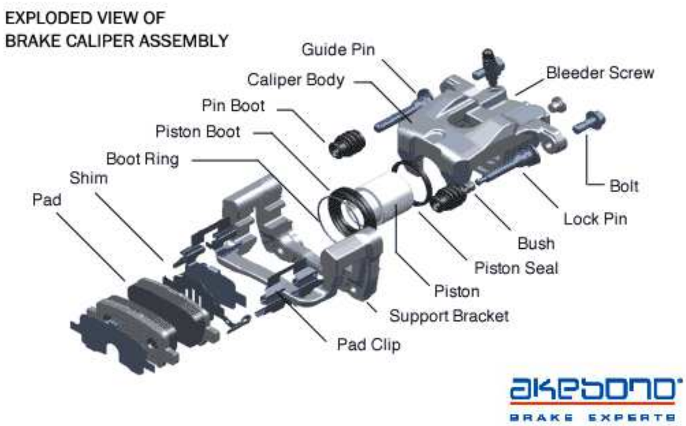

Figure 2.1: Exploded View of Brake Caliper Assembly

Figure 2.2: Brake Disc and Caliper

2.2. Heat Generation

As mentioned before, the function of the brakes is to reduce the vehicle kinetic energy. For disc brakes, this kinetic energy is mainly transformed into heat through a frictional sliding contact between the brake disc and pads. This friction force between the pads and the disc generates a resisting torque on the wheel. This resisting torque initially transforms the rotational inertial energy of the wheel into heat which slows down the wheel rotation with respect to the vehicle translational motion. This reduced wheel rotational speed generates in turn a frictional force at the wheel-ground contact patch which reduces the vehicle translational velocity. During the transient phase of the braking phase, elastic energy is being built up in the tire while it is deforming in order to build up the friction force at the tire-ground contact patch. Next to this, some of the heat is absorbed by elastic deformations in the chassis and suspension. Therefore, the initial part of the heat generation process does not translate immediately into vehicle deceleration. On top of this, at high deceleration braking there is a significant amount of slip at the tire-ground contact patch which generates heat as well. However, for normal braking situations, this amount of heat generation is negligible with respect to the amount of heat generated in the friction brakes. Especially for Alpine descent situations this heat generation is negligible since Alpine braking consists of moderate continuous braking.

In Figure 2.3 below an overview of all the moments acting on the wheel during a brake event are shown. The translational inertia of the vehicle generates a frictional force at the tire-ground contact patch when the brakes are applied. This frictional force together with the wheel rotational inertia should be overcome by the brake torque.

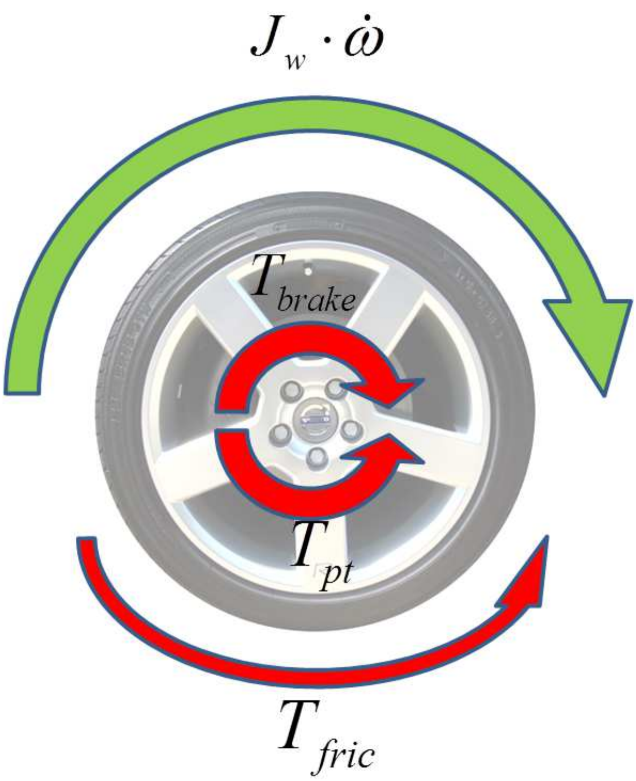

Figure 2.3: Moments acting on a braking wheel

$$J_w \cdot \dot{\omega} = -T_{fric} - T_{pt} + T_{brake} \qquad \text{Eq. 2.1}$$

Figure 2.3 in combination with Equation 2.1 above describes the dynamics of a single wheel. T_{fric} is generated at the tire-ground contact patch due to the difference in wheel rotational and vehicle translational velocity. As mentioned before, this braking moment transforms the vehicle kinetic energy into heat. The amount of heat generation by a frictional contact is given by [6]:

$$\dot{Q} = F_{fric} \cdot v_{slide} \qquad \text{Eq. 2.2a}$$

or for rotational movement:

$$\dot{Q} = T_{brake} \cdot \omega \qquad \text{Eq. 2.2b}$$

in which:

$$T_{brake} = F_{fric} \cdot r_{eff} \qquad \text{Eq. 2.2c}$$

By making use of Equations 2.1 to 2.2c the heat generation process in the brake system can be described.

2.3. Disc – Pad Heat Distribution

During a braking event, kinetic energy is transformed into heat by means of the friction brakes. This heat is generated within the contact surface between the brake disc and pad. In the following section, some general modeling approaches for this contact surface are discussed. Generally, the contact between the disc and pad can be modeled as a perfect or an imperfect contact. The perfect contact postulation assumes that the disc and the pad have identical surface temperatures. The imperfect contact postulation assumes a temperature discrepancy between both parts.

In reality, brake events are very transient and of a short duration. Therefore perfect contact can often not be achieved and mostly imperfect contact is used for modeling the disc-pad interface [7]. In the imperfect contact postulation, a third body constituted by detached particles is present between the disc and pad surfaces. The heat is assumed to be generated at the pad surface and dissipated into the disc through this third body. This causes a discrepancy between the disc and pad surface temperature [8]. A detailed description of the disc-pad interface and modeling of this third body can be found in [9]. This imperfect contact postulation complies more with experimental observations where there is a temperature difference between the disc and the pad. The challenge lies however, in determining the thermal resistance of this third body [8].

A part of the heat generated during braking is immediately dissipated into the surrounding air by means of convection. According to [4] this fraction is only in the order of 1%. The fraction of heat that is absorbed by the brake disc is, according to [7] in the order of 93%. The remaining part of the heat is absorbed by the pad. These findings are compared to a more theoretical model. The exact distribution of heat between the brake disc and pad is dependent on the properties of each specific pad-disc combination. In [7], a theoretical model for the heat partition coefficient between the brake disc and pad for an imperfect contact is given by:

$$\sigma = \frac{\xi_d \cdot S_d}{\xi_d \cdot S_d + \xi_p \cdot S_p} \qquad \text{Eq. 2.3}$$

In this equation, S is the contact surface area of the disc (S_d) and pad (S_p) respectively and the thermal effusivity ξ is calculated according to:

$$\xi = \sqrt{k \cdot \rho \cdot c_p} \qquad \text{Eq. 2.4}$$

in which k is the thermal conductivity and c_p the thermal capacity.

This heat partition coefficient represents the fraction of the total amount of heat generated that travels into the brake disc. The remaining part goes into the pad and into the surroundings. This approach also has some inaccuracies like the overlook of thermal inertia (the responsiveness to a change in temperature) and the assumption of a linear temperature gradient across the third body [8].

When inserting material properties of a normal brake disc/pad combination into Equation 2.3 above, it is observed that 98,8% of the total energy is absorbed by the brake disc. This is significantly larger than the 93% that is assumed in [7]. Therefore, the exact heat distribution between brake disc and pad will be tuned experimentally later on.

2.4. Disc Temperature

In the last decade many investigations on disc thermal behavior have been performed. Also within VCC disc thermal models have been and are being developed varying from simple lumped mass models to advanced Finite Element models. Therefore the disc temperature evolution has not been investigated extensively within this report. The disc model should be kept as simple as possible in order to be replaced by more advanced models in a later stage of the project. In [6] a disc model is proposed consisting of slabs in circumferential direction. Due to the generally large angular velocity of the brake disc, the heat input is assumed to be spread out evenly in angular direction. The brake disc absorbs by far the largest portion of the brake energy. This is conducted within the disc material and spread to the surroundings, mainly through convection and radiation. The exact behavior of this is a complicated process and highly influenced by the disc shape and the aerodynamics of the wheel hub. More detailed investigation of brake disc thermal behavior and its dependency on shape and aerodynamics can be found in [10].

For estimating the fluid temperature however, only the disc surface temperature is of importance in order to model the interaction between disc and pad. Therefore only a lumped mass model is developed for the disc that is matched to the measurements.

2.4.1. Disc Heat Input

The heat that is absorbed by the brake disc is spread throughout the disc by means of conduction and causes the disc temperature to increase. The temperature increase of the disc lumped mass model is calculated by:

$$c_p \cdot m_{disc} \cdot \frac{dT}{dt} = \dot{Q}_{in} - \dot{Q}_{out} \qquad \text{Eq. 2.5}$$

The heat input $\dot{Q}_{in}$ is caused by the brake torque:

$$\dot{Q}_{in} = \sigma \cdot T_{brake} \cdot \omega \qquad \text{Eq. 2.6}$$

The thermal capacity of a brake disc is temperature dependent. In Figure 2.4 below temperature dependency of the thermal capacity of iron is shown [11]. The importance of this was mentioned in [12]. In [12] a linear c_p-Temperature dependency was assumed. In the operating range of the disc (0 degrees Celsius – 700 degrees Celsius) this assumption is justified, as can be seen in Figure 2.4 below. This variable value for c_p is included into the disc lumped mass model.

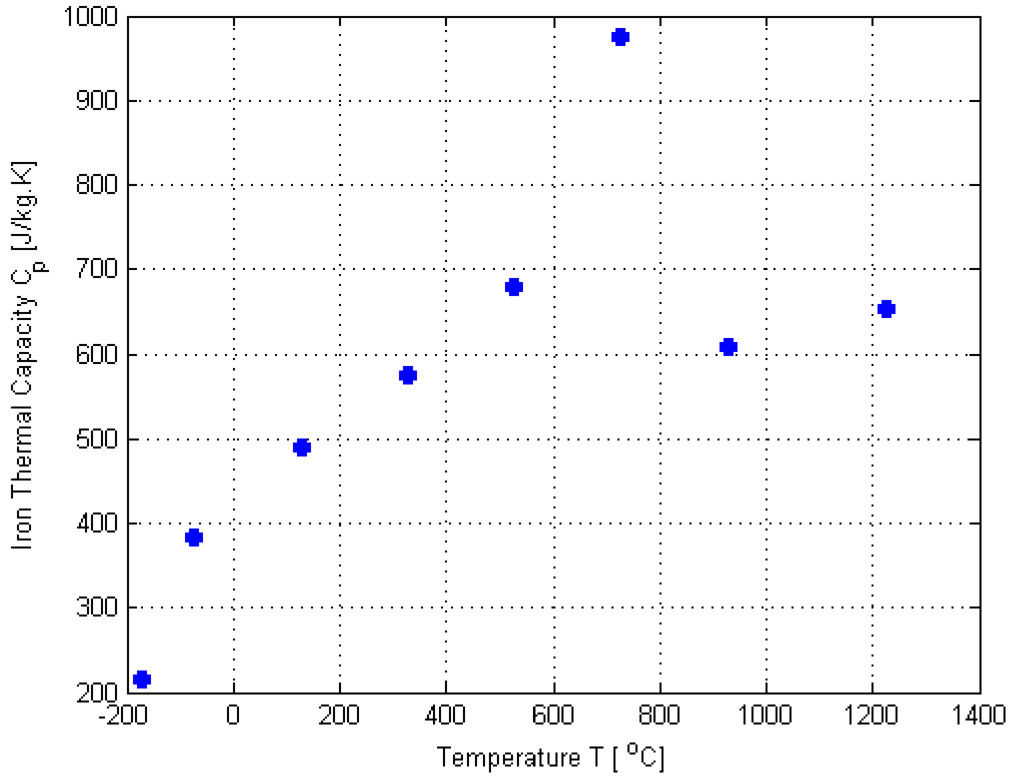

Figure 2.4: Temperature dependency of iron thermal capacity

2.4.2. Disc Heat Output

$\dot{Q}_{out}$ is the heat output of the disc. While driving, convection is the main contributor of this heat loss and can according to Newton be described by [11]:

$$\dot{Q}_{conv} = h_{av} \cdot A_s \cdot \Delta T \qquad \text{Eq. 2.7}$$

in which:

$$h_{av} = \frac{1}{A_s} \cdot \int h \cdot dA_s \qquad \text{Eq. 2.8}$$

The convective coefficient h is dependent on the shape of the disc and the airflow around it, which in turn is mainly dependent on the vehicle aerodynamic shape, rim shape and the vehicle speed. A second way in which the disc spreads heat to the environment is by means of radiation. Radiation is given by the Stefan-Boltzmann law as [11]:

$$\dot{Q}_{rad} = \varepsilon \cdot \sigma_b \cdot A_s \cdot T_s^4 \qquad \text{Eq. 2.9}$$

This radiation is spread to the other parts of the brake system and into the surroundings. Next to the emitted radiation, the disc also receives some radiation from the pad, caliper and backplate.

2.5. Pad Temperature

2.5.1. Heat Input during Braking

Next, the distribution of the heat over the contact area between disc and pad during braking is investigated. An exact temperature distribution of the sliding contact surface involves a very complicated process and is dependent on many factors.

On microscopic level, so-called hot-spots occur. [13], [9]. This is due to the actual contact interface that is restricted to a few small areas at any particular time. These areas are distributed over the nominal contact area. Due to this limited contact interface, very high temperatures arise at these contact points: so called hot-spots arise. However, due to the small sizes, the amount of heat generated at each particular hot-spot is rather low, despite the high temperatures that are reached. Therefore, this heat does not penetrate deep into the material. At a small distance away from the surface already a very homogeneous temperature distribution is present [13], [9].

On a macroscopic level, the contact areas at the disc-pad interface move back and forth. This is influenced by wear and thermal expansion. Thermal expansion tends to decrease the contact area, which increases wear, which in turn tends to increase the contact area. The interaction between both effects influences the evolution of the contact pressure distribution.

For modeling purposes, two distinct approximate models are suggested: uniform wear and uniform pressure [14], [7]. For new pads, the uniform pressure model would give the closest approximation. The pressure between the pad and the disc is the same over the complete surface. When looking at the heat distribution over the pad surface, two distinct increases in temperature could be noted on a macro scale. Firstly, the relative sliding velocity between the disc and pad increases for increasing distance with respect to the wheel center. Due to this velocity difference, the pad surface temperature on the outer radius is higher than on the inner radius since heat generation is related to the sliding velocity. Secondly, the brake disc surface temperature is lower than the pad surface temperature at the initiation of the braking. The pad only covers a small angle of the contact 'ring' from the disc. On the leading edge of the pad, relatively cool disc area comes into contact with the pad area. This piece of the disc ring heats up considerably during the sliding contact with the brake pad and reaches its maximum at the trailing edge of the pad. The rest of the revolution the disc has time again to cool down before entering the leading edge of the pad contact again. Therefore, the second distinct increase in temperature can be noted between the leading edge and the trailing edge of the pad. These two temperature increases give a temperature field as is illustrated in Figure 2.5 below.

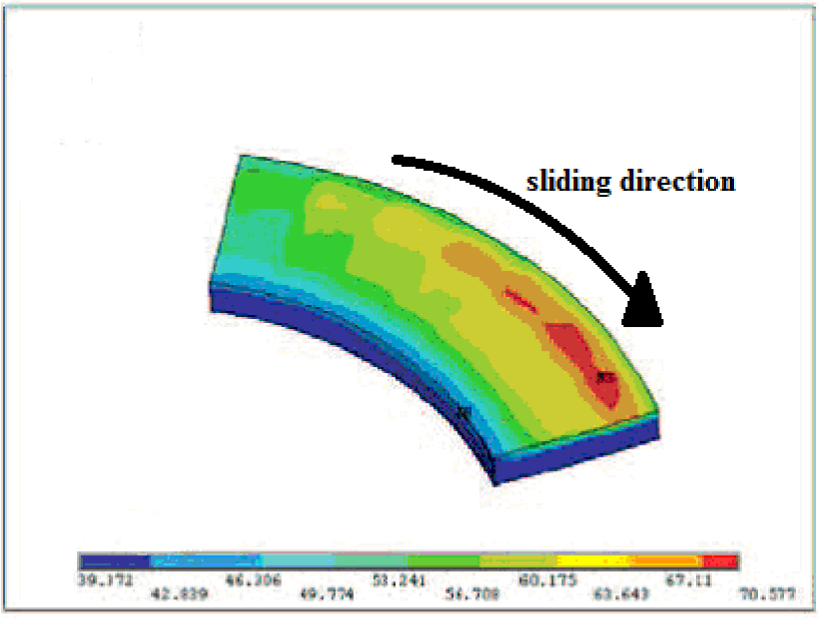

Figure 2.5: Illustration of pad surface temperature field during braking [15]

After the course of time, the high temperature at the upper side and trailing edge of the pad speeds up the pad wear in these regions. This increased wear reduces the contact pressure in this region. This reduced contact pressure in turn decreases heat generation and thus wear until the pressure is distributed in such way over the pad that the pad wear or heat generation is more or less constant over the complete pad surface. The uniform wear model becomes the most valid. Since most of the time, the pads have already worn in the logical choice would be to make use of the constant wear model in which a constant heat input is given to the complete pad. However, during discussions with VCC engineers it came forward that during high pressure braking, the outer radius of the pad tends to get warmer than the inner radius. This is due to the limited caliper stiffness. During low pressure braking, the heat distribution is more equal. Therefore a combination between the constant pressure and constant wear model is expected to give the most accurate result. In the constant wear model, the heat flux into the model is a constant. In the constant pressure model, the heat flux into the model varies in radial direction with *1/r* [7].

In longitudinal direction (from the pad leading edge to the pad trailing edge) it is assumed that the pressure is evenly distributed. This is a reasonable assumption due to the Hammerhead design of the VCC brake pads [5]. When braking, the frictional force at the pad surface generates a moment, since the friction force is not applied in the pad's center of mass. This moment would tend the leading edge of the pad to be pushed more firmly against the disc than the trailing edge, inducing uneven pad wear in longitudinal direction. In conventional pad design this moment caused the pressure in the leading edge to be roughly 1/3th larger than the average pressure. However, the Conti-Teves Hammerhead design (see Figure 2.6 below), counteracts this moment by pulling the pad to keep it into position instead of pushing. This pulling force generates a counteracting moment around the pad backside. Therefore, in modern pads, the pressure gradient in longitudinal direction is much smaller. The pressure at the leading edge of the pad is roughly 1.033 times the average pressure. This effect partly counters the larger wear at the pad trailing edge which has been discussed above.

Figure 2.6: Brake pad Hammerhead

2.5.2. Pad Heat Exchange after Braking

The major heat source for the brake pad after a brake event is radiation from the disc. The amount of radiation absorbed by the pad from the disc is given by [11]:

$$Q_{abs,pad} = \alpha_{pad} \cdot F_{ij} \cdot Q_{rad,disc} \qquad \text{Eq. 2.10}$$

This corresponds to the fraction of the radiation of the disc that is intercepted and absorbed by the pad. If the disc would radiate homogeneously, the view factor, or the fraction of radiation emitted by the brake disc that is directed towards the pad, would correspond to the solid angle that the pad front surface area covers of the sphere surrounding the disc.

Since the distance between disc and pad is negligibly small, and under the assumption of homogeneous radiation from the disc, the view factor can be approximated as the ratio of the pad front surface area and the total disc surface area:

$$F_{ij} \approx \frac{A_{s_{pad,front}}}{A_{s_{disc,total}}} \qquad \text{Eq. 2.11}$$

A first means of heat loss for the pad is convection. It is a complicated process to accurately model the convective cooling of the complete pad since this is dependent on vehicle speed, ambient temperature etc. Next to this, convective cooling of the pad does not occur homogeneously but varies with location on the pad outer surface. Therefore an average convective coefficient for the pad is used. This dynamic convective coefficient can be related to vehicle speed according to [16]:

$$\overline{h} = const \cdot v^n \qquad \text{Eq. 2.12}$$

In this equation n is generally 0.5 for laminar and 0.8 for turbulent flows [16]. Therefore, the average dynamic convective coefficient should only be determined for one speed. According to Equation 2.12 above, this can be expanded to other speeds as well with limited deviation.

However, for automobiles, the aerodynamics are more complicated than this. The cooling airflow around the brake system varies significantly for different speeds. Optimal speeds exist for a specific vehicle, in which a proper airflow is present around the brake system. This can give a cooling coefficient at a specific speed that is higher than at the surrounding speeds. The opposite can also occur: specific speeds at which the cooling airflow around the brakes is worse than the speeds around it. This makes the situation in reality significantly more complicated than the assumption of Equation 2.12.

A second means of heat dissipation for the pad is conduction. The heat coming into the pad during and after braking enters through the pad front surface. This heat is conducted through the pad material towards the pad backplate. The heat spreads within the back plate and is in turn conducted to the caliper through the so-called 'Hammerheads' through which the pad is connected to the caliper. A part of the heat is conducted to the brake piston through the shim, which is attached to the backside of the backplate. The importance of the backplate and shim cannot be underestimated. The backplate of the brake pad has a high conductivity in order to spread the heat as effectively as possible. The shim is composed of multiple layers of rubber and metal [17]. The function of these layers is to isolate the piston – and thus the brake fluid – as effectively as possible. Therefore, the backplate should be modeled as both a thermal resistance and a heat sink. A final means of heat dissipation for the pad is radiation into the surroundings.

2.6. Pad Wear

Once the heat enters the pad surface due to friction braking or disc thermal radiation, this heat is spread through the pad material. Firstly the heat is conducted into the pad material towards the backplate. Secondly heat is radiated outwards and thirdly heat is spread into the air through convection. The heat that is conducted to the backplate is of significant importance, since it is this heat that eventually will reach the brake fluid. Before modeling the heat flux through the pad, the pad thickness should be estimated. In order to do so, inclusion of a pad wear model is investigated.

Examples of pad wear estimation models can be found in [13], [18] and [19]. In [18] a law of wear is determined by means of empirical data. This wear model requires a significant amount of tuning. In [13] Archard's law of wear is discussed which is given by:

$$\dot{z} = K_{wear} \cdot p_{contact} \cdot v_{slide} \qquad \text{Eq. 2.13}$$

where:

- $\dot{z}$ is the wear rate
- K_{wear} is the wear coefficient for which several models exist
- $p_{contact}$ is the contact pressure
- v_{slide} is the sliding velocity

In [19] the pad wear properties are investigated for different initial velocities and temperatures. It can be seen that for different kinds of braking, the pad wear is quite consistent. Only when the temperature goes above 320 degrees Celsius, the pad-wear increases significantly. This can be seen in Figure 2.7 below.

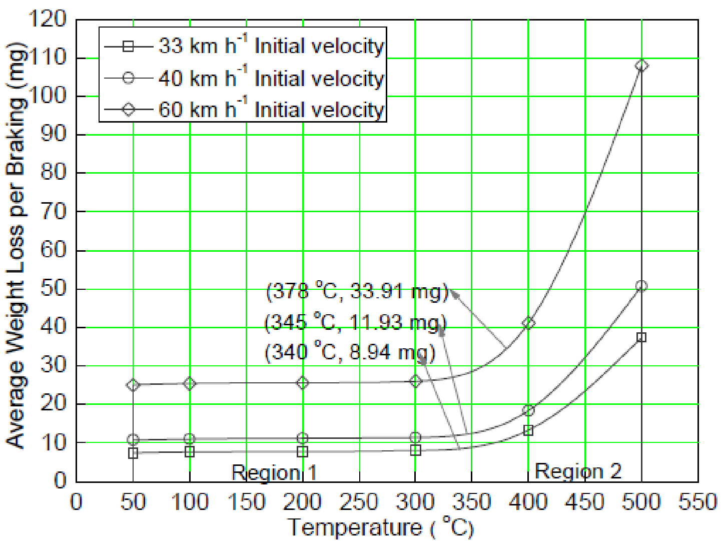

Figure 2.7: Pad material wear for different temperatures [19]

Figure 2.7 above shows test results of average pad wear against temperature for different initial velocities. When looking at normal operating temperatures ($<320^oC$) it can be concluded that the pad wear is related to the total energy absorbed by the brakes. The initial velocity of the braking does not change the pad wear against energy absorbed relation. In Table 2.1 the ratios between the three initial velocities for pad wear and kinetic energy absorbance are shown. It can be seen that these are almost identical for each pair of initial velocities. Therefore, the pad wear per unit energy absorbed can be considered only a function of contact pressure and temperature.

Initial Velocities	Ratio of pad wear	Ratio of kinetic energy absorbed ($\sim v^2$)
40 *km/h* / 33 *km/h*	1,47	1,46
60 *km/h* / 40 *km/h*	2,25	2,27
60 *km/h* / 33 *km/h*	3,3	3,3

Table 2.1: Ratio of pad wear and kinetic energy absorbed between different brake initiation velocities

2.7. Fluid/Caliper Temperature

2.7.1. Fluid Heat Input

The fluid receives heat from the pad backplate through the piston by means of conduction. This travels deeper into the fluid hose and is spread towards the surrounding caliper by means of conduction/convection.

2.7.2. Caliper Heat Flow

The final part of the brake system, the caliper, acts as a heat reservoir that helps the rest of the brake system to get rid of their excess heat. The caliper receives heat from the pad backplate and the fluid through conduction/convection. Next to this, the caliper receives heat from the disc through radiation. The heat is spread throughout the caliper by means of conduction and spread into the surrounding air by means of convection and radiation.

3. Preliminary Test Data Investigation

3.1. Wind Tunnel Testing

In order to get an initial understanding of how the heat distributes throughout the brake system and towards the brake fluid, an empirical investigation is performed on a vehicle that is scheduled for wind tunnel testing. In this wind tunnel experiment the SimAlp test is performed. On this vehicle, thermal couples are attached at different locations of the caliper, at the front- and backside of the brake pad and within the brake fluid. For each of these locations, the temperature evolution over time during the wind tunnel experiment is investigated in order to get a clearer overview of how the brake fluid temperature is influenced by the different brake system components.

In Figure 3.1 below, the temperatures of the different brake system components during the wind tunnel experiment are shown.

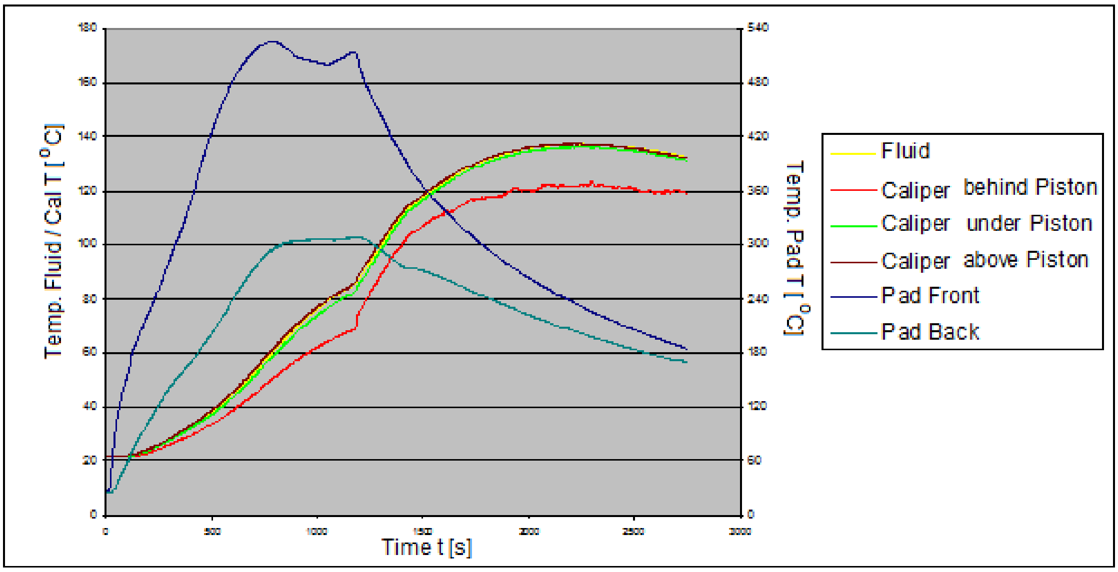

Figure 3.1: Initial Wind Tunnel Investigation

A very important and remarkable observation in the test data is the brake caliper temperature. It can clearly be seen that the caliper around the piston has almost the same temperature as the fluid. The part of the caliper that is located behind the piston has a slightly lower temperature. The heat goes through the pad into the brake piston and subsequently into the fluid. As the fluid temperature increases, this heat is spread towards the surrounding caliper. Due to the high conductivity of the caliper material and the very large contact area between the fluid and the caliper, the caliper temperature immediately follows the fluid temperature and spreads the heat into the environment by means of convection with the surrounding air and radiation. This process also happens in the opposite way. It can be seen that some part of the caliper heats up slightly earlier than the fluid. Part of the heat goes from the pad backside directly to the caliper which distributes the heat towards the fluid. This observation allows modeling of the fluid and the surrounding caliper as if it were one piece. Due to the combination of different materials, complex geometry etc it is difficult to determine accurate values for m and C_p of this 'object'. Next to this, the exact heat exchange of this 'object' is difficult to determine since some of the heat will travel deeper into the hydraulic system. Therefore, the fluid and surrounding caliper are modeled empirically and tuned to match the test data.

3.2. SimAlp Testing I

Besides the Wind tunnel Experiment some old and new SimAlp tests are investigated. In Figure 3.2 below, the test data of an old SimAlp test are shown. With this specific vehicle setup, the SimAlp test has been repeated 7 times. For one of these runs, the temperature measurements are displayed below. It has been observed that the other runs show very similar behaviour as the example shown in Figure 3.2.

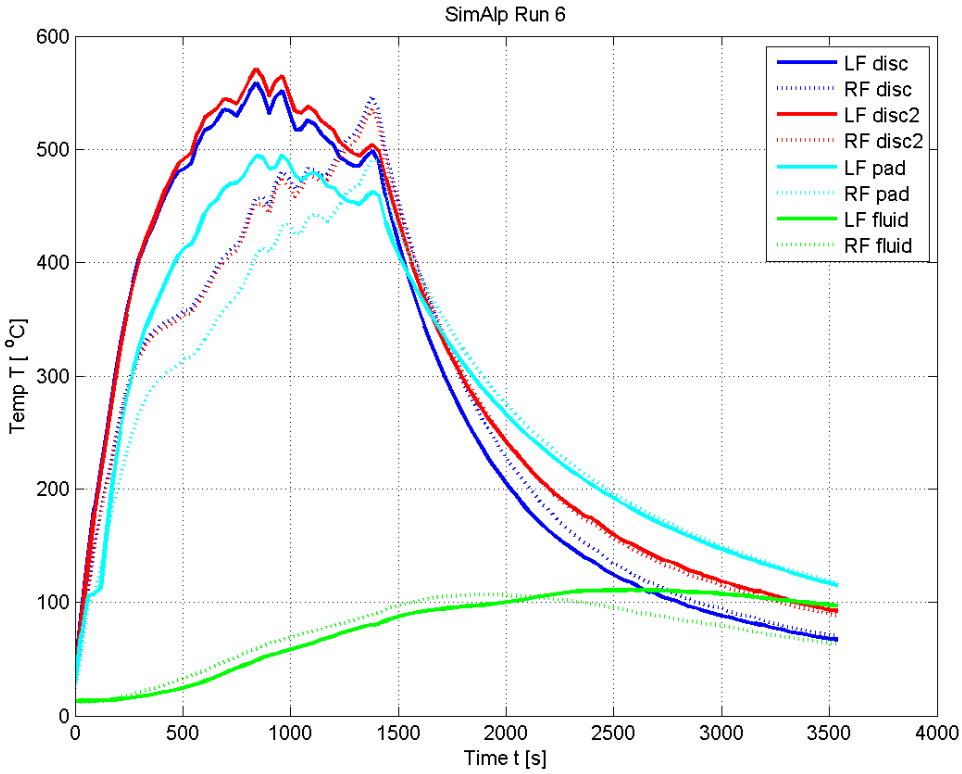

Figure 3.2: Old SimAlp Test Data

Investigation of old SimAlp data immediately revealed a consistent discrepancy between the left and the right front wheel. For each of the tests the left front disc heats up much faster than the right front disc. In the end of the braking phase however, the temperature difference between left and right decreases significantly. A proper explanation for this has not been found.

A second observation is that two different thermal couples in the disc give rather homogeneous results even though there is some deviation. The difference between the measured disc temperatures at two locations tends to be at most 20 degrees. Both measurements tend to follow the same shape, which indicates a rather homogenous heating up of the disc in circumferential direction, as was expected. The heat soaking behaviour of the disc seems to be slightly influenced by the location of the thermal coupling. This is due to the fact the vehicle is standing still. Therefore, there is some variation in the amount of fresh air that reaches different parts of the disc. For example, if the thermal couple would be exactly at the location of the pad, the cooling would go slower as when the couple would be located near the ground where more fresh air reaches the disc. Generally they tend to follow each other though with temperature variations in the order of only 20 degrees.

Since the heat flow between the disc and the pad during the heat soaking phase is of interest, the temperature evolution of the disc at the pad location is of interest. Another SimAlp experiment is scheduled, during which 3 runs are performed with the same vehicle and brake system. During one of the heat soaking phases it is assured that the thermal couple in the disc is facing the pad. During one other run, the thermal couple is as far away as possible from the pad. This allows investigating the largest temperature difference that occurs in the disc during the heat soaking phase and indicates to what extend the lumped mass postulation is justified. This SimAlp investigation is discussed in Section 3.3.

A third observation is the cooling behaviour of the disc and pad. During braking, the disc temperature increases more than the pad temperature, resulting in a slightly higher disc temperature at the end of the braking phase. However, during the heat soaking phase, the disc cools down considerably faster than the pad. The total amount of thermal energy stored in the pad is negligible compared to the disc. Next to this, the pad only covers a small portion of the disc, while the disc covers the complete pad front surface. Due to these two facts it is a reasonable assumption that the disc is not influenced by the pad. The pad on the other hand, is dependent on the disc temperature, since heat exchange between disc and pad is an important heat input/output for the pad.

A forth, and very important observation is that the brake fluid at the left front wheel tends to reach a higher maximum temperature and tends to reach its peak later in time than the brake fluid at the right front wheel. This indicates that more energy is absorbed at the left front wheel. According to [5], the hydraulic response of the brake system is not instantaneous. It is dependent on many factors, like the vacuum booster, the brake fluid viscosity etc. A wheel that is connected to the primary circuit, tends to be actuated faster than a wheel which is connected to the secondary circuit. However, to avoid the difficulty of having to incorporate hydraulic lag of the secondary circuit, it is decided to only model the brake corner that is connected to the primary circuit. It should be taken into consideration that this is not necessarily the warmest brake corner and can be either the left or right front wheel, dependent on the vehicle type.

3.3. SimAlp Testing II

At a later stage, a second SimAlp test was planned. Use has been made of the occasion to investigate the influence of the location of the thermal couple in the disc on soaking behaviour. Three runs are performed. For the Left Front wheel, the location of the thermal couple during the heat soaking at standstill is noted down for each run. Next to this, it is ensured that during one of the runs, the thermal couple is located exactly at the pad location. The result of these 3 runs is shown in Figure 3.3a and Figure 3.3b below for the Front Left and for the Front Right wheel.

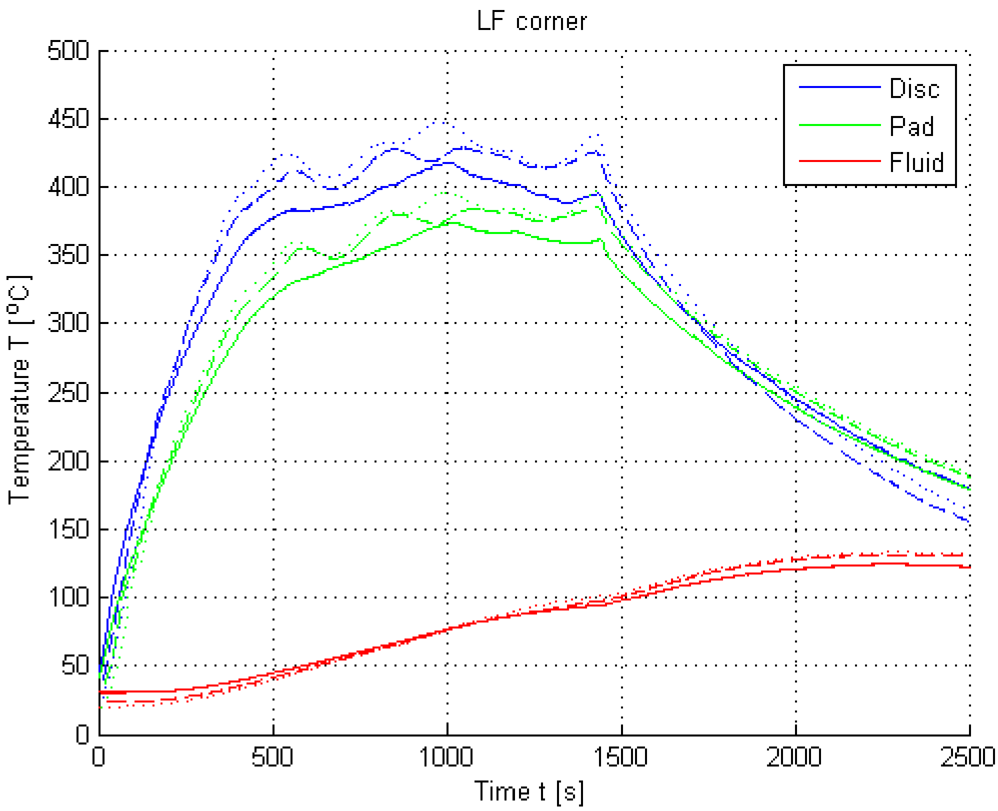

Figure 3.3a: SimAlp experiment with different locations of thermal couple during soaking (LF corner)

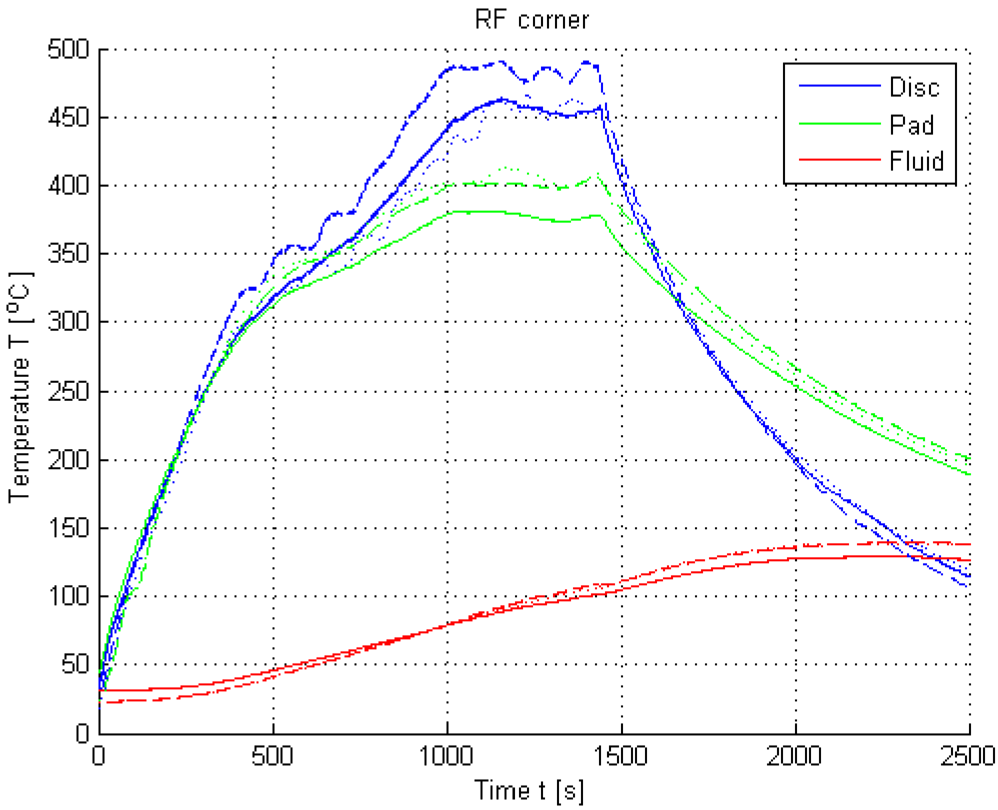

Figure 3.3b: SimAlp experiment with different locations of thermal couple during soaking (RF corner)

It is noted that the temperature evolution is rather consistent for each of the runs. The disc cools down slightly slower at the pad location, as has been expected. A more striking observation for this test is however, that the RF wheel now consistently reaches higher temperatures than the LF wheel. This is not corresponding to previous observations. However, the shape of the temperature evolutions in the LF and RF corner respectively are consistent with previous SimAlp tests.

4. Brake System Temperature Model

4.1. General Description

The Brake System Temperature Model or BSTM is created in Matlab/Simulink. The different parts of the brake system like disc, pad, caliper and fluid are modeled separately. In between each part, their respective heat exchange processes are added. Since only the part of the brake system in proximity of the wheel is of interest, no brake booster or pedal model is included. The input for the model is the primary circuit pressure, which is the pressure in the hydraulic circuit after the brake booster. This setup allows replacing/upgrading individual parts of the BSTM later on. In Figure 4.1 below, a flowchart of the BSTM is shown. The parts within the dotted lines form the Temperature Estimation Model, or TEM that estimates the temperature of different parts of the brake system. This flowchart shows the different subsystems of the BSTM and in which direction the information flows between these parts. The dark arrows show interaction with the surroundings. The input comes from the vehicle CAN-bus. The output comes from the HMI and can either be passive or active intervention.

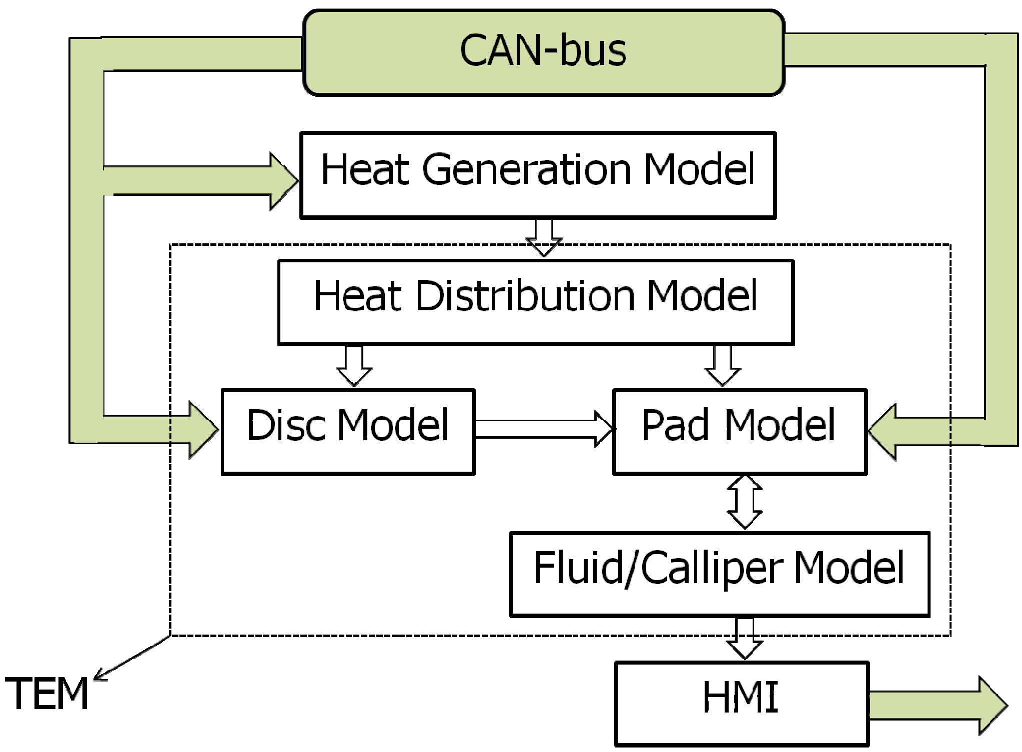

Figure 4.1: BSTM Flow Chart

4.2 Heat Generation

4.2.1. Heat Generation Modeling Options

In Section 2.2, the heat generation process during braking has been described. The next step is to make use of this knowledge in order to estimate the amount of heat generated in a vehicle. In order to estimate the amount of heat dissipated by the brakes, two distinct approaches are possible.

The first option is a Feed Forward Heat generation model. In this model, Equation 2.2b is integrated over time to estimate the amount of heat generated by the brake system. For this model the brake torque and the wheel rotational speed need to be known. The wheel rotational speed is readily available on the vehicle CAN-bus. For estimating the brake torque, the hydraulic pressure in the Primary circuit is used. From this pressure, the brake torque is estimated by means of the brake system parameters. This Feed Forward can give a very good estimation of the brake torque during normal operation conditions. One important limitation of this Feed Forward model is modeling the exact dynamics of the pad-disc contact surface. The Feed Forward Model assumes a constant friction coefficient and effective radius. However, these parameters are in fact variable in between different brake applications and even within one single brake application. The exact behavior of these parameters is very difficult to predict.

The second option is to run a vehicle dynamics model in real time parallel with the actual vehicle. From this vehicle dynamics model the amount of kinetic/potential energy dissipated by the brakes is determined. Since the brakes transfer the vehicle kinetic/potential energy into heat, the amount of energy dissipated by the brakes corresponds to the amount of heat generated during braking. This is done by comparing the measured vehicle deceleration with the deceleration that the vehicle would have if the brakes were not applied, based on the vehicle parameters and external conditions like: vehicle mass, rolling resistance, drag, internal friction, slope, vehicle speed and vehicle deceleration. One major concern in this approach is that many vehicle parameters cannot be known very accurately:

- Rolling resistance is influenced by the road surface and tire pressure
- Vehicle mass is influenced by the amount of fuel, passengers and luggage
- Vehicle drag is dependent on many factors:
    - Open window
    - Trailer
    - Bicycle carrier
    - Rooftop-box
    - …

Next to these vehicle parameters, the gradient of the road needs to be known. This is not available on the vehicle CAN-bus. Without knowledge of the slope it is not possible to estimate the amount of potential energy absorbed by the brakes during an Alpine descent.

4.2.2. Feed Forward Heat Generation Model

Time integration of Equation 2.2b gives the total amount of heat that is generated in one specific brake corner. On the vehicle CAN-bus signals for brake pressure and the wheel rotational velocity are available. Since all the brake parameters are known, the brake torque in one brake corner can be estimated from this according to:

$$T_{brake} = 2 \cdot p_{fluid} \cdot A_{piston} \cdot n_{pistons} \cdot \mu_{fric} \cdot r_{eff} \qquad \text{Eq. 4.1}$$

A very accurate Simulink model of the brake system is available within VCC. This model incorporates effects like hydraulic lag, caliper efficiency, brake fluid hose expansion and brake distribution between front and rear axle, which are not incorporated in the simplified formula above. Therefore, the VCC model will be modified for estimating the amount of energy dissipated by the brakes. CAN-bus signals of the wheel speed and primary circuit pressure, p_{fluid}, as are used as input. The outputs are the brake torque and the amount of energy absorbed for each brake corner.

According to [6] the friction coefficient is temperature dependent with a peak around 400 degrees. According to [13] the friction coefficient decreases linearly with time. However, in reality the friction coefficient is dependent on many other parameters as well, as will be investigated more thoroughly later on. In [20] a method for using a dynamic friction coefficient is proposed. This is not incorporated in the VCC brake model though.

4.2.2.1. Verification Testing

An accurate estimation of the amount of heat generated by the brake system is of crucial importance for the BSTM. If the input of the estimation model already shows significant deviation from reality, an acceptable estimation of the brake system temperature will be impossible. Therefore, the Feed Forward Heat Generation Model is verified by means of in-vehicle testing. The testing is performed at the IDIADA test-track in Barcelona, Spain.

A series of brake events is performed. During this braking, the CAN signals for the wheel speed and primary circuit pressure are recorded. These signals are used as input into a Simulink model that determines the Brake torque from the primary pressure and determines the amount of heat generated by integrating Equation 2.2b over time. Next to this, additional measurement sensors are used to accurately measure the vehicle speed and deceleration. Additionally, a pressure sensor is installed at each of the four wheels. These are used to verify the reliability of the signals on the vehicle CAN-bus.

4.2.2.2. Initial CAN-signal Investigation

As a first step, the quality of the CAN signals is investigated. The brake pressure from the external individual wheel pressure measurement sensors and the primary pressure signal that is available on the CAN-bus are compared for several of the test cases which are described below. For one of the tests, this comparison is shown in Figure 4.2 below. It can be seen that the CAN signal for primary pressure corresponds very accurately to the wheel pressure for one of the hydraulic circuits. For the secondary circuit there is some difference. This effect has also been noticed during other vehicle tests. Sometimes the secondary circuit receives more pressure, sometimes less. It has been decided to focus the BSTM on the wheel that receives the same pressure as the primary pressure for the sake of simplicity. It should be noticed however that this is not necessarily the brake corner that will reach the highest temperature!

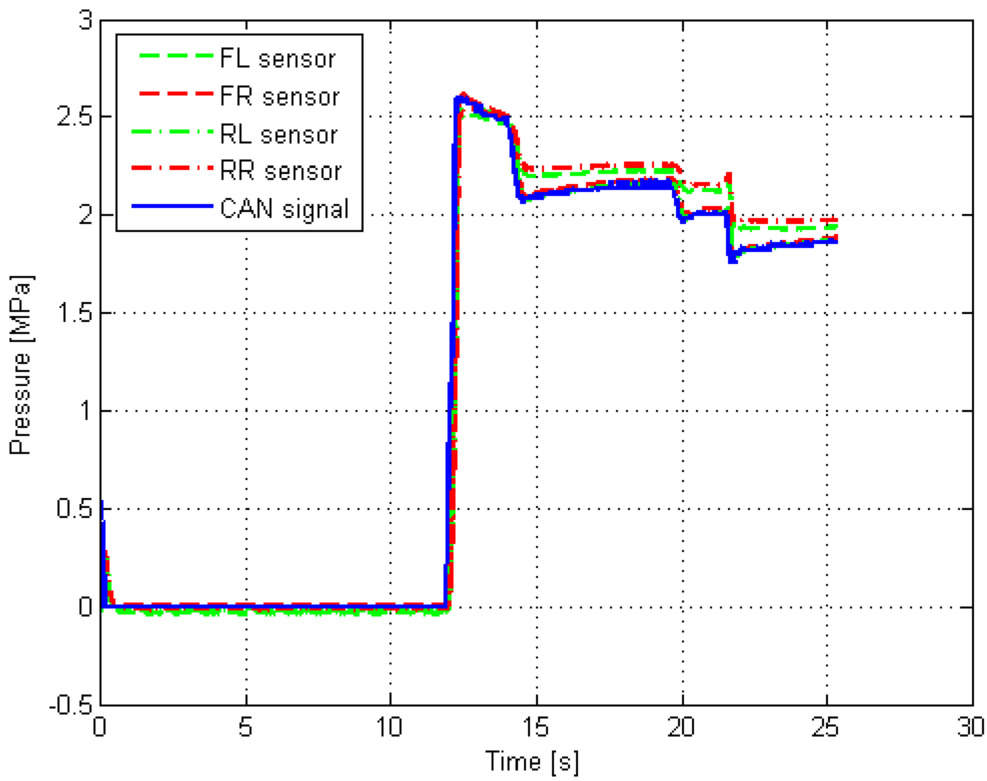

Figure 4.2: Pressure signal quality investigation for step brake input

Secondly, the same procedure is repeated for the velocity signals. Velocity on the CAN-bus and the velocity as measured by an external GPS device are recorded. On the CAN-bus, two velocity estimations are available: one based on the wheel rotational speeds, the other based on integration of the accelerometer. It is observed that both the CAN-signals and the GPS device give a very similar result.

Thirdly, the individual wheel speeds are compared to the vehicle speed. Again it is seen that all the signals contain little noise and show very little deviation with respect to the expected values.

Fourthly, the vehicle acceleration signal is investigated. The vehicle acceleration is measured by means of an external sensor. On the CAN-bus, 2 signals are available for the vehicle acceleration: The output from an accelerometer and a signal with a lower resolution that is based on the derivative of the wheel speed signal. It is observed that a small offset is present between the different signals. This is probably due to a small calibration error in the accelerometers and should be taken into account. However, when braking, this offset is negligible. Besides this small offset the acceleration signals are deemed reliable. The different acceleration signals for one of the tests are shown in Figure 4.3 below.

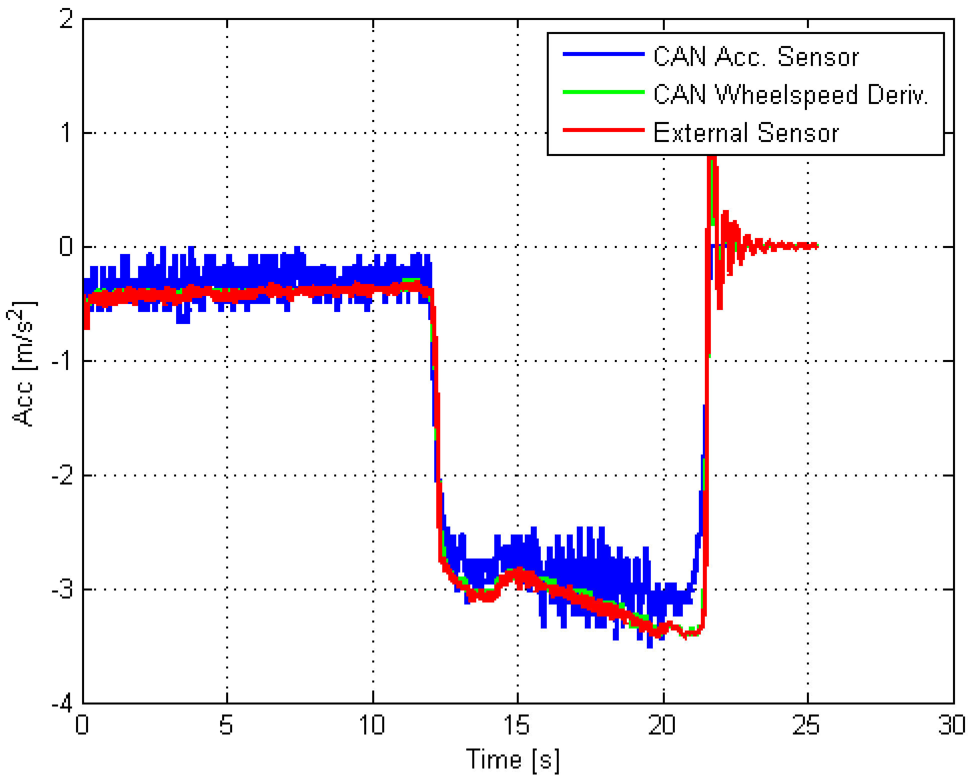

Figure 4.3: Acceleration CAN signal quality investigation

4.2.2.3. Performance of Feed Forward Heat Generation Model

Once it is determined that the CAN signals can be relied upon, the actual verification of the Feed Forward Heat Generation Model can be initiated. For verifying the Feed Forward heat generation model, the following test-series is performed:

For each run the vehicle is accelerated to above 100 km/h and starts coasting down in neutral gear. This is to avoid interference from the power train. Since the exact vehicle mass is measured at the start of the test and the vehicle's aerodynamic parameters are known, the theoretical vehicle deceleration due to drag, internal friction and rolling resistance can be matched very accurately to the measurements. This allows writing down a very exact formula for the drag forces as a function of velocity. The drag forces can be written as:

$$F_{drag} = \frac{1}{2} \cdot C_d \cdot \rho \cdot A_f \cdot v^2 + C_{fric} \cdot v + m \cdot g \cdot f_r \qquad \text{Eq. 4.2}$$

The work done by the drag forces can subsequently be written as:

$$W_{drag} = \int F_{drag} \cdot v \cdot dt \qquad \text{Eq. 4.3}$$

The rolling resistance and internal friction coefficient are tuned in order to exactly match the model to the experimental coasting down behavior for the specific vehicle and underground. In Figure 4.4 the measured and theoretical velocity against time is shown. It was found that the theory matches the experimental data very accurately for a rolling resistance coefficient, f_r, of 0.014 and an internal friction coefficient, C_{fric}, of 4 N/m/s.

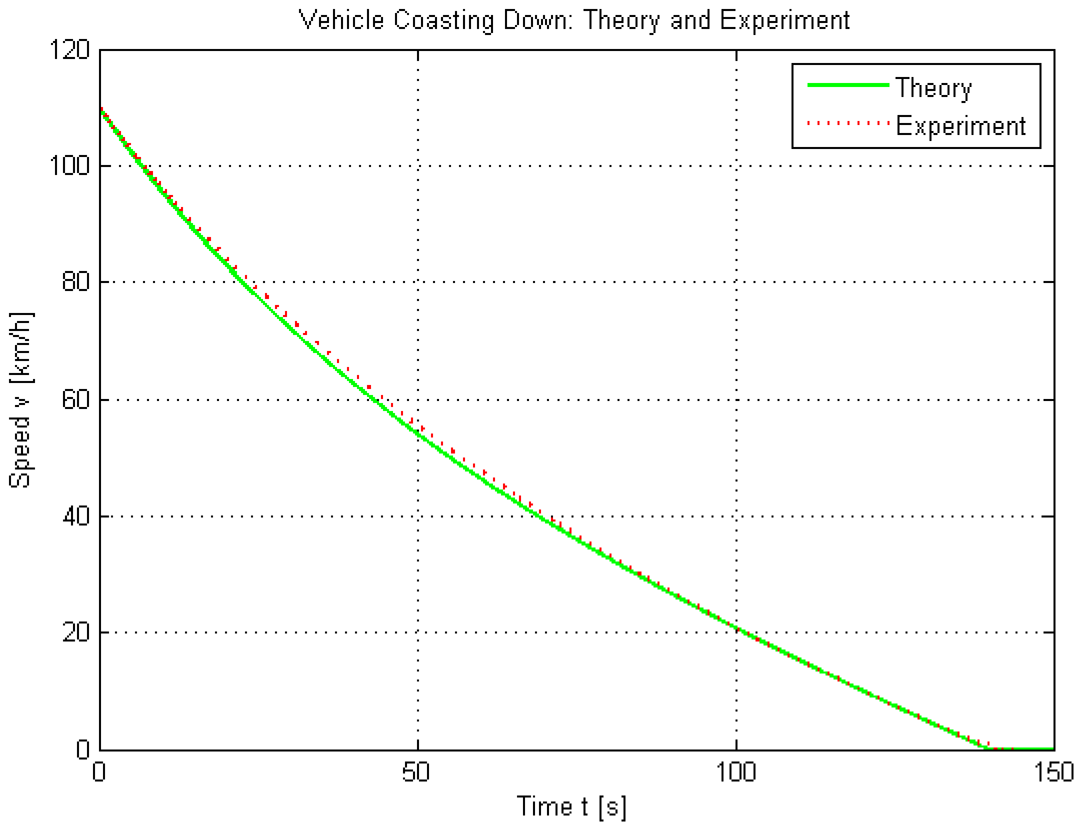

Figure 4.4: Coasting down in neutral gear: theory and experiment

During the subsequent runs, the driver coasts down the vehicle from above 100 km/h. As soon as the vehicle speed drops below 100 km/h, the driver applies the brakes. This procedure is repeated for different levels of constant braking, as well as for linearly increasing braking and highly transient brake pulses. From the recorded CAN-signals the amount of energy absorbed by the brakes is estimated by means of the Feed Forward Heat Generation Model

Since the braking happens in a very controlled manner, the amount of energy absorbed by the brakes can be determined theoretically as well. The work done by the drag forces can be determined accurately by combining Equation 4.2 and Equation 4.3 above.

$$W_{drag} = \int \left(\frac{1}{2} \cdot C_d \cdot \rho \cdot A_f \cdot v^2 + C_{fric} \cdot v + m \cdot g \cdot f_r \right) \cdot v \cdot dt \quad \text{Eq. 4.4}$$

During each run the total kinetic energy of the vehicle is reduced to zero. According to the first law of Thermodynamics, the amount of work done is equal to the change in kinetic energy.

$$W_{tot} = \Delta E_{kin} = W_{drag} + W_{brake} \quad \text{Eq. 4.5}$$

Since the drag force for each speed is known, the amount of work done by the drag forces can easily be determined by looking at the vehicle velocity profile over time and inserting this into Equation 4.4. During each test the total amount of work done is equal to the sum of the work done by the drag forces and the work done by the brakes. Since the initial and final velocities for each run are the same, the total loss of kinetic energy is the same for each run and is given by:

$$\Delta E_{kin} = \frac{1}{2} \cdot m \cdot v_{init}^2 + J_w \cdot \omega_{init} \quad \text{Eq. 4.6}$$

When inserting the velocity profile into Equation 4.4 for the first run, it is seen that the work done by the drag forces corresponds to the initial kinetic energy of the vehicle, as expected.

For the test vehicle, the total amount of work done by the drag forces to coast down the vehicle from 100 km/h to 0 km/h is 846430 J as calculated by means of Equation 4.4. The initial kinetic energy of the vehicle at 100 km/h, this is found to be 846701 J according to Equation 4.6.

In which:

m = 2194 kg

J_w = 3.0809 kg.m^2/rad^2 for the complete chassis (all 4 wheels)

Since both numbers lie very close to each other, with only a 0.03 % difference, it can be seen that the drag work has reduced the vehicle kinetic energy to zero.

According to Equation 4.5, the total work done can be described as the sum of the work done by the drag force and the work done by the brake force. This corresponds to the total change in kinetic energy of the vehicle. Since the vehicle comes to a full stop at the end of each test, the change in kinetic energy corresponds to the initial kinetic energy. In the resulting Figures below, this is represented by the green dotted line. Ideally, the total work done by the brake system as estimated by Feed Forward Model should be equal to this at the end of each maneuver.

The next step is to determine the heat generation rate of the brake system. This is done by the Feed Forward Model which uses the CAN signals for wheel speed and brake pressure. The rate of heat generation is determined by means of Eq. 2.2b. Integration over time of this gives the total amount of heat generated in one brake corner. Additionally, the external sensors are used to determine the rate of heat generation. At each point in time, the vehicle deceleration due to drag/friction/rolling resistance is determined by rewriting Equation 4.2 and inserting the instantaneous velocity. This deceleration is deduced from the measured vehicle deceleration in order to obtain the deceleration due to braking. By multiplying this deceleration with the vehicle mass, the brake force is obtained. This is the Feed-Back Heat Generation Model which will be discussed later on. Only this time the acceleration signal is taken from an external sensor which is calibrated better than the in-vehicle accelerometer. By multiplying this with the vehicle speed, the heat generation rate of the brake system is obtained. The time-integration of this gives the again total heat generated by the brake system.

In Figures 4.5 and 4.6 below, the resulting heat generation rate and total heat generated by the brake system are shown as determined by the Feed-Forward and the Feed-Back Heat Generation Model. Additionally, the total heat generated is shown as determined by the change in kinetic energy. This is shown for several of the tests. It can be seen that all results lie very close to each other, especially for steady state braking situations. The total heat generated lies very close to the total heat generated as calculated from the vehicle kinetic energy. However, for transient braking, the Feed-Forward Model appears to slightly overestimate the total heat generated. However, further investigation reveals that this is probably due to the tire, suspension and chassis elastic deformation. When applying the brakes, the vehicle does not decelerate instantaneously. During the initiation period, the brake energy is used to build elastic deformation in the tires and chassis. Only when the tire and chassis have deformed sufficiently, the friction forces between tire and road start decelerating the vehicle. Therefore, the Feed Forward Model appears to overestimate the heat generated for transient braking. However, in fact the Feed-Back Model and the change in kinetic energy underestimate the heat generated for transient braking.

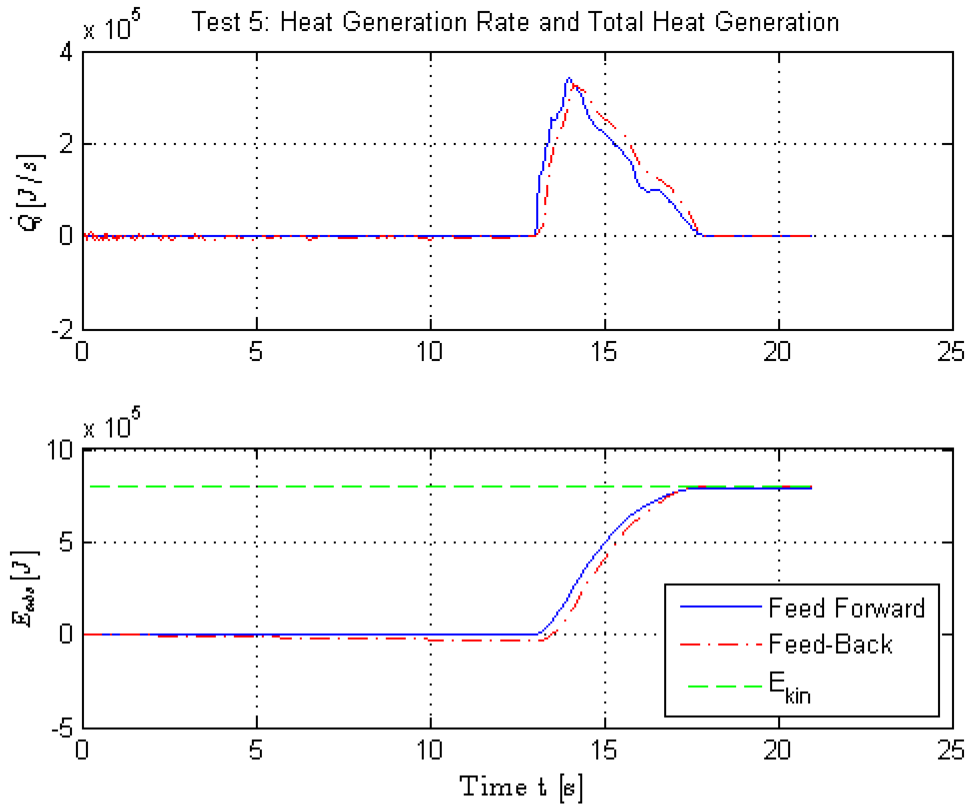

Figure 4.5: Heat generation for Feed Forward and Feed-Back Model (Constant pressure braking)

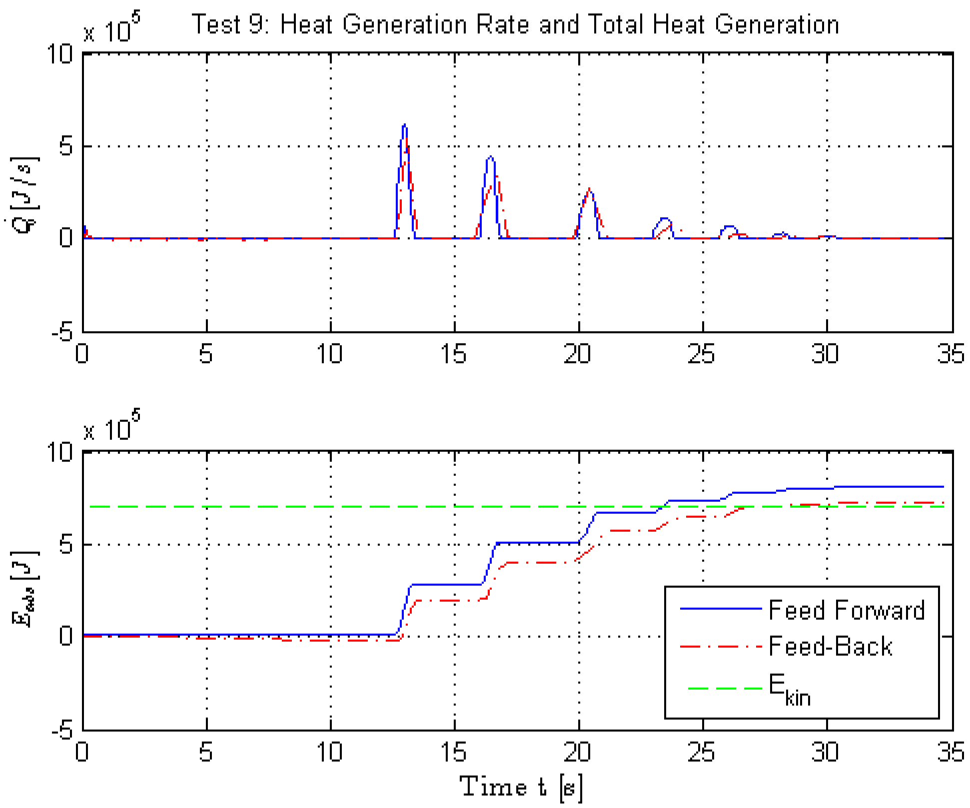

Figure 4.6: Heat generation for Feed Forward and Feed-Back Model (Transient braking)

4.2.3. Friction Coefficient Investigation

One important limitation of the Feed Forward Heat Generation Model is the friction coefficient. The friction coefficient is in reality dependent on many variables. In the Feed Forward Model a constant friction coefficient is assumed. In the following section it is investigated more thoroughly how the friction coefficient varies in reality.

4.2.3.1. Brake Pad Friction Behaviour

By means of old AMS tests it can be observed that the friction coefficient is not constant throughout one brake application. Based on the pad type, the friction coefficient can be progressive, degressive or constant. In Figure 4.7 below, an example of an old AMS test is shown, clearly showing the pad progressivity for the front pad, and pad degressivity for the rear pad at high temperatures.

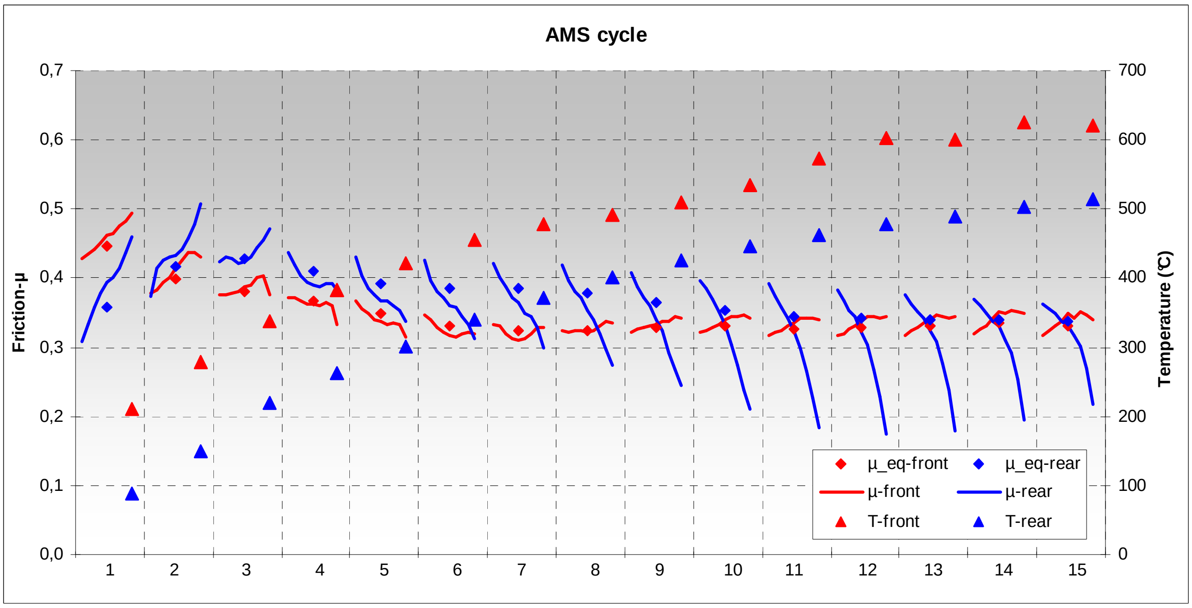

Figure 4.7: Friction coefficient during AMS test

Next, the influence of this pad progressivity on the Feed Forward Model is investigated by looking deeper into the IDIADA test data. In Figure 4.8 below, a constant pressure braking during the IDIADA testing mentioned above is plotted. The blue line is the brake force, as estimated from the brake pressure, the Feed Forward Model. The green line shows the brake force estimation, based on the vehicle deceleration, the Feed-Back Model. In this clearly the progressivity of the brake pad can be seen. Throughout the constant pressure brake application, the actual brake force increases linearly. This is due to an increase in friction coefficient throughout the brake application.

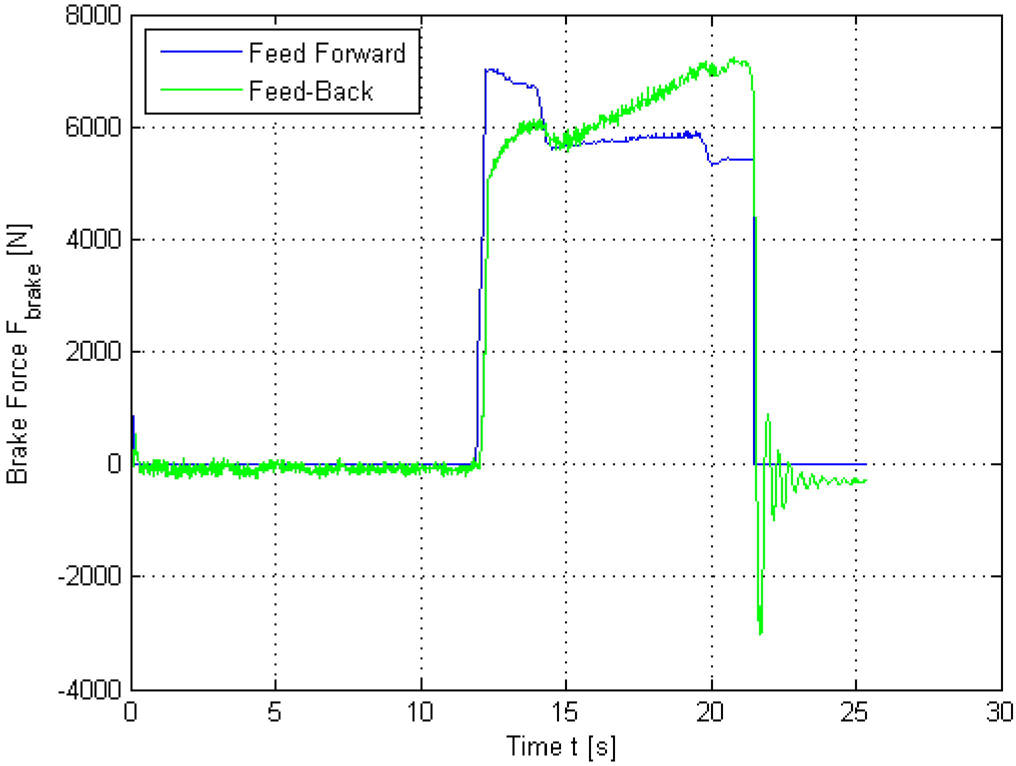

Figure 4.8: Brake force for constant pressure braking according to Feed Forward and Feed-Back Model

From Figure 4.8 above it can clearly be seen that the friction coefficient increases throughout the brake application for the specific pad which has been used during the test. The brake force estimation based on the vehicle deceleration (Feed-Back Model) shows an increase over time, while the brake force estimation based on the brake pressure (Feed Forward Model) and with the assumption of a constant friction coefficient is constant.

4.2.3.2. Temperature Dependency

In Figure 4.7 above, not only the variation of friction coefficient within one brake application is shown. It can also clearly be seen that the friction coefficient changes with brake initiation temperature. How and to what extend the average friction coefficient varies with temperature is dependent on the type of pad.

4.2.3.3. Brake History Dependency

The variation in friction coefficient is also investigated for the SimAlp testing performed at the Hällered test track. Since the test is run at constant speed the acceleration is roughly zero. Therefore, the brake force during a SimAlp test can be calculated as:

$$F_{brake} = F_{push} - F_{resist} \qquad \text{Eq. 4.7}$$

In Figure 4.9 below, the ratio between brake pressure and brake force is shown throughout 3 different SimAlp tests which have been performed without replacing the pads in between each test. It can be seen that for the new pads, the friction coefficient drops significantly at high temperature since the required pressure to maintain a specific brake force increases from 400 Pa/N to 600 Pa/N. For the second run with the same pad pair the friction coefficient decreases already less. For the final run, the friction coefficient stays more or less constant. This clearly indicates the dependency of the friction coefficient on pad history.

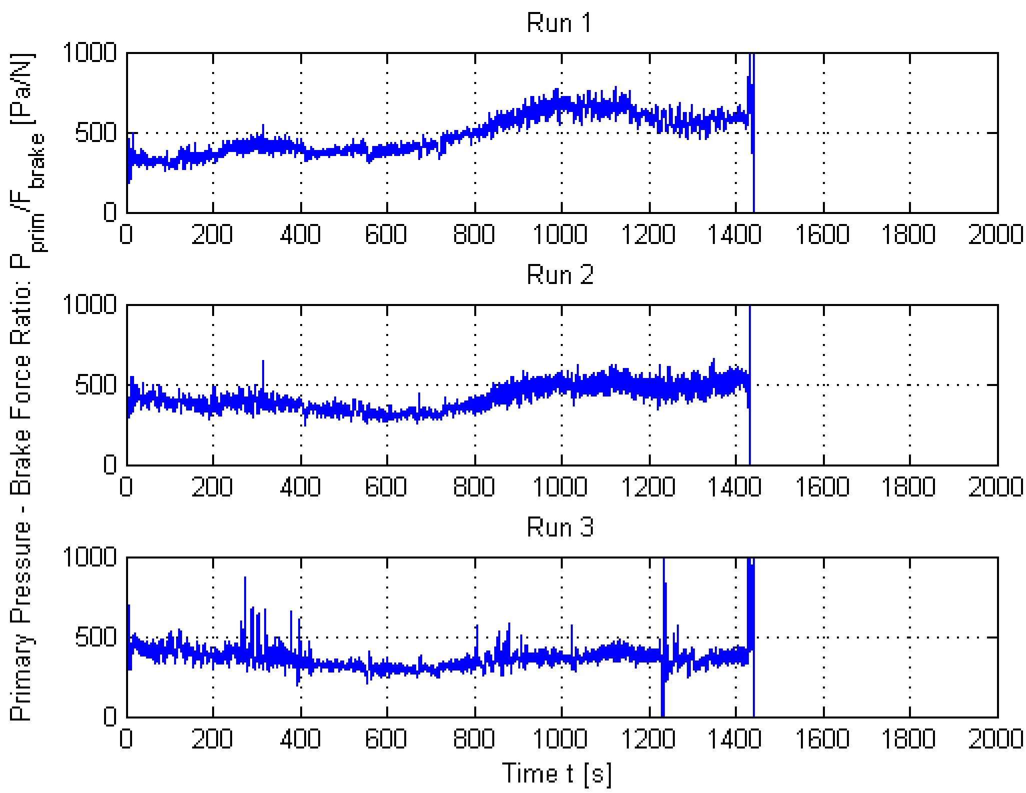

Figure 4.9: Influence of brake history on friction coefficient

During another SimAlp test, two runs have been performed with the same pad type. One of the runs has been performed with a completely new pad and one run has been performed with a pad that has been worn 50 %. This has been established by milling away half of the pad material. The primary pressure – brake force ratio for both SimAlp runs is shown below. It can clearly be seen that for the half worn pad, the friction coefficient drops at high temperatures while the new pad has a much more constant friction coefficient. This indicates that the pad properties are not constant throughout the complete pad life. It can also be seen that the worn pad was not bedded in properly. This can be seen in the beginning where a decrease in friction coefficient occurred. After a while however, the friction coefficient recovers and settles at the same value as the new pad. At high temperatures the friction coefficient of the worn pad decreases again.

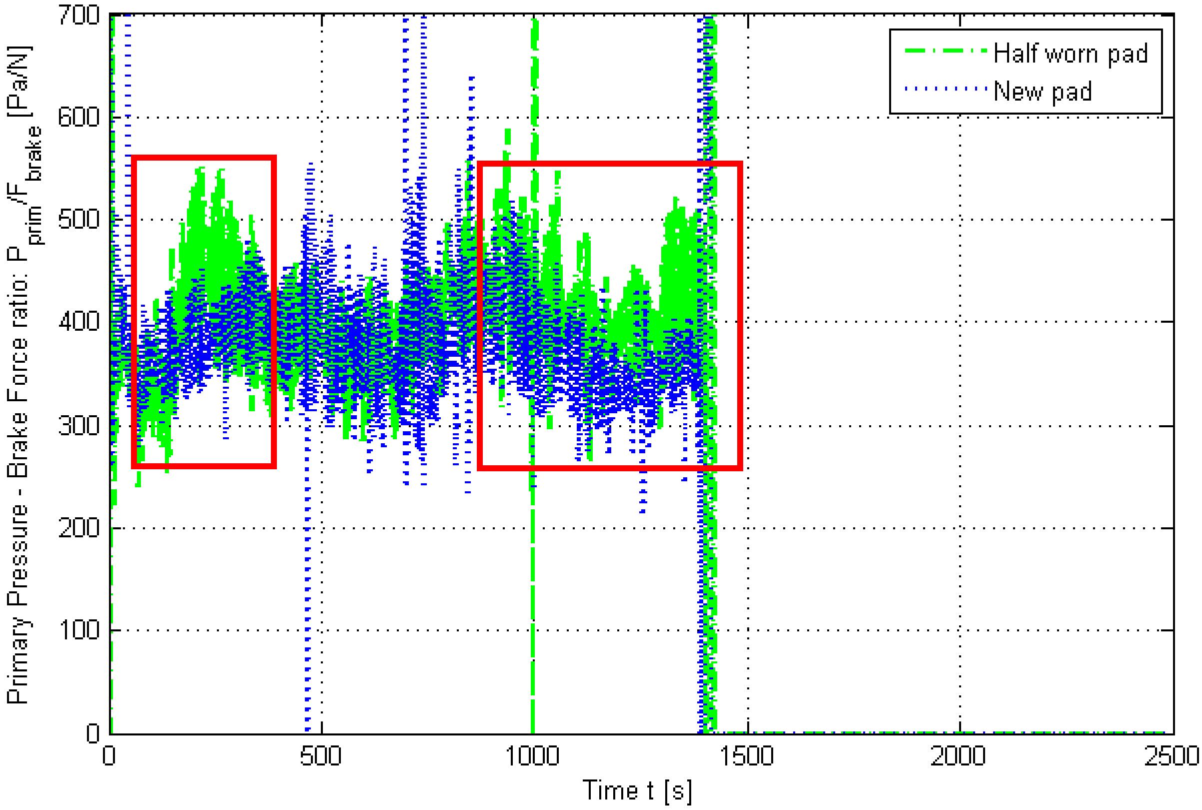

Figure 4.10: Investigation of friction coefficient for new and half worn pad

From all the Figures above it is noticed that the pad friction coefficient is highly variable, with a dependency on amongst others:

- Temperature
- Clamp Force
- Pad history
- Sliding velocity

Therefore, it is very difficult to obtain a theoretical friction coefficient model that captures all this. This forms an important limitation for the Feed Forward Heat Generation Model and clearly indicates the need for a Feed-Back Heat Generation Model. This is investigated in the next section.

4.2.4. Feed-Back Heat Generation Model

The Feed-Back Heat Generation Model based on the vehicle deceleration is shown in the previous section to give accurate results IF idealized conditions are present. The heat generation model based on vehicle deceleration needs to know following parameters:

- Road gradient
- Vehicle mass

- Vehicle drag properties
- Power train torque
- Vehicle deceleration

Of these parameters, an estimation of the power train torque is available on the vehicle CAN-bus. The vehicle deceleration is measured directly with a deceleration sensor and is available on the CAN-bus. The slope, vehicle mass and vehicle drag properties however, are unknown:

- The slope is not measured directly
- The vehicle mass and drag properties are dependent amongst others on:
    - Passengers and luggage in vehicle
    - Trailer
    - Rooftop box

On the vehicle CAN-bus, the vehicle deceleration is determined by a deceleration sensor and by taking the derivative of the wheel speed. The slope could theoretically be deduced from the difference between these two according to:

$$\theta = \sin^{-1}\left(\frac{acc_{slope}}{g}\right) \qquad \text{Eq. 4.8}$$

In which:

$$acc_{slope} = acc_{wheelsensor} - acc_{sensor} \qquad \text{Eq. 4.9}$$

In current production cars, the vehicle deceleration signal often shows an offset. When this happens, it becomes more difficult to give a reasonable estimate of the slope and consequently the model becomes unusable for estimating the heat generation during an Alpine descent. Solving this problem this lies outside the scope of this project. Therefore the Feed-Back Heat Generation Model is developed for flat road only. In order to make this model usable for an Alpine descent situation, the road gradient should be known.

The vehicle mass and drag properties are unknown as well, since these are highly dependent on the number of occupants and external devices attached to the vehicle, like rooftop boxes or trailers.

The development of an estimation model for these parameters lies outside the scope of this project. Therefore, in order to verify the possibility of using a Feed-Back Heat Generation Model, it is assumed that both the vehicle mass and vehicle drag properties are known. This is achieved by weighting the vehicle at test initiation. The vehicle drag properties are taken from available vehicle data and no external device that would corrupt these parameters is attached to the vehicle.

The FB Heat generation model estimates the heat according to:

$$\dot{Q}_{FB} = F_{brake} \cdot v \qquad \text{Eq. 4.10}$$

In which the brake force, F_{brake} is estimated according to:

$$\begin{aligned} F_{brake} &= m\dot{Q}_{FB} = F_{brake} \cdot acc + T_{powertrain}\dot{Q}_{FB} \\ &= F_{brake} \cdot R_w - F_{drag} \end{aligned} \qquad \text{Eq. 4.11}$$

The fraction of this brake force that is attributed to one specific brake corner is derived from the brake system geometry.

4.3. Disc – Pad Heat Distribution

For the heat distribution model, it is decided to make use of an imperfect contact assumption. The theoretical value for σ has been determined by means of Equation 2.3. This σ represents the fraction of the heat generated during a brake event that is absorbed by the disc. The remaining part is absorbed by the pad.

Equation 2.3 gives for the material properties that are available for the brake system used, a fraction of 98,85% that goes to the fluid and consequently a fraction of 1,15% of the heat that goes into the pad. These numbers are slightly different from the rough estimation of 93% of heat that goes into the disc according to [7]. Therefore, it is decided to make use of a correction factor by which 1-sigma (fraction of the total heat that is absorbed by the pad) is multiplied. This correction factor is used to match the BSTM to test data.

4.4. Disc Temperature

As mentioned in Section 2.4, the disc is modeled as a lumped mass in which the thermal capacity is temperature dependent. The heat input into this lumped mass model is described according to Equation 2.6. The heat output of this lumped mass model is composed of convection and radiation. It is decided to implement this by making use of so called *S*-functions. The temperature drop of a lumped mass object can according to Newton's law of cooling be described as [5]:

$$\frac{T_2 - T_{amb}}{T_1 - T_{amb}} = e^{-S \cdot t} \qquad \text{Eq. 4.12}$$

In which S is the cooling coefficient. The cooling of the brake disc is assumed independent on the state of the brake pad. This is to simplify the model and is justified by the fact that the disc absorbs almost 100 times more energy than the pad. The disc cools down due to convection, radiation and conduction. While driving, convection is by far the dominant mode of heat loss. Therefore, radiation is assumed negligibly small.

From experimental testing, the *S*-value can be determined easily for different speeds. This is done by heating up the brake disc and subsequently driving at constant speed while continuously measuring the disc temperature. Five measurements, evenly spread in time are taken and for each velocity, the value for *S* is calculated be means of Equation 4.12 above for each combination of these 5 measurement points. The average from these 10 estimations for S is chosen. This process is repeated for standstill, 18 km/h, 36 km/h, 50 km/h, 70 km/h, 90 km/h and 150 km/h. In each time step the value for S is taken from a lookup table. With this value the decrease in temperature is determined. The eventual increase in brake disc temperature due to braking is added to this. Integration gives the complete temperature profile of the brake disc. In Figure 4.11 and 4.12 below the disc temperature evolution while driving at a specific speed is shown as well as the matched *S*-curve according to Newton's law of cooling. It can be seen that theory and experiment follow each other very closely. It is found that the cooling coefficient is roughly linearly related to vehicle speed.

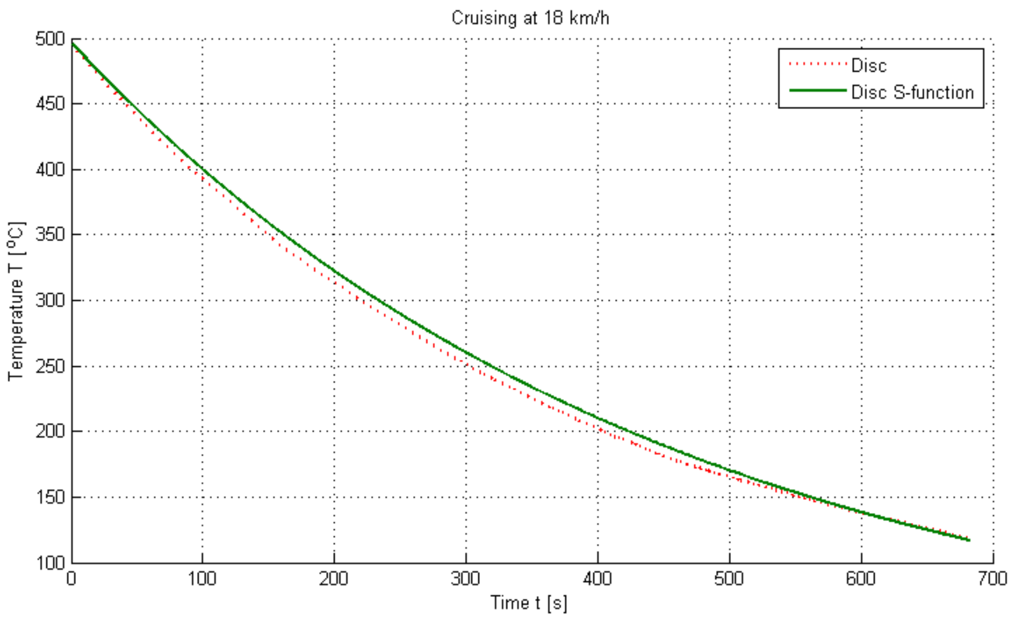

Figure 4.11: S-function and measurement while cruising at constant velocity

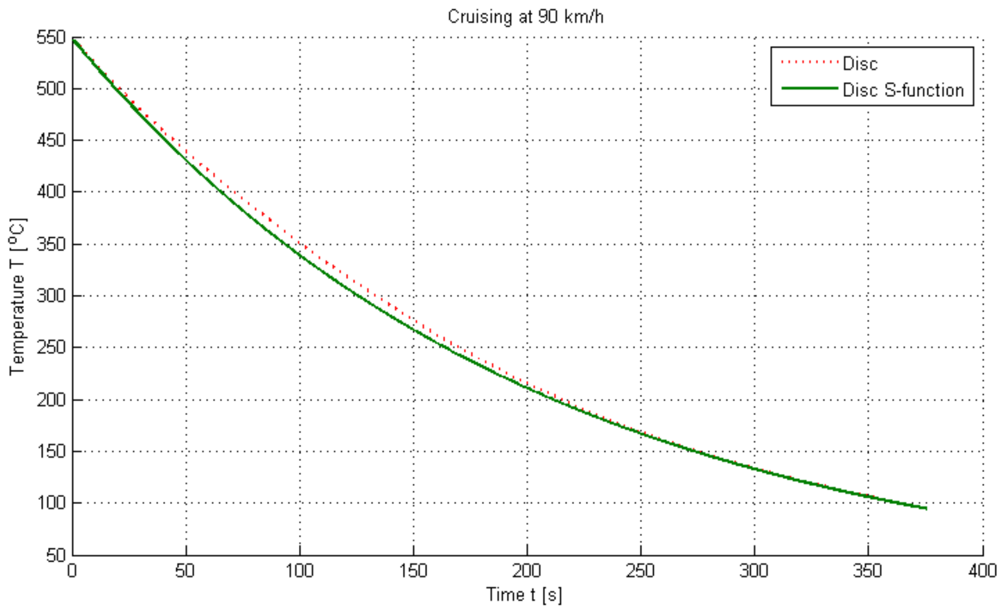

Figure 4.12: S-function and measurement while cruising at constant velocity

As can be seen in Figure 4.11 and 4.12 above, experiments have shown that the disc follows this natural logarithmic curve very closely. Only for heat soaking while standing still, a larger deviation was observed. This is due to the fact, that, while standing still, there is no more forced convection: only free convection. Therefore, the role of radiation in the cooling process is no longer negligible. At standstill it is found that only taking 40 % of the convective cooling and adding the radiation out of the disc, according to Equation 2.9, to this gives a good estimate for the disc temperature evolution. From 18 km/h onwards, only the S-curve is considered. The transition between both models is achieved by adding a smooth transition from 0 to 18 km/h in which the convective cooling is increased from 40% to 100% and the radiative cooling is reduced from 100 % to 0%.

4.5. Pad Temperature

4.5.1. Pad Front Surface Temperature

In Section 2.5 above the different possible temperature distributions on the pad surface have been discussed. A very promising model for heat input into the pad would be a combination of constant pressure and constant wear model. For low pressure braking, constant wear is used, for emergency braking constant pressure. For the first model, the heat input is constant over the entire pad surface. For the second model, the heat input increases linearly in radial direction. As will be explained below, it is decided to develop the constant wear model. According to the constant wear model, the heat input into the pad per unit area is given by:

$$\dot{q} = \frac{A_{piston} \cdot P_{fluid} \cdot v_{slide}}{A_{pad,\,front}} \qquad \text{Eq. 4.13}$$

4.5.2. Heat Flow Through Pad

The pad material acts as a buffer zone between the heat generation zone (disc-pad interface) and the brake fluid. Therefore, it is important to capture the heat flow towards the brake fluid through the pad well. For this reason it is decided to make use of a Finite Difference Model (FDM) for the pad.

In Section 3.1, a wind tunnel experiment is performed in which the temperature distribution at different locations of the brake system is investigated. During this experiment it came forward that the temperature of the caliper above and below the fluid is exactly the same. Even though the outer radius of the pad may be warmer than the inner radius, the heat tends to spread more evenly towards the back of the pad. In the back plate the temperature difference at different radii is diminished. Therefore, it is decided to develop the constant wear model. This will hardly influence the accuracy of the model but allows to make a 1-D model for the pad instead of a 2-D or even 3-D model. This will reduce the calculation effort of the model.

A Finite Difference Model is set up by means of the Energy Balance Method, as explained by [11]. In Figure 4.13 below a schematic depiction of the 1-D pad FDM is shown. The pad is divided into slabs. The total number of slabs is fixed. The thickness of each slab however, is variable. This allows easy implementation of the pad wear model. As the pad wears off, each slab becomes thinner.

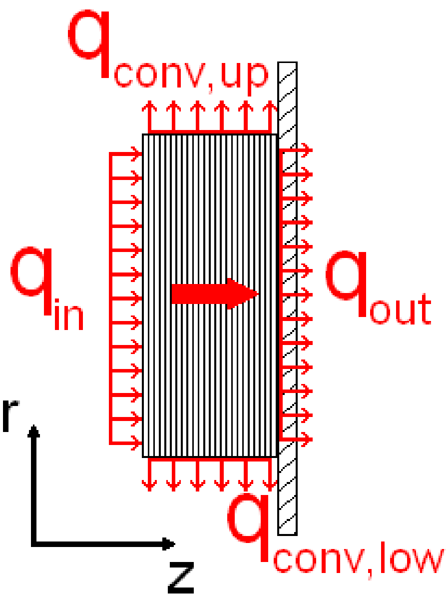

Figure 4.13: Pad Finite Difference Model

The Energy Balance Equation (EB) of each slab of pad material can be written as:

$$\rho \cdot c_p \cdot \Delta z \cdot h_{pad} \cdot \frac{dT}{dt} = \dot{Q}_{in} - \dot{Q}_{out} \qquad \text{Eq. 4.14}$$

The EB for the inner slabs (slab *m* out of $m_{\max}$ slabs) can be written as:

$$\begin{aligned} &\rho \cdot c_p \cdot \Delta z \cdot h_{pad} \cdot \frac{dT(m)}{dt} = \\ &k \cdot h_{pad} \cdot 1 \cdot \frac{T(m-1) - T(m)}{\Delta z} + \\ &k \cdot h_{pad} \cdot 1 \cdot \frac{T(m+1) - T(m)}{\Delta z} - \\ &\left(\dot{q}_{conv,upper} + \dot{q}_{conv,lower}\right) \cdot \Delta z \end{aligned} \qquad \text{Eq. 4.15}$$

The EB for the front slab (the first slab) can be written as:

$$\begin{aligned} &\rho \cdot c_p \cdot \Delta z \cdot h_{pad} \cdot \frac{dT(1)}{dt} = \\ &k \cdot h_{pad} \cdot 1 \cdot \frac{T(2) - T(1)}{\Delta z} + \dot{q}_{in} \cdot h_{pad} - \\ &\left(\dot{q}_{conv,upper} + \dot{q}_{conv,lower}\right) \cdot \Delta z \end{aligned} \qquad \text{Eq. 4.16}$$

The EB for the rear slab can be written as:

$$\begin{aligned} &\rho \cdot c_p \cdot \Delta z \cdot h_{pad} \cdot \frac{dT(m_{max})}{dt} = \\ &k \cdot h_{pad} \cdot 1 \cdot \frac{T(m_{\max} - 1) - T(m_{\max})}{\Delta z} - \\ &\dot{q}_{out} \cdot h_{pad} - \left(\dot{q}_{conv,upper} + \dot{q}_{conv,lower}\right) \cdot \Delta z \end{aligned} \qquad \text{Eq. 4.17}$$

The heat input into the front slab q_{in} consists of two parts:

- Heat transfer from/to the disc (Radiation in both directions)
- Heat input due to braking ($(1-\sigma) \cdot \dot{Q}_{LF}$)

The convective heat losses on the upper and lower surface are assumed zero and heat output at the pad backside through the pad back plate into the fluid/caliper assembly is modeled as conduction through a thermal resistance:

$$\dot{q}_{out->fluid} = \frac{T_{pad,back} - T_{fluid}}{R_{therm}} \qquad \text{Eq. 4.18}$$

Some heat that reaches the pad back plate does not travel to the fluid but instead travels directly to the caliper through the hammerheads. This heat flow is dependent on the temperature difference between the back plate and the caliper and is therefore limited by the capacity of the caliper to lose heat. This heat flow is modeled as heat flow out of the pad backside by means of convection. This is given by:

$$\dot{q}_{out->conv} = h_{dyn} \cdot v \cdot \left(T_{pad,back} - T_{amb} \right) \qquad \text{Eq. 4.19}$$

This way all the additional heat losses are modeled as an evenly distributed convective heat loss through the pad backside. This assumption deviates from reality but it has been found that doing so gives good correspondence with measurements.

4.6. Pad Wear

In Section 2.6 a more detailed investigation of pad wear has been described. Since the amount of testing and verification that is required for this model is not within the scope of this document, a different approach has been used. The developed pad wear model is based on the assumption that following information is available, as is expected to be on future VCC-models:

- Signal is given when new pads are being installed
- Signal is given when the pad has worn one millimeter
- Signal when there is only 2 millimeter of pad material left

These signals are used in a fuzzy logic algorithm to estimate the remaining pad thickness. In Section 2.6 it has been observed that for normal operating temperatures, the pad wear per unit energy absorbed is independent of speed. It is only influenced by brake pressure and transient behavior. A driver who applies the brakes more aggressively will have more wear than a driver who always smoothly increases the brake pressure to its steady state level. The influence of brake pressure in pad wear is easy to model even though it would require a significant amount of testing and verification. Incorporating the influence of the aggressiveness of the brake application would be more difficult. Therefore, a driver dependent pad wear model is implemented. In this it is assumed that the driver has a consistent drive style throughout the pads life.

This algorithm checks the amount of energy absorbed by the pad during the first millimeter of pad wear. As long as the pad wear is less than 1 mm, the pad is assumed to be new. Once the pad wear indicator crosses the sensor that indicates that one millimeter of pad material has worn off, the amount of energy absorbed by the pad during the first mm of pad wear is stored. The pad wear per amount of energy absorbed is simply the same throughout the rest of the pad life, according to:

$$z_{pad} = z_{pad,new} - \frac{E_{abs}}{E_{abs@1mm}} \quad \text{Eq. 4.20}$$

This model uses the assumption that the pad wear during the first mm is representative for the rest of the pad. As limitation it is implemented that the estimated remaining pad thickness cannot be smaller than the pad thickness at the second wear indicator. Once the second pad wear indicator is triggered, the pad thickness is set equal to this. Again the total amount of energy absorbed by the pad is logged. Now the pad thickness is calculated as:

$$z_{pad} = z_{pad,ind2} - \frac{\left(z_{pad,new} - z_{pad,ind1} - z_{pad,ind2}\right)}{E_{abs@ind2} - E_{abs@ind1}} \cdot \left(E_{abs} - E_{abs@ind2}\right) \quad \text{Eq. 4.21}$$

4.7. Fluid/Caliper Temperature

As has been discovered during the Wind tunnel experiment the fluid and surrounding caliper can be modeled as one piece. It is decided to model this as a lumped mass with an empirical value for $m \cdot C_p$. This allows easy empirical matching of the fluid temperature evolution with respect to the measurements. The thermal resistance of the back plate, the thermal capacity of the fluid/caliper ($m \cdot C_p$) and the heat losses of this object are important tunable parameters in the BSTM.

4.8. Heat Flow in BSTM

In Figure 4.1 above an overview of the BSTM can be seen. In this model, there are several parameters which can be tuned in order to match the BSTM to reality. An overview of these parameters is given in Table 4.1.

Parameter	**Unit**
Disc Mass Correction Factor	-
Heat Partition Correction Factor	-
Emissive coefficients of Disc and Pad	-
Pad backplate Thermal Resistance	$m^2 \cdot K / W$
Dynamic Convective Coefficient, Pad	-
Static Convective Coefficient Fluid	-
Fluid/Caliper Thermal Capacity	J / K
Dynamic Convective Coefficient, Fluid	-

Table 4.1: BSTM Tunable Parameters

5. Matching of Temperature Model to Measurements

The BSTM model shown above is matched to the measurements. This is done in multiple phases. In following section the sequence in which this matching is performed is described. In each step, a specific part of the model is 'isolated' from the rest by making use of forced boundary conditions and matched to the measurements. Once each part is matched to reality, these parts are combined and verified.

5.1. Heat Soaking Matching Process

Firstly, the static heat soaking is matched. This is done for the vehicle at standstill. This eliminates the influence of dynamic convection.

5.1.1. Brake Disc

Since the total energy of the pad is negligible compared to the disc, the influence of the pad on the disc temperature is neglected. Therefore, the temperature evolution of the disc during heat soaking is determined by means of the S-functions explained in Section 2.4.

5.1.2. Brake Pad

The temperature distribution over the pad material at the end of the braking situation is used as an initial condition. The measured brake fluid temperature and the measured disc temperature during the soaking process at standstill are used as forced boundary conditions. The goal is to match the pad temperature evolution during soaking of the BSTM to the measurements. This is done by tuning the heat input and heat output of the pad material during the heat soaking phase.

Stage 1: Pad front surface temperature

Firstly, the pad heat input at the front is matched. The measured pad back temperature is used as a forced boundary condition as well as the disc temperature. In reality, the heat flow from disc to pad is complex process of radiation and conduction. By means of iteration it is found that modeling the heat flow between disc and pad as radiation only gives a good match between the model and the measurements. By setting the disc temperature as forced boundary condition it is assured that the correct heat input into the pad is given once the emission coefficients are tuned. It should be taken into account that it is not exactly known where the thermal couples are located in the pad. Therefore, this matching is done rather roughly. The temperature evolution of the model should follow the same shape as the measurements. A small offset is acceptable.

The heat flow between disc and pad consists of two parts: radiation from the disc to the pad and radiation out of the pad. Since the pad-back temperature evolution is used as forced boundary condition, the amount of heat leaving the pad front surface by means of conduction throughout the pad material is correct as well. The emissivity of the pad and the emissivity of the disc are tuned iteratively until the pad front surface temperature evolution matches the measurements.

Stage 2: Pad back surface temperature evolution

Once the temperature evolution of the pad front is tuned during the heat soaking at standstill, the temperature evolution of the pad back is to be tuned. This is done by setting the fluid temperature evolution and the pad front temperature evolution as a forced boundary condition. The pad back temperature evolution at standstill is dependent on the thermal resistance between the pad backside and the fluid. This resistance is tuned until the pad backside temperature evolution of the model matches the measurements.

5.1.3. Brake Fluid

In the previous step, the measured fluid temperature evolution has been used as a forced boundary condition in order to tune the heat input and output of the pad during heat soaking phase. The next step is to match the fluid temperature evolution. This time the measured pad backside temperature evolution is used as a forced boundary condition. The fluid initial temperature is set at the value which occurred at the end of the heat generation phase.

Stage 1: Static convection

The heat input from the pad into the fluid is known, since both the pad back temperature and fluid temperatures are measured and the conductive resistance between both instances has been determined. Therefore, the static convection can be determined by looking at the time at which the fluid reaches its peak temperature. At this point in time, the heat flow from the pad to the fluid is equal to the fluid heat loss due to static convection (since dT/dt = 0 at this point in time). The static convection is given by:

$$\dot{Q}_{conv,stat} = h \cdot A \cdot \left(T_{fluid} - T_{amb}\right) \qquad \text{Eq. 5.1}$$

Since the geometry of the fluid assembly is very complicated, $h \cdot A$ is replaced by an empirical factor *H*. From the measurement data, the time at which the fluid reaches its maximum and the corresponding fluid temperature is read. At this point the total incoming heat equals the outgoing heat. The incoming heat is determined by looking at the respective temperatures of the pad backside and the f1uid at this time as well as the thermal resistance between both, which has already been determined previously. By inserting these data into Equation 5.1 above, the static convective coefficient H is determined.

Using this static convective coefficient ensures that the BSTM fluid temperature evolution reaches its maximum at the same point in time as the measured values.

Stage 2: Matching of $m \cdot C_p$

Once the static convective coefficient is determined, $m \cdot C_p$ of the fluid assembly is tuned. Tuning this will change the height of the BSTM fluid temperature evolution. This is simply a matter of iterating until the fluid temperature curve of the BSTM follows the measured values during heat soaking at stand-still. After these steps, the soaking behavior of the brake system has been completely tuned during the soaking at standstill.

5.2. Heat Generation Matching Process

Once the heat soaking behavior of the brake system has been determined, the heat generation behavior can be matched. Just like during the heat soaking phase, each part of the brake system is isolated by making use of forced boundary conditions. This allows determining the correct heat flow into and out of each part. The brake phase has been split up into two different 'sub-phases': Normal operating temperatures and high operating temperatures.

5.2.1. Normal Operating Temperatures

Stage 1: Pad Front Temperature Evolution

In order to match the pad front temperature evolution, the measured disc temperature evolution and the measured pad back temperature evolution are used as a ‘forced’ boundary condition. Subsequently the heat partition coefficient is optimized until the slope of the pad front temperature evolution is the same as the measurements.

Stage 2: Disc Temperature Evolution

Once the heat partition coefficient has been tuned, the exact heat input into the disc is known. Now it is a matter of tuning the mass of the disc lumped mass model until the slope of the disc temperature evolution follows the measurements. No boundary conditions are needed here, since the disc temperature evolution is independent of other components of the brake system. It should be taken into account that this tuning is done for a new disc. After the course of time, the disc will wear off, reducing its mass. This will alter the behavior of the disc and calls for the need to implement a disc wear model. This has not been done within the timeframe of this project.

Stage 3: Pad back temperature evolution

Next, the heat flow out of the pad is tuned. The measured pad front temperature evolution and the fluid temperature evolution are used as a forced boundary condition. The dynamic convection coefficient is tuned until the model matches the measured pad backside temperature evolution. After this tuning, the dynamic convection has been tuned for 36 km/h. In the current model a linear relation between dynamic convection and speed is assumed. This might induce some error. However, during testing, it seemed that this error is negligible.

Stage 4: Fluid temperature evolution

The final tuning that is required for normal operating temperatures is matching the fluid temperature evolution. Since the fluid heat input and output have been matched completely for static conditions, the only factor left to match is the dynamic convective coefficient out of the fluid. In order to do so, the pad backside temperature evolution is used as a forced boundary condition. The fluid dynamic convective coefficient is tuned until the fluid temperature evolution matches the measurements.

After this step, the model has been completely matched to the measurements for normal operating temperatures!

5.2.2. High Operating Temperatures

For high operating temperatures, the situation becomes significantly more complex. Firstly, friction coefficient of the pad may alter at high temperatures, changing the total heat input. Secondly, and more importantly, the material properties change at high temperatures. For both the pad and the disc, the thermal capacity and conductivity are temperature dependent. This significantly complicates the problem. To limit the amount of tunable variables, but at the same time obtain reasonable accuracy, it is decided to only alter the material thermal capacity.

Stage 1: Pad temperature evolution at high temperatures

Matching the pad model to the measurements at high temperature is difficult. The specific heat capacity and the thermal conductivity of the pad material are likely to be temperature dependent. However, these curves are not known. In Figure 5.1 below, an example of thermal capacity and conductivity for a brake pad material is shown [6]. These curves are for AP Racing brake pads however, not for normal passenger vehicle brake pads. Therefore it is likely that these curves will deviate significantly for other brake pads.

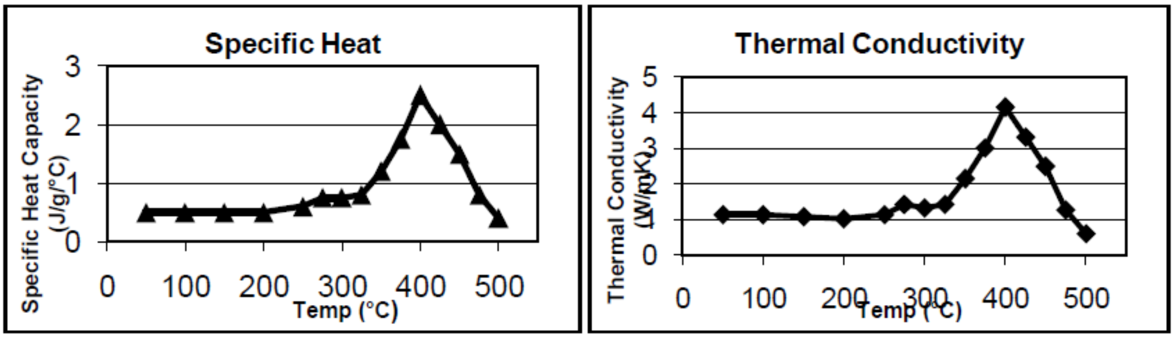

Figure 5.1: Specific Heat and Thermal Conductivity for AP racing brake pads [1]

Since it is hardly possible to come up with a proper estimation of these curves for normal brake pads, it is decided to approach this by reducing the amount of heat entering the pad. At a temperature of 400 degrees it is assumed that 40% of the energy 'dissolves'. By doing so, the pad temperature evolution at high temperatures also matches the measurements within a reasonable range, without the need of incorporating the temperature dependency of the pad material properties. It is clear that this approach is not very accurate and more research in this is desirable. However, for estimating the amount of heat travelling to the fluid, this gives a reasonable approximation.

At this point, the BSTM is matched completely to measurements, based on a SimAlp experiment. In Figure 5.2 below the BSTM model for Test Vehicle 1 is shown compared to the measurements after the matching process that has been based on this specific SimAlp run.

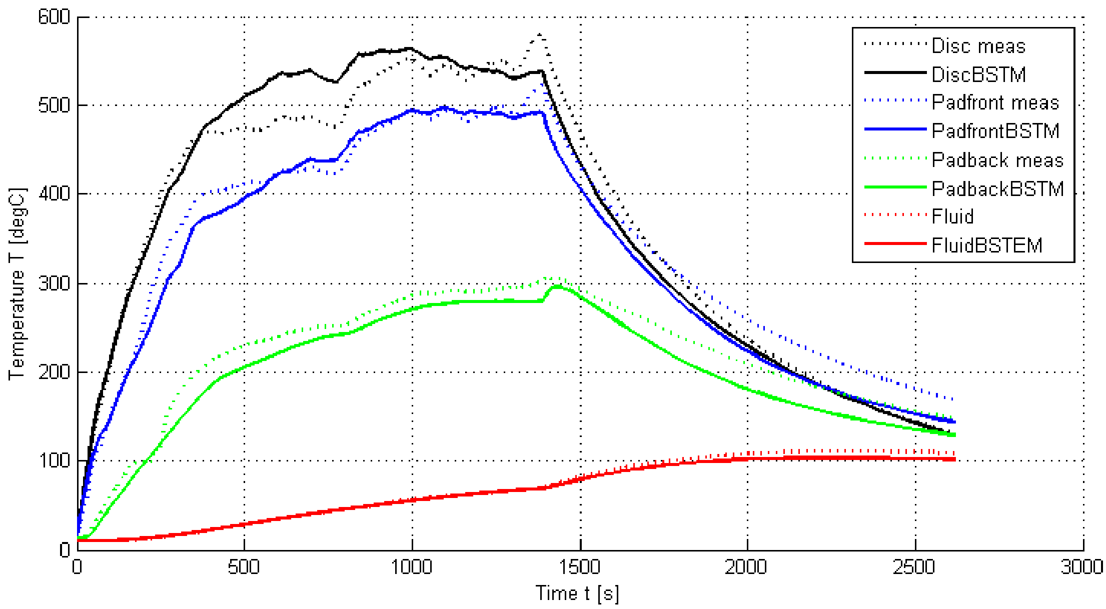

Figure 5.2: New Pad – SimAlp

6. Verification

6.1. Heat Generation Model Verification

6.1.1. Feed Forward Heat Generation Model Accuracy

In Section 4.2 above, the Feed Forward Heat Generation Model was found to be very accurate. However, some offset from test data was noticed. Therefore, during the SimAlp test, the total brake force estimation of the Feed Forward Model was compared to the push force between both vehicles. Since the SimAlp occurred at a constant speed of 36 km/h, the drag and rolling resistance at this speed could easily be subtracted from this push force in order to obtain the brake force. It came forward that the brake system model overestimates the total brake force with roughly 25%, compared to the push force between both vehicles. This indicates that the available brake system data are not sufficiently accurate. Again, this clearly shows the need for a Feed-Back Heat Generation Model.

6.1.2. Feed Forward/Feed-Back Heat Generation Model Comparison

On Test Vehicle 2, the FF and FB Heat generation model are run simultaneously over longer vehicle runs. In Figure 6.1, the total amount of heat generated for both models is shown. Since there is also a brake Torque signal available directly on the vehicle CAN-bus, this is also taken along in the testing.

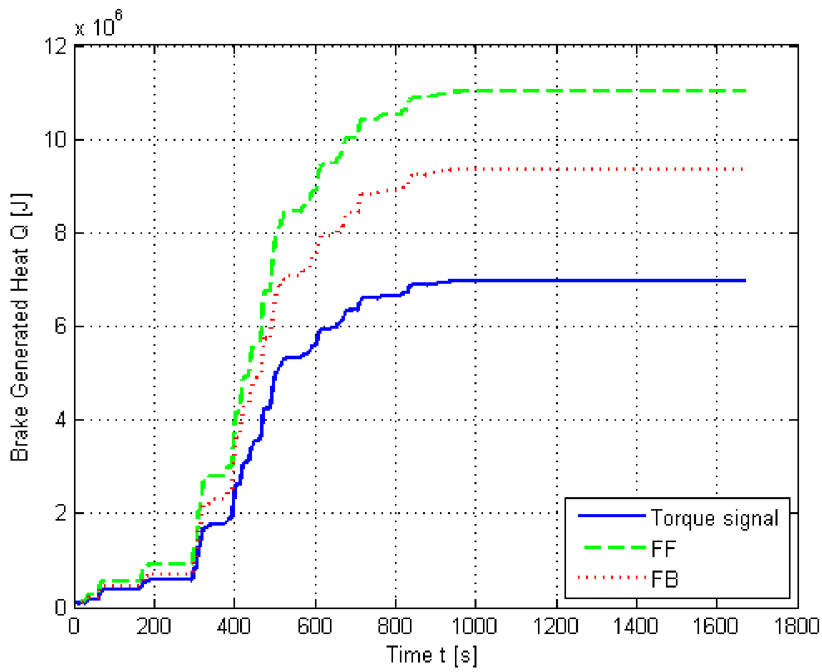

Figure 6.1: Test Vehicle 2: Integrated heat input over time for 3 different models

It is noticed that the brake torque signal available on the vehicle CAN-bus is very inaccurate and is not possible to use. The estimation based on this signal significantly underestimates the brake generated heat. Secondly it is seen that the FB model gives a lower estimation compared to the FF model. This was expected since the FB model does not take into account the portion of the brake energy that is spread into the tires, suspension and chassis. Both models show large similarity though. From this it can be concluded that the power train torque signal that is available on the CAN-bus and is necessary to make a Feed-Back model possible is reliable. This shows the potential that the Feed-Back model has.

For each measurement, during which the vehicle was braking ($p_{prim} > 0$), the ratio of the FB and FF model is plotted. This is shown in Figure 6.2.

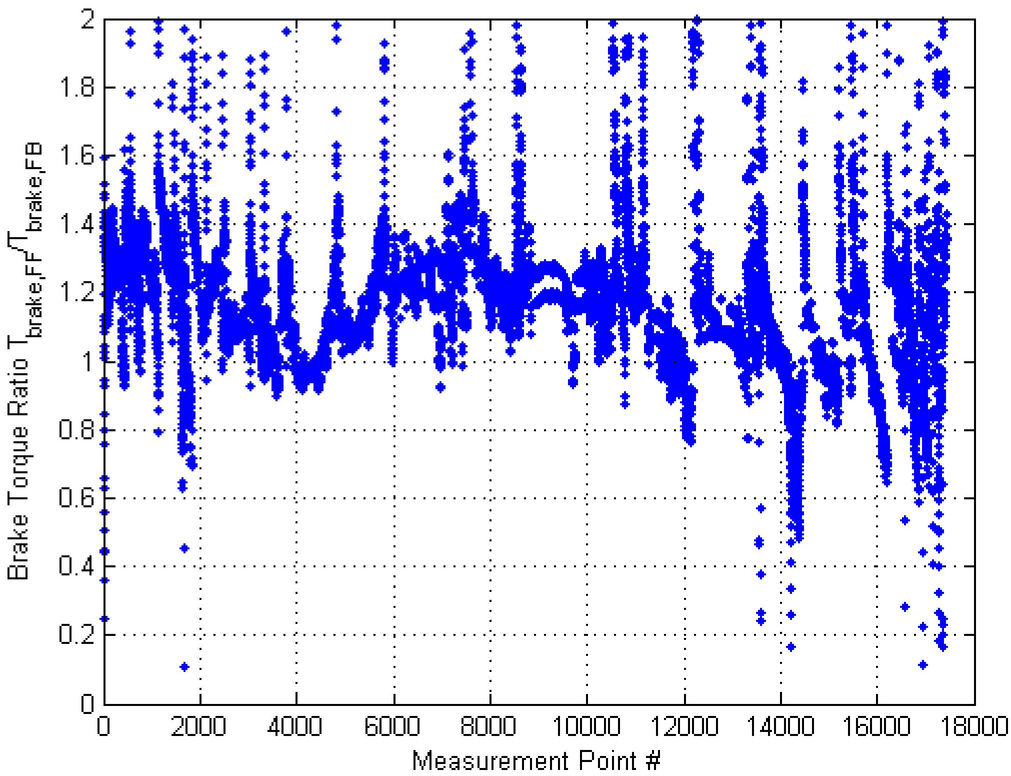

Figure 6.2: Estimated Brake Torque Ratio for Feed Forward and Feed-Back Model

It is observed that the ratio is rather constant over time, in the order of 1.25. Part of this difference can be explained by the heat absorption in deformation of the suspension, chassis and tire as well as frictional heat generation in the tire. However, friction coefficient in the Feed Forward Heat Generation Model seems to be overestimated, since the 25% difference is too large to be explained by absorption effects. Therefore, the actual brake torque will have the shape of the Feed Forward Model, but its magnitude will be closer to the Feed-Back Model. It is noticed that later in time, when high temperatures are reached, there is a slight drop in the ratio. This indicates a small decrease in friction coefficient at high temperatures.

A second way this is investigated is by looking at the ratio of total heat generated in the Feed Forward and Feed-Back model. It can be seen that this value settles at a ratio of roughly 1.2, which lies very close to the value obtained above. These effects clearly illustrate the need for a combination of both Heat Generation Models. The FB model is inherently less accurate for a single brake application than the FF model. On the other hand, in special conditions like snow, rain, damaged brake pads or extreme temperatures, the FB model will automatically take these effects into account while the FF model would give a significant offset with respect to the correct result. Therefore, an ideal heat generation model would combine both models in order to consistently have an accurate brake torque estimation. Before this is possible, the FB model needs additional development in order to know the slope, vehicle mass and vehicle resistance properties at all times. This was not possible within the timeframe of this project.

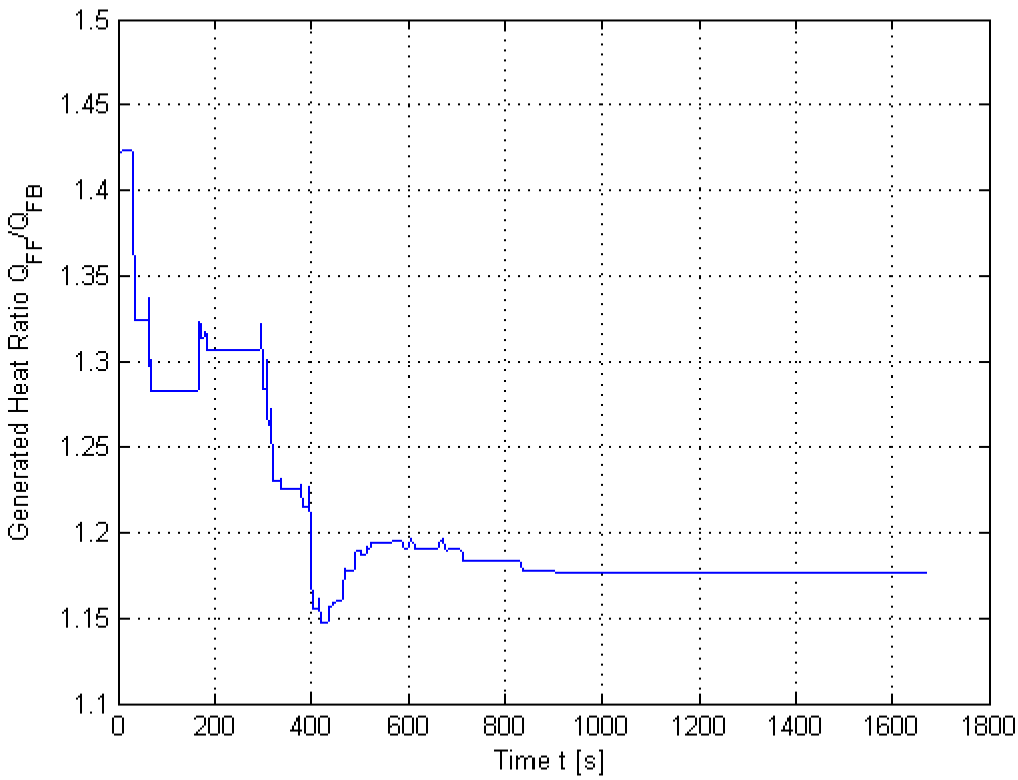

Figure 6.3: Ratio of Generated Heat Estimate for Feed Forward and Feed-Back Model

6.2. Temperature Model Verification

The BSTM has been matched to the SimAlp measurements performed at the Hällered test-track. This matching process has been performed for a vehicle equipped with new brake pads. Afterwards, this model is verified in following tests:

- SimAlp test with half worn pads, which has been performed with Test Vehicle 1.
- Wind tunnel tests which have been performed with Test Vehicle 1.

Later on, the BSTM is implemented into a different vehicle, Test Vehicle 2, by means of a dSPACE system. This is done in a different vehicle type with an almost identical brake system as the previously used vehicle. This is done to verify what amount of tuning is needed to expand the model to other vehicle models. The new vehicle was equipped with identically the same brake disc, but slightly different design of pads and caliper. Next to this, since it concerned a different vehicle type, the wheel hub and consequently the airflow around the brake system was different.

6.2.1. Worn Pad verification

In Figure 6.4 below, a comparison is shown for a SimAlp test between the measured values and the BSTM-estimation when Test Vehicle 1 was equipped with a half worn pad. This is to verify whether the BSTM works throughout the entire pad life. As has been investigated in Section 4.2, the friction coefficient of the worn pad is lower than the friction coefficient of a 'normal' pad at some points, due to insufficient bedding in. Therefore, a Simulation is run in which the heat input is estimated by means of the push force between both vehicles. Since the SimAlp is run at a constant speed of 36 km/h the total resistance force at 36 km/h is subtracted from this push-force. This resistance force is estimated from the aerodynamic drag and rolling resistance of the vehicle. By using this brake force as an input in the BSTM for both the run with new pads as with worn pads, it can be verified if the tuned model works throughout the complete pad life. As has been mentioned earlier, the push-force is 25% lower than the Feed-Forward model. Therefore, this is used as input for the BSTM instead of the Feed Forward Heat Generation Model. In Figure 6.4 below, the BSTM that is tuned for a new pad is shown against the measurements for a vehicle with half worn pads.

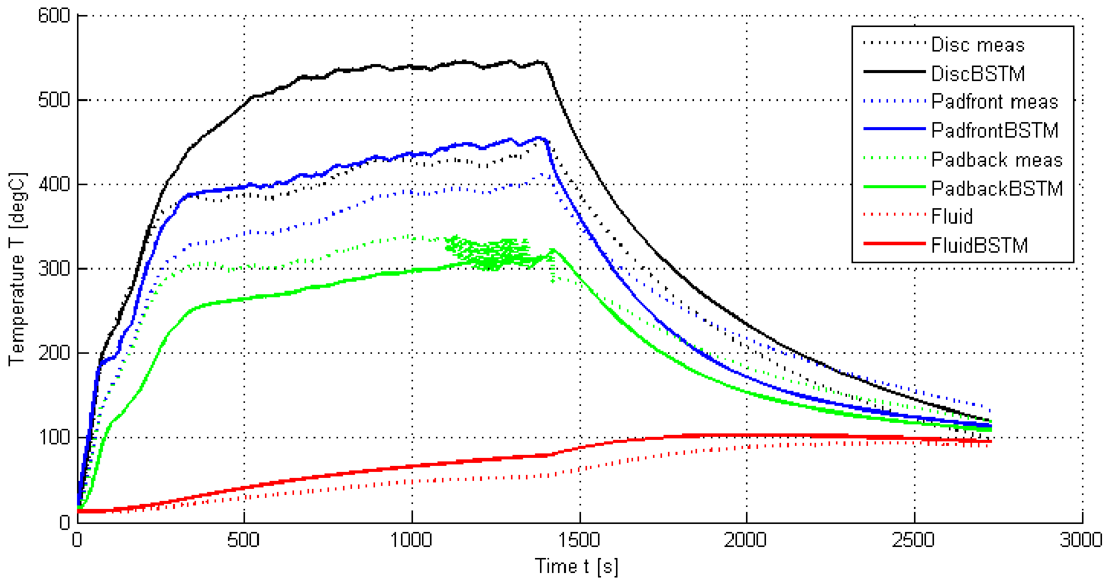

Figure 6.4: Half Worn Pads - SimAlp

It is noticed that the disc temperature of the model initially follows the measurements. At high temperatures however, the BSTM significantly overestimates the disc temperature. A proper explanation has not been found for this. When looking at Figure 5.2 however, it can be seen that the same thing occurs for the experiment with the new pad. At a temperature of 300 degrees Celsius the temperature increase is 'cut off' very suddenly. This is not captured by the model properly since the reason for this is not fully understood. Consequently, the BSTM overestimates the fluid temperature.

During the heat generation phase, the pad back temperature seems rather accurate. However, during the soaking phase it becomes clear that the pad back temperature drops too fast. The thermal resistance at the pad back appears to be too low. It is clear that the pad model needs more investigation since not all effects are completely understood.

6.2.2. Wind tunnel Verification

The BSTM model that is tuned by means of the test results at Hällered is subsequently used to simulate the wind tunnel experiment. The result of this is shown in Figure 6.5 below.

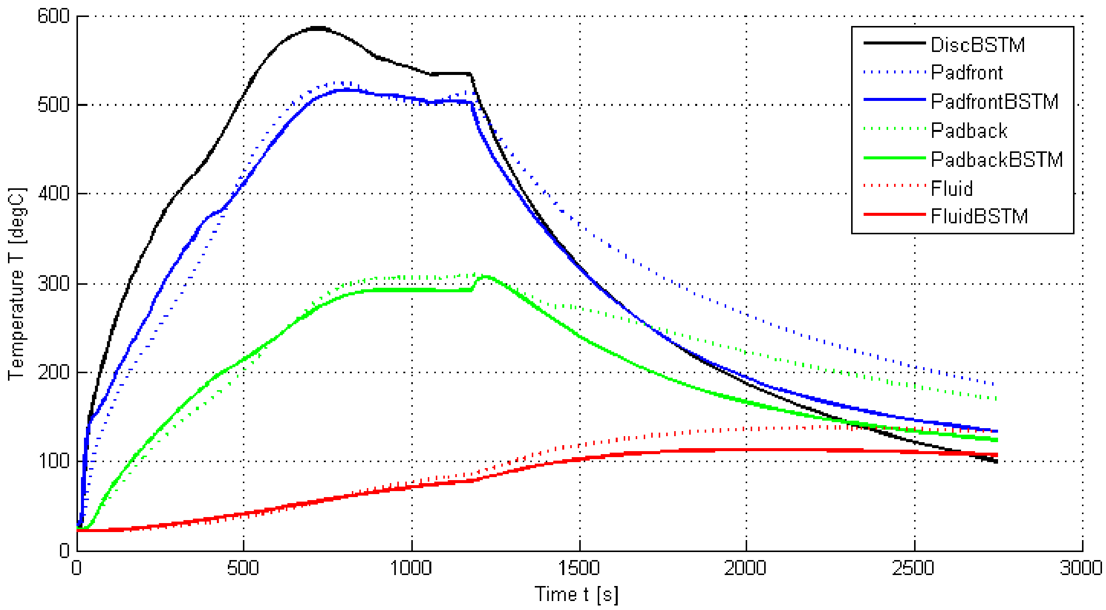

Figure 6.5: Wind tunnel experiment

The soaking behavior of the pad, and consequently the disc in the wind tunnel experiment is different from BSTM (which has been tuned according to the Hällered measurements). A possible explanation for this is the different rims that have been used in both experiments. For more closed rims, the ambient temperature around the brake system increases. This will alter the soaking behavior of the brake system. Since the vehicle in the wind tunnel experiment had more open rims, the cooling during heat soaking is more efficient and consequently the fluid temperature was lower.

6.2.3. Expansion to new Vehicle

The next phase in the verification process is expanding the BSTM to a new vehicle model with an almost identical brake system. Since the brake system is almost identical, ideally, the BSTM should be rather accurate without any significant additional tuning. In Figure 6.6 below, the measured and estimated brake system temperatures are shown for Test Vehicle 2 without any additional tuning. In all the previous tests, the brake torque was available from external sensors: the pushrod during the SimAlp and the driven steel belt during the Wind tunnel experiment. During both experiments this was taken as input into the BSTM. For Test Vehicle 1 the Feed Forward model overestimated the heat generation. From Figure 6.1 it was seen that this is also the case for Test Vehicle 2. Therefore, the Feed-Back Heat Generation Model is used as an input for the BSTM. Since in the experimental setup the vehicle mass and drag properties are known this is justified. In Figure 6.6 below the result of this can be seen, before any modification has been made to the BSTM to match it to Test Vehicle 2.

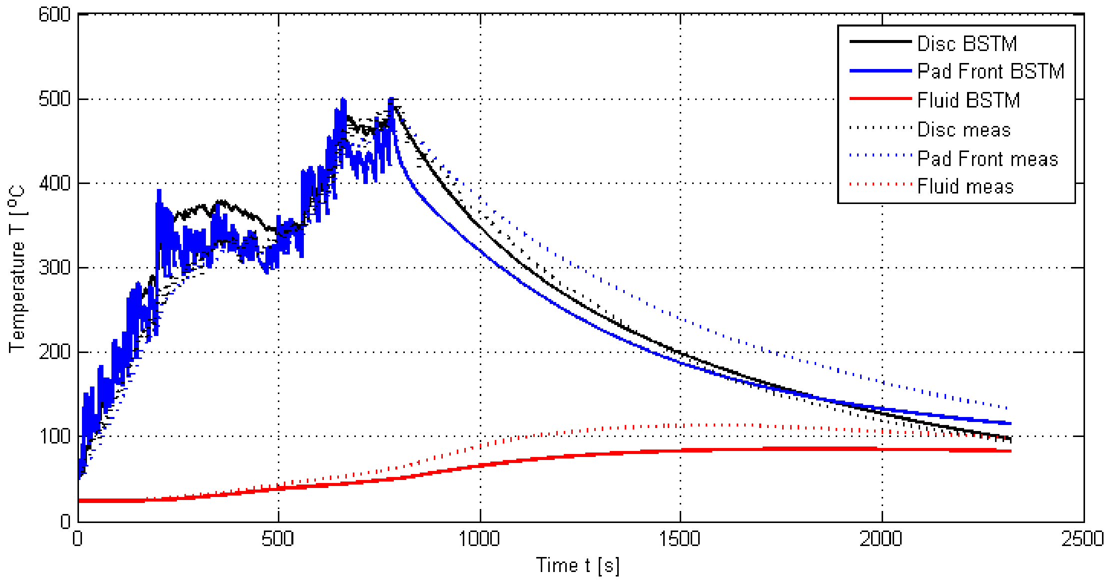

Figure 6.6: Test Vehicle 2 equipped with BSTM before Matching

It can be observed that the Feed-Back Heat Generation Model seems to give a very good result. The disc temperature and pad front temperature give a very close match to the measurements, especially during the heat generation phase. This is as expected, since the used disc is identical to the one used on Test Vehicle 1.

During the Soaking phase, the pad front temperature of the BSTM seems to drop faster than the measurements. Also the fluid temperature increases significantly slower. This can be explained due to fact that a slightly different pad is used in Test Vehicle 1. Consequently, it is possible that the pad backplate has a different layout. Therefore, the amount of heat dissipated from the backplate to the caliper directly as well as the heat travelling to the fluid can be significantly different. Therefore, Test vehicle 2 needs to re-tune the heat flux out of the pad and the fluid behavior in order to match Test Vehicle 1.

It is found that in order to get a proper match between the measurements and the BSTM for Test Vehicle 2, following parameters need to be altered compared to Test Vehicle 1:

- Pad backplate resistance should be increased and the dynamic convection of the pad should be decreased:
- Static and Dynamic convective coefficient of fluid/caliper combination should be altered slightly
- $m \cdot C_p$ of caliper/fluid combination should be decreased.

The convective coefficient of the fluid can be explained by the different airflow for the new vehicle, due to the fact that a different rim is used. The increased thermal resistance of the backplate and the decreased $m \cdot C_p$ of the fluid/caliper combination is due to different backplate design. The thermal resistance between the pad and fluid is improved. At the same time, the fraction of the heat that is guided directly to the caliper through the hammerheads has increased. In Figure 6.7 below, the measurement and BSTM are compared for Test Vehicle 2 after these tuning modifications.

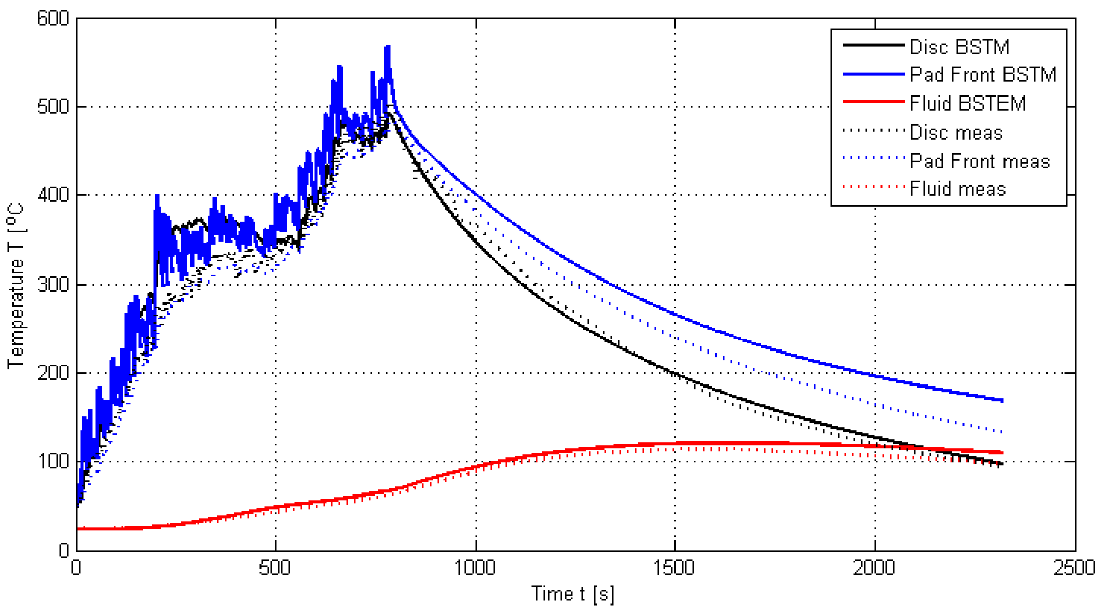

Figure 6.7: Test Vehicle 2 equipped with BSTM after Matching

6.2.4. Long Run Verification

Once the BSTM has been tuned for the new vehicle, a long duration vehicle test is performed in order to verify the performance of the BSTM over longer vehicle drives. More specifically it is verified whether the BSTM starts drifting a away from the measurements after a longer drive. The result of the long duration drive is shown in Figure 6.8 below. The input to the BSTM is the Feed-Back Heat Generation Model. The test run lasts more than two hours in which intensive braking, heat soaking and normal driving phases are repeated multiple times. During this test run extremely high temperature fluctuations occur.

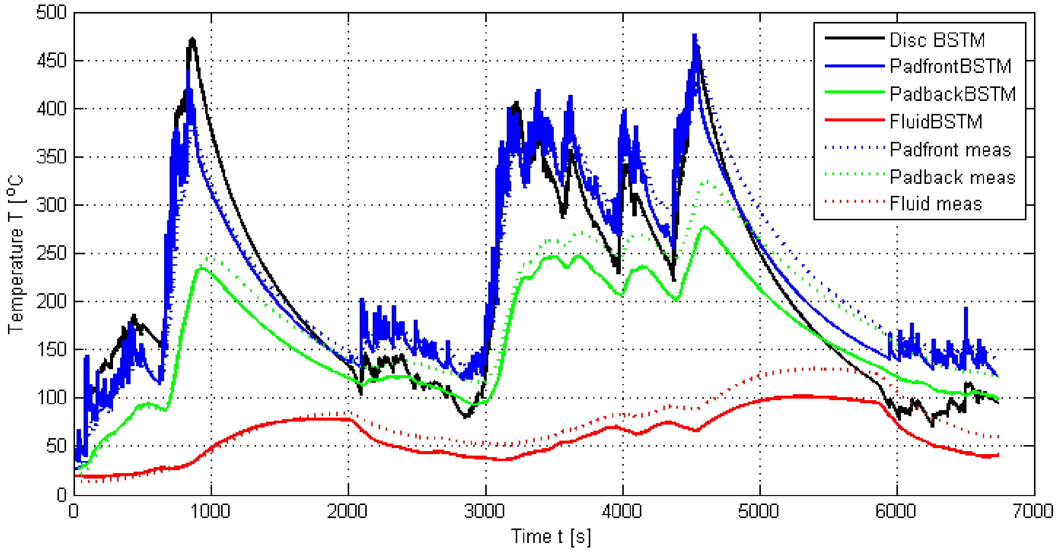

Figure 6.8: Long duration verification drive.

During the long run verification test, the Feed-Back Heat Generation Model was used. It was found that, when the vehicle mass and drag properties are known, as well as the road gradient, this model works surprisingly accurate. It can be seen from the pad front surface temperature evolution that the model follows the measurements very accurately, even after the complete test run. This clearly shows the accurate of the Feed-Back Heat Generation Model and also indicates that the empirically tuned heat partition coefficient works very accurate. The temperature evolution of the pad backside shows a slightly larger difference between simulation and measurements. This is due to the insufficiently modeled pad backplate. If a better model of the pad backplate would be implemented, this would increase the accuracy. Subsequently, this error in the estimated heat flux through the pad backplate induces an error in the fluid temperature estimate as well. Currently the maximum error in fluid temperature estimation is 30 degrees. This is exactly at the boundary of the objective, as was stated in Section 1.3.

7. Human-Machine Interface for Future Implementation

One important concern in the development of new technology is finding how this can be used to our benefit. Being able to estimate the brake fluid temperature is one thing. Now the main question is: How is it possible to benefit from this knowledge? In following Sections several concepts are proposed and a brief selection between those concepts is performed.

7.1. Possible Concepts

Several approaches on how to use this information on brake fluid temperature are possible. For each of these possible approaches following questions should be investigated:

- Q1 Does the system at all times increase vehicle safety?
- Q2 Does the system increase customer satisfaction about the vehicle?
- Q3 Does the system improve customer perception of the overall quality of the vehicle?
- Q4 Is implementation of the system technically feasible?
- Q5 Is implementation of the system economically feasible?

The BSTM informs about the current state of a vehicle. Whenever the brake fluid is overheated and vaporization becomes a feasible possibility, action is required. After reflection and discussion with colleagues, a list of possible actions has been come up with, grouped according to their respective nature.

Active intervention	**Passive intervention**
A1 Active cooling system of brake fluid	P1 Warning light for temperature indication
A2 Reduction in available engine torque	P2 Warning sound for temperature indication
A3 Change in brake pedal feeling	P3 During descent: Use gear shift indicator to enhance engine braking
A4 During descent: Change automatic gearbox setting to enhance engine braking	
A5 Use valves to isolate front wheels: only braking on rear wheels	
A6 Automatic brake assist in Emergency Situation	

Table 7.1: HMI Concepts

Each proposed concept is matched to the questions indicated above. If one or more of the questions are answered with 'no', the concept is discarded.

	Q1	Q2	Q3	Q4	Q5
A1	Y	Y	Y	N	N
A2	N	N	N	Y	Y
A3	Y	/	/	Y	Y
A4	Y	/	/	Y	Y
A5	Y	Y	Y	Y	Y
A6	Y	Y	Y	Y	Y
P1	Y	/	/	Y	Y
P2	Y	/	/	Y	Y
P3	Y	/	/	Y	Y

Table 7.2: HMI Concepts Evaluation Matrix

It can be seen that active HMI concepts A1 and A2 are discarded. All the passive HMI concepts are kept. These concepts are investigated more deeply.

Firstly, it is investigated WHEN overheating of the brake fluid may occur. Clearly, a first condition is a large energy input into the brake system. It is seen that this heat slowly travels towards the brake fluid. When driving a significant amount heat is dissipated into the environment due to forced convection. Even with very large energy inputs, this keeps the brake fluid temperature significantly lower due to the large cooling rate. Only when the vehicle is standing still and forced convection no longer occurs, the heat travels towards the brake fluid without being dissipated at a sufficient rate. Therefore, brake fluid overheating will very rarely occur during a ride itself. Only when the vehicle has been standing still for some time the brake fluid is likely to reach critical temperatures.

7.2. Concept Discussions

7.2.1. Warning Light or Sound for Temperature Indication (P1 and P2)

Warning the driver is the most obvious HMI concept. If the fluid would reach critical temperatures while driving such a system would be unacceptable. Since, a warning light while driving is perceived as a serious vehicle failure by the driver and would negatively influence driver perception of the overall quality of the vehicle. Next to this, especially with sound, sudden warnings while driving could upset the driver creating a dangerous situation. The risk of vapor lock is only present in very specific cases. It has been tried to reach very high brake fluid temperature during normal driving, but this turned out to be very difficult, if not impossible. It has been found that brake fluid vaporization only may occur when the vehicle is parked for a while after an extended period of intensive braking.

Since critical brake fluid temperatures occur only after a vehicle standstill, a warning light or sound would be more acceptable. When starting the vehicle, the driver can be asked through the information display to wait some time in order to allow the brake system temperature to decrease. This informs the driver directly on the brake system condition. However, since this does not occur while driving, this will not induce negative perception of the driver with respect to vehicle quality.

Since the BSTM warning would only occur very few times in a vehicle lifespan – if at all – it is difficult to declare a symbol/relevant sound for this. Therefore, the best proposal is just a text appearing in the driver information display asking to wait a certain amount of time before starting the vehicle in order to allow the brake system to cool down sufficiently.

7.2.2. Gear Shift Indicator (P3)

The gear shift indicator HMI would only work for vehicles with a manual gearbox. Many of these vehicles already have a gearshift indicator in order to reduce the fuel consumption. When the BSTM detects excessive braking during an Alpine descent, the gear shift indicator can ask the driver to shift down. This will automatically induce engine braking.

7.2.3. Change in Brake Pedal Feeling (A3)

With brake-by-wire systems becoming the dominant brake system set-up in future vehicles, it is very simple to modify brake pedal feeling. This allows a gradual increase in sponginess of the brake pedal feeling as the brakes heat up. By allowing this gradual increase the driver gets continuous feedback of the brake system condition without having the driver perceiving the brake system to be of poor quality.

However, critical brake system temperatures usually occur after soaking periods, when the vehicle has been standing still. When the driver starts driving and the brake pedal feeling is spongy from the beginning there is a risk that the driver is unaware of this since he has no reference of previous brake applications in mind with sharper pedal feeling.

One other disadvantage is that the pedal feeling may change sooner than would be the case with a conventional brake system that is overheating. This may be interpreted by the driver as if the brake system would be of poor quality, negatively affecting customer perception of the brake system quality.

7.2.4. Change Automatic Gearbox Setting (A4)

Automatic gearboxes can be tuned to have more engine-braking while descending. A possible application of this is to replace gently brake applications in a brake by wire system by shifting down the gearbox instead when the vehicle detects it is descending a slope. Such a system could be implemented at all times, not just at high brake fluid temperatures and would thus be able to prevent these situations completely. However, such a system requires a complete development process on its own and would be able to operate completely independent of the BSTM information.

7.2.5. Use Valves to Isolate Front Wheels (A5)

Due to hygroscopic effects the brake fluid boiling point decreases over time. Since it is not possible to know the exact conditions under which the brake fluid has been operating or how old the brake fluid is, the boiling point of the brake fluid is unknown. If the BSTM system is not able to accurately predict the risk of brake fluid boiling, it is impossible to implement a reliable warning system for the driver. The likelihood of false alarm or missed detection is too high, causing the system to decrease customer satisfaction.

Critical brake fluid temperatures will only occur at the front wheels where most of the braking was performed. Therefore, an option is, when critical brake fluid temperatures are detected, to use the ABS and ESC valves to isolate the front wheels and only brake on the rear wheels. This approach allows the front wheels to cool down and is not negatively influenced by the uncertainty on brake fluid boiling temperature since it can be used to keep the temperature below the 'worst-case' boiling point. Next to this, in a brake-by-wire vehicle, the pedal feeling can be altered when the front wheels are isolated. A challenge in this concept lies in assuring vehicle stability at all times.

7.2.6. Automatic Brake Assist in Emergency Situation (A6)

The BSTM can be made this way that it is able to detect situations in which brake fluid vaporization is theoretically possible. If such a system is recognized, the Active intervention system is set on standby. Otherwise, it is switched off completely. During the time the emergency system is active, it constantly monitors the vehicle pitch angle (slope), the primary circuit pressure, the wheel speeds, vehicle deceleration and the brake pedal travel. The system estimates in real time what the wheel speed and vehicle acceleration should be theoretically to the real values. If a large discrepancy is detected here (if there is a large brake pressure and the vehicle nearly decelerates) and/or the brake pedal acceleration indicates an anxious driver, the system is activated. The exact control of this system is done by means of fuzzy logic and the thresholds are tuned experimentally.

Once the system intervenes it either sends maximum brake pressure to the non-locked circuit. An alternative is that it automatically applies the hand brake in the severe case when both circuits are locked or to immediately isolate the front wheels, in order to allow rear wheel hydraulic braking. Applying the hand brake could cause problems with respect to vehicle stability. This should be taken into account during development/testing. However, since vapor lock is likely to occur at very low vehicle speed this reduces the stability problem significantly.

8. Results and Conclusions

- A brake torque estimation model based on the primary pressure (Feed Forward) and a brake torque estimation model based on the vehicle deceleration (Feedback) give similar results. This leads to the possibility of combining both models in order to optimize the brake torque estimation in all circumstances.

- The Feed Forward Model needs to know the friction coefficient between the brake pad and the brake disc. This friction coefficient is variable and difficult to predict in reality.

- Before the Feed-Back Heat Generation Model can be used, a proper slope, mass and resistance estimation model needs to be developed since these quantities are not measured directly in the vehicle. If these quantities would be known, the Feed-Back Model gives a sufficiently accurate estimate.

- The lumped mass model for the brake disc gives a reasonable accuracy. It is important to incorporate the temperature dependency of the material properties, since this has a significant influence. From the disc temperature evolution at high temperature it is seen that this is not completely correct in the model, since material properties of pure iron are assumed.

- The Finite Difference Thermal Model of the brake pad is a good way to model the heat flux through the pad. Due to the many parameters involved and the large variation in pad material properties, more investigation is needed in order to fully understand the thermal behavior of the brake pads.

- The backplate of the pad is an important parameter for the heat flux to the fluid. This backplate and shim is usually composed of different material layers to isolate the fluid as much as possible and to guide as much heat as possible to the caliper. Therefore, a large difference in thermal resistance of the back plate can exist between 2 different pad types.

- The fluid and surrounding caliper have almost identical temperatures. Therefore these both can be modeled as one lumped mass. By empirically setting the material properties, this lumped mass can be matched to measurements. This allows incorporation of different effects, like the amount of heat that is 'lost' at the pad backplate without knowing the actual quantity of these effects.

- Due to the high complexity of the thermal processes involved in the brake system, the BSTM needs tuning for every specific vehicle - rim – brake system combination. The amount of tuning work required is limited however and a rather straight forward process.

- The BSTM can easily be upgraded by replacing individual parts of the model. This allows the CAE models that are currently under development to be implemented into the BSTM

- The BSTM is less accurate for worn pads, but gives reasonable indication. However, in current vehicle-park it is not known when a brake pad has been replaced. This is required before the BSTM can be used for worn pads as well.

- Many effects are not yet fully understood. Next to this, there is a large variability in between different runs and a significant degree of uncertainty. Therefore the BSTM can give a good indication of the temperatures of the different components, but it is not ensured that the estimation is accurate.

- Several active and passive HMI concepts are investigated. In order to select or create a final concept from this, considerable additional investigation is required.

9. Future Work

The BSTM is still under development and additional work is required before (parts of) the BSTM can be implemented into production vehicles. Therefore improvement, additional development and final verification is required.

9.1. Heat Generation Model

The Feed forward Model and the Feed-Back Model should be combined and expanded. The Feed-Forward Model should be tuned for each vehicle specifically, since it was observed that inserting the available parameters resulted in an error of up to 25%. For the Feed-Back Model, separate models for estimating the vehicle mass, vehicle resistance properties and road gradient are to be developed and verified. Finally, thought should be given on how to combine and expand both models:

- Use Feed-Forward Model to tune the Feed-Back Model and vice-versa
- Implement ESP, Spin Control and Traction Control into the Heat Generation Model

Integrating the accelerometer signal and comparing this to the velocity signal can further improve the accuracy of the Feed-Back Heat Generation Model. It can be investigated if this can be used to remove the stead state offset of the accelerometer. This could create the possibility of using the acceleration signal in order to estimate the vehicle mass, drag properties and road gradient.

9.2. Temperature Estimation Model

Different parts of the BSTM need additional updating:

- Brake disc material properties (C_p) at high temperatures need to be determined. Find out why the 'dent' in the temperature-increase of the brake disc occurs at different temperatures in between different SimAlp tests.
- Include disc wear into the model. A worn disc has less mass and will subsequently heat up faster.
- Brake pad material properties (k, C_p) at high temperatures need to be determined.
- A pad wear model should be included. It should be thought through on how to tackle this, since the vehicle is not equipped with pad wear indicators.
- A better understanding of how the heat travels through the pad backplate and shim is needed. During a future SimAlp, an additional thermal sensor should be added in the pad backplate. Also it can be interesting to verify the amount of heat that is conduction directly to the surrounding caliper through the hammerheads.

- A large discrepancy between the left front and right front wheel temperature evolution has been noticed. It is not fully understood why there is such a large discrepancy.
- For tuning the cooling behavior of the brake disc at standstill, it is important to know the location of the thermal sensor in the disc. Therefore, during a future SimAlp test, it is advised to make sure the thermal sensor in the brake disc at stand still is exactly facing the brake pad.
- Verification during moisture conditions is required.

9.3. Human Machine Interface

The HMI proposed concepts are just very rough initial ideas. It is important to first finish the Heat Generation Model and BSTM. Afterwards, an optimal HMI can be developed completely.

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