Numerical Efficient Thermal Network for Calculating the Brake Disc Temperature

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Abstract: The simulation of the brake disc temperature is an important tool in the development of passenger cars. Nowadays thermal models of brake discs are real-time applications, running on electronic control units (ECUs) of cars to improve the vehicle safety in several ways. These models are often working with full empirical methods, leading to large deviations between calculation and measurement. To meet the requirements of automotive safety integrity levels (ASILs), these thermal models cannot rely on the state of the art ambient air temperature sensors, which causes unacceptable deviations. Focusing on numerical efficient thermal simulations, a new approach of a semi-analytical thermal network for simulating the brake disc temperature with minimal effort is proposed. The thermal network is based on lumped parameters, using two thermal capacity nodes and a constant ambient temperature. Using semi-analytical correlations, the model can be adapted to different geometries and car lines effortlessly. The empirical parameters of the model result only from two standardized tests. These parameters are used to evaluate the estimation accuracy in real driving situations. Additionally, the adaptability is tested for two different car lines and four brake disc dimensions. These tests are initially performed with unchanged parameters and afterwards with refitted parameters. The model shows a good estimation for the tested load cases. Compared to the state of the art, the proposed model is less accurate than complex finite element method (FEM) models and computational fluid dynamic (CFD) approaches, but shows a higher accuracy and better adaptability than other lumped parameter models with comparable numerical effort. Hence, possible applications can be dimensioning the brake system in the development process of new car lines or a real-time simulation on the latest ECU in the vehicle.

Key words: Brake disc, numerical efficient thermal simulation, lumped parameter, downhill test, performance test, real driving situations.

1. Introduction

Numerical efficient thermal models in the state of the art are often afflicted with large estimation errors. The usually full empirical parameterization leads to huge effort when changing brake disc geometries and car lines.

In this paper a semi-analytical physically based model approach for fast thermal simulations is introduced, using only two nodes for differentiation between the geometries inside the brake disc. Focusing on the thermal simulation, the model is parameterized by two standardized tests, and tested under real drive situations and different car lines and brake discs. A simple replica of the vehicle longitudinal dynamic simulation is used for calculating the friction power.

2. Previous Work

The scope of recent investigations is to get more insights of the thermo-mechanical mechanisms in solids and fluids by using complex computational fluid dynamic (CFD) and finite element method (FEM) simulations. Typical examples are Refs. [1] or [2].

In the following, the state of the art is summarized with the focus on fast thermal simulations.

A commonly used empirical correlation for convective heat transfer can be found in Ref. [3]. Since the calculation of the convection is full empirical and not geometry dependent here, this approach is limited when changing brake disc geometries. A more analytical approach for thermal networks with a small
For a heat map presented in Ref. [4], providing an experimental equation for the air velocity within the rotor vanes, which is used in this paper. A more detailed approach is performed in Ref. [5], using more nodes. A numerical efficient thermal model for commercial vehicles and short repeated braking sequences is presented in Ref. [6]. This approach considers only one node. It has not been tested for downhill tests or real drive conditions. Two-dimensional simulations are performed in Ref. [7], using FEM and providing a good comparison of all established models concerning the heat partition ratio between brake disc and brake pad. A fast-thermal model of a brake disc is described in Ref. [8], using Newton’s law of cooling and focusing on the brake pad and brake fluid temperatures. The approach from Limpert [3] is also used in Ref. [9] for calculating the brake disc temperature of an electronic parking brake system. It shows small errors during the heating sequence while the errors become larger during the cooling time. A new approach with eight nodes for the brake disc is presented by Day [10], using the approach for convectional heat transfer described in Ref. [3]. The generated results are not within the targeted range of accuracy of this paper.

\[
\begin{align*}
\begin{bmatrix}
m_{1} \cdot c_{p,1} & 0 \\
0 & m_{2} \cdot c_{p,2}
\end{bmatrix}
\begin{bmatrix}
\frac{dT_{1}}{dt} \\
\frac{dT_{2}}{dt}
\end{bmatrix}
- \begin{bmatrix}
\frac{1}{R_{1,2}} - \frac{1}{R_{1,2}} \\
\frac{1}{R_{1,2}}
\end{bmatrix}
\begin{bmatrix}
T_{1} \\
T_{2}
\end{bmatrix}
- \begin{bmatrix}
\sigma \cdot \varepsilon_{1} \cdot \sum_{i} A_{r,1,i} \\
\sigma \cdot \varepsilon_{2} \cdot \sum_{i} A_{r,2,i}
\end{bmatrix}
\begin{bmatrix}
0 \\
0
\end{bmatrix}

= \begin{bmatrix}
\sum_{i} \alpha_{1,i}(T_{1}, T_{c}) \cdot A_{c,1,i} \\
\sum_{i} \alpha_{2,i}(T_{0}, T_{c}) \cdot A_{c,2,i}
\end{bmatrix}
\begin{bmatrix}
(T_{1} - T_{0}) \\
(T_{2} - T_{0})
\end{bmatrix}
\end{align*}
\]

The mass \( m \) and the thermal capacity \( c_{p} \) are representing the thermal mass of an element. \( R_{1,2} \) is the conduction resistance between the different elements. The calculation of \( R_{1,2} \) can be found in Ref. [11] and is based on stationary conduction. The term for radiation is derived from the Stefan Boltzmann law for grey bodies (see Ref. [11] for details), by considering the sum of all outer surface areas \( A_{r} \). All
areas have the same constant emissivity factor $\epsilon$, which is multiplied by the Stefan Boltzmann constant $\sigma$.

\[ P_{\text{brake}} = \frac{1}{2} \left( m_{\text{car}} + \frac{J_{\text{rot}}}{r_{\text{dyn}}^2} \right) \frac{d}{dt} v_{\text{car}}^2 + m_{\text{car}} \cdot g \]

(2)

$J_{\text{rot}}$ indicates the rotational masses (e.g. wheels, gearbox etc.), which are transformed with the dynamic roll radius $r_{\text{dyn}}$ to a virtual mass. Together with the car mass $m_{\text{car}}$, the brake power can be calculated in consequence of a change in the vehicle velocity $v_{\text{car}}$. The geodetic height is represented by $h$ and the gravity constant by $g$. Aerodynamic and rolling resistances, such as powertrain losses are summarized in $\Sigma P_{\text{loss}}$. They are reducing the insertion of friction power in the brake system. Steering resistances in curves are not considered.

The braking power $P_{\text{brake}}$ is divided between the axles by the brake power distribution (BPD) and divided by four, considering two brake discs with two friction ring sides in each axle, using:

\[ \dot{Q} = P_{\text{brake}} \cdot \frac{\text{BPD}}{4} \]

(3)

The BPD changes, depending on the deceleration. For a brake application controlled by the ECU (e.g. ABS) it depends on the dynamic wheel loads otherwise it results out of the hydraulic ratios and the friction coefficients inside the brake system, here considered constant. For further details see Ref. [12]. The resulting energy input in one friction ring side $\dot{Q}$ is split up by the heat partition ratio $\gamma$ into the brake disc and by $(1 - \gamma)$ into the brake pad. For all appropriate heat partition ratio models in Ref. [7], a simulation with a high discretization of brake pad and brake disc is needed. The brake disc discretization is much lower compared to Ref. [7] and the brake pad is not considered here. Hence, the heat partition ratio can not be modelled with the approaches compared in Ref. [7] and is therefore determined experimentally in this paper.

For the calculation of the convection and radiation, the geometry is divided into three different geometries.
shown in Fig. 2.

For calculation of the forced convection for the geometries in Figs. 2b and 2c, the approach from Ref. [5] is used by considering the cross flow of the cooling air direction. The cooling air flow is assumed to flow uniform over all areas and the cooling air speed is calculated as a product of the constant factor

\[ v_f \]

and the vehicle velocity. The Nusselt correlations for free convection can be found in Ref. [11]. The geometry of the cooling channel for free convection is divided in cylindrical surfaces for the pins and a heated vertical gap. The used correlations can be found in Ref. [11]. The heat transfer coefficients depend on the characteristic length, which represents the length of the convective air flow over the geometries (compare Ref. [11] for further details). In this paper a new approach for the characteristic length \( L_{\text{char}} \) of the friction surface in Fig. 2c is proposed, shown in Fig. 3.

With this geometrical transformation the characteristic length of the friction ring, depending on the inner \( r_{\text{in}} \) and outer radius \( r_{\text{out}} \), can be calculated by:

\[ L_{\text{char}} = 2 \left( r_{\text{out}} - r_{\text{in}} \right) \]  

(4)

4. Experiments and Discussion for Different Test Scenarios

For the determination of the empirical parameters, the measurements from a downhill test and a performance test are used. The performance test 1 in Fig. 5 is used for the estimation of the heat partition ratio \( \gamma \). The downhill test in Fig. 6 is a better indicator for the cooling behavior and is therefore used to estimate the constant cooling air velocity factor and the boundary condition to the brake hub. The standstill cooling is used to define the brake hub conduction factor \( k_f \). The empirical parameters differ in the following case between FA (front axle) and RA (rear axle) only in \( k_f \). To identify the empirical parameters, the identification process in Fig. 4 is performed iteratively until the model shows the closest performance compared to the measurements.

In the following graphs, the brake disc temperatures are averaged for each axle. Measurements of the height profile are only available for the downhill test.
A different performance test is used to check the model behaviour, as seen in Fig. 7. The test is performed with the same vehicle and brake discs as the tests above and the simulation parameters stay the same. The test includes different load conditions of the vehicle and different vehicle velocities.

Fig. 4 Iterative parameter identification process.

Fig. 5 Performance test 1—simulation (Sim) and measurement (Mea) of the brake disc temperature for the front axle (FA) and the RA.
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The following real driving situations are measured under different conditions. The changes include different rims, a lower ambient air temperature and changed road conditions from dry to wet. The simulation parameters stay the same.

Fig. 8 shows a real driving situation with many short brake applications and speed ranges comparable to the performance tests. Fig. 9 shows some high-speed brake applications followed by a standstill cooling, while Fig. 10 displays comparable brake applications followed by a cooling sequence at a vehicle speed of 50 km/h. The decelerations values of all three tests are higher than the deceleration values of the downhill test and lower than the deceleration values in the performance test.

As shown in Figs. 7-10, the performance of the proposed model is good but there are still some deviations.

Heating sequences and cooling influences after a high-performance brake application seem to be underrated. One explanation for this behavior could be, that the temperature profile for each thermal node is simulated assuming a uniform temperature distribution. It is possible that the average temperature of the cooling channel and the temperature of the thermocouple in the mean radius of the cooling channel can differ. This may be caused by an inhomogeneous pressure distribution in the contact
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Besides some differences in the temperature rising, leading to large estimation errors at about 500 s, Fig. 8 shows a good correlation of the brake disc cooling behaviour. The deviations are decreasing during the following 100 s. The missing energy input is caused by an uphill drive of the car, which could not be considered in the simulation, due to the not-measured height profile. The decreasing of the error is caused by the downhill drive afterwards. In the downhill test in Fig. 6, it can be observed, that the temperature deviations of FA and RA are larger in the measurements compared to the simulation, reaching the maximum deviation at about 1,000 s. Both phenomena can be

Fig. 8 Real driving situation with many short brake applications—simulation (Sim) and measurement (Mea) of the brake disc temperature for the FA and the RA.

Fig. 9 Real driving situation with high-speed brake applications and standstill cooling—simulation (Sim) and measurement (Mea) of the brake disc temperature for the FA and the RA.
caused by variations in the friction coefficients, which lead to a change in the BPD. This cannot be considered by using a constant friction coefficient. To omit these errors of the friction power estimation, one possibility is to change to a hydraulic based approach, using only the hydraulic pressure as an input. Therefore, a simulation of the friction coefficient is needed. To get a good estimation for all operating conditions in this case, further research is required.

The biggest temperature deviation can be seen during standstill cooling in Fig. 9, caused by an overestimation of the simulated cooling. Because of the lower environmental air temperature and the shorter heating time of the brake system and the wheel house, this error cannot be explained satisfactorily. One possible explanation could be a cooling wind during the downhill test standstill cooling. Furthermore, a strong oxidation of the brake disc surfaces, caused by the changed weather conditions, could be observed during the standstill phase after the high-speed applications. This oxidation could lead to a higher isolation of the hot brake disc core from the environment and could cause a lower cooling. The interaction between oxidations and the cooling behaviour of brake discs could be researched in order to understand these effects. Even if these interactions are fully understood, it is not feasible to detect all environmental conditions needed for the modelling in manner of economic real-time simulations. Since the scope of this approach is to consider only simple and numerical efficient temperature simulations, these effects must be tolerated.

5. Experiments and Discussion for Different Car Lines and Brake Disc Geometries

In order to check the model performance for different brake disc geometries and car lines, further tests were simulated and compared to real car measurements. Table 1 shows the deviations between simulation and measurement. The large sedan (LS) with the big ventilated brake disc on the FA (B,F,V) and the big ventilated brake disc on the RA (B,R,V) is listed for the performance test 1 (P1), performance test 2 (P2) and the downhill test (D). These tests are shown in Figs. 5-7. They are also measured and simulated for two different car lines with their corresponding brake discs. All simulations have been performed using the same empirical parameters (constant param.), resulted from the adaption on the large sedan brake discs (fitting on Figs. 5 and 6). Additionally, the parameters are refitted for each brake disc to check the minimal reachable deviations in these cases. Starting with the parameters from the large sedan (compare Figs. 5 and 6) the process in Fig. 4 is done in a comparable way. The refitting process has included minor changes in $\gamma$, $k_v$ and $k_T$. For the
small massive brake disc $k_T$, a larger variation is needed. This is caused by the full connection of the massive brake disc with the brake hub unlike the half side connection in a ventilated brake disc (compare Fig. 1c). The changes are listed simplified in the columns param. change.

The mid-sized sedan (MS) is tested with mid-sized ventilated brake discs on the FA (M,F,V) and mid-sized ventilated brake discs on the RA (M,R,V). The small sedan (SS) is tested with small ventilated brake discs on the FA (S,F,V) and small massive brake discs on the RA (S,R,M).

The results are listed in form of three differences between measurement and simulation. The absolute difference between the maximum temperatures is named DMT. The maximum absolute temperature difference is called MTD. The arithmetical average of the absolute difference temperature is listed as ATD.

The car lines differ in vehicle mass, BPD and wheel house aerodynamic. All brake discs vary in geometries, thermal capacities, surface areas and characteristic lengths.

Comparing the results from the figures and the table for the large sedan, one can see that the specified differences in Table 1 are a good indicator for the model performance.

### Table 1 Deviations between measurement and simulation for different car lines, brake disc geometries and tests, using the parameters for the large sedan (see Figs. 4 and 5)—constant param.—and refitted param. with their respective change; DMT = absolute difference in maximum temperature; MTD = maximum absolute temperature difference; ATD = average absolute temperature difference.

| Car line | Brake disc | Test | DMT | MTD | ATD | Refitted param. | Param. change |
|----------|------------|------|-----|-----|-----|----------------|--------------|
|          |            |      |     |     |     | DMT | MTD | ATD | $\gamma$ | $k_F$ | $k_T$ |
| ( )      | ( )        | ( )  | ( ) | ( ) | ( ) | ( ) | ( ) | ( ) | ( )    | ( )   | ( )   |
| LS       | B,F,V      | P1   | 8.9 | 25.5| 10.6|     |     |     |       |       |       |
| LS       | B,F,V      | P2   | 3.6 | 23.3| 6.3 |     |     |     |       |       |       |
| LS       | B,F,V      | D    | 16.3| 53.6| 13.9|     |     |     |       |       |       |
| LS       | B,R,V      | P1   | 2.0 | 29.9| 7.4 |     |     |     |       |       |       |
| LS       | B,R,V      | P2   | 3.8 | 21.5| 6.7 |     |     |     |       |       |       |
| LS       | B,R,V      | D    | 36.4| 55.9| 16.5|     |     |     |       |       |       |
| MS       | M,F,V      | P1   | 16.2| 28.1| 11.4|     |     |     |       |       |       |
| MS       | M,F,V      | P2   | 18.9| 36.7| 21.3|     |     |     |       |       |       |
| MS       | M,F,V      | D    | 1.0 | 56.4| 21.5|     |     |     |       |       |       |
| MS       | M,R,V      | P1   | 34.6| 39.4| 15.0|     |     |     |       |       |       |
| MS       | M,R,V      | P2   | 17.0| 21.8| 11.0|     |     |     |       |       |       |
| MS       | M,R,V      | D    | 0.9 | 39.6| 14.8|     |     |     |       |       |       |
| SS       | S,F,V      | P1   | 23.5| 38.7| 11.1|     |     |     |       |       |       |
| SS       | S,F,V      | P2   | 10.8| 33.1| 9.5 |     |     |     |       |       |       |
| SS       | S,F,V      | D    | 30.0| 69.0| 19.5|     |     |     |       |       |       |
| SS       | S,R,M      | P1   | 35.5| 55.9| 26.4|     |     |     |       |       |       |
| SS       | S,R,M      | P2   | 16.9| 35.7| 11.8|     |     |     |       |       |       |
| SS       | S,R,M      | D    | 7.9 | 83.0| 45.7|     |     |     |       |       |       |

Using the same parameters for the other car lines the model has large estimation errors. Almost all differences can be reduced by refitting the parameters for each brake disc.

The biggest errors can be observed for the downhill tests, especially for the small sedan. Even with refitted parameters, the maximum deviation is higher than 70°C. This deviation is located in the heating phase (compare Fig. 5, time range 400-800 s) with an overestimation of both axle temperatures. The deviations are diminished close to zero before the last hard brake application. This could be explained by the replacement of the brake pad by the experimental heat partition ratio $\gamma$, which is parameterized using the
high-power test 1. The temperature distributions between brake pad and brake disc are completely different for soft long term and hard short-term brake applications. The heat partition ratio should change respective to the changing temperature distribution. While other car lines do not show the same deviations, further investigations are necessary to clarify, whether this is a systematic error in the model or if this is a special error only occurring for the small sedan.

6. Conclusions

In this paper, a new lumped parameter model with two nodes is proposed. The model is physically based, but uses some empirical correlations. For the estimation of the empirical parameters, two training load cases, namely a high power and a downhill test, are used. The adjusted model is then tested with different real driving situations, car lines and brake disc geometries.

The simulated temperatures are mostly in good correlation compared to the car tests.

A larger deviation of the simulation can be observed during a stand still phase after a real drive load case. If changing brake disc geometry and car lines without adapting the empirical parameters, the errors are increasing. Minor changes in the empirical parameters lead to a significant error reduction.

Further research should focus on a friction power estimation, based on hydraulic pressure by using a friction coefficient estimation model.

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