

A Monolithic Approach to Simulate the Cooling Behavior of Disk Brakes

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ABSTRACT

In the present paper we introduce a monolithic CFD approach to simulate the cooling-down characteristics of disk brakes. To ensure a strong coupling between fluid and solid domain the overall transient thermal problem is solved within a single flow solver during the complete cooling-down process. We employ a fully implicit second order solution procedure.

The experimental configuration consists of an inertia dynamometer including a generic 17 inch vented front disk with caliper, dust shield, bearing and knuckle. The validation is carried out for three different air flow velocities, with and without dust shield. The temperature is monitored via two thermocouples embedded into outer and inner rotor cheeks. In order to quantify the cooling-down characteristics, regression analysis are conducted on the temperature curves. The obtained cooling coefficient serves as comparison between measurement and computation.

Within the proposed monolithic CFD approach the whole transient thermal problem considering radiation, convection and conduction is solved for the fluid and solid domain, simultaneously in the frame of one single flow solver (i.e. monolithic approach, no partitioned coupling). Employing moving mesh for the solid domain leads to a realistic rotational symmetric temperature distribution over the brake disk. We present graphs of total heat flux, radiation heat flux and heat transfer coefficient over time. The validation is conducted on the basis of measured and computed cooling coefficients considering three different air flow velocities and two dust shield configurations. An influence of starting temperature for the rotor has been also confirmed by experiments and simulations, respectively. The dynamometer test rig enables high repeatability without disturbances usually occurring in vehicle integrated brake corners.

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INTRODUCTION

To prevent brake systems of passenger cars from overheating the cooling capability of the brake components has to be investigated in the early stage of vehicle development. Too high brake disk temperatures can reduce overall brake system performance, e.g. pedal travel increase, pad fading, reduced wear life, brake fluid boiling or rotor cracking. One possibility to overcome a critical rotor temperature issue exists in improving the cooling air flow. A precise prediction of cooling capability is essential for the subsequent flow optimization loops.

Quantifying brake cooling characteristics can be done throughout several ways. In times in which computing power has still not been sufficiently developed, empirical equations had a major impact. Expressions for NUSSELT numbers and heat transfer coefficients were a great help for the

dimensioning in the early development process, e.g., [4, 6]. With the steady increase of computing power over the last decades the use of numerical simulations for such kind of problems gained substantially significance. Employing a pure CFD approach helps to understand the complex flow pattern surrounding a spinning brake disk, see [10] for instance. Solving only the fluid domain opens also up the possibility computing an averaged surface heat transfer coefficient for the rotor that can be used to quantify the cooling capability. Solving the thermal problem solely for the solid domain allows speeding up computations. Such an approach is described in [8] incorporating a finite difference heat transfer model for an aluminum rotor using RadTherm [5]. Combining the solution process for the fluid and solid domain via coupling strategies leads often to improved accuracy. That kind of partitioned approach is introduced in [9], for instance, describing a solution procedure involving

the Lattice-Boltzmann-Code EXA PowerFlow [7] and PowerTherm.

Most of today's solution strategies which can be found in literature are using partitioned coupling consisting of a first pure CFD computation and a subsequent pure transient structural analysis incorporating mapped heat transfer coefficients which serve as boundary condition for the solid domain. The cooling-down temperature curve obtained in such a way may be used to quantify cooling capability. One possibility is to measure time the rotor needs for cooling from a pre-defined initial temperature down to a certain final temperature. A further approach is to perform regression analysis determining curve fitting parameters for validation purpose, see [11] for instance.

In our present paper we introduce a monolithic solution procedure enabling a strong coupling between fluid and solid domain during the complete cool-down process of a brake disk. That kind of solution strategy allows to estimate the cooling-down characteristics within a very high degree of accuracy. To reduce disturbances, i.e. side wind, moisture and ambient temperature, and also to keep the measurement setup easily comprehensible the reference experiment has been performed on an inertia dynamometer. The test rig enables also high repeatability. The capability of the introduced approach regarding full vehicle simulations covering influences of brake ducts and wheel types will be part of future investigations.

DEFINITION OF TEST CASE

Validations are carried out on the basis of measurements performed on an inertia dynamometer. In Figure 1 the used test rig is shown (right side) with the corresponding simulation model (left side) including the 17 inch vented front brake disk, the caliper, dust shield, bearing and knuckle. The geometry of the experimental chamber was virtually reconstructed for simulation purposes. A more detailed representation of the inner compartment is given in Figure 2. The air flow enters the chamber with a predefined velocity and temperature through the upper inlet and leaves the test section at the outlet. During cooling the heated disk rotates with constant speed and the temperature course over time is monitored via two thermocouples, embedded into outer and inner cheek, respectively, see Figure 3. Between brake disk and test rig a 3mm thick Teflon slice is inlaid to prevent conduction into the dynamometer components.

Figure 4 shows the used dust shield including two vent-holes and wrap-around feature. The experiments are performed with and without dust shield.

In order to achieve similar air flow conditions compared to disk brakes integrated into driving passenger vehicles, the dynamometer measurements are performed with constant air flow through the test chamber and constant rotating disk speed at the same time. Equivalent vehicle drive speeds are computed out of rotational velocities considering the dynamic

rolling radius. All three scenarios are listed in the following table:

Table 1. Air flow velocities and disk rotational speed for measurement and simulation

	Inlet air flow velocity [m/s]	Disk rotation [rpm]	Equivalent vehicle drive speed [kph]
1.	3.33	160	20
2.	8.33	400	50
3.	13.33	640	80

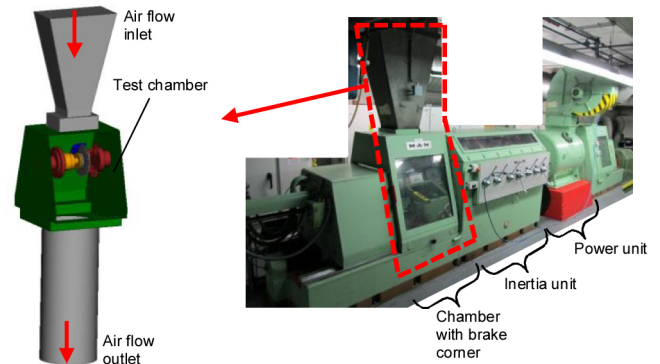


Figure 1. Inertia dynamometer (right) and corresponding simulation model (left)

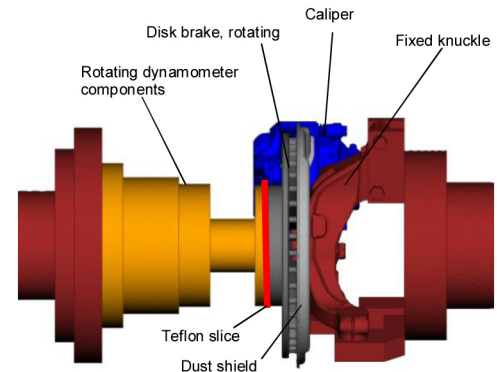


Figure 2. Inner compartment with disk brake, caliper, dust shield and knuckle

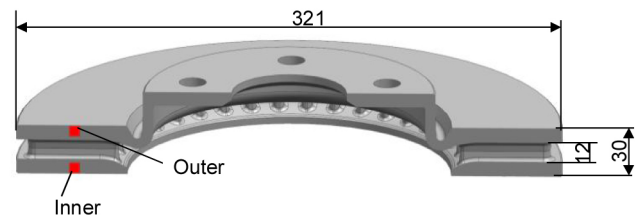


Figure 3. Section view of generic 17 inch vented rotor with positions of thermocouples red colored (material is gray cast iron)

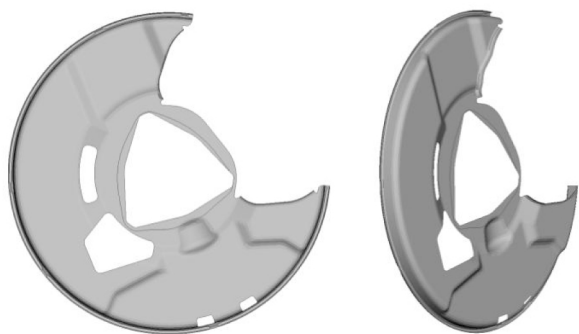


Figure 4. Side view and spatial representation of the used dust shield with 2 vent-holes and wrap-around feature

MEASUREMENT TECHNIQUES AND EXPERIMENTAL RESULTS

During preliminary test runs the question raised in which way the rotor has to be heated-up obtaining appropriate and realistic initial boundary conditions for the simulation. We choose an 80 snub brake-scenario that leads to almost quasi-stationary temperature distributions. Each single snub starts from an initial speed decelerated to a final speed. In case of the three applied velocities we used 99.6 to 80kph, 77.6 to 50kph and 62.6 to 20kph. After the last snub is reached the speed is decreased a last time to the final speed, the caliper remains opened and the temperature is monitored keeping the rotational velocity constant. The complete procedure is described in [Figure 5](#) for 80kph, without dust shield, for instance.

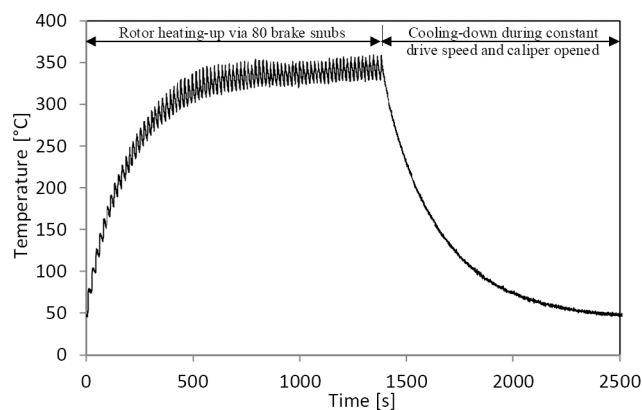


Figure 5. One measurement cycle of heating-up and cooling-down for 80kph, without dust shield

Temperature sampling-rate is adjusted to 2Hz. We employ two thermocouples of type K, class 1, with a size of 3mm × 3mm, embedded into rotor cheeks, see [Figure 3](#).

The cooling-down temperature curve is evaluated throughout regression analysis applying the following exponential function:

$$T(t) = T_{end} + (T_{start} - T_{end}) \cdot e^{-bt} \quad (1)$$

This leads to three regression parameters T_{start} , T_{end} and b , the cooling coefficient. Note that regression evaluation is performed always for the overall cooling-down time. To overcome issues regarding measurement deviations the complete measurement procedure, i.e. heating-up and cooling-down is repeated four times. Afterwards an averaging over the five cooling coefficients is carried out obtaining a single value per rotor cheek. For the evaluation we either use the mean cooling coefficient per rotor side or we average once again over both cheeks to get an overall final cooling coefficient.

[Figure 6](#) shows the measured cooling coefficients over speed. Notice that we always changed rotation speed and air flow velocity simultaneously according to [Table 1](#). Removing the dust shield leads to increased air flow to the rotor and thus to higher values. The cooling coefficient depends linearly on the drive speed for both dust shield configurations. In the right lower corner the corresponding straight line equations are given. The comparison reveals lower cooling values for mounted dust shield also accompanied by reduced slope.

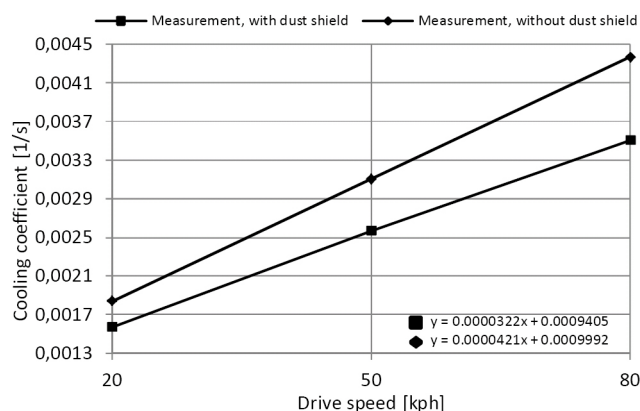


Figure 6. Measured average cooling coefficients

Corresponding to [Figure 6](#) the same results are listed numerically in [Table 2](#) together with their deviations derived from the five repeated measurements. The comparison illustrates that a maximum variance occurs for the lowest velocity of 20kph and both dust shield configurations.

Table 2. Evaluation of the measured cooling coefficients, b , with the corresponding deviations from the mean values (1/s)

Speed	Without dust shield	With dust shield
20 kph	0.00184 ± 0.00009	0.00158 ± 0.00010
50 kph	0.00311 ± 0.00002	0.00257 ± 0.00003
80 kph	0.00437 ± 0.00003	0.00351 ± 0.00004