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Replicating Brake Cooling Tests by using Coupled CAE Simulations

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Abstract

The present paper provides an overview of the different steps implemented to develop a simulation procedure which aims at replicating the Alpine descent brake cooling performance test. The Alpine descent brake cooling performance test is a driving scenario where brakes are continuously applied at down-hill driving for a prolong time when going down from a mountain. Results are presented and compared to experimental data obtained from the on-road testing. The comparison shows that the developed method is capable of predicting trends and deltas when doing case studies at concept phase of vehicle development. One of the examples of such studies presented in this paper is the effect of different rim designs on the brake cooling performance.

1 Introduction

Friction brakes are used to convert the kinetic and sometimes potential energy of the vehicle into thermal energy, hence, allowing the vehicle to decelerate or stop when that is required. The thermal energy goes into heating up the different components of the brake system and parts around it, and ultimately it is dissipated via radiation and convection into the environment [1]. In case if the dissipation process is not fast enough or the energy input is too high, the temperatures in the system can increase to the levels when the brake performance will become affected. Overheating of the brake system components is a safety issue as it can result in many negative effects including brake fading, thermal cracking or even brake fluid vaporization [2].

When the transition to hybrid and battery electrical vehicles (HEV & BEV) took hold, it was a common misunderstanding that the friction brakes will become obsolete or less important as electrical motors may be used for energy recuperation converting vehicle kinetic energy into electricity. However, even though the friction brakes are used less frequently in city driving, they are still required for the extreme scenarios. Hence, good brake cooling characteristics are still important for those type of vehicles.

Traditionally, the brake systems are tested experimentally using different test benches as well as full vehicle testing. However, nowadays a lot of the tests can be replicated by CAE simulations which are cheaper and can also provide information that is difficult to obtain in real life tests. Moreover, the simulations can be performed at much earlier stages, such as concept, avoiding possible later problems in vehicle development, reducing the lead time for parts and permitting to reduce the total length of vehicle development.

To ensure the reliability of the simulation results it is of high importance to validate the CAE methods against the experimental data. The present study summarizes and discusses one of such methods being developed to at least partially replace one of the

standard brake tests at Volvo Car Corporation. One example of how this method can be applied is also going to be provided.

2 Experimental setups

There are several tests that are typically performed to ensure good brake cooling performance for passenger vehicles. One of such tests is Alpine descent, represented by the Figure 1. The Alpine test consists of two phases:

- 1) downhill driving on a slope of around 10% with constant braking, maintaining a velocity of 10 m/s,
- 2) thermal soaking when the vehicle stands still with engaged brakes while heat dissipates into the brake system and its environment with natural convection.

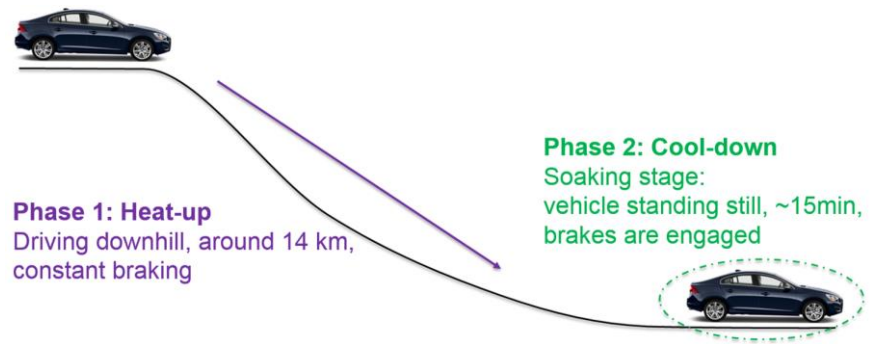


Figure 1. Alpine descent test scenario

Several thermocouples are usually used to monitor the temperatures of the different parts, but the main focus in this test is on the brake fluid temperature. If the brake fluid reaches its boiling point, it will result in losing the braking function and, accordingly, the boiling scenario is unacceptable.

Testing all possible vehicle variants in the Alps is obviously impossible for several reasons, and therefore, there are simplified variants of the Alpine decent test which can be performed. The vehicles can be tested on a dynamometer test bench, inside the wind tunnel or on a test track. The two latter test scenarios are going to be used in this paper to compare the experimental results against the simulation ones.

2.1 Wind tunnel test

Wind tunnel tests provide a highly controlled environment with precise air speeds and temperatures, moreover, the exact braking force can be measured and maintained. For this paper the tests were performed in a full-scale closed-circuit wind tunnel at Volvo Car Corporation test lab facilities [3]. Obviously, there are a number of differences in the test setup compared to the typical aerodynamic test. First of all, since high torque to the wheels is required, the test needs to be performed on a dynamometer part of the wind tunnel test section and not on the moving ground system. This means that not only the ground under the vehicle is standing still, but also the rear wheels are fixed in their positions, see Figure 2. To minimize the boundary layer effects, the suction scoop, moving belts and distributed suction zones in front of the vehicle were switched on during the heat-up phase of the test. Another big difference was the running engine of the vehicle which means that additional tubes had to be connected to the tail pipes of the vehicle to remove the exhaust gasses.



Figure 2. Test object in the wind tunnel

The test object was a full-size production sedan type vehicle (Volvo S60) with 300 mm brake discs. One of the front wheelhouses was equipped with around 120 thermocouples measuring surface and air temperatures, including 8 measuring points inside the brake disc itself and one sensor measuring the brake fluid temperature inside the caliper. Information on the exact setup used during the wind tunnel investigations can be found in [4].

2.2 Test track investigation

Wind tunnel tests were mainly used to validate the simulation procedure developed, however, in reality the wind tunnel tests are quite expensive and require a lot of preparation work. Hence, most of the investigations for the brake cooling performance are actually done on a flat test track. In this case, the tested vehicle is braking, and the second vehicle is constantly pushing the first one with a controlled force. This is done to replicate the effects of the gravitational force during the real test. Such investigations are the closest option to real life Alpine decent as they include testing on actual roads with all complications and uncertainties provided by the environment. Also, temperatures are compensated to mimic temperature gradient between top and bottom of the mountain. Normally, to ensure a faster process the number of thermocouples in these tests is limited to brake fluid temperature sensor and one sensor in each of the brake pads.

In order to show how the simulation results can be compared to the test track results, six separate tests were conducted during 2 consecutive days.

3 Numerical simulations

It has been shown before that the airflow through and around the brake disc is highly affected by the surrounding parts in the wheelhouse [5]. Moreover, it has been observed on multiple occasions that the temperatures of the brakes and air around them on two different sides of the car can vary significantly due to the packaging of the engine compartment [4]. Therefore, it is of high importance to use a fully detailed geometry for the simulations of the airflow around the brakes.

Standard aerodynamic simulations do not include the energy equation or any temperature related effects. In addition to that, brake discs are usually simulated with moving wall boundaries, which can significantly affect the amount of mass flow through the discs and, hence, the calculated cooling performance. Consequently, the standard aerodynamic procedure needs to be extended and developed to address these issues. To accomplish this, a simplified setup was firstly tested to check for the mesh independent solution and boundary conditions for the area inside and around the brake discs.

Later, to simulate the Alpine decent test in the CAE environment a coupled semi-transient method was developed. In this method the airflow and, hence, the convective heat transfer coefficients (HTCs) were calculated using commercial CFD code: StarCCM+ v11. In parallel, a separate thermal solver (TAITherm v12) was used to calculate conduction and radiation.

3.1 Simplified numerical setup

To obtain an appropriate mesh and boundary conditions settings a simplified test case with just a single hot rotating disc inside the domain was used. Figure 3 shows the computational domain and brake disc geometry with a section cut to demonstrate the vanes inside the disc. The boundaries of the domain were assigned as velocity inlet, pressure outlet and symmetry plane for all the rest of the domain surfaces. Three different rotation techniques were tested for the brake disc itself:

- moving wall (MW) for all disc surfaces,
- multiple reference frame (MRF) for the vanes region, separated from the main domain by the interface surfaces seen in Figure 3, with moving wall for all external brake disc surfaces,
- rigid body motion (RBM) for inner vanes with moving wall for the rest of the disc geometry.

A mesh dependency study was performed to ensure mesh-independent solution.

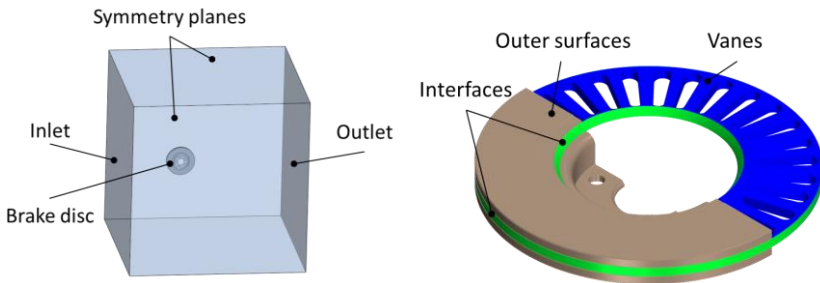


Figure 3. Simulation domain and brake disc geometry

Information on the exact models and simulation settings used can be found in [6].

3.2 Full vehicle coupled setup

Alpine decent test takes around 50 minutes of real time, and this governs the approach that must be used in order to simulate it numerically. The method developed included coupling of CFD and thermal solvers, this coupling was performed using the CoTherm v1.1 software.

The following subsection will include the summaries of the aerodynamic and thermal setups for the method developed as well as a short description of the coupling process. More information on this method is available in [7].

3.2.1 Aerodynamic simulation setups

A modified version of a steady state RANS aerodynamic procedure was utilized. The modifications included the following:

- using central symmetry plane to simulate only one half of the car,
- removing excessive cells in regions that are not important, especially in the wake of the vehicle,
- adding refinement regions inside and around the front wheel-house,
- introducing the rotational region inside the disc vanes to obtain the correct mass flow through them.

The latter was done using multiple reference frame (MRF) method, as it was shown by simplified studies to provide good enough accuracy for significantly less cost as the rigid body motion approach.

From the physical modelling perspective, the standard procedure was modified by using incompressible ideal gas model as well as including energy equation and taking gravity into account to simulate the temperature gradients in the air and buoyancy force effects. These effects are indisputably important for the cool-down phase of the test, when the natural convection is driving the flow. Later, it will also be shown that they cannot be ignored for the heat-up phase as well.

The same volume mesh was used for both heat-up and cool-down simulations with total cell count of around 35-45 million predominantly hexahedral cells depending on the vehicle variant being tested. The boundary conditions for the vehicle surfaces as well as the virtual wind tunnel were changed in between the two phases to account for changing conditions in the real-life test.

3.2.2 Thermal simulation setups

Thermal model includes all parts that are in close proximity to the hot brake disc and participate in the heat dissipation from it: brake disc, brake pads with shims and back-plates, caliper with all internal parts, wheel rim and tire, dust shield, wheel hub and knuckle. A typical mesh for the model contained 3-5 million cells, an example can be seen in Figure 4.

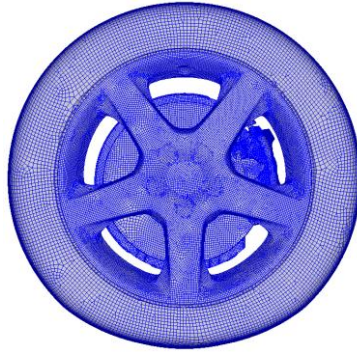


Figure 4. Meshed geometry of a typical TAItherm model

To ensure the correct heat transfer inside the model, all contiguous parts are connected using thermal links. During the heat-up phase, the full friction surfaces of the disc and pads are connected using generic thermal links with the energy source added in between them. This allows to split the amount of energy generated during the braking event that goes to the disc and pads without introducing any fixed ratio for these energy fluxes. For the cool-down phase this node is removed and the thermal link is modified to only connect the pad to the corresponding section of the brake disc.

The brake fluid was simulated as a solid with fluid material properties, as it was found to produce better results compared to simulating it as a single fluid node.

3.2.3 Coupling procedure

The coupling process in between two codes is relatively straightforward. The CFD simulations are used to provide the thermal solver with the convective surface heat transfer coefficients and reference temperatures. The thermal solver uses this data to compute the convective heat fluxes from the surfaces. Calculating the energy fluxes due to convection, conduction and radiation in the model, the thermal solver is advancing in the simulation time to obtain new temperature distribution inside the parts and on their surfaces. The surface temperatures are then mapped back to the CFD simulation to recalculate the air flow around hot parts. The process is looped to obtain a gradual change in convective energy fluxes and ensure a converged solution for every time interval.

When the vehicle is stopped in between heat-up and cool-down phase the solid temperatures are being mapped from one thermal model to another one and the new buoyancy driven airflow is being recalculated based on the surface temperatures and updated boundary conditions of the virtual wind tunnel.

For some of the brake cooling simulations the assumption of heat transfer coefficients being independent of surface temperatures can be used [8], however, it will be shown here that for the Alpine descent such approach cannot be used even for the heat-up phase, as the vehicle speed is too low, and hence, the flow is strongly affected by the surface temperatures.

3.3 Rim design comparison

The developed simulated Alpine decent procedure can be used for many case studies including changing materials or part shapes/dimensions inside the brake system and around it. One interesting option is to look at different rims and how they affect the brake cooling performance, and an example of such study is presented below.

A newly developed vehicle variant prototype was simulated with three different options for the wheel rims, see Figure 5. Remaining vehicle geometry and parameters were unchanged. Three rims included a baseline design, that was relatively open, and two other rims, referred later simply as rim 2 and rim 3. The two additional rims had more covered designs, since such variants are known to be beneficial for reduction of aerodynamic drag of the vehicle [9].



Figure 5. Three rim designs investigated (baseline – left, new designs – middle and right)

4 Results and discussions

4.1 Simplified simulations

The main goal for the simplified studies was to eliminate possible mesh dependency of the results in the region around the brake disc. Eight different meshes were tested to find the meshing strategy inside the disc vanes and around the braking system. These findings were later applied to the wheel-house region meshing in full-vehicle simulations.

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The second goal for the simplified simulations was to test different rotation simulation techniques for the vanes inside the disc. Table 1 summarizes results of this investigation.

Table 1. Comparison of the rotation simulation methods

	MW	MRF	RBM
Mass flow through the vanes, kg/s	6.34E-03	4.56E-02	4.75E-02
Average HTC for the disc vanes, W/m ² K	14.9	49.4	55.7
Average HTC for the friction surfaces, W/m ² K	21.3	29.6	29.0
CPU-hours required for the simulation, h	9	10	115

As it was expected, just a moving wall (MW) rotation does not produce satisfactory results as the vane surfaces will be assigned incorrect boundary conditions resulting in almost negligible mass-flow through the disc. The MRF and RBM methods are close in terms of their results. However, rigid body motion requires running transient simulations and then averaging of the results, hence, increasing the computational power required 10 times compared to the MRF approach. Keeping in mind that full-vehicle simulations will be run more than 100 times for recalculating the flow, the MRF approach was considered to be more appropriate.

Additional tests for heat transfer and pumping effect dependency on the different simulation parameters as well as solver settings were performed and can be found in [6].

4.2 Testing a fixed HTCs approach

To test a popular approach of considering heat transfer coefficients independent of the temperature and only dependent on the vehicle velocity, the average temperatures of different hot parts were assigned to corresponding boundaries in a CFD simulation and the “averaged” HTCs were calculated. These convection coefficients were mapped to a thermal solver and the simulations of the heat-up phase were performed. It may be pointed out that even though the HTC values for particular boundary nodes are not changing and the reference fluid temperature is fixed, the convective heat fluxes are not constant, since they also depend on the surface temperatures.

The results of this investigation are presented in Figure 6. It can be seen that the brake disc temperatures are developing in a similar way, even though minor differences can be observed in the curvature and the maximum temperatures the disc reaches in the end of the heat-up phase of the test. However, the temperatures for the brake fluid show

quite a big difference between the two methods. Hence, even though average HTC approach may simplify the method and decrease the computational cost, it was considered to be unacceptable for simulating this particular test.

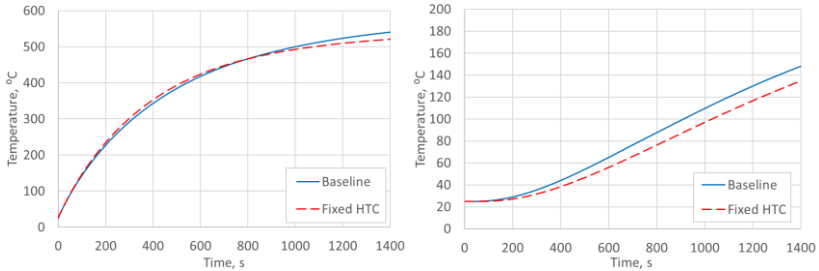


Figure 6. Brake disc temperature (left) and brake fluid temperature (right) comparisons

4.3 Experimental validation of the results

The results from the wind tunnel investigations were mainly used to tune the method and obtain good correlation between the simulation and reality. Results can be found in [4]. The following figures are presenting the correlation between the simulations and the data obtained from the test track runs.

Figure 7 and Figure 8 show the temperature history of a measurement point inside the brake pad and inside the brake fluid respectively. These measurements were obtained during 6 different tests and the color-coding represents the fact that the experiments were conducted on two different days with slightly different environmental conditions. It may be seen that during day 1 there was a problem with the pad thermocouple failing during the cool-down phase of the test. This problem was fixed for day 2. The figures show one of the main problems with such tests which is the repeatability as the results are being affected by the test track features, weather and many other factors.

Comparing simulation and experimental results, it can be noted that the general behavior is captured well. There is a difference in brake fluid temperatures during the cool-down phase which reach the maximum temperature slightly faster in the simulation as compared to the experiment, but the maximum temperature value itself is close to the experiments.

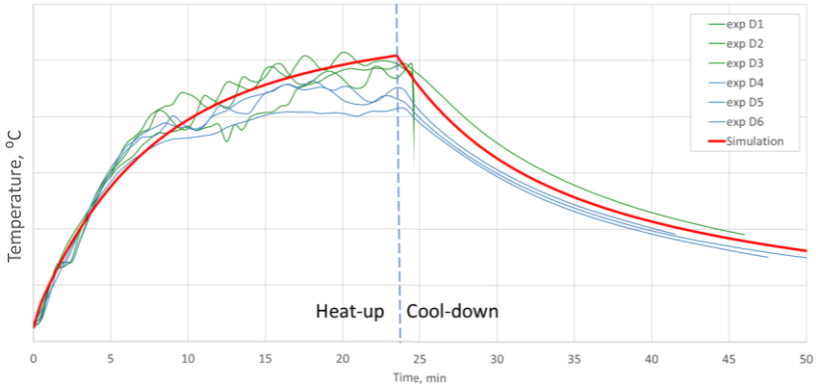


Figure 7. Results comparison for the brake pads temperatures.

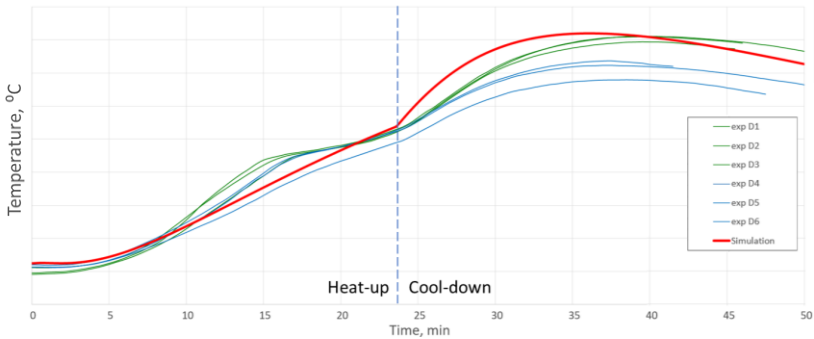


Figure 8. Results comparison for the brake fluid temperature.

4.4 Effects of the rim designs on the cooling performance

The results of the case study investigating three different rim designs are presented in Figure 9. A bold dotted red line represents the maximum requirement temperature for the Alpine decent test scenario, which should not be exceeded due to safety reasons. One can see that covering the rims has a significant effect on the brake fluid temperature and both aerodynamic rims perform much worse compared to the open baseline rim from the brake cooling performance perspective.

The simulations show that rim 3 should not be used on the particular vehicle variant, while rim 2 is predicted to be one of the borderline designs when the brake fluid is getting really close to the maximum allowed temperature. An obvious recommendation here would be to test this rim experimentally before adding it to the official rim selection.

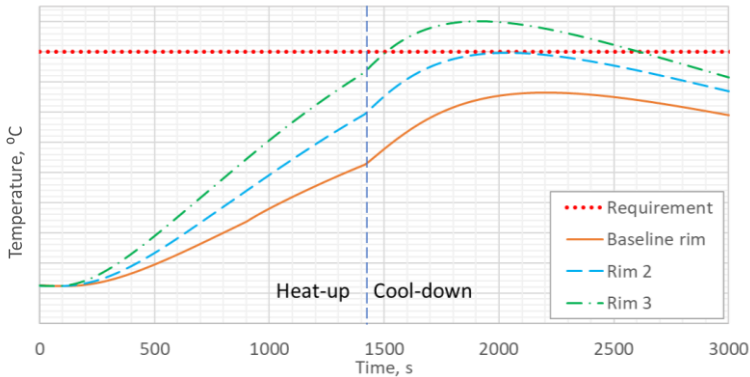


Figure 9. Effects of the rim design on the brake fluid temperatures.

This result shows one more time that the trade-offs are always needed when designing a vehicle, since improving one area most often will negatively affect the other one.

5 Conclusions

The main idea of the developed coupled semi-transient procedure for simulating the Alpine decent test is that it can be used in the concept phase of vehicle development. It has been shown that the method produces reliable results and can be used for predicting trends and deltas in between different case scenarios. Though, due to a large number of simplifications and uncertainties the exact matching between simulations and experiments could not be achieved.

Another important benefit of the method is that it allow to get a different view on the brake system heat dissipation problem, providing not only the temperature gradients, for any point in the system at any time, but also the heat flux distributions for radiation, conduction and convection for every part surface or part contact.

As the computational resources become more affordable, the amount of tests that can be replicated by CAE simulations will increase, however, much more work is needed to improve the existing methods and develop new ones before they can actually replace physical testing.

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