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PREDICTION OF THE COOLING FACTORS OF A VEHICLE BRAKE DISC AND ITS INFLUENCE ON THE RESULTS OF A THERMAL NUMERICAL SIMULATION

M. PEVEC[1)*], I. POTRC[2)], G. BOMBEK[2)] and D. VRANESEVIC[1)]

[1)]Product Development - Brake & Suspension, Cimos d.d., cesta Marezganskega upora 2, Koper 6000, Slovenia
[2)]Faculty of Mechanical Engineering, University of Maribor, Maribor 2000, Slovenia

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ABSTRACT−In the process of developing the brake disc, it is necessary that we predict the suitability of the design. In this manner, we can affirm that even the first prototype will satisfy all of the customer homologation requests. Usually those comprise different sequential braking tests in which the maximal achieved temperature is the criterion that governs brake disc suitability. The knowledge of how to predict the behavior of a brake disc in the early pretesting phase has a significant impact on development costs and time. The common method that is used for predicting the temperatures in the brake disc during braking is numerical simulation analysis. With the help of Computational Fluid Dynamics, the flow through a vehicle ventilated brake disc of known geometry was determined, and the wall heat transfer coefficients for all vehicle speeds and brake disc temperatures were calculated. The results were then imported into a thermal numerical simulation of a sequential-braking vehicle test. The results showed that the consideration of cooling factors has a significant impact on temperature courses. To obtain accurate results from the numerical simulation and to simulate the vehicle test precisely, the proper wall heat transfer coefficients must be considered. The proposed method produces more accurate numerical results and enables the development engineer to develop suitable brake disc geometry in the early pretesting phase.

KEY WORDS : Ventilated brake disc, Heat transfer coefficient, CFD and FEM, Temperature loads

1. INTRODUCTION

During braking, the kinetic energy and the potential energy of a moving vehicle are converted into heat. The majority of this energy is absorbed by the brake disc and then transferred to the surrounding air. Solid brake discs dissipate heat slowly. Therefore, ventilated discs are currently commonly used in vehicle brake systems to improve cooling by enhancing the air circulation. The ventilated brake disc behaves as a centrifugal fan, drawing cool air from the inboard side, passing it through the disc passages and exhausting it at the outer diameter. It has been noted that ventilated disc brakes generally exhibit convective heat transfer coefficients that are approximately twice as large as those associated with solid discs (Limpert, 1975).

Brake discs are developed for every specific vehicle because the braking power is dependent on vehicle weight, maximal speed etc. To date, the usual procedure for developing the brake disc was to copy the design from a similar vehicle or just to repeat the previous design with slight modifications. If the design proved to be satisfactory during the vehicle tests, it was approved. Nobody actually knew for sure whether it would pass the homologation vehicle test procedure until the test was performed. Development engineers are now trying to predict the behavior of the brake disc before it is physically tested. Currently, numerical analysis has become a powerful tool in the process of developing a brake disc. There is a variety of computer programs for performing finite element calculations that offer insight into the operation of a brake disc.

In the last few years, thermal analyses of brake discs have become very popular, and many papers have been written on this topic. A thermomechanical analysis of a train brake disc using an estimated heat transfer coefficient was conducted (Reibenschuh and Oder, 2009), and the stresses due to temperature loading were determined. A similar analysis was conducted on vehicle brake discs (Gotowicki *et al*., 2005). Again, the average convection coefficient determined by the experimental work of the brake industry was considered. The experimental results are between 90 and 100 W/m^2 K, taking into consideration the average disc temperature and speed. The latest research is focusing on the improvement of cooling using computational fluid dynamics (CFD). Until now, though, the research has been limited mostly to commercial and train brake discs (Galindo-Lopez and Tirovic, 2008), although the majority of brake discs are installed in

*Corresponding author. e-mail: miha.pevec@cimos.eu

passenger vehicles. Some research has been performed on passenger vehicle brake discs as well. The geometry of cooling channels was optimized to promote brake cooling (Palmer *et al*., 2009). It is reasonable to continue the work in this field, not only to investigate the optimal channel geometry for maximal cooling efficiency but also to include CFD calculations in the thermal numerical analysis. This is a new method that attempts to make numerical analysis as accurate as possible.

The presented methodology will take the calculated cooling factors for known geometries into consideration and provide more accurate results for the thermal course during a standard sequential braking test. The results of the temperature distribution from the numerical analysis that considers cooling will be compared with the same analysis without cooling.

In the first part of the paper, the cooling factors for brake discs will be calculated for two different brake disc regions. The factors will then be organized in such a way that they can be used as a boundary condition in a thermal analysis. In the second part, the thermal analysis of 10 sequent brakings will be made. First, the analysis will be conducted without consideration of cooling. The second identical analysis will take the previously calculated cooling factors into consideration. The results of the two analyses will then be compared to investigate the accuracy of the method that considers cooling.

2. HEAT DISSIPATION FROM A BRAKE DISC

The evaluation of heat transfer coefficients requires consideration of the individual contributing mechanisms, which for a brake disc involves conduction, convection and radiation. In terms of convective heat transfer, the mechanism can be considered in two parts. The first involves convection from all external surfaces. The second, and of particular interest, involves convection through the radial passages (Galindo-Lopez and Tirovic, 2008). In this analysis, those two regions will be named rest for the external surfaces and vent for radial passages.

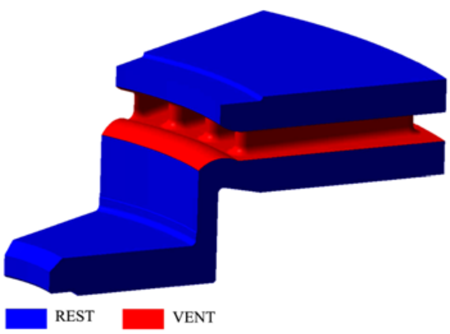

Figure 1. Two separated regions ("rest" and "vent") for the heat transfer coefficient calculation.

Table 1. Disc temperatures for the numerical calculation of heat transfer coefficients.

Disc temperatures [ºC]
25
50
100
200
300
400
500
600
700
800

3. NUMERICAL CALCULATION OF HEAT TRANSFER COEFFICIENTS

The airflow through and around the brake disc was analyzed using the ANSYS CFX software package (ANSYS, 2009). Afterward, the heat transfer coefficients considering convection and radiation were calculated and organized in such a way that they could be used as a boundary condition in the thermal analysis. Averaged heat transfer coefficients for the two separate regions had to be calculated for all rotation speeds and temperatures. The analysis was made for vehicle speeds of 5, 25, 50, 75, 100, 125, 150, 175 and 200 km/h with every disc temperature shown in Table 1. That means the calculation was repeated 90 times.

3.1. 3D Model of the Brake Disc

The design and basic dimensions of the passenger vehicle front brake disc are shown in Figure 2. The brake disc has 41 cooling vanes equally spaced, which enables the use of symmetry with a basic angle of 8,78º. This model was chosen because serial production of this disc existed in the Cimos factory, which enables the simple production of test specimens. A 3D model with three cooling vanes was chosen for the CFD numerical calculation, which means that the 3D model comprises an angle of 26,34º.

3.2. CFD Analysis of the Brake Disc

3.2.1. Mesh model

The model for the calculation represents a 26,34º cake-slice-like region around the brake disc section with a height of 4x the brake disc height and a length of 3x the disc radius, as shown in Figure 3. Knowing that the most important region of the model is the area around the brake discs walls, the mesh was sized to provide good quality in this region. The rest of the model was more roughly meshed, with no particularly care paid to the regularity of

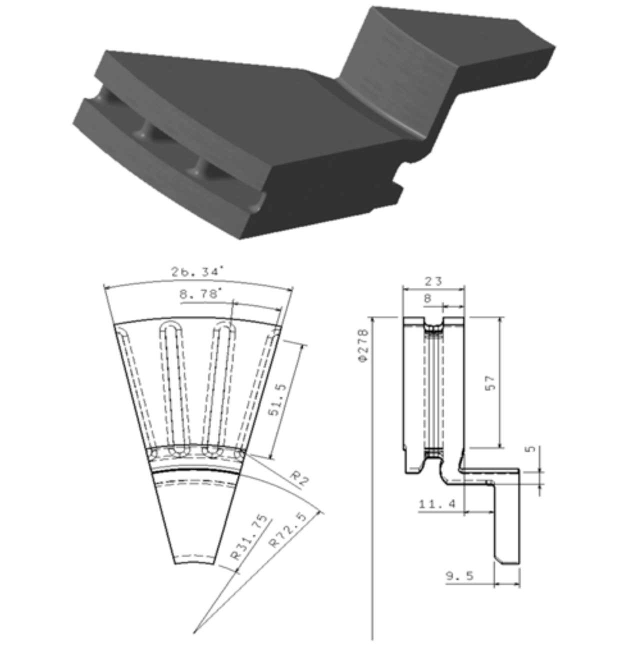

Figure 2. Geometry and basic dimensions of the analyzed brake disc.

the mesh because it serves only for visualization of the inflowing and outflowing air patterns.

The model was meshed using the patch-confirming tetrahedron mesh method, which produced 649272 elements and 125091 nodes. A tetrahedron mesh was used in this analysis because of its versatility and the ease of automatic mesh generation, but further research should be performed to compare it with other mesh types and improve its quality.

3.2.2. Boundary conditions

The model applies periodic boundary conditions to the section sides. Because the brake disc is made from sand-cast grey cast iron, the surface roughness is taken to be 100 μm. The disc surfaces are heated uniformly. The disc model is attached to an adiabatic shaft whose axial length spans that of the domain. The air around the disc is considered to be 30ºC, and open boundaries with zero relative pressure were used for the upper, lower and radial ends of the domain.

A range of constant angular speeds was modeled using a rotating frame of reference about the disc and shaft axis. In Figure 3, it is shown as a periodic boundary condition. This was necessary because the solver does not allow the rotation of the instance, only the rotation of the boundary conditions around the instance.

Material properties were taken from the Ansys material data library for air at 25ºC. The reference pressure was set to be 1 atm because the brake disc operates in open air. The turbulence intensity was set to a low value because the relatively slow airflow and turbulent model used was k-ε

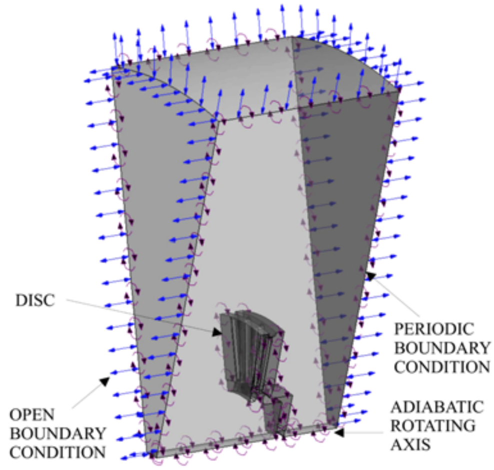

Figure 3. Boundary conditions for the CFD analysis of the brake disc.

(ANSYS, 2009), as suggested in similar analyses (Chi, 2008). Because radiation was also taken into consideration, the emissivity was set to be 0,55, with a diffuse fraction of 1, which are the physical properties of the material used for brake disc production, grade EN-GJL-250 grey cast iron (Galindo-Lopez and Tirovic, 2008). The buoyancy effect was neglected because the air pumping action of the brake disc is almost entirely dependent on the rotation action of the brake disc (Galindo-Lopez and Tirovic, 2008).

The Ansys CFX solver automatically calculates the heat transfer coefficient at the wall boundary using the following equation:

$$h_c = \frac{q_w}{T_b - T_{nw}} \tag{1}$$

where h_c is a specified heat transfer coefficient, q_w is the heat flux at the wall boundary, T_b is the specified boundary temperature (that is, outside the fluid domain) and T_{nw} is the temperature at the internal near–wall boundary element center node (ANSYS, 2009).

To enable universal application of the results, the solver had to be set to compute the convective heat transfer coefficient relative to the ambient (constant) temperature:

$$h_c = \frac{q_w}{T_b - T_\infty} \tag{2}$$

where T_∞ is the constant ambient temperature.

3.3. Results of the CFD Analysis

The result converged on the preset residual target of $1 \cdot 10^{-5}$ in approximate 130 iterations, which indicates that the mesh and boundary conditions were properly chosen.

The air streamlines shown in figure 4 have a direct influence on the heat transfer coefficients shown in figure 5. It can be noticed that where the cooling air has a greater velocity, the heat transfer coefficients are higher. Figures 4

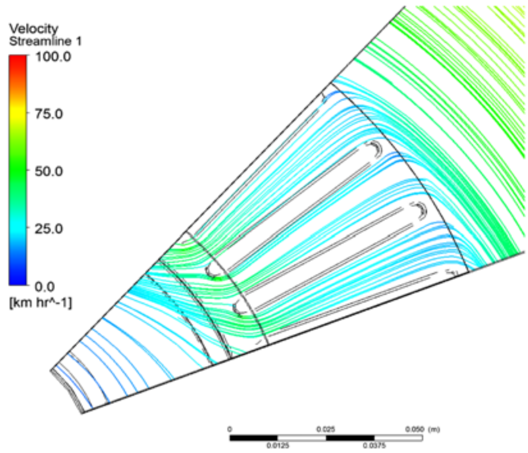

Figure 4. Streamlines of cooling air through a ventilated brake disc.

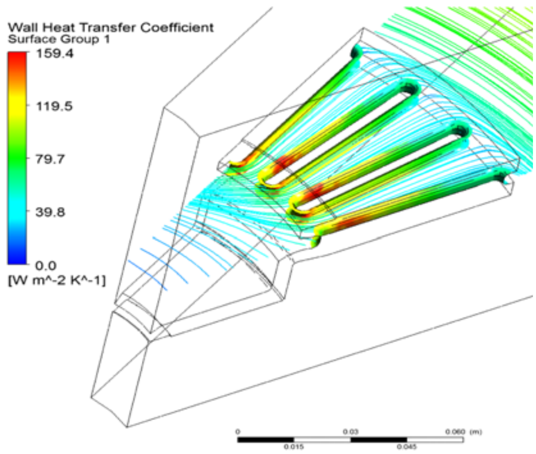

Figure 5. Wall heat transfer coefficient factors as a result of air flow through the brake disc.

and 5 represent the air flow at a vehicle speed of 100 km/h and a brake disc temperature of 300°C.

If we examine figure 5 more carefully, we can notice that the outer end of the cooling vanes is less efficient at cooling than the inner end. This suggests that the design of the cooling vanes is not yet perfected because the pressure drop is too high. This opens up an opportunity for further research.

3.3.1. Preparation of the results for input into the thermal analysis

For further application of the calculated data in this research, the wall heat transfer coefficients had to be arranged into a form that could be used as inputs into a thermal simulation. The wall heat transfer coefficient had to be arranged into averaged values at specific disc temperatures and speeds.

Figure 6 shows the arranged wall heat transfer coefficient values that can be used as input in the following thermal numerical analysis of brake discs. The film coefficients are presented as a function of vehicle speed and brake disc temperature. Between the calculated values, the results were averaged with the help of Microsoft Excel. The quadratic form of the curves is a consequence of thermal radiation, and the offset of the curves is a result of the enlarged forced convection at higher vehicle speeds. Because the course of wall heat transfer coefficients is clearly evident, the number of calculation repetitions can be drastically lowered by defining curved courses with functions. Further research should be performed to determine the accuracy of this method.

4. THERMAL CALCULATION OF THE BRAKE DISC

4.1. Testing Procedure Description

In the following thermal analysis, the AMS brake test (ESSE, 2005) was simulated using Abaqus finite element software (Dassault Systems, 2008).

This procedure was chosen because of its widespread use by almost all of the European car manufacturers. It must also be noted that every sequential braking test would be appropriate for the type of analysis that is presented in this paper.

The AMS procedure is a standard brake evaluation

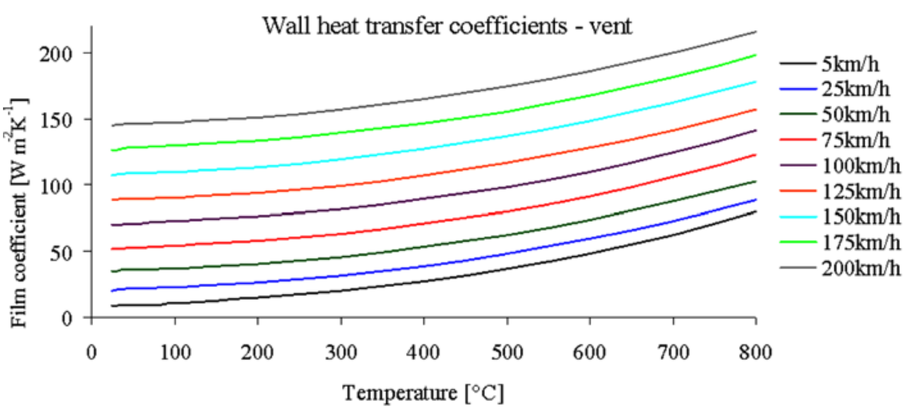

Figure 6. Averaged and arranged data for thermal analysis input.

vehicle test initially performed by Auto-Motor-Sport magazine to compare competitive vehicle brake performance (ESSE, 2005). It consists of 10 basic cycles, each including one braking with deceleration of 1 g from 100 km/h and one cooling, while the vehicle is accelerating back to 100 km/h. The whole calculation is a sequence of 10 basic cycles. Because of vehicle acceleration, a new variable for vehicle speed had to be introduced in the numerical simulation, *"FV1"*, which is changing throughout the analysis. This is necessary because the cooling intensity is dependent upon the air flow around the disc and therefore upon the vehicle speed. This dependency is represented by the Film Coefficient Property being dependent upon *FV1*. The output of the thermal analysis is a temperature distribution at various time intervals.

The course of *FV1* is shown in figure 7. The basic cycle begins with *FV1* = 100 km/h, which then linearly falls to 0 km/h at the end of the braking phase (t = 2,8 s), at which point the acceleration phase starts, and *FV1* again rises to 100 km/h in 15,9 s. The total basic cycle time is thus 18,7 s, and the total AMS procedure time is 187,8 s.

4.2. Load and Boundary Condition Determination

The brake disc is symmetric, so only a 26,34° sector was modeled. Temperature-dependent material data were determined in the Cimos material laboratory for laminar gray cast EN-GJL-250. Some of basic physical properties at 20°C are shown in table 2.

Great care was taken to produce as regular and as undistorted a mesh as possible. Therefore, all of the small fillets were removed, and more important ones were replaced by a chamfer. The mesh is produced from 3 separate parts (hub disc, free disc and vanes) that are later

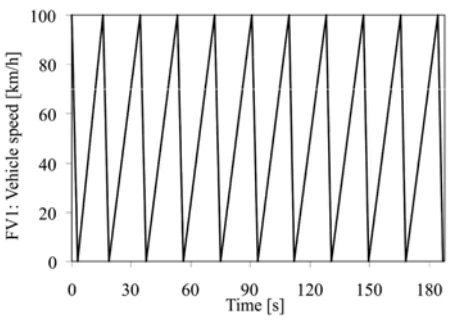

Figure 7. AMS test – *FV1*, course of time.

Table 2. Material properties.

Material properties	Value
Heat conductivity – λ [W/m K]	45
Density – ρ [kg/m^3]	7200
Specific heat – c_p [J/kg K]	537

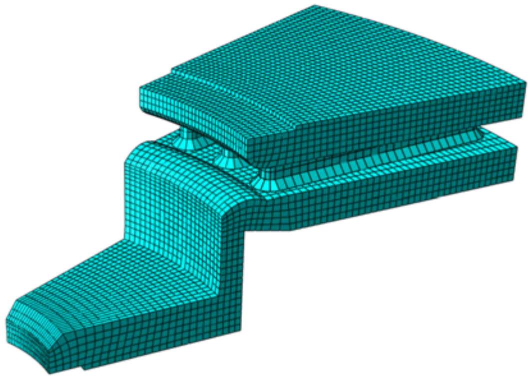

Figure 8. Mesh of a segment of the brake disc.

assembled using TIE constraints. Element type DC3D8 (linear heat transfer brick) is used in the thermal analysis. The mesh is shown in Figure 8.

Cooling due to heat convection and radiation was modeled with the Surface Film Condition. The film coefficient is defined as a property table in which the coefficient itself is dependent upon the part temperature and the vehicle speed, as discussed in the previous chapter.

The disc is divided into two areas: cooling vanes and the rest of the disk. The "vent" film condition property is applied to cooling vanes and inside surfaces and the "rest" property to the rest of the disc. The surrounding air temperature (sink temperature) is defined to be 30°C. The initial disc temperature is 100°C, as defined in the AMS standard (ESSE, 2005). The side area of the modeled sector is adiabatic to model the symmetry of the part.

During each braking, a Surface Heat Flux with the beginning value of 2,7 W/mm^2 is applied to the surfaces under the brake pads, which then linearly falls to 0 W/mm^2 at vehicle standstill. The heat flux was averaged and was predicted to be constant within the radial direction of the disc. The surface plotted in Figure 9 represents the actual contact between the brake disc and the brake pad in the discussed brake disc. The brake pads have an outer diameter that is 0,5 mm smaller than the outer diameter of the brake disc, and the inner diameter is 12 mm larger than the inner diameter of the brake disc without counting the cold-ring section. The same brake pad dimension is used for both sides of the brake disc.

The heat flux was taken to be constant in the radial direction because of the simplification of the analysis. In addition, the rotation of the brake disc and the heat flux were not taken into the account for the same reason, and the analysis performed is stationary.

The surface heat flux acting on the brake disc due to vehicle braking was calculated using basic vehicle data that were available, such as gross vehicle mass, rear-to-front brake distribution, share of heat absorbed by the brake disc and basic brake disc geometry, as shown in Figure 2.

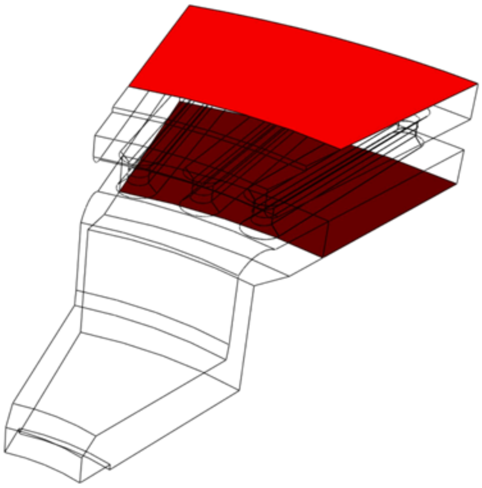

Figure 9. Load – surfaces under brake pads.

The heat flux was calculated using equation 3:

$$Q(t)=\frac{P_{disc}-\left(\frac{P_{sidc}}{t_{stop}}\cdot t\right)}{A_{friction}}=\begin{cases}\dot{Q}=2,7\frac{W}{mm^2},t=0s\\ \dot{Q}=0\frac{W}{mm^2},t=t_{stop}\end{cases} \quad (3)$$

where P_{disc} is the braking power at each front disc, t_{stop} is the time of braking to standstill, and $A_{friction}$ is the area of the friction ring that is the entire rubbing path of the disc where the brake pads contact the brake disc.

$$P_{disc}=\frac{m_{vehicle}\cdot|a|\cdot d_{f/r}\cdot v_{AMs}\cdot \eta_{disc}}{2} \quad (4)$$

The braking power is calculated using $m_{vehicle}$, which is the gross vehicle mass, a is the deceleration, $d_{f/r}$ is the front-to-rear brake distribution, v_{AMS} is the initial velocity of the braking procedure and η_{disc} is the share of heat absorbed by the brake disc.

Because we know the brake pad geometry, it is easy to calculate the area of the friction ring:

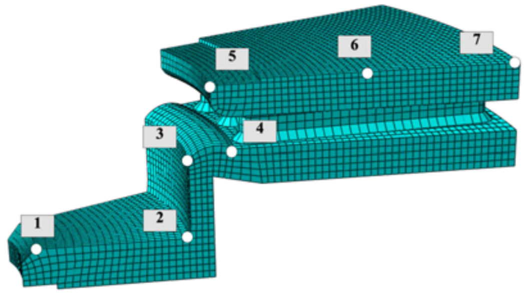

Figure 10. Points for the temperature plot.

$$A_{friction}=2\cdot\Pi\cdot(r_o^2-r_i^2) \quad (5)$$

where r_o represents the outer diameter of the friction ring and r_i represents the inner diameter of the friction ring.

The heat flux has the greatest value at the beginning of braking and then falls linearly to 0 at the end of braking, remaining 0 when the vehicle is accelerating. Another cycle repeats itself when the vehicle is braking again.

5. ANALYSES OF RESULTS

The temperature courses are presented at points on the selected brake disc shown in figure 10.

5.1. Analysis of the Results without Considering Cooling

The first analysis was conducted with no cooling consideration. That means that all entering energy stays in the brake disc. The consequence is a constantly rising temperature, as shown in figure 11. On the points immediately below the brake pads, the temperature has a typical course. It rises when the vehicle is braking and then it falls as the temperature is conducted to the other part of the brake disc. The points far away from the friction ring do not follow this repetition, and the temperature rise is continuous. The maximal temperature is 825ºC, which is above 700ºC, the recommended maximal operational

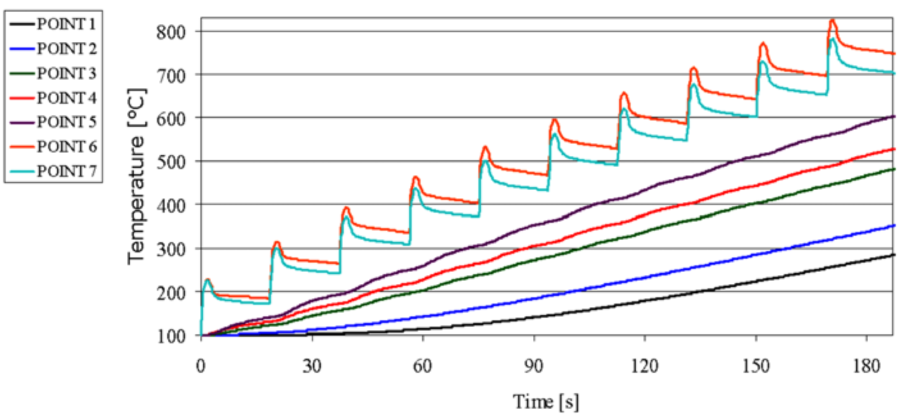

Figure 11. Temperature plot of the numerical simulation without considering cooling.

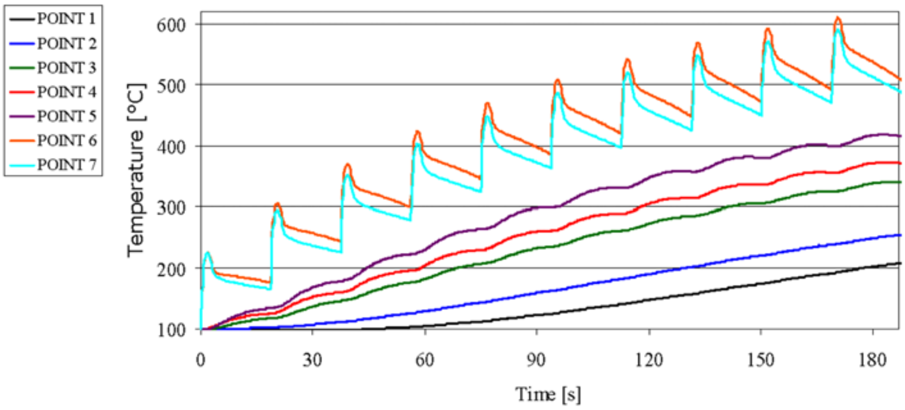

Figure 12. Temperature plot of the numerical simulation considering cooling.

temperature of the brake disc. Above this temperature, excessive braking fade appears, and the material structure starts to change. The conclusion of such an analysis would be that the brake disc design is not adequate.

5.2. Analysis of the Results Considering Cooling

The second analysis was identical to the first one, but the cooling was considered. The temperature- and speed-dependent surface film condition with values obtained in the previous air flow analysis was applied. If we examine figure 12, it is evident that the maximal temperature is lower than the maximal temperature in the previous analysis (figure 11). At higher disc temperatures, the cooling gets more intense, and the temperature drop at the end of the braking phase is higher. The temperature course is not linear as in the previous analysis, but it has a curved shape as a result of cooling. If we were to continue the brake cycles, such as is performed in a thermal capacity procedure, which comprises 25 brakings (Cimos Developlent Group, 2009) the braking temperature would eventually stabilize at a certain value. The disc maximal temperature is 610ºC, which is below the 700ºC boundary value. Therefore, the brake disc design meets the requirements. It was later found that the same brake disc had a maximal temperature of just below 600ºC during the physical vehicle test (Cimos Developlent Group, 2009). This result proves the accuracy of this numerical simulation method.

6. DISCUSSION OF RESULTS

The AMS procedure is an extreme braking test. Brakes mounted on a passenger vehicle are hardly ever subjected to such extreme brakings and temperatures in their lifetime. The brake disc after the AMS vehicle test is usually destroyed, cracked and overheated. It is difficult to reach temperatures higher than those obtained in the AMS test because the brakes fail sooner. The temperatures at points immediately below the brake pad are not to be taken as necessarily dangerous because they exist just a few microns below the surface for a relatively short period of time. Nevertheless, microcracks on disc surfaces such as those shown in Figure 13 are a consequence of this local overheating. The figure shows the actual brake disc damage after the AMS test on a dynamometer. It is clearly observed that the crack exists in the radial direction of the brake discs because the cooling vanes are oriented in the same direction. The areas that are supported by a cooling vane have better conduction than the areas that are not supported by a cooling vane. This leads to a differences in temperatures that cause cracks to form because of the extreme loads and friction.

Figure 13. Microcracks on the disc surface.

The microcracks form on the friction surface below the brake pads where the temperature is the highest. With the addition of high stresses as a consequence of high loads during the brake operation (Yildiz and Duzgun, 2010), the combination of both can lead to a failure. Therefore, it is essential to construct a brake disc in such a way that the temperatures reached are as low as possible.

The experiment was performed in the Cimos brake systems laboratory on a LINK 3900 full NVH and performance Dynamometer. It was performed with the fixed knuckle fixture. The test was an imitation of the AMS

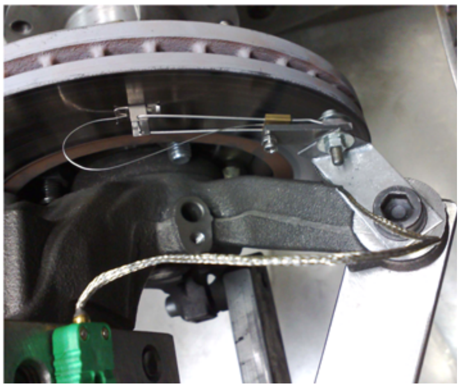

Figure 14. Place of temperature measurement in the experiment.

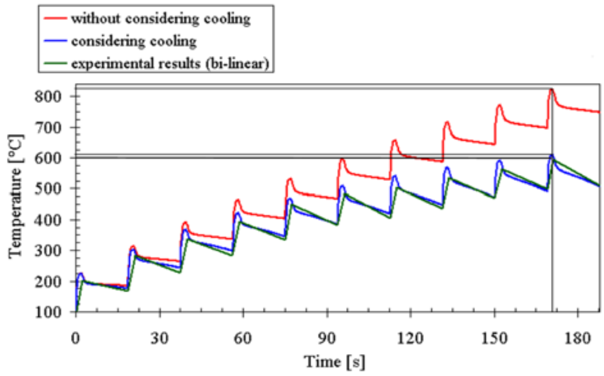

Figure 15. Comparison of the maximal temperatures (POINT 6) of both cases with the experiment.

vehicle test. Figure 14 shows the actual temperature measurement location, which is in the middle of the friction plate and corresponds with the point 6 in Figure 10. The temperature during the experiment was measured with a rubbing thermocouple, such as is shown in Figure 14, to obtain the actual surface temperature and to avoid any stress concentrations that occur when using embedded thermocouples. This location was chosen because the highest temperatures during the braking exist in the middle of the friction plate, which corresponds to the middle of the brake pads as well. It is also the location defined by the AMS standard specification (ESSE, 2005).

To make the values easier to handle and for a better comparison with the numerical methods results, only the temperatures at the beginning and end of each braking phase were measured for this analysis. The results are shown in Table 3. The values represent the peak values measured during each braking and not the end values, which are slightly lower. For better visualization and comparison of the results of the dynamometer test, the experimental values are plotted in Figure 15. The values were linearly connected; therefore, a line is plotted in Figure 15.

Table 3. Experiment results.

Stop no.	Brake disc temperatures [°C] – rubbing plate thermocouple – front right disc	
	Start of braking	End of braking
1	98	203
2	167	284
3	227	338
4	284	398
5	334	449
6	383	484
7	405	505
8	437	535
9	469	562
10	497	593

Beside the experimental values, the maximal disc temperatures on point 6 for both numerical analyses with and without considering cooling are also plotted. The maximal temperature in the analysis without cooling is almost one-third higher than the maximal temperature in the analysis with cooling. The difference is 215ºC, which is immense.

This analysis also shows that the numerical method prediction methodology used in this analysis has a very good correlation with the experimental results. The end temperature at all of the stops is within 10%, which can be determined from Figure 15.

According to company internal regulations (Cimos Developlent Group, 2009), the results from a numerical analysis should have an accuracy over 90%. With others variables in the brake test that cannot be described as homogeneous, such as the material structure, vehicle air resistance and wheel rolling resistance, it is necessary that we estimate the cooling effect of the brake disc as accurately as possible.

7. CONCLUSION

The wall heat transfer coefficients for a known disc geometry were calculated for several vehicle speeds and temperatures using the ANSYS CFX software package. The results were arranged in such a way that they could be used as a boundary condition for a thermal numerical simulation. Two identical thermal numerical analyses were conducted: one without considering the cooling and the other considering the cooling. The results of temperature plots were compared to determine the significance of properly considering the cooling factors.

The results showed that proper cooling factor prediction is essential for accurate numerical calculation predictions of vehicle brake test temperatures. The maximal temperature of the brake disc in a simulation without cooling consideration reached 825ºC, which is above the recommended brake disc operating temperature of 700ºC (Cimos Developlent Group, 2009). The maximal temperature of the brake disc in a simulation considering cooling reached 610ºC, which is 215ºC less than the value obtained in the parallel analysis. The simulation considering cooling was more accurate based on the experimental results shown in Table 3, in which the maximal temperature at the 10^{th} stop reached 593ºC (Cimos Developlent Group, 2009).

To assure that the errors of the thermal numerical simulation are below the requested 10% of the experimental vehicle testing results, it is necessary to accurately predict the disc wall heat transfer coefficients.

Further research should be made to determine whether the wall heat transfer coefficients can be generalized for certain types of brake discs and what the effect of cooling vane geometry is on the wall heat transfer coefficients.

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