PRESSURE INFLUENCE ON HEATING OF VENTILATING DISC BRAKES FOR PASSENGER CARS

by

Nadica R. STOJANOVIĆ[a], Jasna D. GLIŠOVIĆ[a], Oday I. ABDULLAH[b,c], Ivan Lj. GRUJIĆ[a*], and Saša Ž. VASILJEVIĆ[a,d]

[a] Faculty of Engineering, University of Kragujevac, Kragujevac, Serbia
[b] Department of Energy Engineering, College of Engineering, University of Baghdad, Baghdad, Iraq
[c] Hamburg University of Technology, Hamburg, Germany
[d] High Technical School of Professional Studies, Kragujevac, Serbia

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The braking system is one of the most important elements in vehicle systems from the aspect of vehicle safety, besides the steering system and the internal combustion engine. During the braking process, the disc and pads absorb a large amount of kinetic energy that converted to heat. Owing to this frictional heating, it is necessary to compute the temperature distribution that will be appeared during the braking process, which is the main goal of this research paper. There are many factors that can be influenced to the distribution of frictional heat generated. One of the significant factors is the applied pressure by the brake pad on the braking disc. The results proved that when increased the applied pressure then the frictional heat generated increased too. It was developed a new finite element model based on observed data from real vehicle. It was used ANSYS/WORKBENCH 14.5 software to perform the numerical analysis, module Transient Structural. Parts that are the most disposed to the thermal stress are braking pads. Also, it was found time period from 0 to 0.1 second is the most critical period during the whole braking period, because in this period, temperature rises rapidly, the maximum temperature occurred at 1.338 seconds, and after that it falls.

Key words: *braking system, heat generated, finite element method, thermal stresses*

Introduction

The main task of the braking system, whether it's a passenger or heavy duty vehicle, is to stop or reduce the speed of the vehicle. In real exploitation conditions, it can be realized in three characteristic ways [1]:

– rapid braking, maximum braking in some dangerous situations,
– moderate short-term braking, under normal driving conditions (road speed adjustment), and
– moderate long-term braking, when a vehicle is driven on long downhill.

Of course, in addition to these three characteristic braking modes, the braking system has the task to ensure a permanent braking of the vehicle when the vehicle is parked.

* Corresponding author, e-mail: ivan.grujic@kg.ac.rs

Braking is achieved by friction force that occurs between two contacting elements (the brake disc and brake pads). It is clear that the frictional heat generated occurs due to the friction process during braking [2].

A high amount of the kinetic energy is converted into heat energy under braking conditions. The most important factors that affected the thermal behaviour of the brake system are [3]:

– thermal properties of the materials,
– brake ventilation, and
– frictional characteristics of brake pads.

Aleksendric and Duboka [4] showed that the experimental value of the coefficient of friction varies during the fading procedure. The reason is great influence of the temperature, contact pressure and speed variation on the value of coefficient of friction.

Friction that occurs between brake pads and discs creates heat. The heat generated on the parts can have a significant influence on tribological behaviour in the contacting zone [2]. Therefore, the heat significantly affects the performance of the braking system, where the most vehicle's kinetic energy is converted into heat.

Brake discs are made from various types of high quality cast irons, and it rarely made from steel. Cast iron is more appropriate than steel because it can be easily welded and easily formed into the desired shape. They are also easy to handle and have high wear resistance. Also, contact between the cast iron and the friction materials gives satisfied value of the coefficient of friction. Furthermore, the cast iron has good thermal and mechanical properties (thermal conductivity, young modules, *etc.*) at high temperatures. In addition to all this, the price of cast iron is low compared with other types of available materials.

Based on the characteristics of the disc material, this makes the thermal capacity of a disc limited compared to the large amount of frictional heat that generated during the braking process. So in this case, the temperature of the pad increases dramatically, and this could further lead to increase the temperature of the brake fluid, which can lead to evaporate the brake fluid.

However, researches that have been studied the transfer of heat through the braking pads indicated that the temperature value decreases with increasing distance from the contact surface with the disc [5].

The minimum recommended thickness of the brake pad by the manufacturers is 2 mm [6]. Based on a study that achieved by Zhu *et al.* [5], the surface temperatures may exceeds 50 °C at a distance of 2 mm from the contact zone.

The design of the brake disc is considered essential item to obtain a successful product, when the disc has radial ribs, this improvement in the design of the brake disc will play an important role to improve the brake cooling because it works as ventilator [7]. The shape of the disc ribs plays a very important role that affecting the rate of cooling and provides good resistance against the high temperatures [8]. Studies achieved by other authors showed that the ventilating discs are more suitable for critical operating conditions than those with full cross-section [9]. The ventilated brake disc is a better solution because it contains additional surfaces that exposed to the surrounding environment. Disc brake geometry plays an important role in heat transfer [10]. Also, it is necessary to know that temperature significantly affects the contact and thermal behaviours of the disc brake assembly. Large deformation and high contact pressure are observed in the disc-pad model under the thermal effect [11, 12].

In the early stages of product development, numerical analysis has the ability to investigate and study the behaviour and performance of the automotive brakes under different working conditions and diagnosis of any defect exists in the design of the brake system. Based on the numerical analysis, it can be obtained a wide range of results for example temperature distribution, contact pressure, thermal stresses as well as critical temperatures [13]. Also, it can be investigated the effect of selected material to produce the product and to find the optimal one [14].

Today, due to the fast progress in the technology field, the numerical analysis can be used instead of the experimental work in order to develop the products; this can reduce the cost, and the time needed to develop any product. The characteristics of the selected materials (thermal and mechanical properties) for the products are considered the necessary elements to achieve the numerical analysis. As stated in the study Belhocine and Omar [9] where are concluded that gray cast iron FG15 is the most appropriate material for the braking disc.

Based on a detailed analysis of the research which focused to study the problems of the automotive brakes, it was found that the thermo-elastic instability problem is the most critical problem which affecting the behaviour and performance of the automotive brakes [15].

The aim of the paper is to explore the potential of the numerical analysis to determine the temperature field of the brake disc and brake pads during the braking process applying different pressures. Three cases with different values of applied pressures will be used. The reference value for the applied pressure is 1 MPa, according to Belhocine [10]. In order to show the influence of applied pressure on the frictional heating that appeared in the braking disc and braking pads. It was adopted two values of applied pressures, by which one is 10% lower than reference value, and other one 10% higher than reference. Besides that, analysis is performed for braking system till stopping. The results presented the temperature distribution at any instant during the braking operation, and how to define the frictional heat generated (heat flux). The results showed that the level of temperature for the disc rotor increased with time of the braking. The main outcome from this research paper is to calculate accurately the temperature distribution based on the friction that occurs during the braking process. While the remaining input data is taken from experimental research. In order to determine temperatures that occur during the braking process till stopping, it was developed a new model to find the solution of the structural-thermal coupling.

Heat transfer

The main problem of disc brake heating can be attributed to the reduction of the friction coefficient. Figure 1 shows the effect of temperature on a coefficient of friction. This figure covered a wide range of working temperatures of the brake system. It can be divided this thermal curve into two zones: the stable thermal zone and the critical thermal zone. Under the long-term braking conditions, the temperature will increase dramatically, so small cracks can occur on the disc which may lead to the fracture of the disc [4]. As can be seen, when the temperature of the disc exceeds 700 ºC, the coefficient of friction decreases rapidly, and this leads to increase of the stopping time [16].

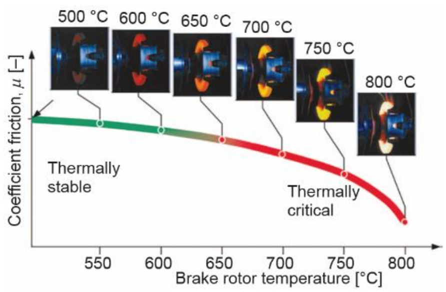


Figure 1. Influence of temperature on coefficient of friction [16]

Figure 2 illustrates the energy balance of disc brake during the braking process. Some amount of

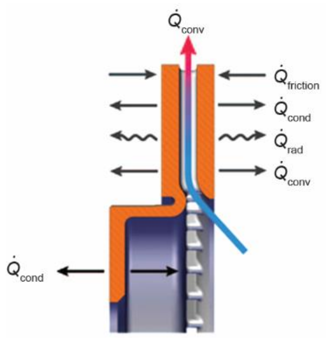


Figure 2. Energy balance of disc brake [16]

the total heat generated during the braking process is accumulated in the braking disc, and the other amount of heat will transfer to the surrounding ambient by radiation and convection, while the remaining of heat is transferred to the adjacent components by conduction [16].

Equations for conduction, convection and radiation [17] are given by eqs. (1)-(3):

$$\dot{Q}_{\text{cond}} = h_{\text{cond}} A_{\text{cond}} [T_{\text{d(int)}} - T_{\text{c(int)}}] \tag{1}$$

$$\dot{Q}_{\text{conv}} = h_{\text{conv}} A_{\text{conv}} \left(T_{\text{d}} - T_{\infty}\right) \tag{2}$$

$$\dot{Q}_{\text{rad}} = \varepsilon \sigma A_{\text{rad}} (T_{\text{d}}^4 - T_{\infty}^4) \tag{3}$$

While the equation of energy balance [18] is defined:

$$\dot{Q}_{\text{cond}} - \dot{Q}_{\text{conv}} - \dot{Q}_{\text{rad}} = 0 \tag{4}$$

Table 1. Material characteristics of brake disc and brake pads

	Disc	Pad (frictional material)
Density, [kgm^{-3}]	7100	2300
Elastic modulus, [GPa]	118	20
Poisson ratio, [–]	0.32	0.3
Thermal conductivity, [Wm^{-1}°C^{-1}]	53.3	3
Specific heat, [Jkg^{-1}°C^{-1}]	490	1200
Thermal expansions, [°C^{-1}]	10.85×10^{-6}	10×10^{-6}

Material characteristics

The material of the disc is grey cast iron [19], with very good thermo-physical properties; all material properties used in this analysis are isotropic. All specifications of the selected materials for the numerical analysis are listed in tab. 1.

Determination of the time braking

The vehicle speed in the moment when braking starts is v_1 = 120 km/h. The constructive parameters of tire are 185/65 R15, from which can be calculated the wheel radius, and it is R = 0.31075 m. For the numerical analysis it is necessary to determine the value of the wheel angular speed:

$$\omega = \frac{v_1}{R} \tag{5}$$

Further, it is obtained that the value of the wheel angular speed that corresponds to the vehicle initial speed which is ω = 107.26 rad^{-1}.

The time necessary for the vehicle to stop can be calculated according to the following equation that found by Demic and Lukic [20]:

$$t_k = \frac{m\delta}{\sqrt{BKA}} \left[\text{arctg}\left(\sqrt{\frac{KA}{B}} v_1 \right) - \text{arctg}\left(\sqrt{\frac{KA}{B}} v_2 \right) \right] \tag{6}$$

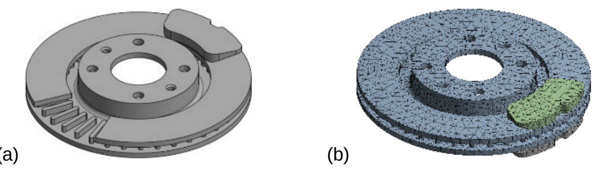
The calculation and values of necessary data to find the stopping time are given in tab. 2.

Table 2. Calculation data for the braking time determination [20, 21]

Name	Calculation and values
Final vehicle speed	$v_2 = 0$ km/h
Rolling resistance	$B = mgf = 7732.242$ N
The influence of the inertial masses	$\delta = 1$
Air resistance	$KA = 1/2\ \rho c_x A = 0.369$ N
Vehicle mass	$m = 1126$ kg
Rolling coefficient	$f = 0.7$
Air density	$\rho = 1.225$ kgm^{-3}
Drag coefficient	$c_x = 0.3$
Front surface of the vehicle	$A = 2.01$ m^2
Time necessary for vehicle to stop	$t_k = 4.77$ s

The 3-D model and finite element formulation

The most loaded parts of the braking system during the braking process are the brake disc and brake pads, therefore only these parts will be included in the numerical analysis, while the other parts of the brake system will not be involved in the analysis. The 3-D model of the brake system (assembly) was built, as shown in fig. 3. The types of final elements that used for contact model (elastic model) in this analysis are shown in tab. 3. The type of mesh are tetrahedrons, fig. 3(b), and the number of elements, as well as the number of nodes, are shown in tab. 4. The degrees of freedom (DOF) for the selected elements are listed in tab. 3. The contact surfaces are surfaces of the brake pads, and the target surfaces are surfaces of the brake disc.



(a) (b)

Figure 3. The 3-D model and mesh of (a) ventilated disc and (b) brake pads

Table 3 Types of finite elements used and their description [22]

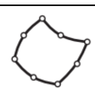
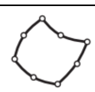
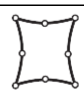
Title	Description	DOF	Illustration
SOLID227	3-D 10-Node Coupled-Field Solid 10 nodes 3-D space	UX, UY, UZ, TEMP, VOLT	
CONTA174	3-D 8-Node Surface-to-Surface Contact 8 nodes 3-D space	UX, UY, UZ, TEMP, VOLT, MAG	
TARGE170	Contact 3-D Target Segment 8 nodes 3-D space	UX, UY, UZ, TEMP	
SURF154	3-D Structural Surface Effect 4 to 8 nodes 3-D space	UX, UY, UZ	
MPC154	Structural Multipoint Constraint 2 or 3 nodes 3-D space	UX, UY, UZ, ROTX, ROTY, ROTZ KEYOPT Dependent	

Table 4 Number of finite elements and number of nodes

		Number of nodes	Number of elements
Disc		38938	21096
Pad			
Initial case	Inside	1684	840
	Outside	1660	824

Boundary conditions

The contact algorithms and the values of factors that used in each kind of algorithm are very effective, in the case when the frictional contact is simulated, there are more than one algorithm can be used but it should select the type of algorithm carefully to ensure obtain accurate results. The types of algorithms are [23]:

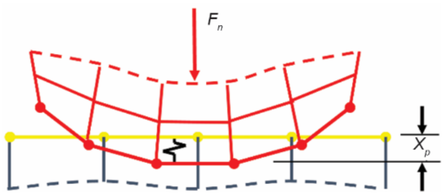


Figure 4. Contact between two parts [24]

– Penalty method represents the method that use contact stiffness to establish the relationship between the two contacting surfaces, the tags given in eq. (5) are shown in fig. 4:

$$F_n = k_n x_p \tag{7}$$

– Augmented Lagrangian method is consisted from series of penalty methods. This method is better than penalty method because usually leads to better conditioning and is less sensitive to the magnitude of the contact stiffness. However, in some analyses, the augmented Lagrangian method may require additional iterations, especially if the deformed mesh becomes too distorted:

$$F_n = k_n x_p + \lambda \tag{8}$$

– Lagrange multiplier is used for adding the contact traction to the model as additional degrees of freedom. The use of the Lagrange multiplier requires additional iterations to stabilize contact condition. This can lead to the increasing the computational time compared to the augmented Lagrangian method.
– Pure Lagrange is very similar to the Lagrange multiplier method. But this method is specific, because this method does not require the contact stiffness.

Internal multipoint constraint is used when in case of different types of contacts. This method is good if there is no separation or bonded contacts.

In this paper, for defining the coefficient of friction, the algorithm which used is Augmented Lagrange. The reason to use the Augmented Lagrange is the convenience of this algorithm during the calculations of structural problems between contact surfaces where the frictional sliding exists.

Further in paper, three cases will be analysed, where only the applied pressure will be different, where the values of the applied pressure of selected cases are:
– Case 1: 0.9 MPa;
– Case 2: 1 MPa, and
– Case 3: 1.1 MPa.

The pressure is constant for all three cases during braking process. It is important to say that other boundary conditions will be the same for all three cases; such are material characteristics, environment temperature, speed during the braking process, coefficient of friction, *etc.*

Results and discussions

During the braking process, the frictional heating will be appeared in contacting surfaces of the braking system. The observed braking disc is vented disc, and as the result of such design, the cooling is constant, and it is very difficult to warm up to such an extent that can lead to be a real trouble. This is one of the advantages of the design of disc of brake.

This behaviour can be seen during all the period of barking until the vehicle stop, which is investigated deeply in this paper. However, during the braking process, there are very dynamical changes in the disc temperature as shown in fig. 5. Also, there is other advantage that prevent the disc from warm up, which is the disc surface is greater than surface of braking pads.

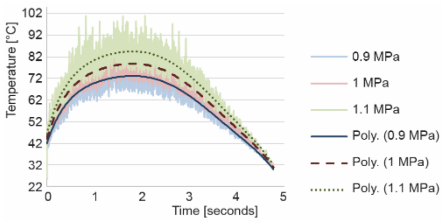


Figure 5. The braking disc heating during the braking process in respect to the applied pressure

In other words, the disc contact surface is not entire in contact with pads, but one part that was in one moment in contact with pad; the next moment is not, and due to this the rate of cooling is constant, and it cannot warm up.

The highest amount of heat that generated on the contact surface of the braking pad, and that on the contact surface of the outer braking pad, which can be seen in fig. 6. The difference between them is around 4 °C, due to the phenomenon of umbrellas [25, 26]. Where the disc is bending to the outer side, and due to this, the higher pressures occurred in outer side more than the inner side. Braking pads are manufactured from such materials that are preventing heat to transfer to the surface that is on the outer side from a contact surface, where heat is generated. In this way, the other parts of the braking system are protected from heating as well the braking fluid is protected because in this case, the hydraulic brakes are observed. On this way, it will not come to the *brake pedal deterioration*.

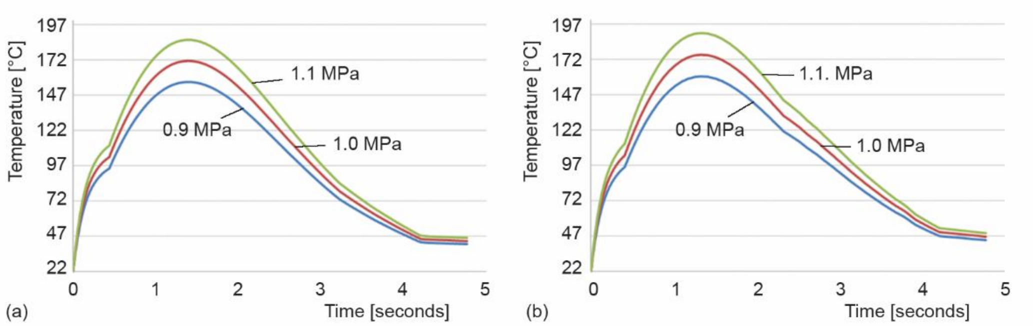


Figure 6. The heating of inner (a) and outer (b) braking pad in respect to the applied pressure

Generally, all cases have the behaviours for the temperature, where the temperature increased during the braking process until a certain point and after this point the temperature

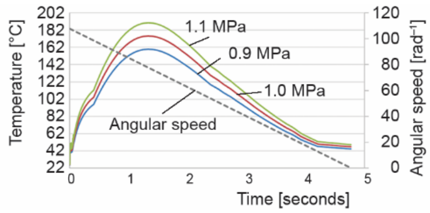
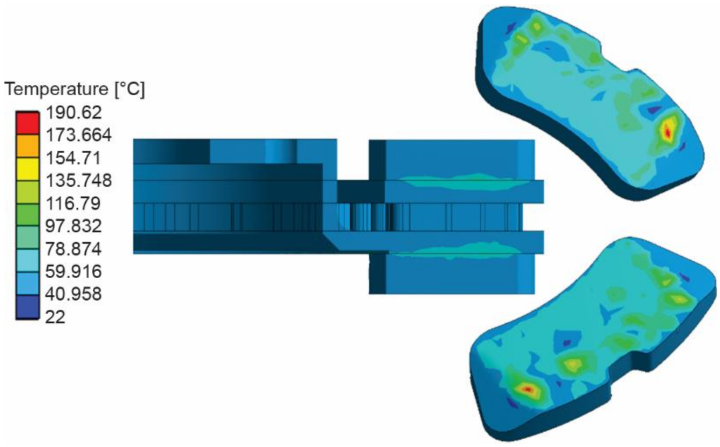


Figure 7. The heating in respect to the applied pressure

decreased. The only difference between them the values of temperature, where the highest temperature appeared when applied the highest pressure and the lowest temperature appeared when applied the lowest pressure as shown in fig. 7. The highest temperature occurred, when the applied pressure is 1.1 MPa, which is expected. It can be observed that the most critical is period from 0 to 0.1 seconds, for third case the temperature reached 68 °C, and for first case 59 °C. This is not suitable, when the temperature increased suddenly; this will lead to produce high thermal stresses and under such condition will lead to the element of brake to damage.

The maximal temperature for the first case occurs after 1.331 seconds, and then temperature reached maximum value that is equal 159.95 °C, while for second case occur after 1.337 seconds the temperature reached 175.3 °C. While for the third case, it can be seen that the maximum temperature occurred after 1.338 seconds. The maximum temperature occurred on the contact surface of the outer pad, fig. 8. The temperature on the braking pad contact surface is 190.62 °C, while the maximum temperature on the inner pad at the observed moment is 185.95 °C. Besides that, it can be noticed that maximum temperature appeared on zones that were first in contact with disc, more accurate on the entry side of the pad. In the case when the applied pressure is 10% lower than reference value, the maximum temperature is 6% lower than the maximum temperature when applied the reference pressure [10]. While in the case when the applied pressure is 10% higher than reference value, the maximum temperature is higher with 2.5% than when applied the reference pressure.

Figure 8. Temperature at the moment when the assembly temperature reaches its maximal value (third case)



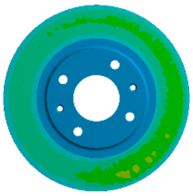
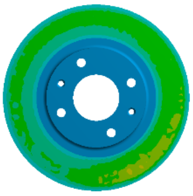
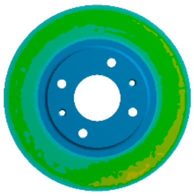
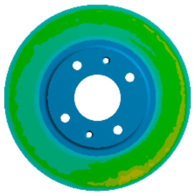
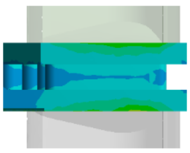
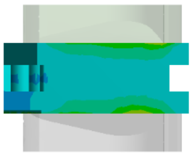
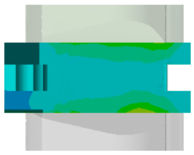
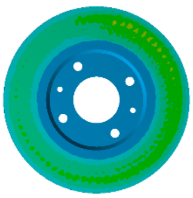
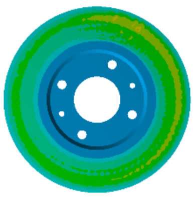
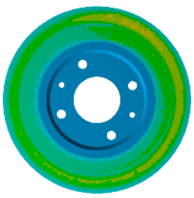
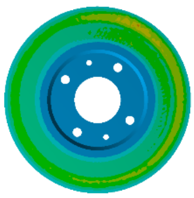
The similar research with braking till stopping it was performed by Yang *et al.* [15], and they have obtained maximal temperature after 2 seconds of braking, while in this paper, the maximum temperature is appeared after 1.337 seconds. The reason for this result is the material properties and the value of initial speed. However, in their research, they have obtained higher values of temperature, because the applied pressure was higher.

At 1 mm depth, the temperature value is around 65 °C, and at 2 mm, the temperature is lower for almost 20 °C than temperature that appeared on the contact surface, what Zhu *et*

al. [5] proved in their research. Where this is logical, if the frictional heat generated (heat flux) [27] for the third case is calculated, the following results are obtained:
– the braking disc heat flux is 91521.41 W/m^2, and
– the braking pad heat flux is 11165.71 W/m^2.

Unlike pads, the maximum temperature on disc appears at t = 2.069 seconds is found to be 100.8 °C. It can be observed that the period between 0 and 1.4 seconds during the braking process has a negative effect on the pads, because of the temperature raised quickly on contact surfaces. However, if the braking disc is observed, the first two seconds are critical, in other words until the sliding speed reached the half value, tab. 5. The results presented also the variation of temperature through the thickness of the disc during the observed period. At the moment when maximum temperature occurred on the disc, the temperature at 1 mm depth is around 48-74 °C, while the temperature at 2 mm depth is around 48-65 °C. Further by depth, temperature is around 39-48 °C.

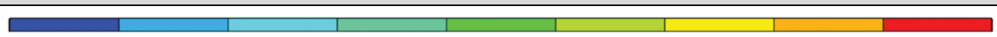
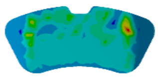
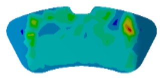
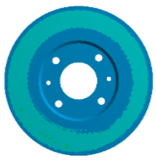
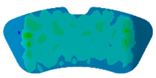
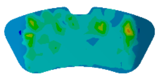
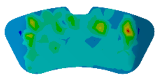
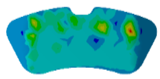
Table 5. The disc temperature field

Temperature, [°C]				
 22 30.76 39.515 48.27 57.026 65.78 74.54 83.293 92.05 100.8				
Outside surface				
				
Internal surface				
Time, [s]	0.52	1.0118	1.5418	2.0236

The amount of frictional heat generated on the braking pad is smaller than the amount of frictional heat that generated in the braking disc. This means that the braking disc should store the highest amount of heat compared with the heat that transferred by convection to the environment. Owing to this fact, the temperature values jumped during the braking process as shown in fig. 5, more precisely, there is a tendency for temperature rising, but because of the disc construction, this is excluded. Besides that, the temperature generated on the contact surface of the braking pad, should not be transferred to the other parts of the braking system, because it exists the possibility of braking fluid heating, which can be very dangerous.

It can be noticed from the results in fig. 7 that the most critical period during the braking process is the period between the starting time until time equal to 1 seconds, tab. 6. Maximum temperatures occurred on contact surfaces of pads, not on the disc, because of the disc has a constant cooling, by its construction.

Table 6. Temperature field on the contact surfaces for the third case

Temperature [°C]					
 22 · 40.958 · 59.916 · 78.874 · 97.832 · 116.79 · 135.748 · 154.71 · 1763.664 · 190.62					
Outer pad					
Disc					
Inner pad					
Time, [s]	0.33727	0.62636	0.91545	1.2045	1.5418

Maximum temperatures occurred on the outer braking pad, as shown in tab. 5. However, if the temperature of the disc surface in the contact with the outer pad is monitored during the braking process, in the observed period, it is around 97 °C. The minimum temperature occurred on the disc, because of the disc construction, where much better cooling of the outer plate is achieved. While higher temperature occurred on the second side of the disc, more accurate on the surface that is in the contact with the inner pad. The obtained results have agreement with results of other researchers that used different method [23], one of the reasons is phenomenon known as the *umbrella effect*. One of the disadvantages of the high temperature that appeared during the braking process is bending the inner side of the braking disc to the outer side

Conclusions and remarks

This research paper investigated the effect of the applied pressure on the surface temperatures during the braking process. It was used finite element method to simulate the brake system during working. Different values of applied pressure were used to show the effect of applied pressure on the thermal behaviour of brake system.

The results showed that when increased the value of the applied pressure the temperatures grow dramatically. So, it can be considered that the contact pressure is one of the most influential factors on the thermal behaviour of the brake system, where most of heat generated was absorbed by the elements of brake system during the braking process. The highest amount of heat was generated on the contact surface of the outer braking pad. The reason for this result is phenomena that known as the umbrella effect, while the temperature of the inner braking pad is slightly lower than the temperature of the outside pad. It was found that the maximum temperature appeared in the contacting surfaces are 159.95 °C, 175.3 °C, and 190.62 °C that corresponding to 1st Case, 2nd Case, and 3rd Case.

The temperature that appears because of the friction during the braking process is growing, and it arise maximal value after 1.338 seconds, and after almost one second it has constant value. After that, it was noticed a linear decrease until reach time 4 seconds. After that, the temperature is constant till the end of the braking process.

The constructive characteristics of the braking disc contribute that outer pad of the disc have less amount of heating compared with the inner side, more accurate the constant temperature is retained during the whole braking process. However temperatures of the inner and the outer braking pad are almost equal, the difference between them is around 4 °C. While on the inner side, the disc temperature increased in the period from 0 to 1 second, and after that from 1 to 2 seconds have almost constant value. Finally the temperature fall exponential to final value. However, the disc temperature at the end of the braking process is around 20 °C higher than the environment temperature.

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Nomenclature

A – front surface of the vehicle, [m^2]
$A_{\rm cond}$ – conduction contact area, [m^2]
$A_{\rm conv}$ – disc wetted area, [m^2]
$A_{\rm rad}$ – disc area emitting heat through radiation, [m^2]
B – rolling resistance, [N]
c_x – drag coefficient, [–]
f – rolling coefficient, [–]
F_n – contact force, [N]
g – gravitational acceleration on earth, [ms^{-2}]
$h_{\rm cond}$ – thermal conductance, [Wm^{-2}K^{-1}]
$h_{\rm conv}$ – convective heat transfer coefficient, [Wm^{-2}K^{-1}]
KA – air resistance, [N]
k_n – contact stiffness, [Nm^{-1}]
m – vehicle mass, [kg]
$\dot{Q}_{\rm cond}$ – conductive heat transfer, [W]
$Q_{\rm friction}$ – friction generated neat, [W]
$Q_{\rm rad}$ – radiative heat dissipation, [W]
$Q_{\rm conv}$ – convective heat dissipation, [W]
R – wheel radius, [m]
T_∞ – ambient air temperature, [K]
$T_{\rm c(int)}$ – carrier interface temperature, [K]
$T_{\rm d}$ – disc temperature, [K]
$T_{\rm d(int)}$ – disc interface temperature, [K]
t_k – stop time, [s]
v_1 – initial speed, [ms^{-1}]
v_2 – final vehicle speed, [ms^{-1}]
x_p – distance between two existing nodes or separate contact bodies, [m]

Greek symbols

δ – the influence of the inertial masses, [–]
ε – emissivity, [–]
λ – Lagrange multiplier component, [–]
ρ – air density, [kgm^{-3}]
σ – Stefan-Boltzmann constant, [Wm^{-2}K^{-4}]
ω – Angular speed, [rad^{-1}]

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