

## **THERMOMECHANICAL STRESS ANALYSIS OF VEHICLES GRAY CAST BRAKE\***

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**Abstract**–The vehicle braking system is considered to be one of the most fundamental safety-critical systems in modern vehicles, as its main purpose is to stop or decelerate the vehicle. The frictional heat generated during braking application can cause numerous negative effects on the brake assembly, such as brake fade, premature wear, thermal cracks and disc thickness variation. In the past, surface roughness and wear at the pad interface have rarely been considered in studies of the thermal analysis of a disc brake assembly using the finite element method. The ventilated pad-disc brake assembly is built by a three-dimensional model with a thermomechanical coupling boundary condition and multi-body model technique. The numerical simulation for the coupled transient thermal field and stress field is carried out sequentially with the thermal-structural coupled method, based on ANSYS software, to evaluate the stress fields of deformations that are established in the disc with the pressure of the pads and in the conditions of tightening of the disc; thus, the contact pressure distributions field in the pads is obtained, which is another significant aspect in this research. The results obtained by the simulation are satisfactorily compared with those of the specialized literature.

**Keywords**– Brake discs, heat flux, heat transfer coefficient, Von Mises stress, contact pressure

### **1. INTRODUCTION**

The braking system represents one of the most fundamental safety-critical components in modern passenger cars. Therefore, the braking system of a vehicle is undeniably important, especially in slowing or stopping the rotation of a wheel by pressing brake pads against rotating wheel discs [1]. Through the last two decades, numerous active safety systems have been developed to enhance handling and stability of ground vehicles, and considerable improvements have been achieved in this regard [2]. The investigation of the localized thermal phenomena such as hot spotting and hot banding [3, 4] requires a fully coupled thermoelastic analysis and thus, is beyond the scope of the current study. This separate work is underway to include the localized thermal effects in the proposed design process and will be reported in the future.

Braking performance of a vehicle can be significantly affected by the temperature rise in the brake components. The frictional heat generated at the interface of the disc and the pads can cause a high temperature. Particularly, the temperature may exceed the critical value for a given material, which leads to undesirable effects such as brake fade, local scoring, thermo elastic instability, premature wear, brake fluid vaporization, bearing failure, thermal cracks, and thermally excited vibration [5].

Gao and Lin [6] stated that there is considerable evidence that shows the contact temperature is an integral factor reflecting the specific power friction influence of the combined effect of load, speed,

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friction coefficient, and the thermo physical and durability properties of the materials of a frictional couple. Lee and Yeo [7] reported that uneven distribution of temperature at the surfaces of the disc and friction pads brings about thermal distortion, which is known as coning and has been found to be the main cause of Judder and disc thickness variation (DTV). Ouyang et al [8] in their recent work found that temperature could also affect the vibration level in a disc brake assembly. In a recent work, Ouyang et al [8] and Hassan et al [9] employed finite element approach to investigate thermal effects on disc brake squeal using dynamic transient and complex eigenvalue analysis, respectively. Braking system is the single most important safety feature of every vehicle on the road. The ability of the braking system to bring a vehicle to a safe controlled stop is absolutely essential in preventing accidental vehicle damage and personal injury. The braking system is composed of many parts, including friction pads on each wheel, a master cylinder, wheel cylinders, and a hydraulic control system [10].

Disc brake consists of a cast-iron disc which rotates with the wheel, caliper fixed to the steering knuckle and friction material (brake pads). When the braking process occurs, the hydraulic pressure forces the piston and therefore pads and disc brake are in sliding contact. Set up force resists the movement and the vehicle slows down or eventually stops. Friction between disc and pads always opposes motion and the heat is generated due to conversion of the kinetic energy [11]. The three-dimensional simulation of thermo-mechanical interactions on the automotive brake, showing the transient thermo-elastic instability phenomenon, is presented for the first time in this academic community [12].

In this work, a model of the thermomechanical behavior of the dry contact between the disc of brake pads at the time of braking phase will be constructed, the strategy of calculation is based on the software ANSYS 11 [13]. This last is elaborated mainly for the resolution of the complex physical problems. The numerical simulation of the coupled transient thermal field and stress field is carried out by sequentially thermal-structurally coupled method based on ANSYS.

## 2. HEAT FLUX ENTERING THE DISC

In a braking system, the mechanical energy is transformed into a calorific energy. This energy is characterized by a total heating of the disc and pads during the braking phase. The energy dissipated in the form of heat can generate temperature rise ranging from 300°C to 800°C. Generally, the thermal conductivity of the material of the brake pads is lower than that on the disc ( $k_p < k_d$ ). We consider that the heat quantity produced will be completely absorbed by the brake disc. The heat flux evacuated from this surface is equal to the friction power. The initial heat flux  $q_0$  entering the disc is calculated by the following formula [14]:

$$q_0 = \frac{1-\phi}{2} \frac{m g v_0 z}{2A_d \varepsilon_p} \quad (1)$$

Figure 1 shows the ventilated disc – pads and the applied forces.

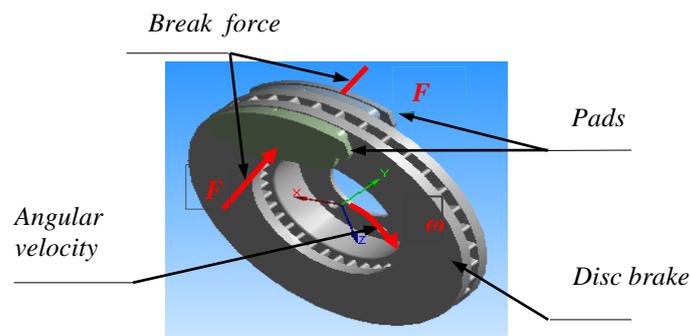


Fig. 1. Disc-pads assembly with forces applied to the disc

The loading corresponds to the heat flux on the disc surface. The dimensions and the parameters used in the thermal calculation are recapitulated in Table 1.

Table 1. Geometrical Dimensions and application parameters of automotive braking

Inner disc diameter, mm	66
Outer disc diameter, mm	262
Disc thickness (TH) ,mm	29
Disc height (H) ,mm	51
Vehicle mass $m$ , kg	1385
Initial speed $v_0$ , km/h	28
Deceleration $a$ , m/s <sup>2</sup>	8
Effective rotor radius $R_{rotor}$ ,mm	100.5
Rate distribution of the braking forces $\Phi$ , %	20
Factor of charge distribution of the disc $\epsilon_p$	0.5
Surface disc swept by the pad $A_d$ ,mm <sup>2</sup>	35993

The disc material is gray cast iron (GFC) with high carbon content [15], with good thermophysical characteristics and the brake pad has an isotropic elastic behavior whose thermo-mechanical characteristics adopted in this simulation in the transient analysis of the two parts are recapitulated in Table 2.

Table 2. Thermoelastic properties used in simulation

Material Properties	Pad	Disc
Thermal conductivity, $k$ (W/m°C)	5	57
Density, $\rho$ (kg/m <sup>3</sup> )	1400	7250
Specific heat, $c$ (J/Kg °C)	1000	460
Poisson's ratio, $\nu$	0.25	0.28
Thermal expansion, $\alpha$ (10 <sup>-6</sup> /°C)	10	10.85
Elastic modulus, $E$ (GPa)	1	138
Coefficient of friction, $\mu$	0.2	0.2
<b>Operation Conditions</b>		
Angular velocity, $\omega$ (rd/s)		157.89
Hydraulic pressure, $P$ (MPa)		1

### 3. MODELING IN ANSYS CFX

The rotating disk flow is one of the classical and important problems in fluid mechanics. The rotating disk flows have practical applications in many areas, such as vehicle rotor aerodynamics [16]. The air flow characteristics around the brake components are highly complex and they can vary significantly with the underbody structure as well as the component shapes. Instead of using empirical equations, which are commonly used in the thermal analysis [17, 18], the average heat transfer coefficients are calculated from the measured cooling coefficients by an iteration algorithm. Since the cooling coefficients account for all three modes of heat transfer, the estimated heat transfer coefficients include the equivalent radiation heat transfer coefficient.

The first stage is to create the CFD model which contains the fields to be studied in ANSYS Workbench. In our case, we took only one quarter of the disc, then we defined the field of the air

surrounding this disc. ANSYS ICEM CFD will prepare various surfaces for the two fields in order to facilitate the mesh on which the results towards CFX using the command "Output to cfx" will be exported [19]. After obtaining the model on CFX Pre and specifying the boundary conditions, the physical values that come into play on CFX must be defined in order to start calculating.

The disc is related to four adiabatic surfaces and two surfaces of symmetry in the fluid domain whose ambient air temperature is taken equal to 20 °C [20]. In order not to weigh down calculation, an irregular mesh is used in which the meshes are broader where the gradients are weaker (non-uniform mesh).

Figure 2 shows the elaborate CFD model which will be used in ANSYS CFX Pre.

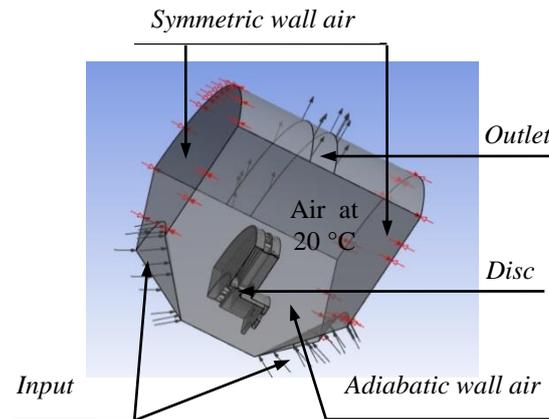


Fig. 2. Brake disc CFD model

In this step, all of the physical characteristics of the fluid and the solid are stated. We introduce into the library the physical properties of used materials. In this study a gray cast iron material (FG 15) was selected with thermal conductivity (57 W/m°C). Since the aim of this study is to determine the temperature field in a disc brake during the braking phase of a vehicle of average class, the temporal conditions are as follows:

- Braking time= 3.5 [ s ]
- Increment time = 0.01 [ s ]
- Initial time = 0 [ s ]

Before starting the calculation and the analysis with ANSYS CFX PRE, it is ensured that the model does not contain any error.

After verification of the model and boundary conditions, we run the calculation by opening the menu "File" and clicking on "Write solver file". The values of the coefficient of exchange will be taken as average values calculated by the minimal and maximum values obtained using ANSYS CFX POST as indicated in Fig. 3.

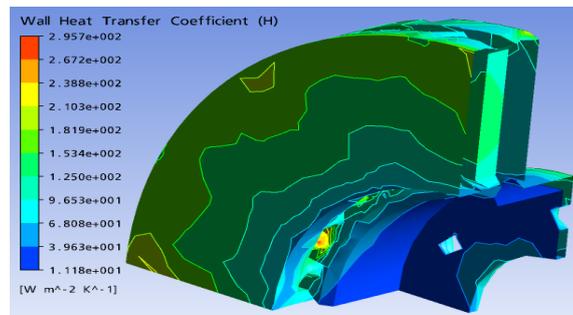


Fig. 3. Distribution of heat transfer coefficient on a ventilated disc in the stationary case (FG 15)

#### 4. MESHING OF THE DISC

The elements used for the meshing of the full and ventilated disc are tetrahedral three-dimensional elements with 10 nodes (isoparametric) (Fig. 4). In this simulation, the meshing was refined in the contact zone (disc-pad). This is important because in this zone the temperature varies significantly. Indeed, in this strongly deformed zone, the thermomechanical gradients are very high. This is why the correction, taking into account the contact conditions, involves the use of a refined mesh.

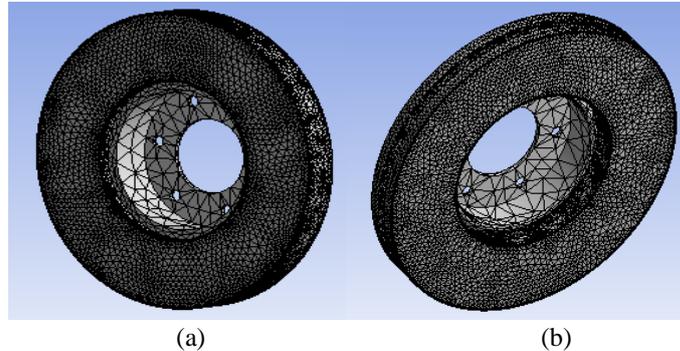


Fig. 4. Meshing of the disc (a) full disc (172103 nodes -114421 elements)  
(b) ventilateddisc (154679 nodes- 94117 elements)

#### 5. LOADING AND BOUNDARY CONDITIONS

The thermal loading is characterized by the heat flux entering the disc through the real contact area (two sides of the disc). The initial and boundary conditions are introduced into module ANSYS Workbench. The thermal calculation will be carried out by choosing the transient state and by introducing physical properties of the materials. The selected data for the numerical application are summarized as follows:

Total time of simulation = 45 [s]

- Increment of initial time = 0.25 [s]
- Increment of minimal initial time = 0.125 [s]
- Increment of maximal initial time = 0.5 [s]
- Initial Temperature of the disc = 60 [°C]
- Material: Gray Cast iron (FG 15).

#### 6. RESULTS AND DISCUSSIONS

The modeling of temperature in the disc brake will be carried out by taking the variation of a certain number of parameters into account, such as the type of braking, the cooling mode of the disc and the choice of disc material. The brake discs are made of cast iron with high carbon content; the contact surface of the disc receives an entering heat flux calculated from the Eq. (1).

##### a) Influence of construction of the disc

Figure 5 shows the variation of the temperature versus time during the total time simulation of braking for a full disc and a ventilated disc. The highest temperatures are reached at the contact surface disc-pads. The strong rise in temperature is due to the short duration of the braking phase and to the speed of the physical phenomenon. Of the two types of discs, starting from the first time step there is quick rise in the temperature of the disc followed by a fall in temperature after a certain time of braking.

We quickly notice that for a ventilated disc out of cast iron FG15, the temperature increases until  $T_{max} = 345$  °C at the moment  $t = 1.85$  s, then it decreases rapidly in the course of time. The variation in temperature between a full and ventilated disc having the same material is about 60°C at the moment  $t =$

1.88 s. It can be concluded that the geometric design of the disc is an essential factor in the improvement of the cooling process of the discs.

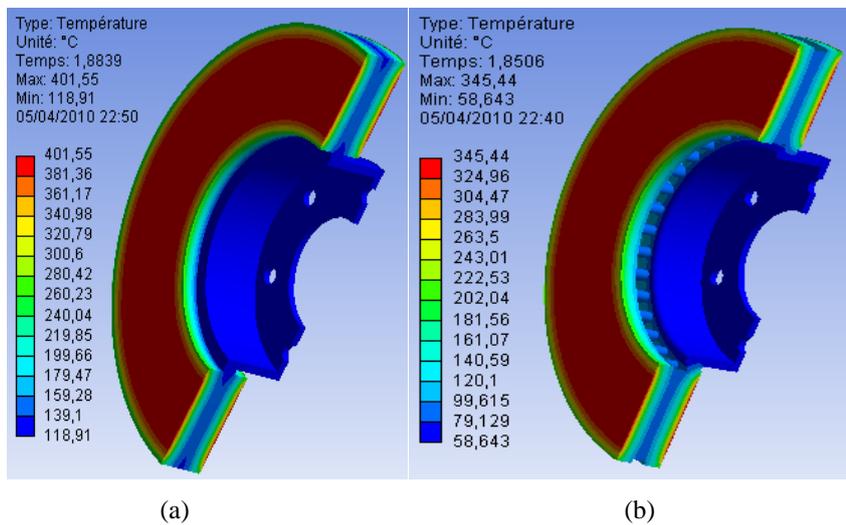
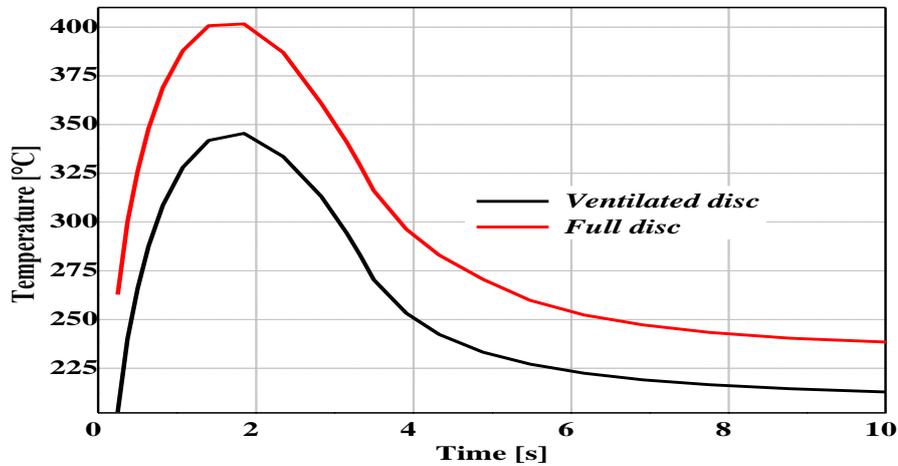


Fig. 5. Temperature distribution of a full (a) and ventilated disc (b) of cast iron (FG 15)

### 7. COUPLED THERMO-MECHANICAL ANALYSIS

The purpose of the analysis is to predict the temperatures and corresponding thermal stresses in the brake disc when the vehicle is subjected to sudden high speed stops as can occur under highway driving conditions [21]. A commercial front disc brake system consists of a rotor that rotates about the axis of a wheel, a caliper–piston assembly where the piston slides inside the caliper, which is mounted to the vehicle suspension system, and a pair of brake pads. When hydraulic pressure is applied, the piston is pushed forward to press the inner pad against the disc and simultaneously the outer pad is pressed by the caliper against the disc [22]. In a real car disc brake system, the brake pad surface is not smooth at all. Abu Bakar and Ouyang [23] adjusted the surface profiles using measured data of the surface height and produced a more realistic model for brake pads. Figure 6 shows the finite element model and boundary conditions embedded configurations of the model composed of a disc and two pads. The initial temperature of the disc and pads is 20°C, and the surface convection condition is applied to all surfaces of the disc and the convection coefficient (h) of 5 W/m<sup>2</sup>°C is applied at the surface of the two pads. The FE

mesh is generated using three-dimensional tetrahedral element with 10 nodes (solid 187) for the disc and pads. About 185901 nodes and 113367 elements are used (Fig. 7).

In this work, a transient thermal analysis will be carried out to investigate the temperature variation across the disc using ANSYS software. Further structural analysis will also be carried out by coupling thermal analysis.

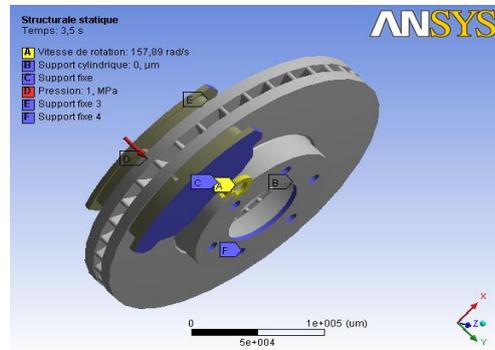


Fig. 6. Boundary conditions and loading imposed on the disc-pads

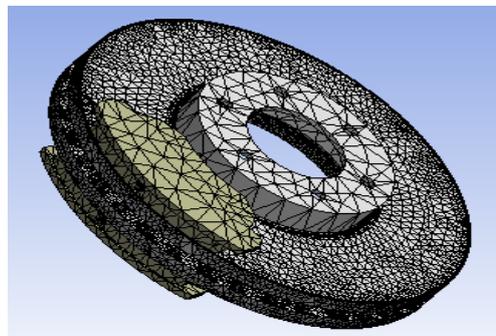


Fig. 7. Refined mesh of the model

**a) Thermal deformation**

Figure 8 gives the distribution of the total distortion of the whole (disc-pads) for various moments of simulation. For this figure, the scale of values of the deformation varies from 0  $\mu\text{m}$  with 284  $\mu\text{m}$ . The value of the maximum displacement recorded during this simulation is at the moment  $t=3,5$  s which corresponds to the time of braking. A strong distribution is observed which increases with time on the friction tracks and the external crown and the cooling fins of the disc. Indeed, during braking, the maximum temperature depends almost entirely on the heat storage capacity of disc; (on particular tracks of friction) this deformation will generate a dissymmetry of the disc following the rise of temperature which will cause a deformation in the shape of an umbrella.

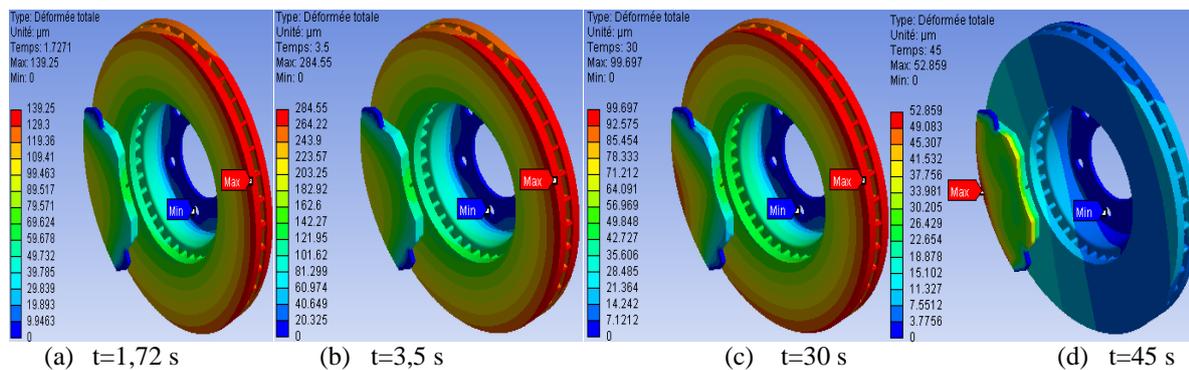


Fig. 8. Total distortion distribution

**b) Von Mises stress distribution**

Figure 9 presents the distribution of the constraint equivalent of Von Mises to various moments of simulation, the scale of values varies from 0 MPa to 495 MPa. The maximum value recorded during this simulation of the thermomechanical coupling is more significant than that obtained with assistance in the mechanical analysis dryness under the same conditions. A strong constraint on the level of the bowl of the disc can be observed. Indeed, the disc is fixed to the hub of the wheel by screws preventing its movement. In the presence of the rotation of the disc and the requests of torsional stress and shears generated at the level of the bowl, stress concentrations are created. The repetition of these requests will involve risks of rupture on the level of the bowl of the disc.

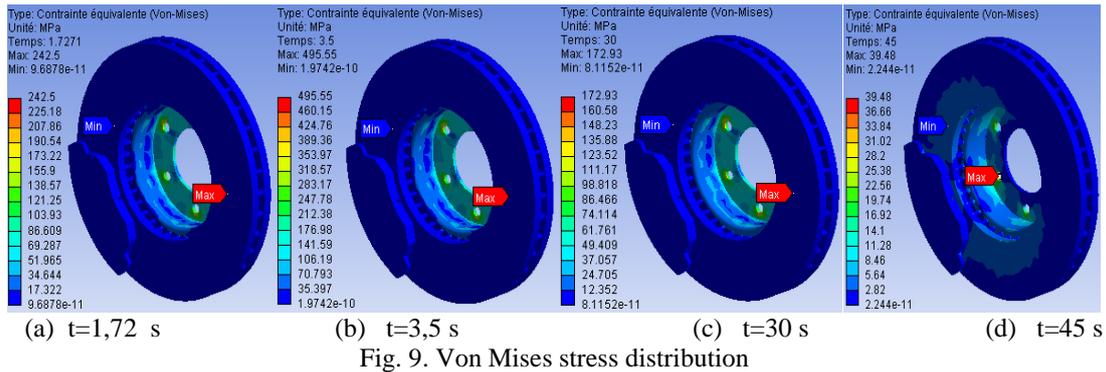


Fig. 9. Von Mises stress distribution

**c) Contact pressure**

Due to thermal deformation, contact area and pressure distribution also change. Thermal and mechanical deformations affect each other strongly and simultaneously. As pressure distribution is another important aspect concerned with this research, it will be studied in the context of uneven temperature distributions. Contact analysis of the interfacial pressure in a disc brake without considering thermal effects was carried out in the past, for example, in Tirovic and Day [24]. In recent years brake squeal analysis has always included a static contact analysis as the first part of the complex eigenvalue analysis [25, 26].

Figure 10 shows the contact pressure distribution in the friction interface of the inner pad taken for various times of simulation. For this distribution the scale varies from 0 MPa to 3,3477 MPa and reaches a value of pressure at the moment  $t=3,5$  s which corresponds to the null rotational speed. It is also noted that the maximum contact pressure is located on the edges of the pad of the entry and goes down towards the exit from the area from friction. This pressure distribution is almost symmetrical compared to the groove and has the same tendency as that of the distribution of the temperature because the highest area of the pressure is located in the same sectors. Indeed, at the time of the thermomechanical coupling 3d, the pressure leads to the not-axisymmetric field of the temperature. This last affects thermal dilation and leads to a variation of the contact pressure distribution.

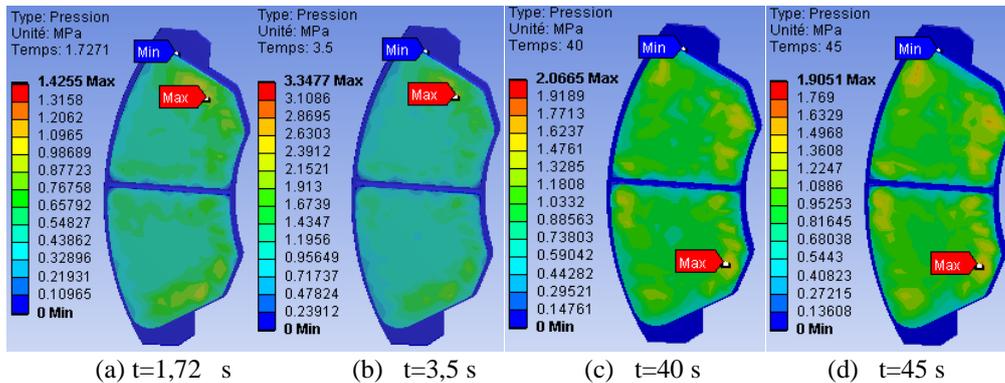


Fig. 10. Contact pressure distribution in the inner pad

## 8. CONCLUSION

In this article, analysis of the thermomechanical behavior of the dry contact between the brake disc and pads during the braking process was presented. The modeling is based on the ANSYS 11.0. We have shown that the ventilation system plays an important role in cooling disks and provides high temperature resistance. The analysis results showed that, temperature field and stress field in the process of braking phase were fully coupled. The temperature, Von Mises stress and the total deformations of the disc and contact pressures of the pads increase as the thermal stresses are added to mechanical stress which causes the crack propagation and fracture of the bow and wear of the disc and pads. Regarding the calculation results, we can say that they are found in the literature investigations. It would be interesting to solve the problem in thermo-mechanical disc brakes with an experimental study to validate the numerical results, for example on test benches, in order to show good agreement between the model and reality.

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