

Finite Element Analysis of Automotive Disc Brake and Pad in Frictional Model Contact

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Abstract: The purpose of this work is to present a study of the thermomechanical behavior of the automotive disc brake during the braking phase. The first analysis is performed on the disc-pad model without the presence of thermal properties. Structural performance of the disc-pad model such as deformation and Von-Mises stress is predicted. The case of thermoelasticity on the same model with the inclusion of convection, adiabatic and heat flux elements were also studied. The prediction results of temperature distribution, deformation, stress and contact pressure are presented. Furthermore, the structural performances between the two analyses (mechanical and thermomechanical) were compared as well. The results of this investigation may assist brake engineers to choose a suitable analysis in order to critically evaluate structural and contact behaviour of the disc brake assembly.

Keywords: Brakes/Clutches, Cast Iron, Fatigue, Residual Stress, Sliding, Steel

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1 INTRODUCTION

When two bodies are in contact with friction, there is a dissipation of energy and thus heat is produced at the level of the contact, which involves a dilation being able to increase the pressure field. This phenomenon increases if the tangential constraints as well as relative sliding speeds between the two bodies are significant. These thermomechanical effects are translated more with the share of time by the formation of zones located with very high thermal gradients at the hot points. There is appearance of thermal deformations and the stress concentrations being able to generate cracks, vibrations, etc.

In the aircraft industry and automobile, many parts are subjected simultaneously to thermal and mechanical requests. Thermomechanical requests can cause deformations and damages, for example, friction in a brake generates heat in the disc which can generate deformations and vibrations. Hwang and Wu investigated the temperature and thermal stress in the ventilated disc-pad brake during single brake by basing their study on multi-body technique and 3D thermomechanical coupling model [1]. Lakkam et al., studied heat dissipation characteristics of different types of brake discs during their operation [2]. The experimental results in this study were employed as thermal properties inside a computer simulation in order to determine the temperature responses and temperature distribution as well as the deformation of such brake discs.

Akhtar et al., employed finite element (FE) method and has explained the transient thermoelastic phenomena of a dry clutch system [3]. The effect of sliding speed on contact pressure distribution, temperature and heat flux generated along the frictional surfaces was analyzed. The clutch system has been simulated by axisymmetric model using ANSYS software. In the study established by Wang and Fu, ABAQUS software was exploited to determine the relation between a brake pad and temperature and thermal stresses on brake disc surface [4]. A procedure was proposed in order to improve heat flow distribution of the brake disc by optimizing the position of the friction block of the brake pad. Saryerli and Bozacı examined in a study, the effect of the pressure distributions at the rotor and the pad interface on the braking torque by using a linear and a nonlinear mathematical model of pressure distribution of basing on imposing assumptions [5].

Abu Bakar et al., carried out a study to examine the interface contact pressure distributions at the rotor and the contact between piston-brake pads at various rotational speeds by using a validated and quite detailed three-dimensional finite element model, while examining only the piston-pad; because it has distributions of pressure more heterogeneous than the

finger pad [6]. Bayas and Aher proposed an analytical method by using ANSYS tool to calculate the influence of heat flux and by that heat flux temperature finds out on the rotor [7]. Sowjanya and Suresh treated a static structural analysis of the disc brake by using ANSYS workbench, where some composite materials were selected to compare the results obtained like the deflection, normal stress, Von-Mises stress [8]. In the research developed by Reddy et al., thermal and structural coupled analysis was carried out to find the strength of the disc brake [9].

Gnanesh et al., investigated thermal-structural analysis of normal and vented disc brake rotor using ANSYS software 13.0 in the case of design with holes and without holes in the rotor [10]. The materials used in the simulation were cast iron, stainless steel and Aluminum metal matrix composites.

Thilak in an attempt investigated the transient thermo elastic behavior of disc brakes in repeated brake where the various results were compared, while the materials used for the rotor were Cast Iron, Aluminum based metal matrix composite and High Strength Glass Fiber composites [11]. While basing on the strength and rigidity criteria, the brake disc design is consequently safe. Tehrani and Talebi performed a thermal stress analysis on two designs of ventilated disc brake of which one is manufactured of functionally graded composite material and the other out of steel alloy [12]. A 3D finite element model and the computer code ABAQUS were used in this study. The advantage of using functionally evaluated material is illustrated thanks to the comparison of the temperatures and stress fields. Albatlan studied and compared the distributions of the temperature caused by mutual sliding of two members of the brake pads, and described a dynamometer system with inertia which was applied to the testing of a disc brake assembly at various operating conditions [13].

Nathi et al., developed a transient thermal analysis using ANSYS software to investigate the temperature variation across the disc using axisymmetric elements; the structural static analysis is also coupled with the thermal analysis in their work for prediction of the effect of stiffness, strength and variations in disc brake rotor design and the stress fields [14]. Youfu et al., used a non-linear method, and finite element to analyze the variation of the temperature of disc brake under conditions of dynamic speed in the braking process [15]. Abdullah et al., used finite element method to study the contact pressure and stress field during the full engagement period of the clutches using different contact algorithms [16]. In addition, parametric sensitivity of the contact pressure was treated in order to indicate the importance of the contact stiffness between floating surfaces brought into play.

Daniel Das et al., using ANSYS software studied the temperature fields and also structural fields of the full disc during short and emergency braking with four different materials [17]. The distribution of the temperature depends on various factors such as friction, external surface roughness and velocity. The angular velocity and the contact pressure induce the rise in temperature of the brake disc. The value of temperature, contact friction power, nodal displacement and deformation for various pressure conditions were taken into consideration in their work, by using the software analysis with four materials in particular cast iron, cast steel, aluminum and carbon fiber reinforced plastic.

In the study established by Yildiz and Duzgun, three different ventilated brake discs were designed and manufactured, and their performances of braking force with those of a full disc were studied in experiment [18]. Then, the stress analyses were carried out by FEA. In these analyses, another different approach, namely the variable loading which is most favorable on brake pads, was also presented. It was noted that this approach has remarkably reduced the stresses on ventilated discs. Consequently, this single approach could be an improvement to eliminate crack formations in ventilated brake discs.

Kang and Cho conducted the geometric analyses of vents in motorcycle disc brakes which affects the surface of the disc [19]. To study the thermal characteristics of the disc brakes, thermal deformation and thermal stress due to the heat flux for a full and ventilated disc, the three-dimensional model of the disc was elaborated using commercial code ANSYS Workbench. Park et al., conducted thermal analysis method to determine the performance of a brake disc during hydraulic braking process [20]. Thus, the thermomechanical analysis led to the evaluation of temperature and the total deformations of the disc. The aim of this paper was to investigate structural and contact behaviors of the disc brake and pads during the braking phase with and without thermal effects. Firstly, total deformation of the disc-pads model at the time of braking and the stress and contact distributions of the brake pads were determined. Later, results of the thermoelastic coupling such as Von Mises stress, contact pressure field and total deformations of the disc and pads were also presented, which may be useful in the brake design process for automobile industry.

2 DESCRIPTION OF A DISC

The friction tracks are known as external when they are situated on the side of the rim and interior when they are located on the side of the axle. The disc consists of a solid ring with two friction tracks, a bowl which is

fixed to the hub and one which is fixed the rim and a connection between the tracks and the bowl (Fig. 1). This connection is required because the ring and bowl part which is fixed at the hub is not on the same plane for reasons of obstruction and housing of the pads and caliper. The junction between the bowl and the tracks is often machined into the shape of a throat to limit the heat flux resulting from the tracks towards the bowl in order to avoid excessive heating of the rim and the tire.

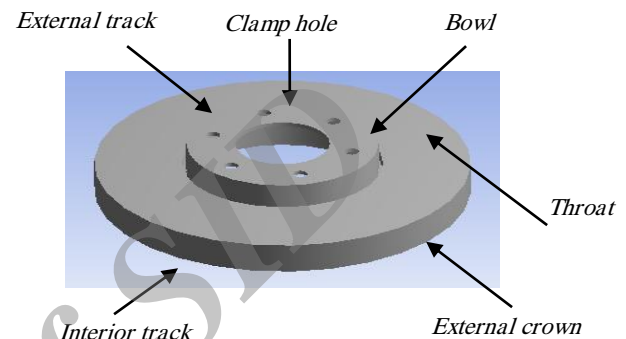


Fig. 1 The full brake disc

The throat area of the bowl is likewise severely solicited. Indeed, the disc tends to be put in cone due to the expansion cone because of dilations of the hottest tracks, but this displacement is retained by the presence of the bowl which is less hot and by that of the clamp, which leads into great stress concentration in this zone. During very severe tests on dynamometric bench, one may notice appearance of a circumferential crack (exterior side and/or internal side of the disc) which is propagated and may cause the brutal rupture of the bowl.

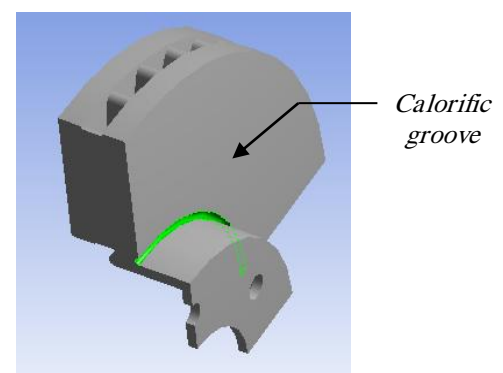


Fig. 2 Calorific groove

The gradients in the groove of the bowl are explained in the same manner. At the beginning of braking, the temperature of the bowl is 20°C while that of the tracks is a few hundred degrees. Moreover, in order to prevent the temperature rise of the hub (what would generate rises in temperature of the tire, very critical for its

behavior), the groove is machined so as not to transmit too much heat to the bowl (Fig. 2). With this machining, the temperature of the bowl effectively decreases, but the heat gradients increase consequently in this zone, where thermal stresses are generated, which explain the rupture of bowl observed during rigorous experimental tests.

3 TYPES OF DISC BRAKE

There are two types of discs: full discs and ventilated discs. The full discs pose simple geometry and thus are simply manufactured; they are generally installed at the rear axle of the car. They are composed quite simply of a full crown connected to a "bowl" which is fixed to the hub of the car (Fig. 3). The ventilated discs, having more complex geometry, appeared more tardily. They are most of the time on the nose gear. However, they are increasingly used at the rear and front cars, upscale composed of two crowns - called flasks - separated by fins (Fig. 4); they cook better than the full discs thanks to ventilation between the fins, which promote convective heat transfer by increasing the exchange surface discs. The ventilated disc comprises more matter than the full disc; its capacity for calorific absorption is thus better.

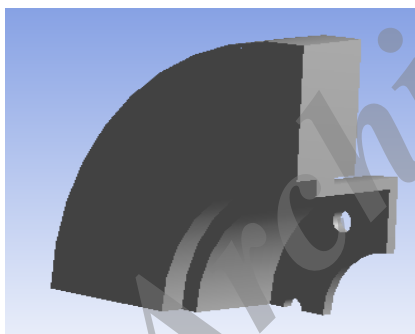


Fig. 3 Full disc

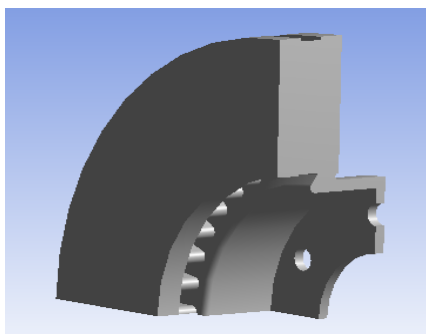


Fig. 4 Ventilated disc.

4 PROBLEM OF BRAKE DISC

The literature review of the phenomena of braking shows that the principal request comes from the sharp variations in temperature induced by the friction of the pads against the disc. Thermal load is due to the temperature rise during braking, leading to gradients of temperature, and thermal cycles because of conduction, convection or radiation. Indeed, the temperature can vary from 20°C to more than 700°C in a few seconds only. These abrupt temperature variations do not make it possible to be homogenized. So the disc experiences very high thermal gradients in the thickness of the friction tracks, but also in the circumferential direction. These last gradients are explained by the fact that the heat flux which enters the disc is localized under the brake pads. Sometimes, it appears what is called of the hot-spots; they are circular zones regularly spaced on the tracks where the temperature is locally higher. Subjected to such thermal cycles, the disc undergoes the anelastic deformations (plastic and even viscoplastic).

The numerical prediction of the thermomechanical fields which are established in the disc, has implemented the method of fundamental calculation which takes into account the essential couplings between the various phenomena, the transitory character of the thermal history of the disc, the anelastic behavior of material, the orthoradial thermomechanical gradients and the rotation of the disc.

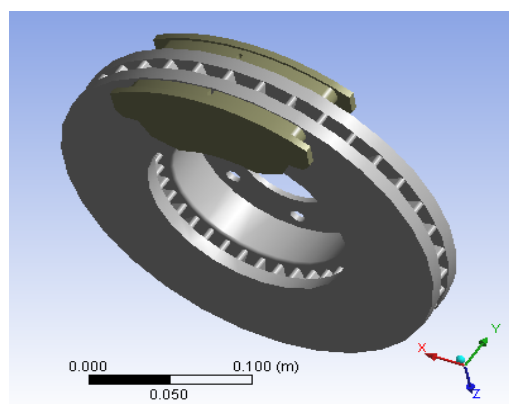


Fig. 5 Model creation on ANSYS WB11

5 FINITE ELEMENT (FE) MODEL

The first step was to prepare a structure model of the brake disc with pads. This was carried out using finite element software (Fig. 5). Then it was meshed and defined by boundary conditions to put on ANSYS

Multiphysics and to initialize the calculation. In this work, a three-dimensional FE model consists of a ventilated disc and two pads as illustrated in Figure 6. Whilst, Figure 7 shows contact zone between the disc and pad, details of the mesh properties are given in Table 1. A frictional contact pair was defined between disc-pad interfaces. Table 2 gives the results of the element of contact where the element types used were Quadratic Quadrilateral Contact (Conta 174) and Quadratic Quadrilateral Target (Targe 170).

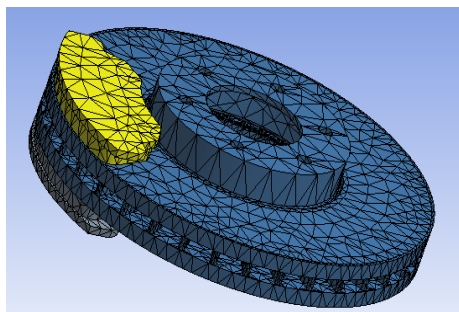


Fig. 6 FE model of a disc-pad assembly

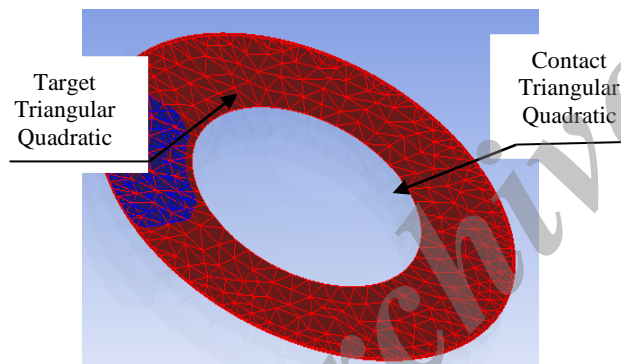


Fig. 7 Contact zone of the disc and pad

Table 1 Results of a mesh type of tetrahedron elements quadratic at 10 nodes

Name	Nodes	Elements
Disc	34799	18268
Pad 1	1446	650
Pad 2	1461	660
Contact zone 1	0	914
Contact zone 2	0	83

The selected material of the disc is Gray cast iron FG 15 with high carbon content and the brake pad has an isotropic elastic behavior whose mechanical characteristics of the two parts are recapitulated in Table 3. Design characteristics of the pieces are also

provided directly by the code ANSYS 11, as given in Table 4 [21]. A commercial FE software, namely ANSYS 11 (3D) is fully utilized to simulate structural deformation, stress, temperature and contact pressure distributions of the disc brake during braking application.

Table 2 Summary of elements types.

Genetic type element	Name ANSYS	Description
Quadratic tetrahedron at 10 nodes	Mesh200	Meshing Facet
triangular quadratic contact	Conta174	3D 8 Node Surface to Surface Contact
Triangular quadratic target	Targe170	3D Target Segment

Table 3 Thermoelastic properties used in simulation

	Disc	Pad
Young modulus E (GPa)	138	1
Poisson's ratio ν	0.3	0.25
Density ρ (kg/m ³)	7250	1400
Coefficient of friction μ	0.2	0.2
Thermal conductivity (W/m°C)	57	5
Specific heat, c (J/Kg. °C)	460	1000

Table 4 Design characteristics of the disc and pads

	Disc	Pad
Volume (m^3)	9.5689e-004	8.5534e-005
Surface (m^2)	0.24237	1.8128 e-002
Mass (kg)	6.9375	0.44975
Faces	205	35
Edges	785	96
Summits	504	64
Nodes	34799	2165
Elements	18268	1014
Inertia moment Ip1	3.5776e-002	2.7242e-005
Inertia moment Ip2	6.9597e-002	1.5131e-004
Inertia moment Ip3	3.5774e-002	1.2863e-004

The coefficient of friction μ is equal to 0.2 at the contact zone. In the case of friction, the latter is of the contact interface, where the shear stresses caused by friction on this level are the origin of this phenomenon. The friction conditions include those parameters which

affect the flexibility and the quality of the produced samples. Changes due to friction may cause changes in the stress-strain distribution [22]. The coefficient of friction depends on many parameters (pressure, sliding speed, temperature, humidity, etc.). It is recalled that ANSYS can either use a method of Lagrange multipliers or augmented Lagrangian method, a penalty method to solve the problem of contact, where the latter is selected in this research [23].

5.1. Determination of hydraulic pressure

During braking, the contact patch between tire and ground begins to slip. This substantially affects both the longitudinal and lateral friction coefficients [24]. In this study, initial mechanical calculation aims at determining the value of the contact pressure (presumably constant) between the disc and the pad. It is supposed that 60% of the braking forces are supported by the front brakes (both rotors), that is to say 30% for a single disc [25]. The force of rotor for a typical vehicle is calculated using the vehicle data contained in Table 5.

Table 5 Vehicle data

Item	Value
Vehicle mass - M [kg]	1385
The initial velocity - v_0 [m/s]	60
Time to stop - t_{stop} [s]	45
The effective radius of the disc - [mm]	100.5
The radius of the wheel - [mm]	380
Friction coefficient disc/pad μ [/]	0.2
Pad Surface A_d [mm ²]	5246.3

Working forces to the brake disc [25]:

$$F_{disc} = \frac{(30\%) \cdot \frac{1}{2} M v_0^2}{2 \cdot \frac{R_{rotor}}{R_{tire}} \left(v_0 \cdot t_{stop} - \frac{1}{2} \left(\frac{v_0}{t_{stop}} \right) t_{stop}^2 \right)} = 1047.36 \text{ [N]} \quad (1)$$

The rotational speed of the disc is calculated as follows:

$$\omega = \frac{v_0}{R_{tire}} = 157.89 \text{ rad/s} \quad (2)$$

Total disc surface in contact with pads is 35797 mm² as depicted in Fig. 8. The external pressure between the disc and pads is calculated by the force applied to the disc, for a flat channel, the hydraulic pressure is [26]:

$$P = \frac{F_{disc}}{A_c \cdot \mu} = 1 \text{ [MPa]} \quad (3)$$

Where A_c is the surface area of the pad in contact with the disc and μ is the friction coefficient. The surface area of the pad in contact with the disc in mm² is provided directly to the ANSYS by selecting this surface as indicated in Fig 9 as green color. In the case of a brake pad without groove, the calculation of the hydraulic pressure is obtained in the same manner.

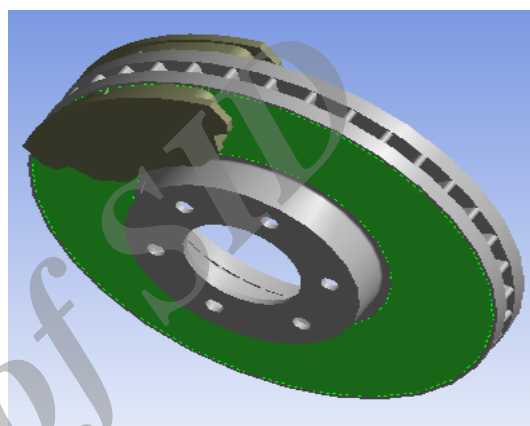


Fig. 8 Contact surface of the disc

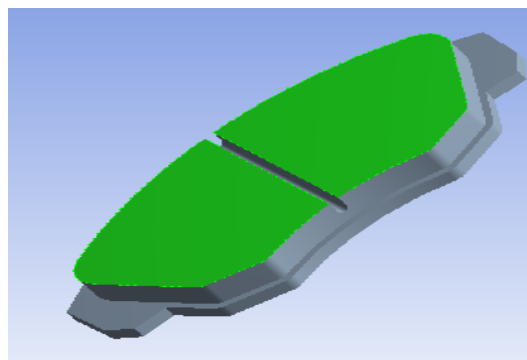
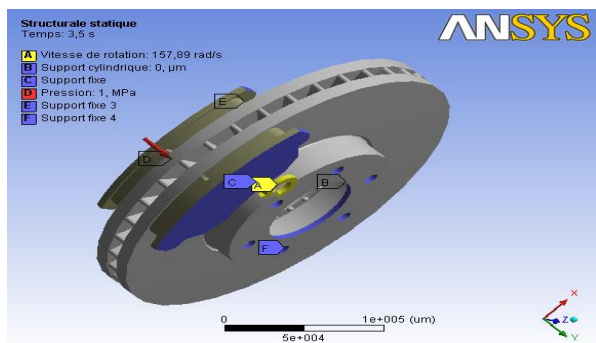


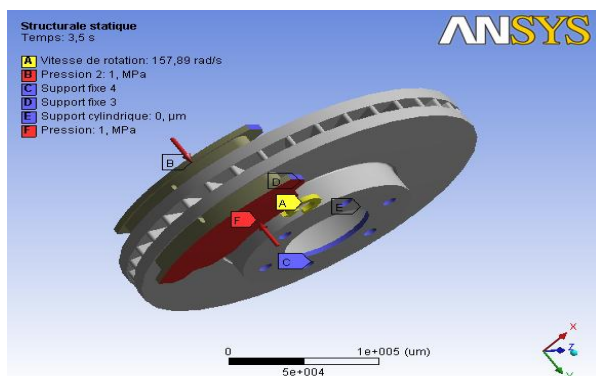
Fig. 9 Contact surface of the pad

5.2. Boundary conditions and loading of the disc and pads

In this FE model, boundary conditions in embedded configurations are imposed on the models (disc-pad) as shown in Fig. 10(a) for applied pressure on one side of the pad and Fig. 10(b) for applied pressure on both sides of the pad. The disc is rigidly constrained at the bolt holes in all directions except in its rotational direction. Meanwhile, the pad is fixed at the abutment in all degrees of freedom except in the normal direction to allow the pads move up and down and in contact with the disc surface [27].



(a) Applied pressure on one side of the pad



(b) Applied pressure on both sides of the pad

Fig. 10 Boundary conditions and loading imposed on the disc-pads

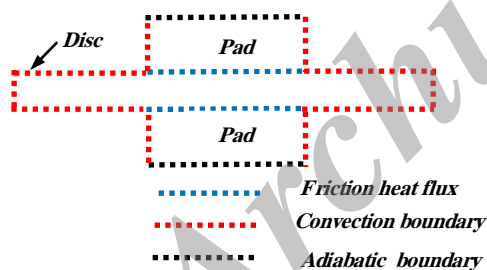


Fig. 11 Boundary condition for thermal analysis of the disc brake

5.3. Thermal boundary conditions

To express the heat transfer in the disc brake model, thermal boundary conditions and initial condition have to be defined. As shown in Figure 11, at the interface between the disc and brake pads, heat is generated due to sliding friction, which is shown as dashed lines. In the exposed region of the disc and brake pads, it is assumed that heat is exchanged with the environment through convection [28]. Therefore, the convection surface boundary condition is applied there. On the surface of the back plate, adiabatic or insulated surface boundary condition is used as presented in Figure 11. The vehicle speed decreases linearly with respect to

time where the variation of heat flux during the simulation is represented in Fig. 12.

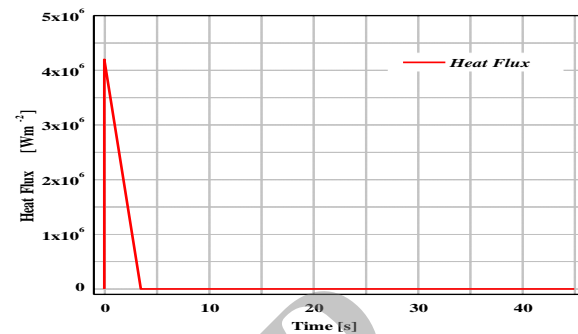


Fig. 12 Heat Flux versus time

6 RESULTS OF MECHANICAL CALCULATION AND DISCUSSIONS

The computer code ANSYS also allows determination and visualization of the structural deformations due to the sliding contact between the disc and the pads. The results of calculations of contact described in this section is related to the displacements or the total deformation during the loading sequence, the field of equivalent stresses Von Mises on the disc, and the contact pressures of inner and outer pad at different braking period.

6.1. Meshing of the Model

The finite element model of the rotor is carried out with a mesh of 20351 elements for a total of 39208 nodes, where the mesh of the disc and pad resulting from ANSYS software is presented in Figure 13.

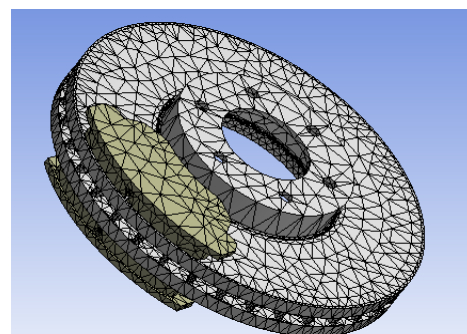


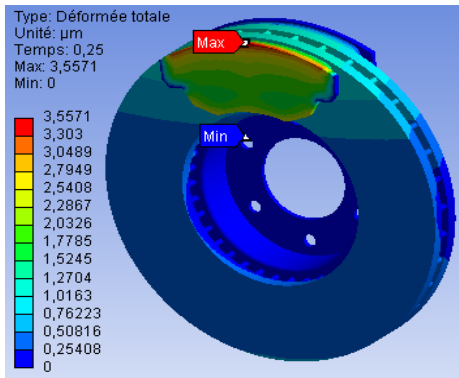
Fig. 13 Voluminal Mesh of the disc and pads (39208 nodes, 20351 elements)

6.2. Disc-pad deformation

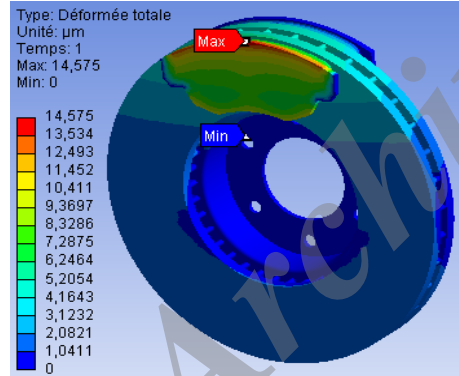
Figure 14 shows disc deformations against braking time. It is noted that the large deformation is always found at outer radius of the disc, that is the area in contact with the pad. From this figures it can be seen

that the highest deformation is $52.8 \mu\text{m}$ and it is predicted at braking time $t = 3.5$ [s] and onwards. As for the pads, the huge deformation is located at the outer radius region (visualized in the red colour) as depicted in Figure 15.

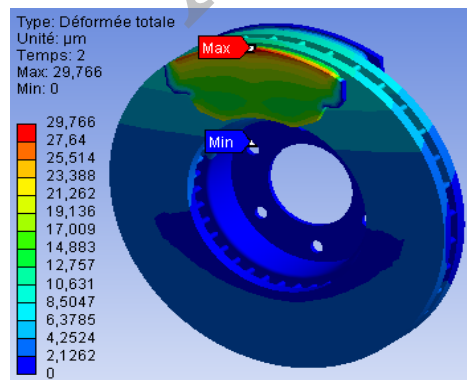
It is also seen that the pads are less deformed compared to the disc and this is shown in Figure 16. At braking time of $t = 3.5$ s and onwards, the maximum deformation of the pads is predicted at $19.1 \mu\text{m}$ which is 64% lower than that of the disc. This is due to the structural stiffness and type of constraints that have been assigned to the disc and pads.



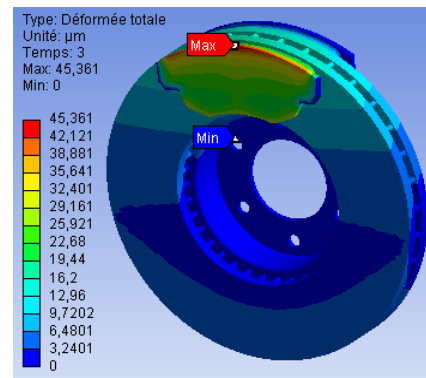
(a): at time $t = 0.25$ [s].



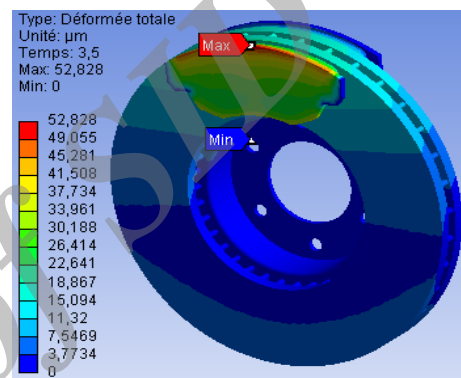
(b): at time $t = 1$ [s].



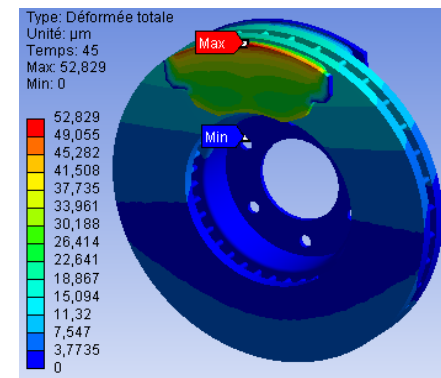
(c): at time $t = 2$ [s].



(d): at time $t = 3$ [s].

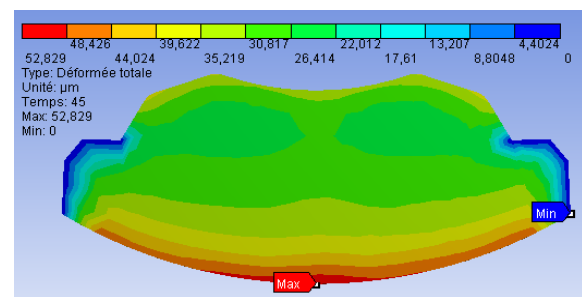


(e): at time $t = 3.5$ [s].

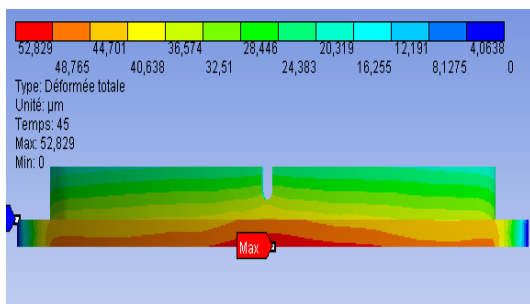


(f): at time $t = 45$ [s].

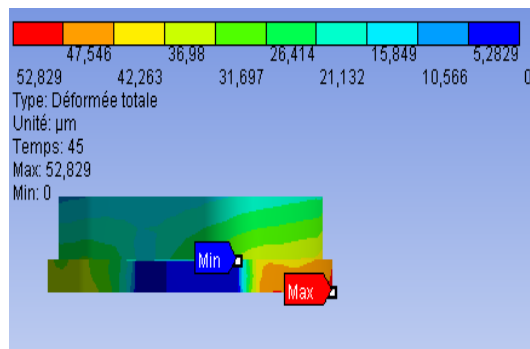
Fig. 14 Disc deformation at different braking time



(a): Front view



(b): View from below.



(c): Right view.

Fig. 15 Pad deformation at braking time $t = 45$ [s].

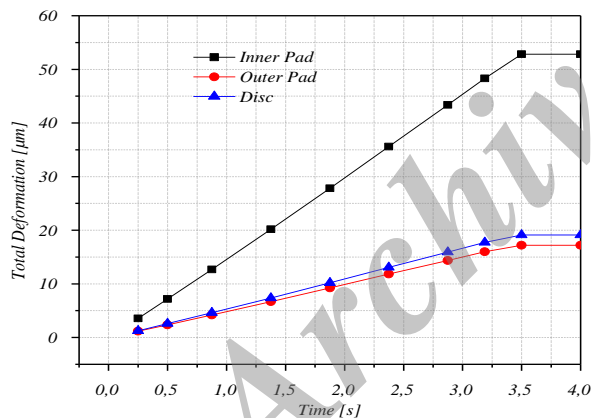
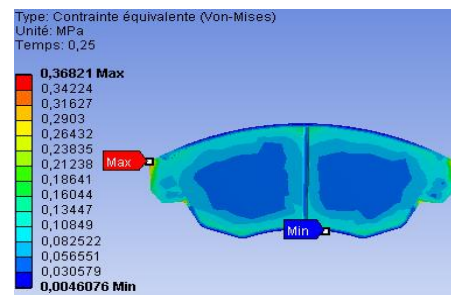


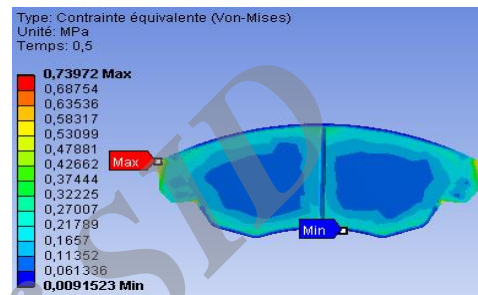
Fig. 16 Deformation of the disc and pads at different braking period

6.3. Stress distribution of the pads

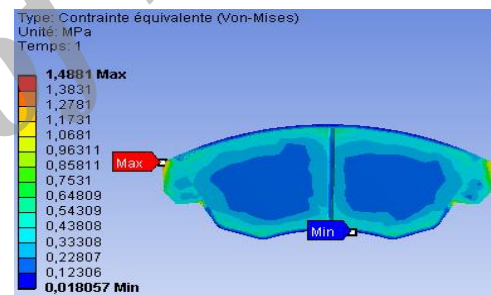
From Fig. 17, it can be observed that the equivalent Von Mises stress is distributed almost symmetrically between the leading and trailing side of the pad. These stress distributions are barely unchanged over braking time except the stress value. It shows that the stress increases gradually and it reaches its maximum value of 5.3 MPa at braking time of 3.5s and onwards. The highest stress is predicted on the left side and the outer radius of the pad whilst the lowest stress is located on the lower radius of the pad and near the groove area.



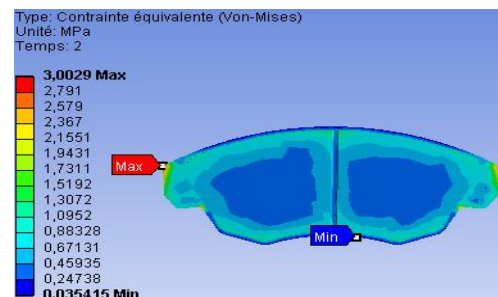
(a): at time $t = 0.25$ [s].



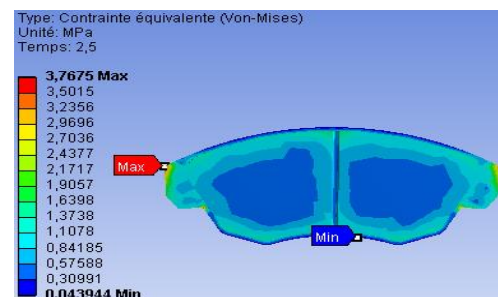
(b): at time $t = 0.5$ [s].



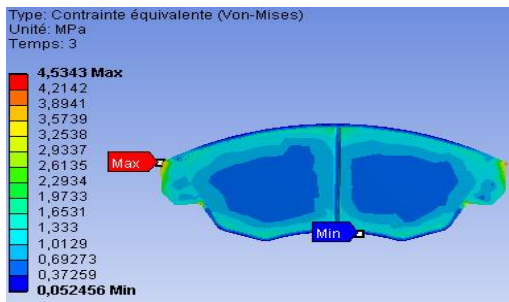
(c): at time $t = 1$ [s].



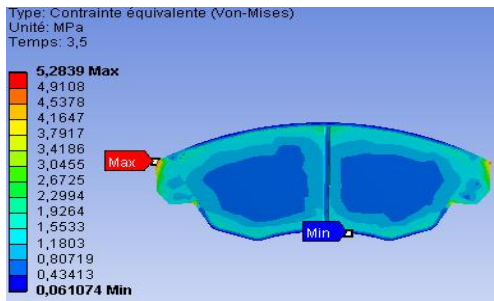
(d): at time $t = 2$ [s].



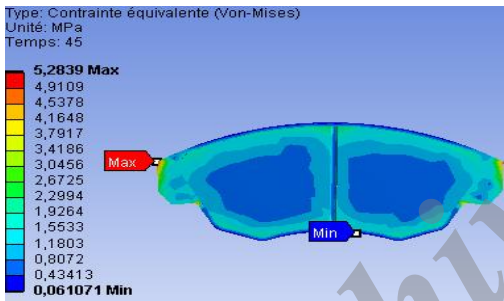
(e): at time $t = 2.5$ [s].



(f): at time t = 3 [s].



(g): at time t = 3.5 [s].



(h): at time t = 45 [s].

Fig. 17 Von Mises stress distribution of the inner pad over braking time

7 RESULTS OF THERMOELASTIC CALCULATION

In this section, structural and thermal analysis are coupled using ANSYS Multiphysics to identify the stress levels and global deformations of the model studied during the braking phase under the effect of temperature. The initial temperature of the disc and pads is set at 20°C, and the surface convection condition is applied to all surfaces of the disc and the convection coefficient (h) of 5 W/m²°C is applied at the surface of the two pads. As shown in Figure 19 that at braking time of 1.7 sec., the disc and pad surface generates quite high temperature, i.e. 346.3°C. However, the upper part of the back plate shows a lower temperature approximately at 90°C. This is caused by the effect of convection ambient air.

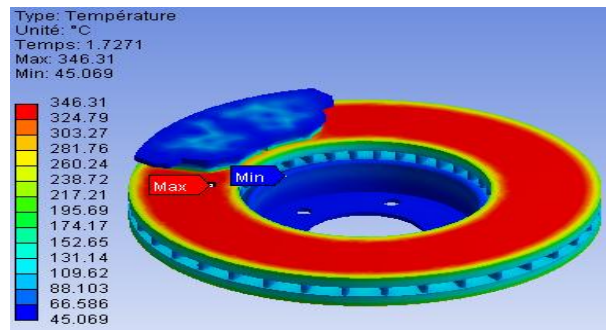


Fig. 19 Temperature distribution of the disc and pads at braking time t = 1.7 [s]

7.1. Deformation and Von Mises stress of the disc-pad model

In this coupled structural-thermal analysis, the aim is to gain a better understanding of the total deformation of the disc when it is not only subjected to the load from the pads but also the expansion induced by the temperature effect. Figure 20 shows the disc deformations at the nodes located on the mean and outer radius of the disc. A clear difference of the disc deformation can be found between these two regions where the outer radius of the disc is recording higher deformation than the mean radius. The curves indicate umbrella phenomenon which is resulting from the heating of non-parallel paths of friction with respect to the initial position.

It is also observed that the disc deformation increases linearly as a function of the disc radius as shown in Figure 21. The highest deformation is predicted at an angular position of 90° and the lowest deformation is located at 270°. The result shows that there are significant differences between the mechanical and thermo-elastic model in terms deformation.

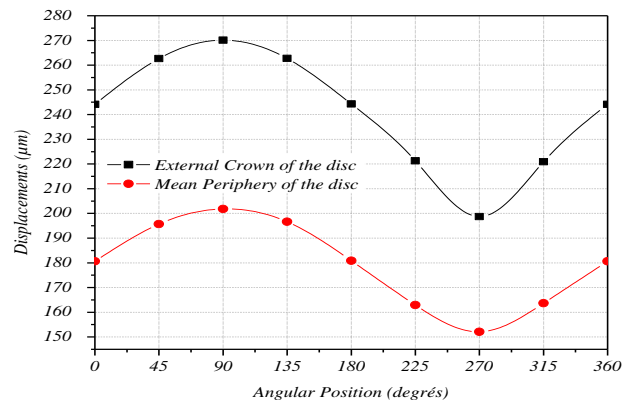


Fig. 20 Disc deformation at the mean and outer radius over angular positions at braking time t = 3.5 [s]

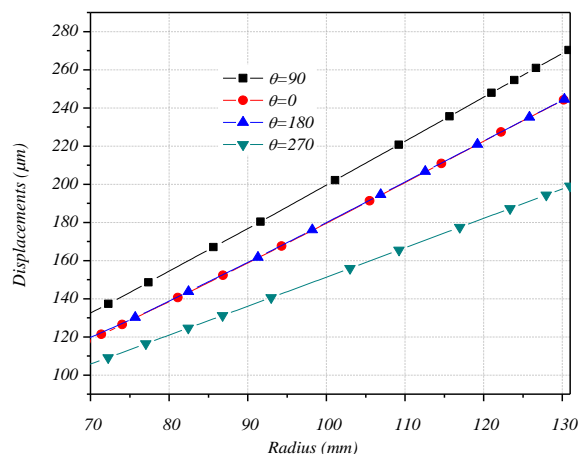


Fig. 21 Disc deformation at different radius and angular positions at braking time $t = 3.5$ [s]

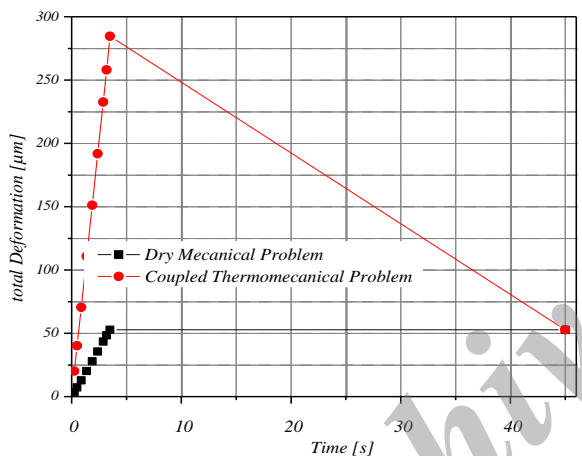


Fig. 22 Disc deformation with and without thermal effects

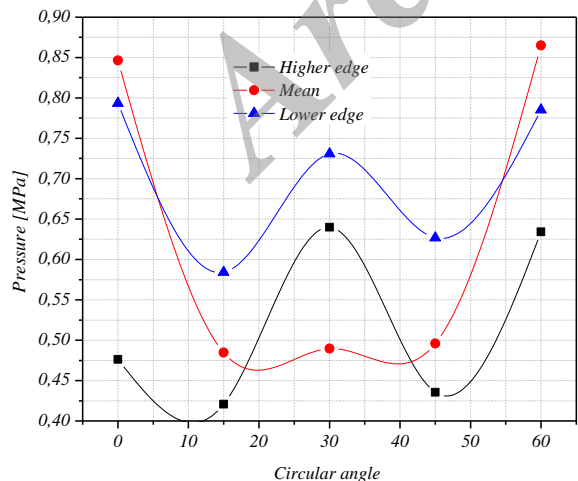


Fig. 23 Distribution of contact pressure along the lower, middle and upper raduis of the pad at time $t = 1.7$ [s]

From Figure 22, it is shown that the disc deforms severely under the effect of temperature. For instance at the braking time of 3.5 sec, the disc with the thermal effect deforms at $280\mu\text{m}$ compared to $53\mu\text{m}$ for the disc without the thermal effect. This clearly indicates that the temperature has a strong influence on the thermos-mechanical response of the brake disc.

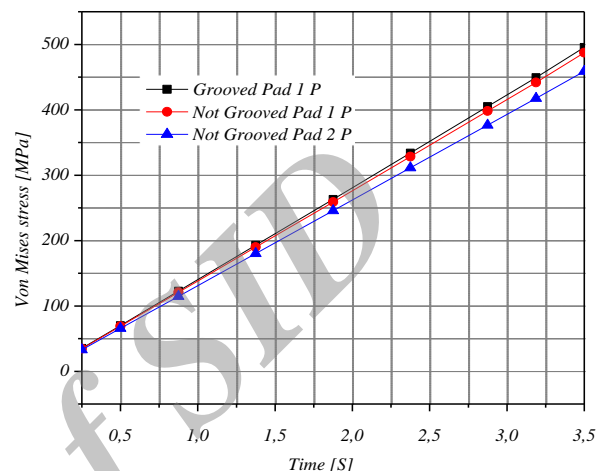


Fig. 24 Variation of von Mises stress versus time at the thermo-mechanical coupling

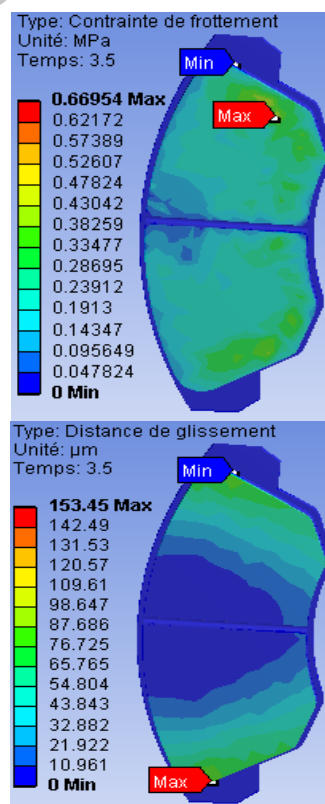


Fig. 25 Friction stress and sliding distance on the inner pad at time $t = 3.5$ [s]

7.2. Contact pressure distribution

Contact pressure distribution that is being plotted in Figure 23 is based on braking time $t = 1.7$ [s] where the pad surface temperature is $T = 346^\circ\text{C}$. It is also seen that the contact pressure curves are almost identical in shape for three different regions of the pad. At 30° angle, contact pressure is predicted to be higher at the lower pad radius followed by the outer pad radius and middle pad radius. The most significant outcome is to see the effect of temperature on contact pressure of the disc-pad model.

In Fig 24, it is found that the presence of the slot affects negatively the stress on the disc surface, unlike the use of the double-piston. Figure 25 shows that the friction stress and sliding distance of the inner pad are symmetrical with respect to the groove and is highest at the edges.

7.3. Deformation of the disc

During a braking maneuver, the maximum temperature achieved on the tracks depends on the storage capacity of the thermal energy in the disc. It is observed in Figure 26, that the maximum displacement is localized on the slopes of friction, the fins and the outer ring. This phenomenon is due to the fact that the deformation of the disc is caused by the heat (the umbrella effect) which can lead to cracking of the disc. In this case, the thermo-coupling analysis is quite important while thermal gradients and expansions generate thermal stresses in addition to mechanical stresses.

7.4. Effect umbrella

The temperature rise of the brake disc provokes a movement of the friction tracks relative to the initial state, where this deformation is called umbrella effect (Figure 27). The maximum total deformation is occurred at the outer rim of the disc, which reaches $284.55 \mu\text{m}$, at time $t = 3.5$ s. An increase in pressure and hence the temperature on a limited contact surface can create this phenomenon (heat deformation), local material fatigue and sometimes cracking of the disc. The umbrella disc effect is not desired because it is under a negative influence on the effectiveness of the brakes. Correct operation of a brake under the influence of a thermal load is limited by certain thermo-mechanical phenomena such as:

- Cracking due to the temperature gradient on the friction tracks, where this may cause disc rupture.
- Deformation of the disc due to heat (umbrella effect) which influences the contact surface, thus reducing the brake effectiveness.

- Disc wear and brake pads
- Influence on the environment of the disc (the clamps, the state oil...).

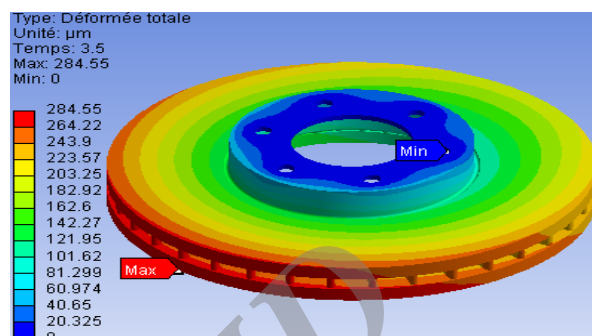


Fig. 26 Maximum total deformation of the disc with thermo-mechanical coupling

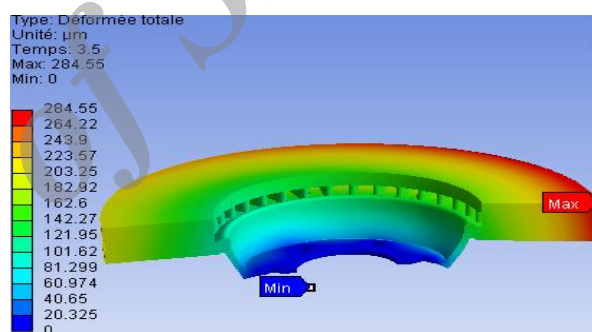
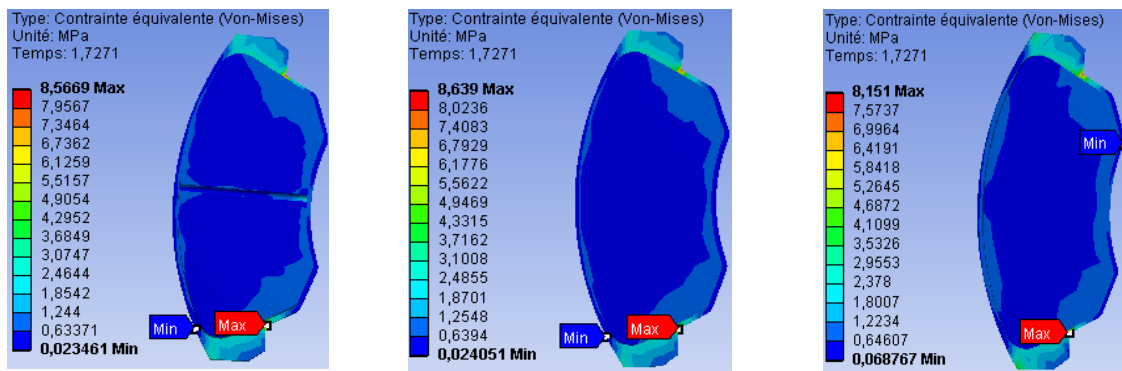


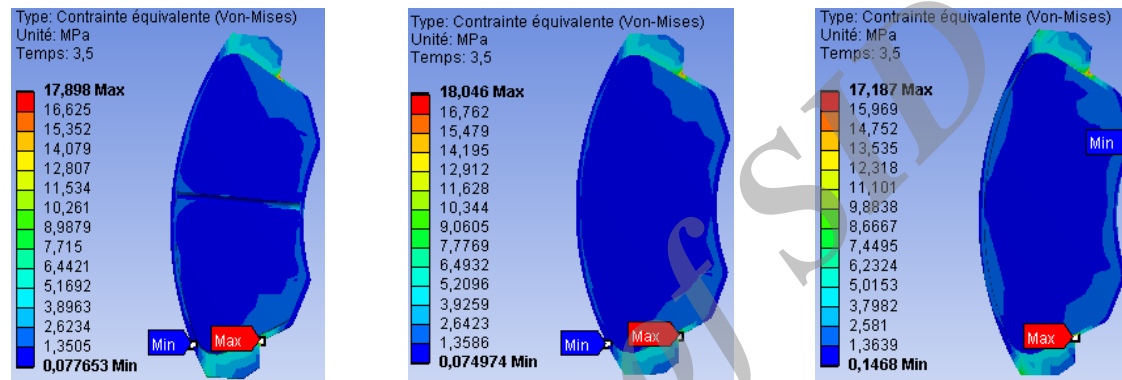
Fig. 27 Effect of an umbrella disc

7.5. Von Mises stress at the inner pad

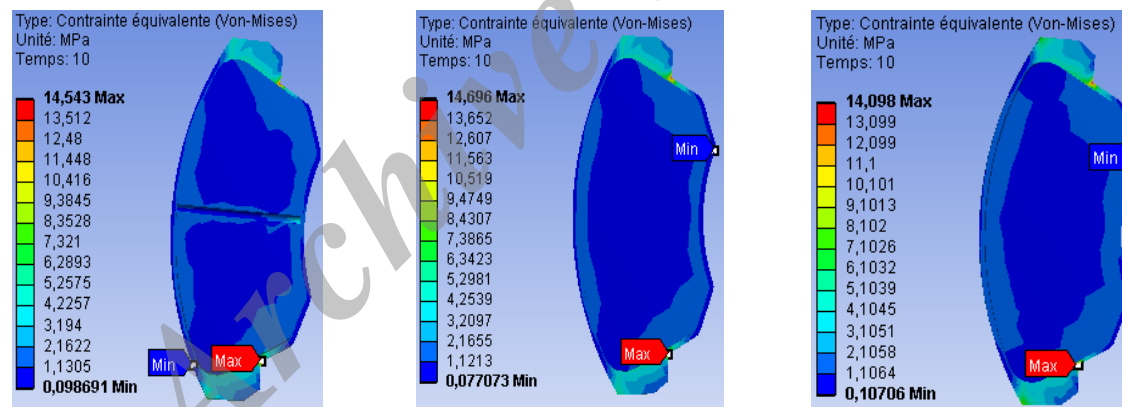
In this section, influence of the pad groove and the loading mode (single and dual piston) on the equivalent Von Mises stress distribution is presented. The stress evolution for three pad designs can be seen in Figure 28(a)-28(g). It is shown that at the beginning of braking time ($t = 1.7$ sec) most of the pad contact surfaces are covered by a dark blue which indicates a lower stress. However, when it comes to braking time $t = 45$ sec the stress level has been increased, where the colour spectrum now becomes almost an ocean blue. At this braking time, the stress distribution is also clearly noticed between those three pad designs. The presence of the groove and the double piston loading provide a positive affects on the stresses of the pad.



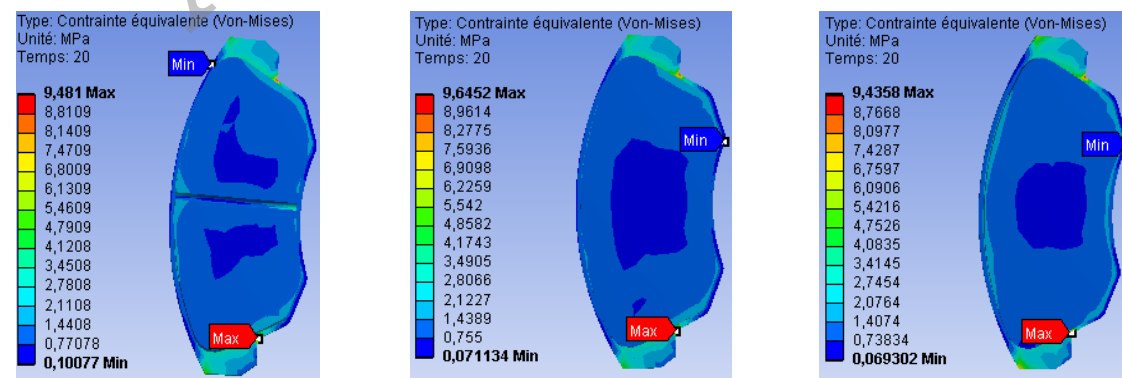
(a): at time $t = 1.7 [s]$



(b): at time $t = 3.5 [s]$



(c): at time $t = 10 [s]$



(d): at time $t = 20 [s]$

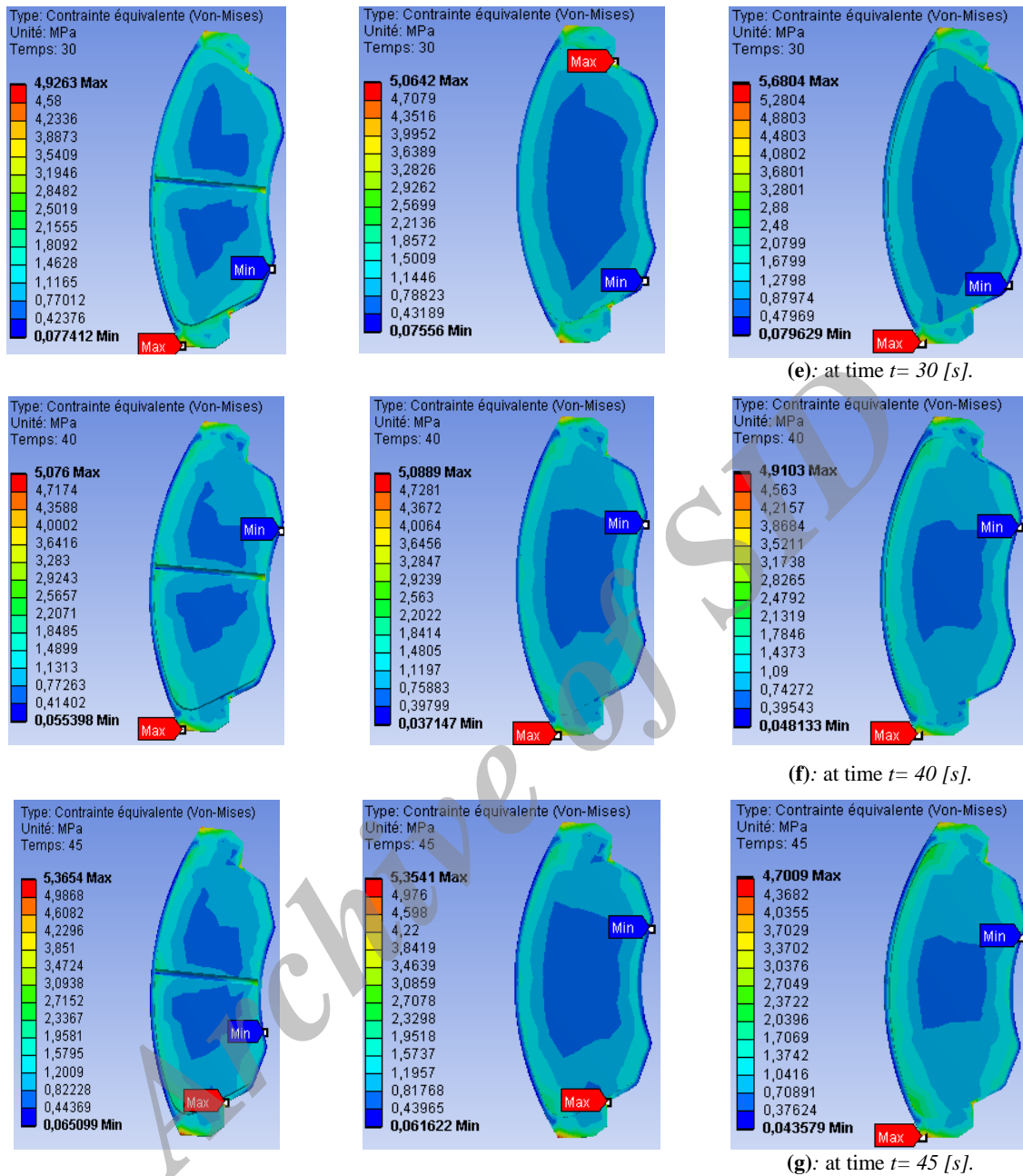


Fig. 28 Distribution of Von Mises stress in the inner pad: Single piston with pad center groove (left), Single piston without groove (center) and double-piston without pad groove (right).

8 CONCLUSION

In this work, a disc-pad model has been analyzed using two approaches, namely mechanical and thermo-mechanical analysis. In addition to this, three pad designs are also simulated to identify its influence on the stress distribution. From the prediction results it can be deduced that:

- Large deformations are occurred at the outer radius of the disc.
- The presence of grooves in the pads influences unfavorably the mechanical behavior of a brake.
- More uniform stress distribution is noticed for the pad without groove and with a double-piston loading.

- Temperature has a significant effect on the structure and contact behaviour of the disc brake assembly. Large deformation and high contact pressure is found in the disc-pad model with the thermal effect.
- The smoothness of the mesh increases the precision of the solution.
- The results obtained for the numerical calculation are comparable with those which one finds in the specialized literature.

Concerning the prospects, one can quote certain research orientations:

- Experimental study to verify the accuracy of the numerical model developed,
- Tribology and vibrations study of the contact disc-pads,
- Study of the dry contact sliding under the macroscopic aspects (macroscopic state of the surfaces of the disc and pads).

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