Thermomechanical Modelling of Disc Brake Contact Phenomena

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In automotive disc brake system, friction heat is not uniformly distributed due to various reasons such as thermal expansion and imperfections in geometry. It is well known that thermoelastic distortion due to frictional heating affects the contact pressure distribution and can lead to thermoelastic instability, where the contact load is concentrated in one or more small regions on the brake disc surface. These regions then take very high temperature and passage of hot spots moving under the brake pads can cause undesirable effects, such as brake fade, thermal cracks, and vibration. The main purpose of this study is to analysis the thermomechanical behavior of the dry contact between the brake disc and pads during the braking phase. The thermal-structural analyye is then used to determine the deformation and the Von Mises stress established in the disc, the contact pressure distribution in pads. The results are satisfactory, when 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 road vehicles. The ability of the braking system to bring a vehicle to safe controlled stop is absolutely essential in preventing accidental vehicle damage and personal injury [1].Therefore, the braking system of a vehicle is undeniably important, especially in slowing down or stopping the rotation of a wheel by pressing brake pads against rotating wheel discs.

An interaction between a brake disc and friction material of automotive brake is characterized by a number of dry contact phenomena. These phenomena are influenced by brake operation conditions (applied pressure, speed, and brake interface temperature) and material characteristics of a friction couple. The coefficient of friction should be relatively high and keep a stable level irrespective of temperature change, humidity, age, degree of wear and corrosion, presence of dirt and water spraying from the road [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, it is under 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 significantly be affected by the temperature rise in the brake components [5]. The frictional heat generated at the interface of the disc and the pads can cause a high temperature during the braking process. 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 [6]. Gao and Lin [7] stated that there is considerable evidence that contact temperature is an integral factor reflecting the specific power friction influence of the combined effect of load, speed, friction coefficient, and the thermo physical and durability properties of the materials of a frictional couple. Lee and Yeo [8] 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 found to be the main cause of Judder and disc thickness variation (DTV). Ouyang et al [9] 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 [9] and Hassan et al [10] employed finite element approach to investigate thermal effects on disc brake squeal using dynamic transient and complex eigenvalue analysis, respectively.

Modelling of disc brake output performance could be very complex if using the conventional analytical modelling techniques, regarding the complexity of requirements imposed to brakes, and highly non-linear phenomena involved in the field of tribological interactions [11, 12]. Thus, prediction of synergistic effects of all influencing parameters on disc brake performance in dynamic operating conditions requires accurate and effective tools. As it is discussed in detail in [13], the conventional analytical approaches cannot be able to handle modelling errors and suffer from lack of accuracy and robustness.

The braking performance is significantly affected by the temperature rise in the process of halting the vehicle. Each moment (time step) during the continuous braking process gives a different value of temperature distribution as a result of the frictional heat generated on the rotor surface which can cause high temperature rise

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(Qi and Day [14]; Hwang and Wu [15]). When the temperature rise exceeds the critical value for a given material, it leads to undesirable effects in the operation of the rotor such as thermal elastic instability (TEI), premature wear, brake fluid vaporization (BFV) and thermally excited vibrations (TEV) (Gao and Lin [16]; Kao, et al. [17]).

Finite element (FE) method for brake rotor analysis has become a preferred method in studying the thermal distribution performance because of its flexibility and diversity in providing solutions to problems involving advanced material properties. Chandrupatla and Belegundu [18] stated that temperature distribution analysis is mostly performed using FE method due to its powerful tool for numerical solutions for a wide range of engineering problems. Day [19] conducted a study using FE to predict temperature, wear, pressure distribution and thermal distortion of a brake drum which is generated during high pressure brake application from two different road speeds and friction materials. Valvano and Lee |20] proposed a thermal analysis on disc brake based on a combination of computer based thermal model and FE based techniques to provide reliable method to calculate the temperature rise and distortion under a given brake schedule.

In this work, we will apply modeling to the thermomechanical behavior of the dry contact between the disc and the brake pads during braking; the computational modeling is carried out using Ansys Workbench 11 [21]. This last is elaborate 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 heat. 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 on the disc ($k_p < k_d$) consider that the heat quantity produced will be completely absorbed by the brake disc. The heat flux evacuated of this surface is equal to the friction power. The initial heat flux $q_0$entering the disc is calculated by the following formula [22]:

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

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

Indeed, the brake disc assumes the most part of the heat, usually more than 90% [23], through the effective contact surface of the friction coupling. Considering the complexity of the problem and average data processing limited, one replaced the pads by their effect, represented by an entering heat flux as shown in Figure 2.

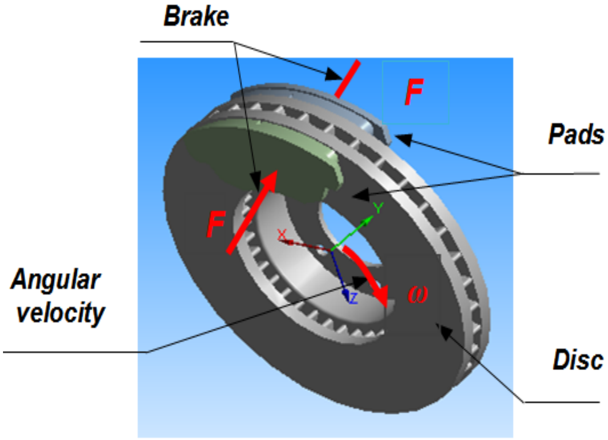


Figure 1.Disc-pads assembly with forces applied to the disc.

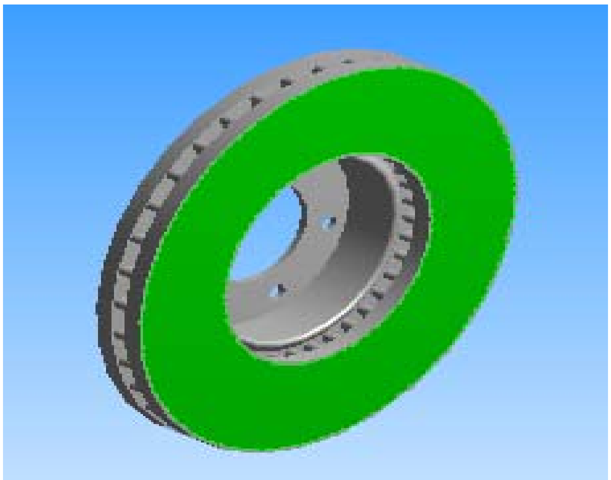


Figure 2. Application of flux.

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 for front disc (Citroën vehicle of type CX GTi Turbo 2)

Item	Values
Inner disc diameter, mm	66
Outer disc diameter, mm	262
Disc thickness (TH), mm	29
Disc height (H), mm	51
Total vehicle mass m, kg	1385
Initial speed v_0, km/h	28
Deceleration a, m/s^2	8
Effective rotor radius R_{rotor}, mm	100.5
Rate distribution of the braking forces Φ, %	20
Factor of charge distribution of the disc ε_p	0.5
Surface disc swept by the pad A_d, mm^2	35993

The disc material is gray cast iron (GFC) ISO strandad with high carbon content [24], with good thermophysical characteristics and the brake pad has an isotropic elastic behaviour, 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, ρ (kg/m^3)	1400	7250
Specific heat, c (J/Kg. °C)	1000	460
Poisson's ratio, ν	0,25	0,28
Thermal expansion, α (10^{-6} / °C)	10	10,85
Elastic modulus, E (GPa)	1	138
Coefficient of friction, μ	0,2	0,2
Operation Conditions		
Angular velocity, ω (rd/s)		157.89
Hydraulic pressure, P (MPa)		1

3. MODELING IN ANSYS CFX

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 [25, 26], 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 (the fluid which is the air and the solid which is the disc) in order to facilitate the mesh on which that one will export the results towards CFX using the command "Output to cfx" [27]. After obtaining the model on CFX Pre and specifying the boundary conditions, we begin calculation on the CFX.The disc is related to four adiabatic surfaces and two surfaces of symmetry in the fluid domain whose ambient temperature of the air is taken equal at 20 °C [28]. An unsteady-state analysis is necessery.

Figure 3 shows the elaborate CFD model which will be used in ANSYS CFX Pre.

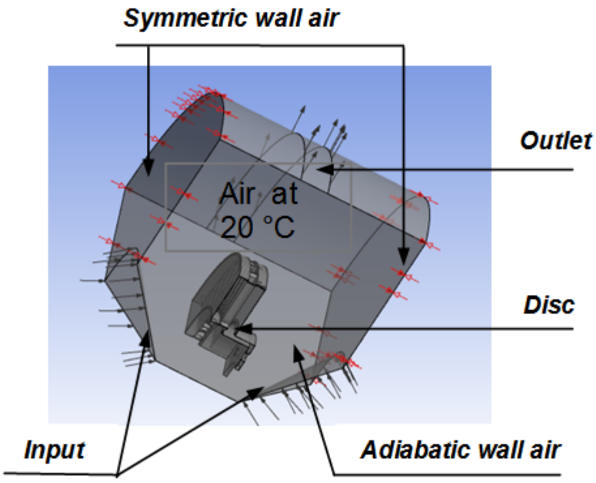


Figure 3.Brake disc CFD model.

In this step, one declares all of the physical characteristics of the fluid and the solid. We introduce into the library the physical properties of used materials. In this study we selected a gray cast iron material (GFC) ISO strandad with its thermal conductivity is equal to (57 W/m°C). Since the aim of this study is to determine the temperature field in a disc brake during the braking phase, we take the following temporal conditions:

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

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

$$h_c = \frac{q_w}{T_b - T_{nw}} \qquad (2)$$

where h_c is a specified heat transfer coefficient, q_w is heat flux at the wall boundary, T_b is the specified boundary temperature (that is, outside the fluid domain) and T_{nw} is the temperature at the internal near – wall boundary element center node. To enable universal usage of the results, the solver had to be set to compute the convective heat transfer coefficient related to ambient (constant) temperature:

The airflow through and around the brake disc was analysed using the ANSYS CFX software package. The Ansys CFX solver automatically calculates heat transfer coefficient at the wall boundary. Afterwards the heat transfer coefficients considering convection were calculated and organized in such a way, that they could be used as a boundary condition in thermal analysis. Averaged heat transfer coefficient had to be calculate for all disc using Ansys CFX Post as it is indicated in Figures.4 and 5.

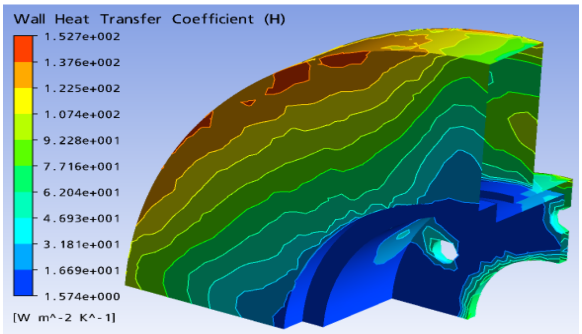


Figure 4. Distribution of heat transfer coefficient on a solid disc in the steady state case (FG 15).

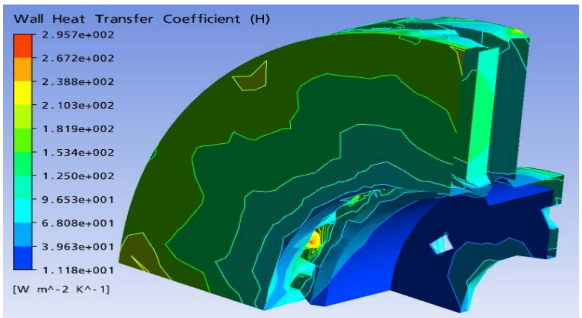


Figure 5. Distribution of heat transfer coefficient on a ventilated disc in the steady state case (FG 15).

4. MESHING OF THE DISC

The elements used for the meshing of the solid and ventilated disc are tetrahedral three-dimensional elements with 10 nodes (isoparametric) (Figure.6). 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 of the contact conditions involves the use of a refined mesh.

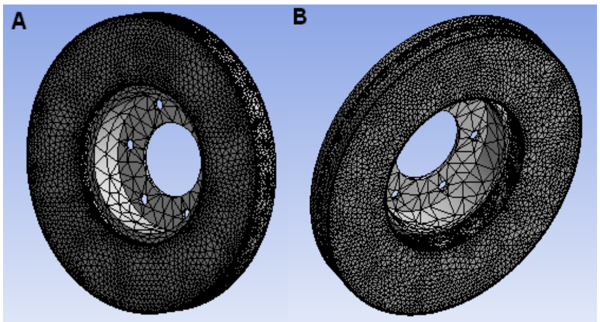


Figure 6. Meshing of the disc (a) full disc (172103 nodes - 114421 elements) (b) ventilated disc (154679 nodes- 94117 elements).

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 the physical properties of the materials. The selected data for the numerical application are summarized as follows:

- Total time of simulation (stopping time)= 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 DISCUSSION

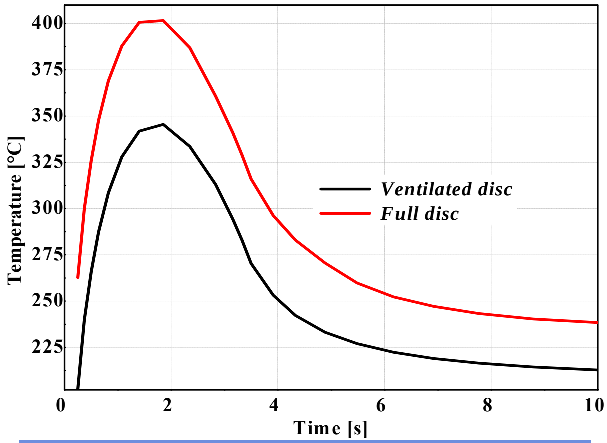
The modeling of temperature in the disc brake will be carried out by taking into account of the variation of a certain number of parameters such as the cooling mode of the disc and the choice of disc material. The brake discs are made of cast iron with high carbon content; the contact surface of the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

6.1 Influence of Construction of the Disc

Figure 7. shows the variation of the temperature versus time during the total time simulation of braking for a solid disc and a ventilated disc. The highest temperatures are reached at the contact surface disc-pads. The high 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, one notices that starting from the first step of time one has a fast 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 made 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 of the same material is about 60 °C at the moment t = 1.88 s.

Comparing the different results obtained from analysis. It is concluded that ventilated type disk brake is the best possible for the present application.



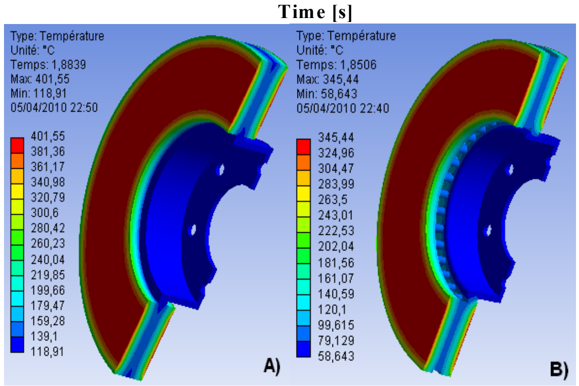


Figure 7. Temperature distribution of a full (a) and ventilated disc (b) of cast iron (FG 15).

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 freeway driving conditions [29]. Figure. 8 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 60°, and the surface convection condition is applied to all surfaces of the disc and the convection coefficient (h) of 5 W/m^2°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. There are about 185901 nodes and 113367 elements used (Figure.9).

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.

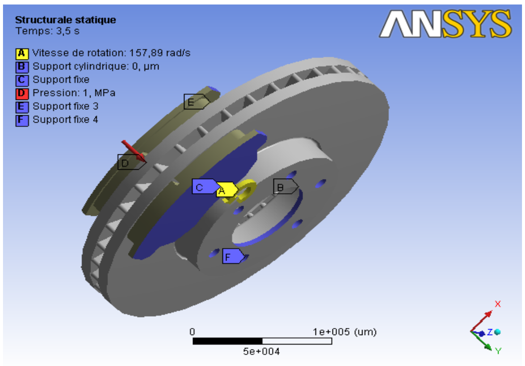


Figure 8. Boundary conditions and loading imposed on the disc-pads.

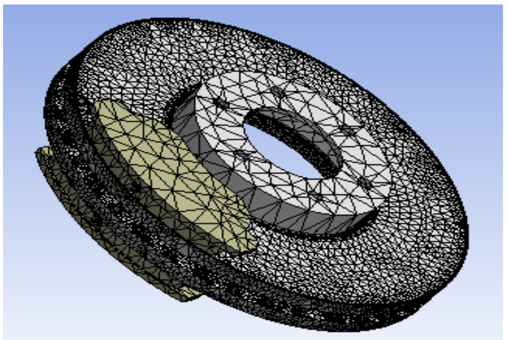


Figure 9. Refined mesh of the model.

7.1 Thermal Deformation

Figure 10 gives the distribution of the total distortion in the whole (disc-pads) for various moments of simulation. For this figure, the scale of values of the deformation varies from 0 μm to 284 μm. 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. One observes a strong distribution 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 the disc (on particular tracks of friction) this deformation will generate asymmetry of the disc following the rise of temperature which will cause a deformation in the shape of an umbrella.

7.2 Von Mises Stress Distribution

Figure.11 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 very significant compared to that obtained with the assistance in the mechanical analysis under the same conditions. One observes a strong constraint on the level of the bowl of the disc. Indeed, the disc is fixed to the hub of the wheel by screws preventing its movement. In the present of the rotation of the disc and the requests of torsional stress and sheers generated at the level of the bowl which are able to create the stress concentrations. The repetition of these effects will involve risks of rupture on the level of the bowl of the disc.

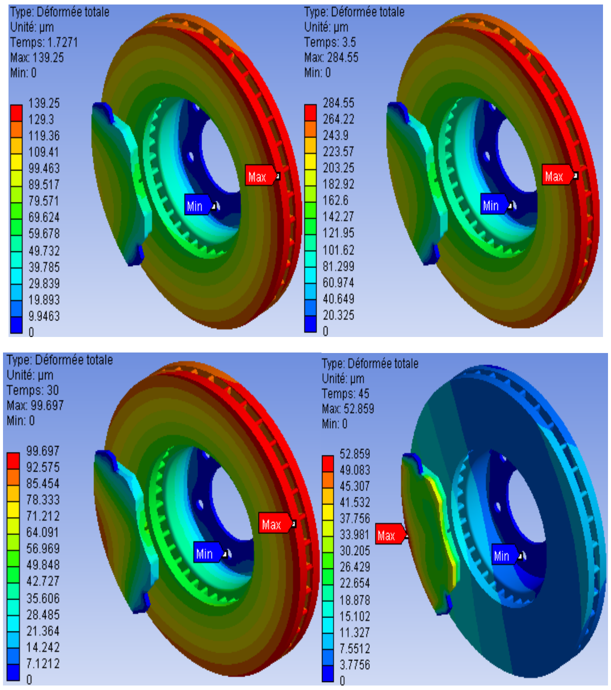


Figure 10. Total distortion distribution.

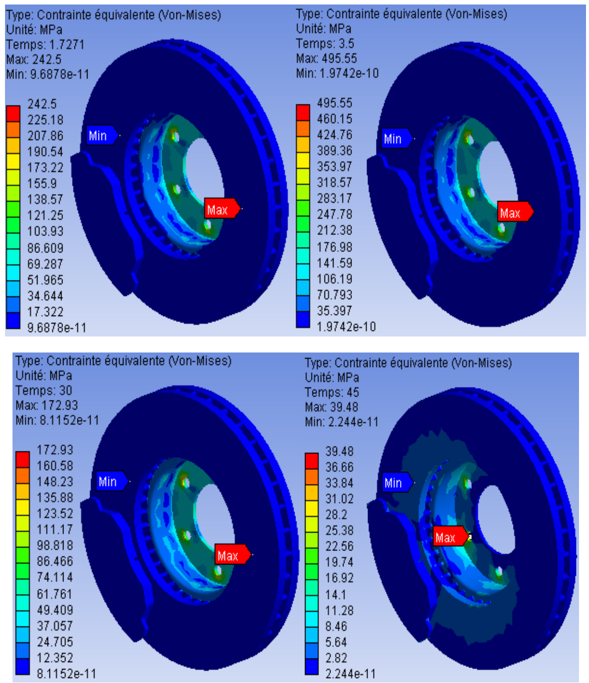


Figure 11. Von Mises stress distribution.

7.3 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, by Tirovic and Day

[30]. Brake squeal analysis in recent years always includes a static contact analysis as the first part of the complex Eigenvalue analysis [31, 32].

Figure.12. shows the contact pressure distribution in the friction interface of the inner pad taken for at various times of simulation.For this distribution the scale varies from 0 MPa to 3,3477 MPa and reached a value of pressure at the moment t=3,5 s which corresponds to the null rotational speed. It is also noticed that the maximum contact pressure is located on the edges of the pad decreasing from the leading edge towards the trailing edge from friction. This pressure distribution is almost symmetrical compared to the groove and it 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 produces the symmetric field of the temperature. This last affects thermal dilation and leads to a variation of the contact pressure distribution

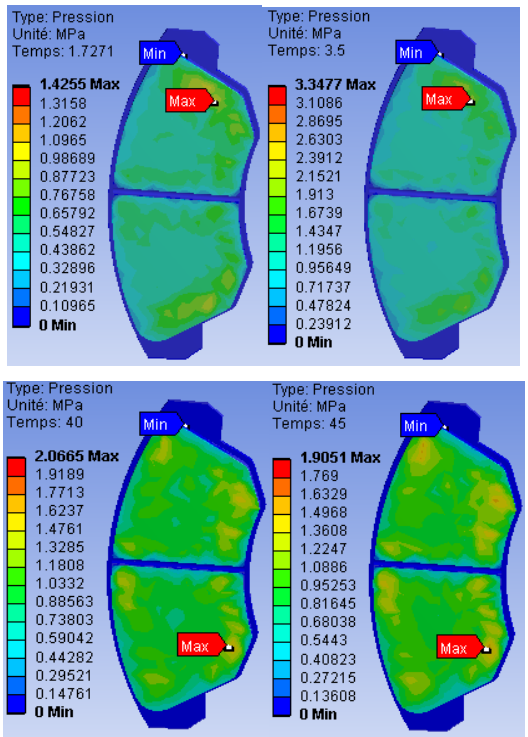


Figure 12. Contact pressure distribution in the inner pad.

CONCLUSION

In this publication, we presented the analysis of the thermomechanical behavior of the dry contact between the brake disc and pads; the modeling is based on the ANSYS 11.0. We have shown that the ventilation system plays an important role in cooling disks. 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 additional to mechanical stress which causes the crack propagation and fracture of the bowl and wear of the disc and pads. 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 a good agreement between the model and reality.

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МОДЕЛИРАЊА ТЕРМОМЕХАНИЧКОГ КОНТАКТА У ДИСК КОЧНИЦАМА

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У аутомобилском систему диск кочница, топлота настала услед трења није униформно расподељена због разних разлога као што су термалне експанзије и несавршености у геометрији. Добро је познато да термоелачна изобличења због грејања насталог услед трења утичу на дистрибуцију контактног притиска и могу да доведу до нестабилности , где су контактна оптерећење концентрисана у једној или више малих зона на површини диска кочнице. Ове зоне су изложене високим температурама што може да изазове нежељене ефекте, као што су слабљење кочења, појава термалних пукотина и вибрација. Основни циљ овог истраживања је термомеханичка анализа сувог контакта између диска и плочица током фазе кочења. Резултати термоструктуралне анализе се потом користе за одређивање деформација и фон Мизесових напона на диску, као и расподела контактног притиска у кочионим плочицама. Резултати су задовољавајући, када се упореде са онима из стручне литературе.