DISC BRAKE ROTOR-THERMAL ANALYSIS

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Abstract

The main purpose of this study is to analysis the thermo mechanical behavior of the dry contact between the brake disc and pads during the braking phase. The simulation strategy is based on computer code ANSYS11. The modelling of transient temperature in the disk is actually used to identify the factor of geometric design of the disk to install the ventilation system in vehicles. The thermal-structural analysis is then used coupling to determine the deformation established and the Von Mises stresses in the disk, the contact pressure distribution in pads. The results are satisfactory compared to those found in the literature.

Keywords: Brake Discs, Heat Flux, Heat Transfer Coefficient, Von Mises Stress, Contact Pressure

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1. Introduction

Passenger cars have been one of the essential transportation for people to travel from one destination to another. 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. Braking performance of a vehicle can significantly be affected by the temperature rise in the brake components. The frictional heat generated on the interface of the disc and the pads can cause 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[1]. Light weight advanced composite materials for vehicle application has been identified in order to reduce fuel consumption in the automobile systems. Over 75% of fuel consumption relates directly to vehicle weight. However, reducing the vehicle weight can improve the cost to performance ratio for the transportation industry. Researchers have shown that the vehicle weight reduction is a promising strategy for improving energy consumption in vehicles, and presents an important opportunity to reduce energy use in the transportation sector[2] .

Gao and Lin[3] stated that there is considerable evidence that shows the contact temperature is an integral factor reflecting the specific power friction influence of combined effect of load, speed, friction coefficient, and the thermo physical and durability properties of the materials of a frictional couple.Lee and Yeo[4] 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[5] in their recent work found that temperature could also affect vibration level
in a disc brake assembly.

In a recent work, Ouyang et al[5] and Hassan et al[6] 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 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[7].

Disc brake consists of 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[8]. 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[9] .

In the study established in 2012 by Park et al.[10], an analysis technique that can estimate the temperature rise and thermal deformation of the ventilated disc considering vehicle information, braking condition and properties of the disc and pad was developed. The analytical process of the braking power generation during braking was mathematically derived. The thermal energy that is applied to the surface of a disc as heat flux was calculated when a vehicle is decelerating from 130 km/h to 0 km/h with deceleration of 0.4 g. Then, the temperature rise and thermal deformation of a disc ware calculated through the thermo-mechanical analysis by using SAMCEF code. The shape of the cross section of the disc is optimized according to the response surface analysis method in order to minimize the temperature rise and thermal deformation.

In this work, we will make a modeling of the thermomechanical behavior of the dry contact between the disc of brake pads at the time of braking phase, the strategy of calculation is base on the software Ansys 11[11]. 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

The brake disk assumes the most part of the heat, usually more than 90%[14], 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 (Figure 1.).

The initial heat flux q_0 entering the disc is calculated by the following formula[15]:

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

Where $z = a/g$: Braking effectiveness, a : Deceleration of the vehicle[ms^{-2}], ϕ : Rate distribution of the braking forces between the front and rear axle, A_d : Disc surface swept by a brake pad[m^2], v_0 : Initial speed of the vehicle[ms^{-1}] ε_p : Factor load distribution of the on the surface of the disc., m : Mass of the vehicle[kg] , g : Acceleration of gravity (9.81)[ms^{-2}].

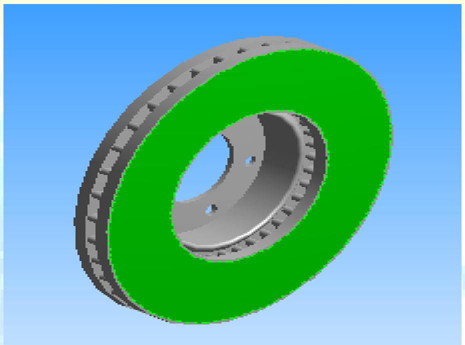


Figure 1. Application of flux

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. Parameters of automotive brake application

Inner disc diameter, mm	66
Outer disc diameter, mm	262
Disc thickness (TH) ,mm	29
Disc height (H) ,mm	51
Vehicule 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	20

Factor of charge distribution of the	0.5
Surface disc swept by the pad	35993

The disc material is gray cast iron (GFC) with high carbon content[16], with good thermophysical characteristics and the brake pad has an isotropic elastic behavior whose thermo- mechanical characteristics adopted in this simulation in the of the two parts are recapitulated in Table 2.

Table 2. Thermoelastic properties used in simulation

Material Properties	Pad	Disc
Thermal conductivity, k	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	10,85
Elastic modulus, E (GPa)	1	138
Coefficient of friction, μ	0,2	0,2
Operation Conditions		
Angular velocity, ω (rd/s)		157.8
Hydraulic pressure, P		1

3. Modelling in ANSYS CFD

The finite volume method consists of three stages; the formal integration of the governing equations of the fluid flow over all the (finite) control volumes of the solution domain. Then discretisation, involving the substitution of a variety of finite-difference-type approximations for the terms in the integrated equation representing flow processes such as convection, diffusion and sources. This converts the integral equation into a system of algebraic equations, which can then be solved using iterative methods[17]. The first stage of the process, the control volume integration, is the step that distinguishes the finite volume method from other CFD methods. The statements resulting from this step express the 'exact' conservation of the relevant properties for each finite cell volume. This gives a clear relationship between the numerical analogue and the principle governing the flow.To enable the modeling of a rotating body (in this case the disc) the code employs the rotating reference frame technique. For the preparation of the mesh of CFD model, one defines initially, various surfaces of the disc in ICEM CFD as the Figure 2. shows it, we used a linear tetrahedral element with 30717 nodes and 179798 elements. In order not to weigh down calculation, an irregular mesh is used in which the mesh is broader where the gradients are weaker (not-uniform mesh), (Figure 3).

The CFD models were constructed and were solved using ANSYS-CFX software package[18]. The model applies periodic boundary conditions on the section sides. As the brake disc is made from sand casted grey cast iron, The disc model is attached to an adiabatic shaft whose axial length spans that of the domain. Air around the disc is considered to be 20 °C, and open boundaries with zero relative pressure were used for the upper, lower and radial ends of the domain. Material data were taken from Ansys material data library for air at 20 °C.Reference pressure was set to be 1 atm, turbulence intensity low and turbulent model used was k-ε. (Figure 4.)

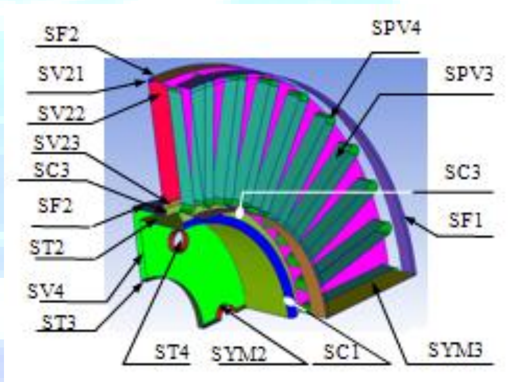


Figure 2. Definition of surfaces of the ventilated disc

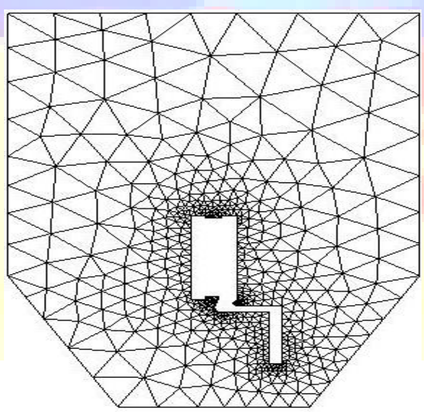


Figure 3. Irregular mesh in the wall

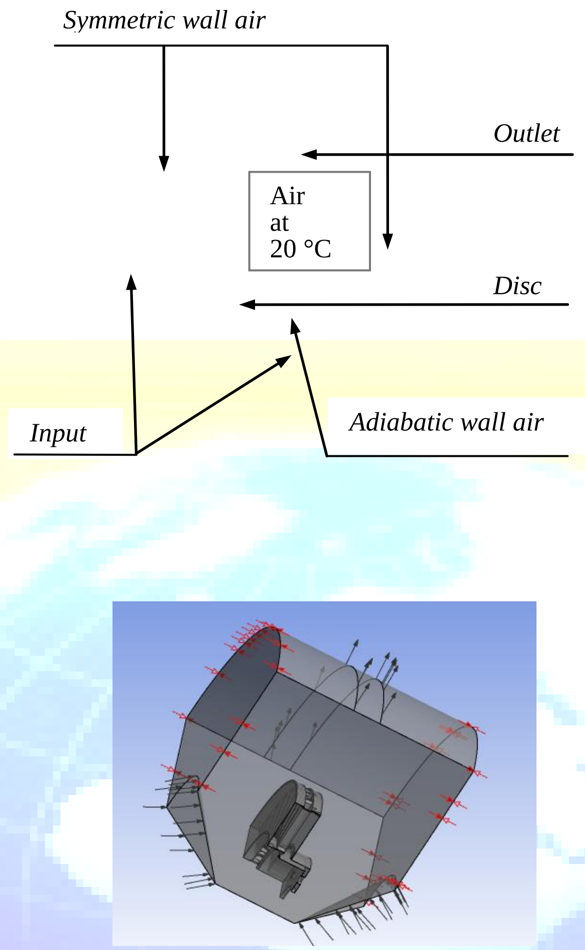


Figure 4. Brake disc CFD model

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 east indicates on Figure 5.

a) Results of the calculation of the coefficient h

The comparison between Figures.6 and 7 concerning the variation of heat transfer coefficient in the non stationary mode for the two types of design full and ventilated, one notes that the introduction of the system of ventilation influences directly the value of this coefficient for same surface what is logically significant because this mode of ventilation intervenes in the reduction in the difference in temperature wall-fluid.

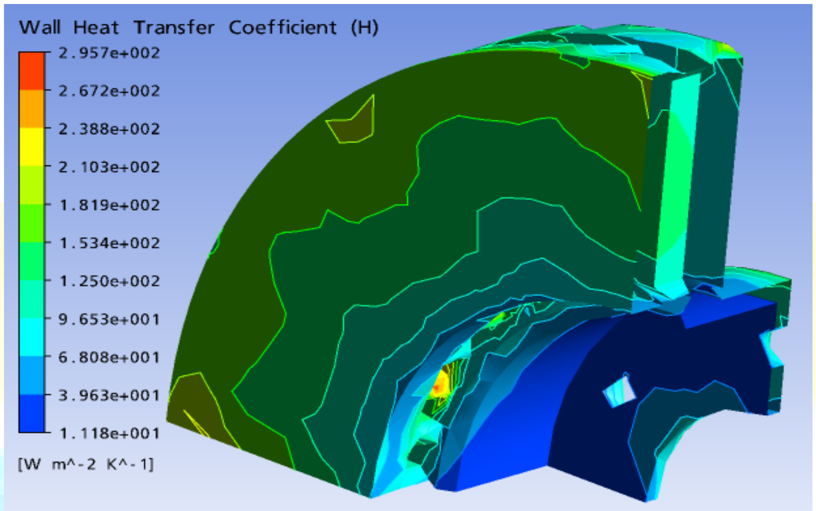


Figure 5. Distribution of heat transfer coefficient on a ventilated disc in the stationary case (FG 15)

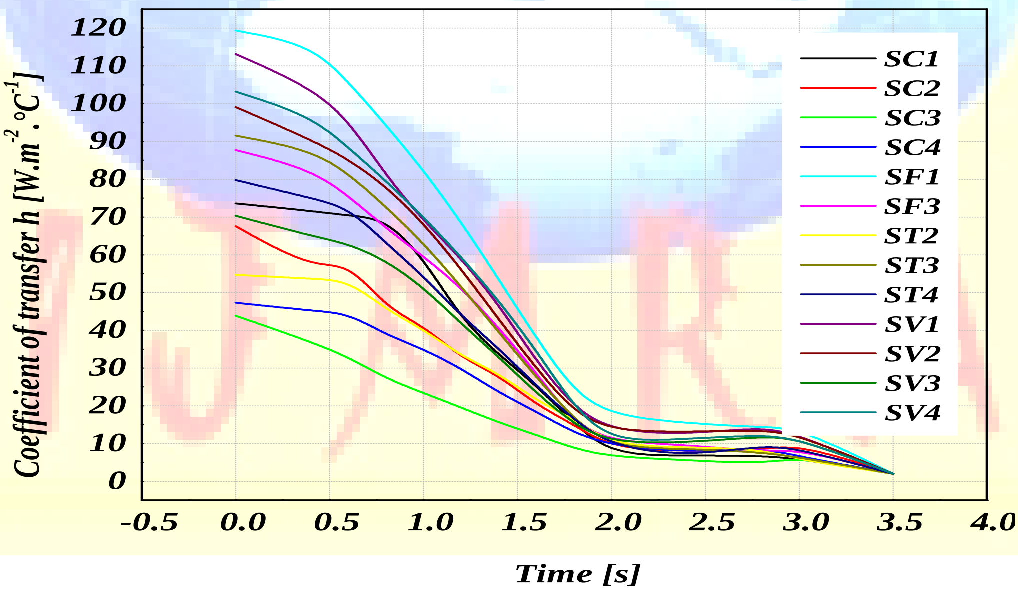


Figure 6. Variation of heat transfer coefficient (h) of various surfaces for a full disc in the non stationary case (FG 15)

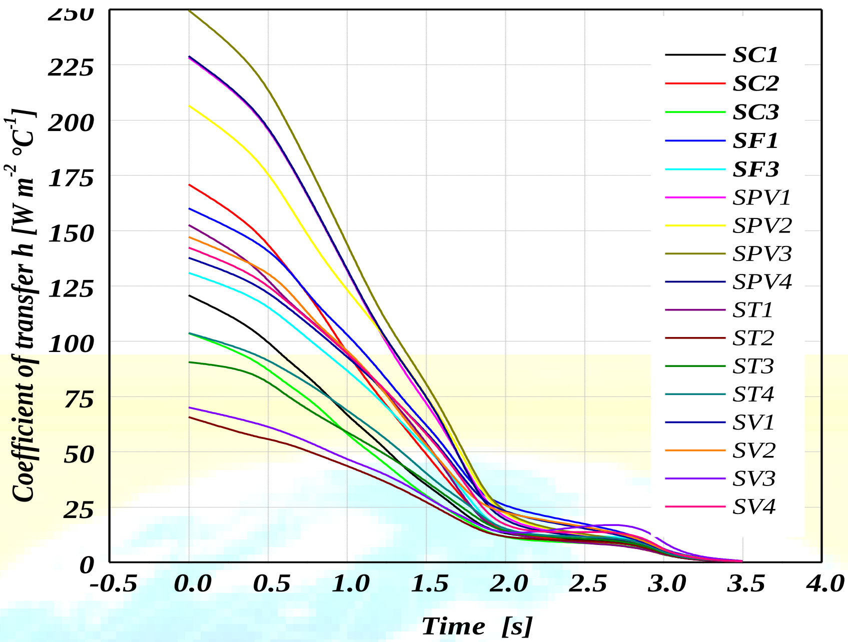
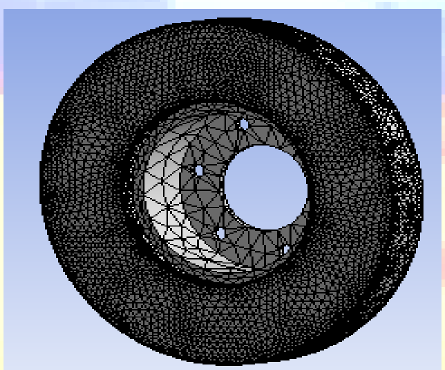


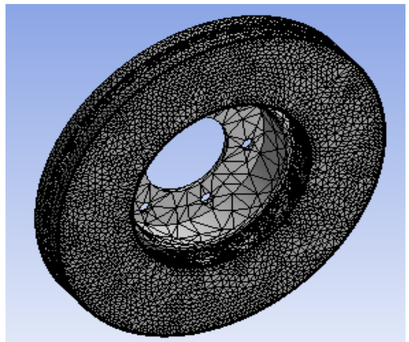
Figure 7. Variation of heat transfer coefficient (h) of various surfaces for a ventilated disc in transient case (FG 15)

4. Meshing of the Disc

The elements used for the mesh of the full and ventilated disc are tetrahedral three-dimensional elements with 10 nodes (isoparamitric) (Figure 8.).



(a) full disc(172103 nodes -114421 elements)



(b) ventilated disc(154679 nodes- 94117 elements)

Figure 8. Meshing of the disc

5. Initial and Boundary Conditions

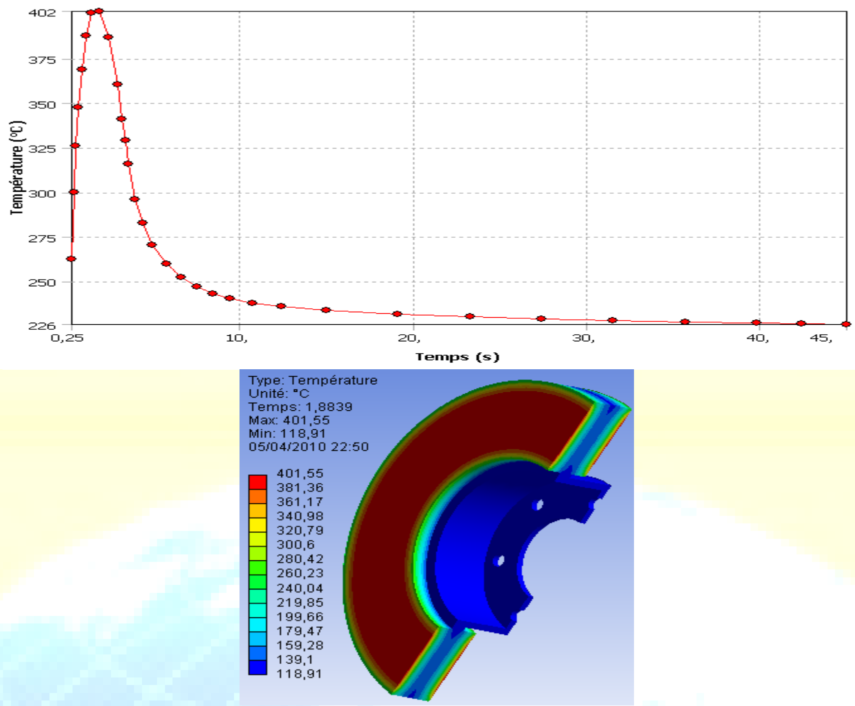
The boundary conditions are introduced into module ANSYS Workbench[Multiphysics], by choosing the mode of first simulation of the all (permanent or transitory), and by defining the physical properties of materials.These conditions constitute the initial conditions of our simulation.After having fixed these parameters, one introduces a boundary condition associated with each surface

- 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]
- Materials: Grey Cast iron FG 15.
- Convection:One introduces the values of the heat transfer coefficient (h) obtained for each surface in the shape of a curve (Figures. 6, 7),
- Flux:One introduces the values obtained by flux entering by code CFX.

6. Results and discussions

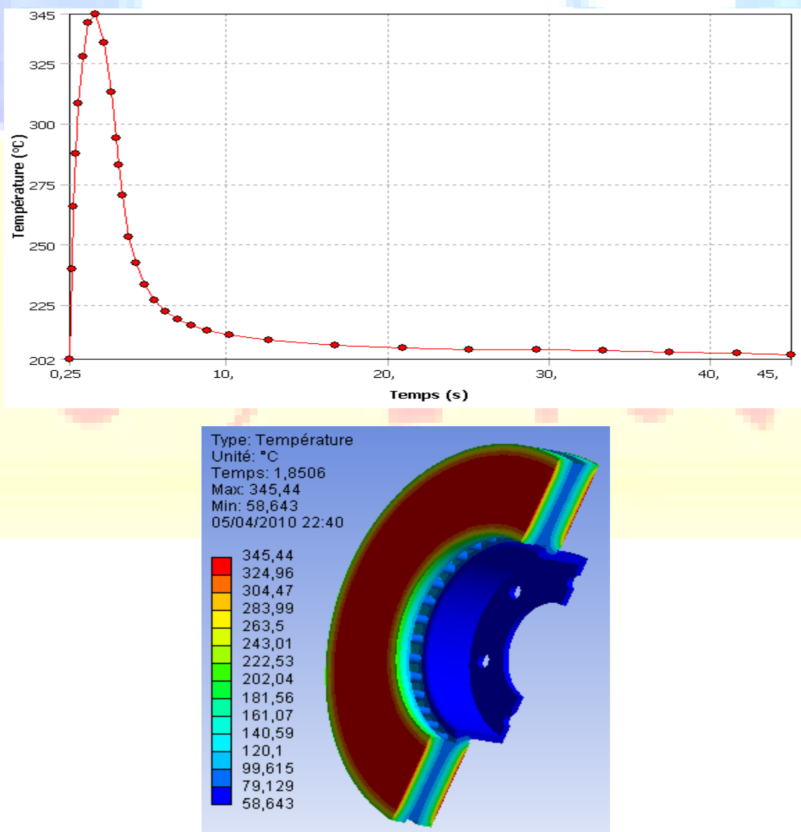
6.1. Influence of Construction of the Disc

Figures.9-10 shows the variation in the temperature according to time during the simulation.From the first step, the variation in the temperature shows a great growth which is due to the speed of the physical course of the phenomenon during braking, namely friction, plastic microdistortion of contact surfaces …



t= 1.8839 s , T_{max}=401.66 °C

Figure 9. Temperature distribution of a full disc of cast iron (FG 15)



t=1.8506 s t, T_{max}=346.44 °C

Figure 10. Temperature distribution of a ventilated disc of cast iron (FG 15)

For the full disc ,the temperature reaches its maximum value of 401,55 °C at the moment t = 1,8839 s, then it falls quickly until to 4,9293 s, as from this moment and until the end t = 45 s) of simulation the variation in the temperature become slow. It is noted that the interval[0-3,5] s represents the phase of forced convection. From the latter, one is in the case of the free convection until the end of the simulation. In the case ventilated disc one observes that the temperature of the disc falls approximately 60 °C compared to the first case. It is noted that ventilation in the design of the discs of brake gives a better system of cooling.

7. Coupled Thermo-Mechanical Analysis

7.1. FE Model and Boundary Conditions

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 autobahn driving conditions[19].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[20]. In a real car disc brake system, the brake pad surface is not smooth at all. Abu Bakar and Ouyang[21] adjusted the surface profiles using measured data of the surface height and produced a more realistic model for brake pads. Figure 11 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 at 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 are used (Figure 12).

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.

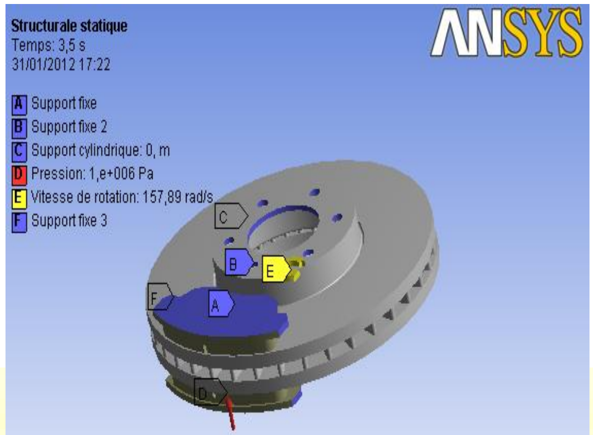


Figure 11. Boundary conditions and loading imposed on the disc-pads

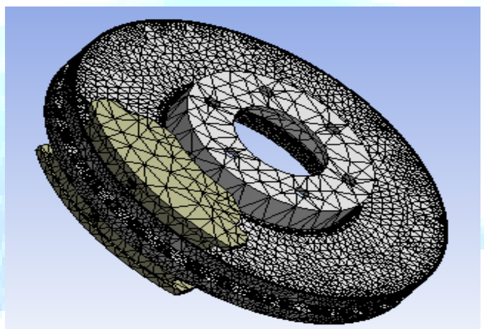


Figure 12. Refined mesh of the model

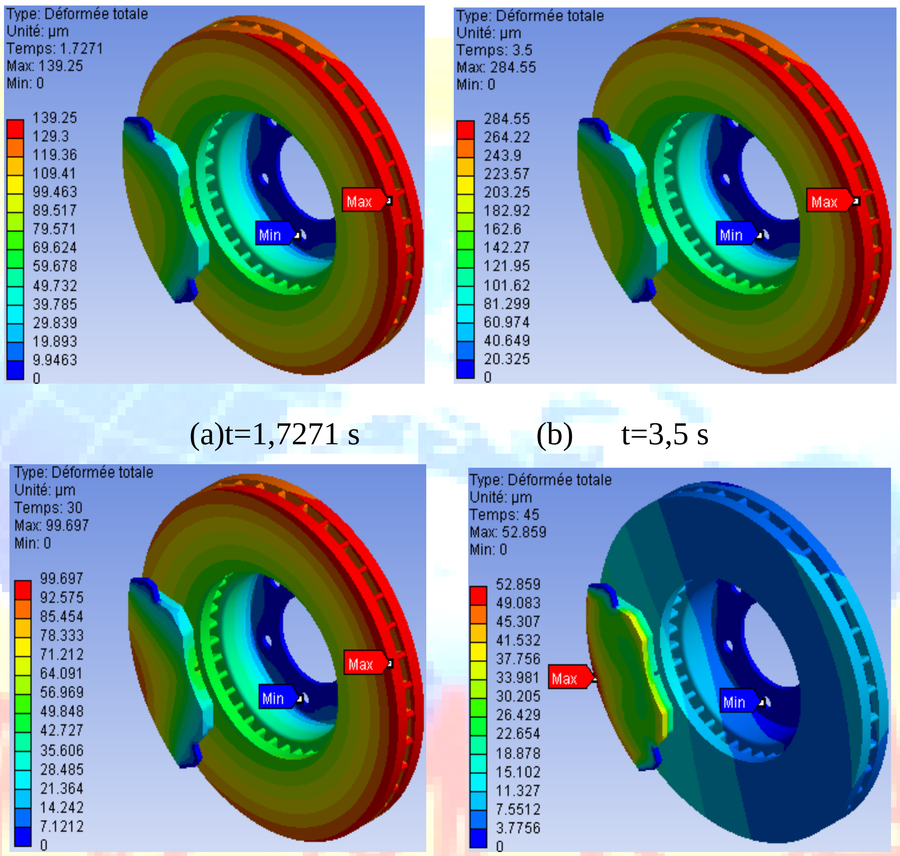
7.2. Thermal Deformation

Figure 13. 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 with 284,55 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 crown external and the cooling fins of the disc. Indeed, during a braking, the maximum temperature depends almost entirely on the storage capacity of heat of disc (on particular tracks of friction) this deformation will generate a dissymmetry of the disc following the rise of temperature what will cause a deformation in the shape of an umbrella.

7.3. Von Mises Stress Distribution

Figure 14. presents the distribution of the constraint equivalent of Von Mises to various moments of simulation, the scale of values varies from 0 MPa to 495,56 MPa.The maximum value recorded during this simulation of the thermomechanical coupling is very significant that that

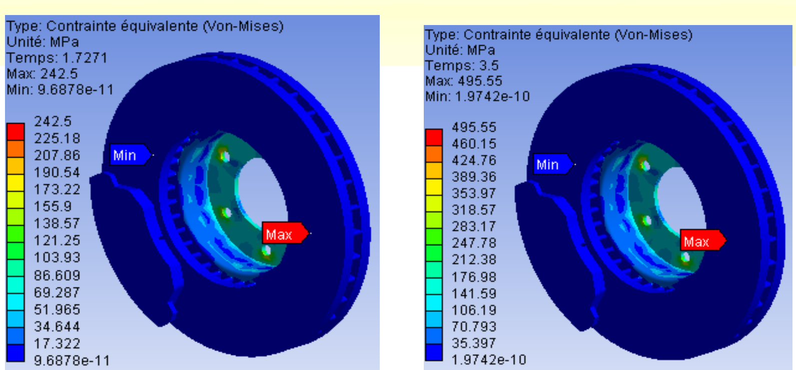
obtained with the assistance in the mechanical analysis dryness under the same conditions. One observes a strong constraint on the level of the bowl of the disc.Indeed, the disc is fixed to the hub of the wheel by screws preventing its movement.And in the present of the rotation of the disc and the requests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



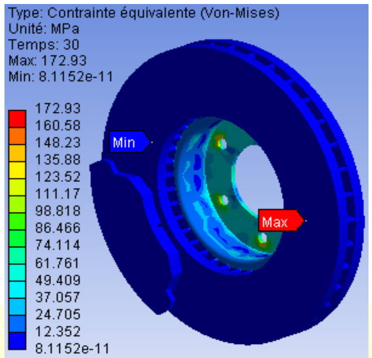
(a)t=1,7271 s (b) t=3,5 s

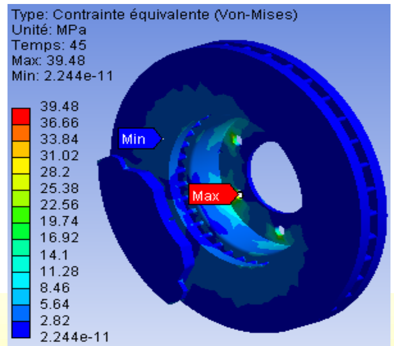
(c) t=30 s (d) t=45 s

Figure 13. Total distortion distribution



(a) t=1,7271 s (b) t=3,5 s





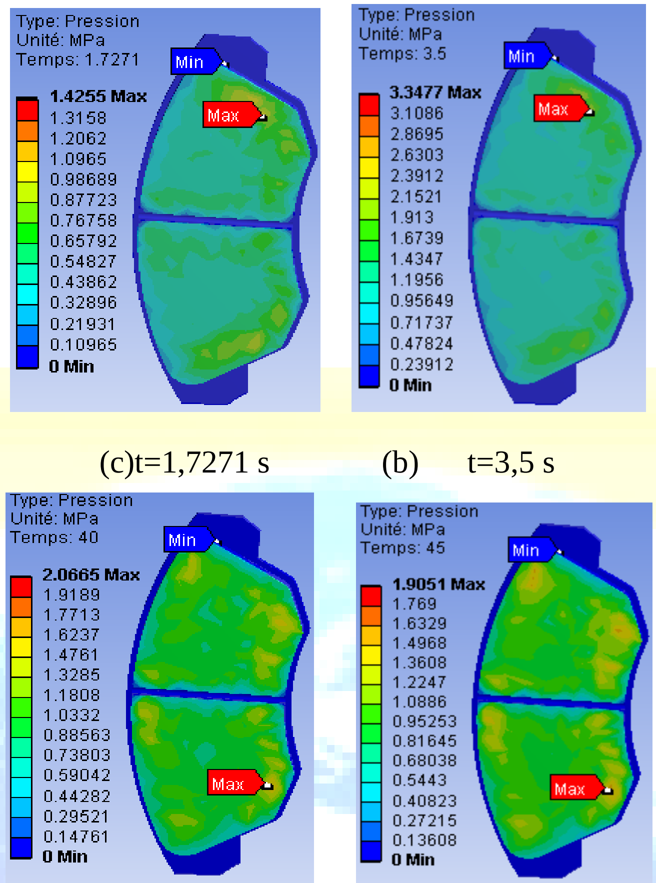
(c) t=30 s (d) t=45 s

Figure 14. Von Mises stress distribution

7.4. 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 in 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[22]. Brake squeal analysis in recent years always includes a static contact analysis as the first part of the complex eigenvalue analysis[23,24].

Figure 15. 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 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 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 carries out to lead to the not-axisymmetric field of the temperature. This last affects thermal dilation and leads to a variation of the contact pressure distribution .



(c)t=1,7271 s (b) t=3,5 s

(c) t=40 s (d) t=45 s

Figure 15. Contact pressure distribution in the inner pad

8. Conclusions

In this publication, we presented the analysis of the thermomechanical behavior of the dry contact between the brake disc and pads during the braking process; 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 a good 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 increases 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. Regarding the calculation results, we can say that they are satisfactory commonly 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 a good agreement between the model and reality.

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