Simulation of fully coupled thermomechanical analysis of disc brake rotor

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Abstract:- Vehicle braking system is considered as 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 (DTV). In the past, surface roughness and wear at the pad interface have rarely been considered in studies of thermal analysis of a disc brake assembly using finite element method.. The ventilated pad-disc brake assembly is built by a 3D model with a thermo-mechanical coupling boundary condition and multi-body model technique. The numerical simulation for the coupled transient thermal field and stress field is carried out by sequentially thermal-structural coupled method based on ANSYS to evaluate the stress fields and of deformations which are established in the disc had with the pressure of the pads and in the conditions of tightening of the disc thus the contact pressures distributions field in the pads which is another significant aspect in this research .The results obtained by the simulation are satisfactory compared with those of the specialized literature.

Key –words:- Brake Discs, Pads, Heat flux, Heat transfer coefficient, Thermo-mechanical coupling, Von Mises stress, Contact pressure

1 Introduction

The thermal analysis is a primordial stage in the study of the brake systems, because the temperature determines thermomechanical behavior of the structure. In the braking phase, temperatures and thermal gradients are very high. This generates stresses and deformations whose consequences are manifested by the appearance and the accentuation of cracks [1,2]. It is then important to determine with precision the temperature field of the brake disc. During the stop braking, the temperature does not have time to be stabilized in the disc. A transient analysis is required. It is also essential to evaluate the thermal gradients, which requires a three-dimensional modeling of the problem. The thermal loading is represented by a heat flux entering the disc through the brake pads. Many studies about the brake disc thermo-mechanical coupling analysis have been done. Choi and Lee developed an axisymmetric finite element model for the thermoelastic contact problem of brake disk and investigated the thermoelastic instability phenomenon of disc brake during the drag-braking processand repeated braking process [3, 4]. Gao and Lin et al. analyzed the transient temperature field and thermal fatigue fracture of the solid brake disc by a three-dimensional thermal-mechanical coupling model [5,6], In 2007, the authors investigated the temperature field and thermal distortion of the ventilated brake disc by axisymmetric model and partial 3D model [7]. In 2008, the authors identified the temperature field of ventilated brake disc in the repeated braking based on the thermo-mechanical coupling and multi-body model [8].

In this study, we will present a numerical modeling in three dimensions to analyze the thermo mechanical behavior of the full and ventilated disc brake .The strategy calculation based on the finite element method will be carried out using code ANSYS 11.

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 rises in temperature ranging from 300 ° C to 800° C. The heat quantity in the contact area is the result of plastic micro-deformations generated by the friction forces.

Generally, the thermal conductivity of material of the brake pads is smaller than of 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 of this surface is equal to the power friction. The initial heat flux q_0 entering the disc is calculated by the following formula [9]:

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

$z = a/g$: 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 surface of the disc.

m : Mass of the vehicle [kg]

Fig.1 shows the ventilated disc – pads and the applied forces. 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. The disc material is gray cast iron with high carbon content FG, 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.

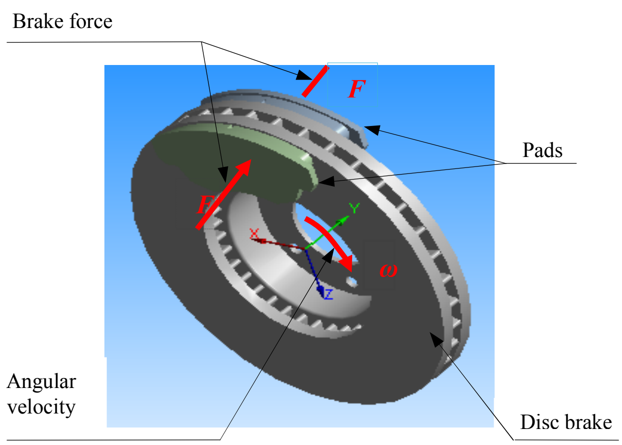


Fig.1. Disc-pads assembly with forces applied to the disc.

Table 1 Geometrical Dimensions and application parameters of automotive braking.

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,
Rate distribution of the braking forces– ϕ-	20
Factor of charge distribution on the disc ε_p	0.5
Surface disc swept by the pad A_d [mm^2]	3599

Table 2 Thermoelastic properties used in simulation.

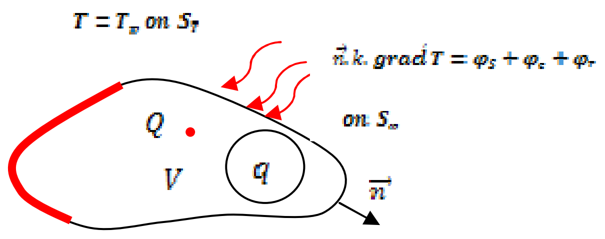
Material Properties	Pad	Disc
Thermal conductivity, k	5	57
Density, ρ (kg/m^3)	1400	7250
Specific heat, c (J/Kg. °C)	1000	460
Poisson.s ratio,, ν	0,25	0,28
Thermal expansion, α (10^{-6} / °C)	10	10,85
Elastic modulus,, E (GPa)	1	138
Coefficient of friction, μ	0,2	0,2
Operation Conditions		
Angular velocity, ω (rd/s)		157.8
Hydraulic pressure,, P (MPa)		1

3 Numerical modeling of the thermal problem

3.1 Finite Element Method

The finite element method is used in many applications to solve partial differential equations [10].It leads to a simple approximation of the unknown factors of the continuous equations. These last will be then to transform into a system of equations of finished dimension, which we can schematically write in the form AU=L, where U is the vector of the unknowns, A is a matrix and L is a vector.

3.2 Form Differential



$S = S_T \cup S_\varphi \;,\; S_T \cap S_\varphi = \emptyset$

Fig.2. Thermal loads applied to a continuous medium.

The system shown in figure 2 is subjected to the following thermal loads:
- A specific heat source Q in [W]
- A voluminal heat source q in [W/m^3]
- Temperature imposed (or prescribed) T_P on a surface S_T
- Flux density φ_S imposed on *a* S_φ surface in [W/m^2]
- Heat transfer by convection φ_c on *a* surface Sφ
- Heat transfer by radiation φ_r on *a* surface *Sφ*

The solution of a thermal problem is to find the temperature field $T(x, y, z, t)$ at any point of the solid so that [11].

$$C_p \dot{T} - div\left(-k.\overrightarrow{grad}T\right) - q = 0 \qquad (2)$$

- With the boundary conditions

$$T = T_p \text{ on } S_T$$
$$\vec{n}.\left(-k.\overrightarrow{grad}\,T\right) = \varphi_S + \underbrace{h(T_f - T)}_{convection} + \underbrace{\varepsilon\sigma\,(T_\infty^4 - T^4)}_{radiation} \text{ on } S_\varphi$$
$$S = S_T \cup S_\varphi \;,\; S_T \cap S_\varphi = \phi \qquad (3)$$

- The initial condition at time $t = t_0$:

$$T(x, y, z, t_0) = T_0(x, y, z) \qquad (4)$$

Where
ρ : Density of material (kg/m^3)
C_p : Mass heat capacity (J/ (kg .K))
$\vec{n}$: Unit normal with *s* directed towards the outside of *v*

This system of equations is written in weak formulation as follows [12], [13], [14]:

$$\int_V T^* \rho C_p \, dV + \int_V \overrightarrow{grad}\, T^*.\left(k.\overrightarrow{grad}\, T\right) dV - \int T^*\left(\varphi_S + h(T_f - T) + \varepsilon\sigma\,(T_\infty^4 - T^4)\right) - \int_V T^* q \, dV = 0 \;\forall\; T^* \qquad (5)$$

T^* is weight function (or function test).
With the initial and the following boundary conditions:

$$T(x, y, z, t_0) = T_0(x, y, z) \text{ and } T = T_p \text{ on} \qquad (6)$$

The temperatures field $T(x, y, z, t)$ has for expression on the whole domain V [11]:

$$T\;(x, y, z, t) = \left[N_1\,(x, y, z) \cdots N_i\,(x, y, z) \cdots N_n\,(x, y, z)\right] \begin{Bmatrix} T_1\,(t) \\ \vdots \\ T_i\,(t) \\ \vdots \\ T_n\,(t) \end{Bmatrix} = [N]\{T\} \qquad (7)$$

$[N(x, y, z)]$: The matrix of interpolation.
$\{T(t)\}$: Vector of the nodal temperatures.
By carrying the following relations in the equation (5):

$$T = [N]\{T\} \qquad (8)$$
$$T' = [N]\{\dot{T}\} \qquad (9)$$

$$\{gradT\} = [B]\{T\} \text{avec} [B] = \left[\{B_1\} \cdots \{B_i\} \cdots \{B_n\}\right] \qquad (10)$$

$$T' = [N]\{\dot{T}\} = \{T^*\}^T [N]^T, \{grad\; T^*\} = [B]\{T^*\},$$

$$\{grad\; T^*\}^T = \{T^*\}^T [B]^T \qquad (11)$$

We obtains

$$\{T^*\}^T\left([C]\{\dot{T}\} + [K]\{T\} - \{F\}\right) = 0 \qquad (12)$$

Where

$$[C] = \int_V \rho C_p [N]^T [N] \; dV \qquad (13)$$

$$[K] = \int_V [B]^T [\lambda][B] \; dV + \int_{S_\varphi} h\,[N]^T [N] \; dS \qquad (14)$$

$$\{F\} = \int_V [N]^T q \, dV + \int_{S_\varphi} [N]^T \left(S_\varphi + hT_f + \varepsilon\sigma(T_\infty^4 - T^4)\right) dS \qquad (15)$$

$[C]$: Thermal capacity matrix (J/K).
$[K]$: Thermal conductivity matrix (W/K).
$\{F\}$: Nodal flux vector (W).
$\{T\}$: Nodal temperatures vector (K).

3.3 Initial Conditions

We assume that the initial temperature of the disc is constant.

$$T(x,y,z,t) = 60\,°C \quad at\ time\ \ t = 0 \tag{16}$$

3.4 Boundary conditions

This is a transient thermal problem with two boundary conditions:

- A heat flux entering the disc localized in the contact zone disc-pad in both sides.
- A heat transfer by convection on all the free surfaces of the disc of which the exchange coefficient h depends on time because rotational speed of the disc varies with time.

The heat exchange coefficient h on each disc surface was calculated and imported using ANSYS CFX module.

4 Presentation of the computing code Ansys

The modules used for this study are:

- ANSYS Workbench: This platform offers a different approach in the construction of model using the original computer code ANSYS [15].It is particularly adapted to handling cases with complex geometry and to the unconfirmed users. In this environment, the user works on geometry and not on the model itself. Before starting the solution, the platform will convert the data introduced by the user into code ANSYS. The generated finite element model is handled by inserting specific commands of ANSYS code.
- ANSYS ICEM CFD: It is mesh generation software for applications in fluid mechanics and mechanical structures.
- ANSYS CFX: This software is designed to perform simulations in fluid mechanics.
- ANSYS Metaphysics: This product contains all modules of ANSYS simulation code.

Fig.3 shows the stages of simulation with ANSYS CFX in Workbench.

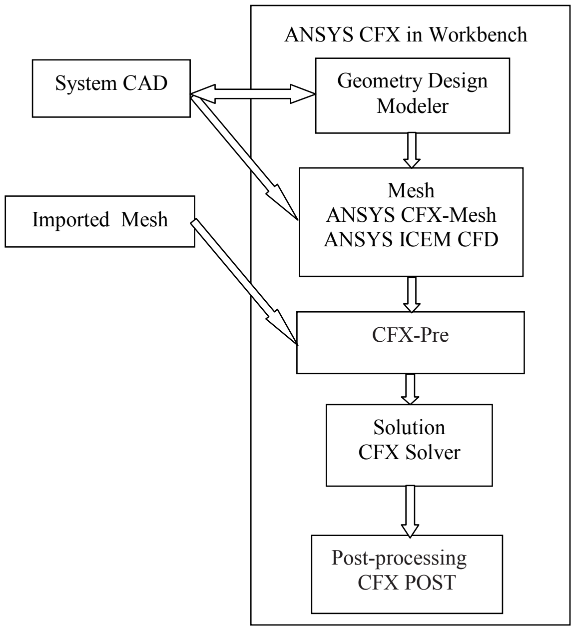


Fig.3. Simulation steps with CFX [15].

5 Determination of the coefficient of exchange by convection (h)

5.1 Introduction

The thermal analysis of the braking system requires a precise determination of the quantity of heat friction produced and as well as the distribution of this energy between the disc and the brake lining. During an emergency braking, all the heat produced with the interface is equal to the heat absorbed by the disc and the brake lining.

At the time of braking process, a part of the frictional heat escapes in the air ambient by convection and radiation. Consequently, the determination of the heat transfer coefficients is essential. Their exact calculation is however rather difficult, because these coefficients depend on the location and the construction of the braking system, the speed of the vehicle travel and consequently of the air circulation. Since the process of heat transfer by radiation is not too important, we will determine using code ANSYS CFX only the convection coefficient (h) of the disc**.** This parameter will be exploited to determine the three-dimensional distribution of the temperature of the disc.

5.2 Modeling in ANSYS CFX

The first stage is to create the model CFD 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 that one will export the results towards CFX using the command "Output to cfx ".After obtaining the model on CFX Pre and specified the boundary conditions, we must define these physical values come into play on CFX to start calculation.

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 [16]. An unsteady-state analysis is necessary.

Fig.4 shows the elaborate model CFD which will be used in ANSYS CFX Pre.

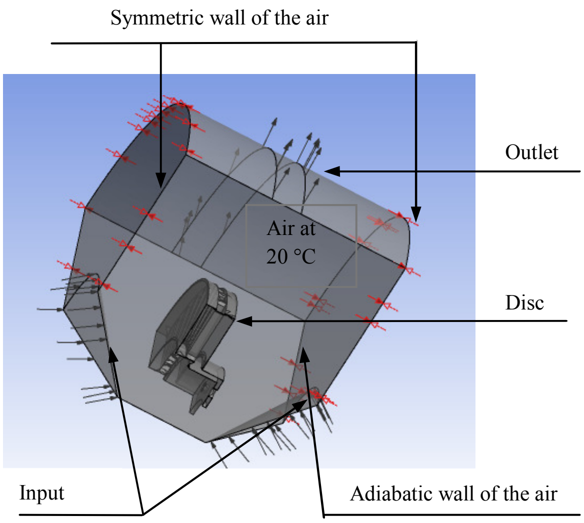


Fig.4. Brake disc CFD model.

a) Physical model

In this step, one declares all of the physical characteristics of the fluid and the solid. After the meshing are defined all the parameters of the different models to be able to start the analysis.

b) Definition of the domains

Initially, one valide the elaborated models and one activate in the option "Thermal Energy " the calculation of heat transfer "Heat Transfer ".

Fluid domain: Speed entry: $V_{ent\ non.st} = V_{ent} - Va{\cdot}t$

Disc domain: Entering flux: $FLUX_{non.st} = (CF)\ (V_{ent\ non.st})$,

CF = 149893,838.

$V_{ent\ non.st} = V_{ent} - Va{\cdot}t$

$FLOW_{non.st}$: Non stationary flux entering.

$V_{ent\ non.st}$: Non stationary speed entering of the air.

c) Definition of materials

We introduce into the computer code the physical properties of used materials. In this study we selected three cast iron materials (FG 25 Al, FG 20 and FG 15).

d) Definition of the boundary conditions

The first step is to select the Inlet and Outlet faces of the heat flux. These options are found in the insertion menu “Boundary Conditions” in the CFX Pre.

The boundary conditions concerning the pads will be also defined. One selects the options “Wall” and "Symmetry ", because there will be the possibility of adjusting a certain number of parameters in the boundary conditions such as flux entering the disc.

e) Application of the interfaces domains

The areas of interfaces are commonly used to create the connection or linkage areas. Surfaces located between the interactions regions (air-disk) are reported as solid-fluid interface.

f) Temporary Condition

Since in this study is to determine the temperature field in a disc brake during the braking phase of a vehicle of average class, we take the following temporal conditions:

- 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.

g) Launch of the calculation

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 average values calculated by the minimal and maximum values obtained using ANSYS CFX POST as it east indicates on the figure 5.

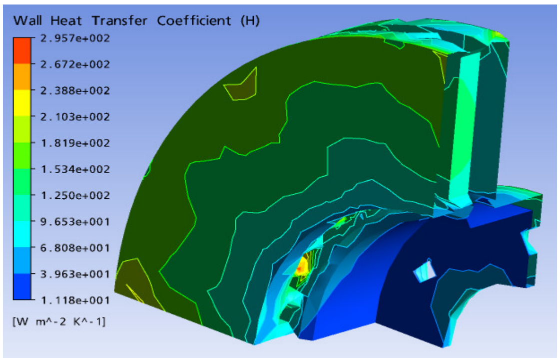


Fig.5.Distribution of heat transfer coefficient on a ventilated disc in the stationary case (FG 15).

h) Results of the calculation of the coefficient h

Figs.6 and 7 show the variation of the heat transfer coefficient (h) of different surfaces respectively for a full and ventilated disc in cast iron (FG 15) in transient state. We found that after a short time all the curves of *h* are decreasing with time.

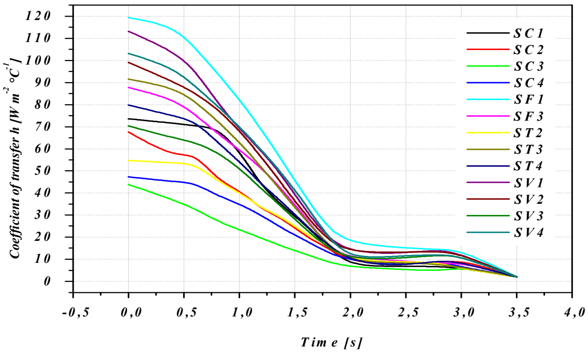


Fig.6. Variation of heat transfer coefficient (h) of various surfaces for a full disc in the non stationary case (FG 15).

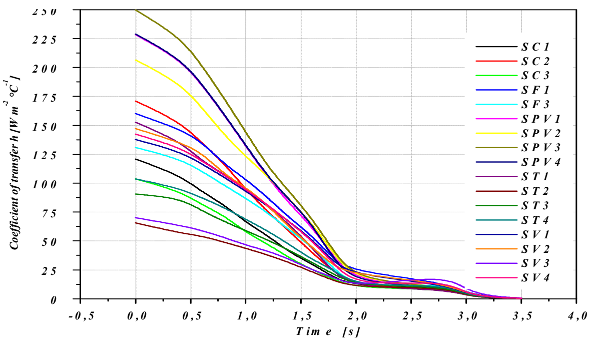


Fig.7. Variation of heat transfer coefficient (h) of various surfaces for a ventilated disc intransient case (FG 15).

6 Determination of the disc temperature

The modeling of the disc temperature is carried out by simulating a stop braking of a middle class car (braking of type 0).

The characteristics of the vehicle and of the disc brake are listed in Table 1.

The vehicle speed decreases linearly with time until the value 0 as shown in figure 8. The variation of the heat flux during the simulation time is represented on Fig.9.

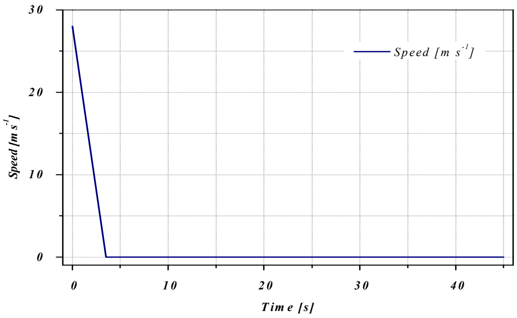


Fig.8. Speed of braking versus time (braking of type 0).

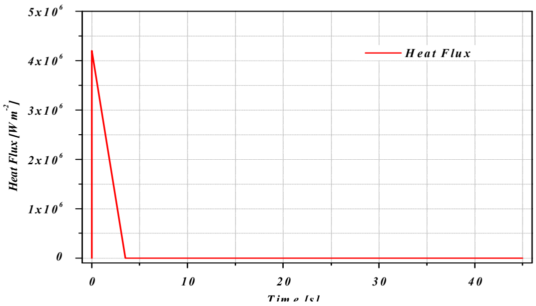


Fig. 9. Heat Flux versus time.

6.1 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.10 and 11).In this simulation, the meshing was refined in the contact zone (disc-pad).This is important because in this zone the temperature varies significantly.

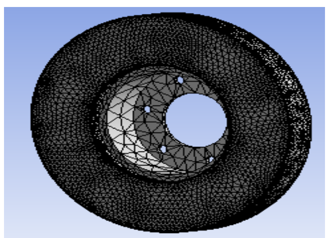


Fig.10. Meshing of a full disc in ANSYS Multiphysics (172103 nodes – 114421 elements).

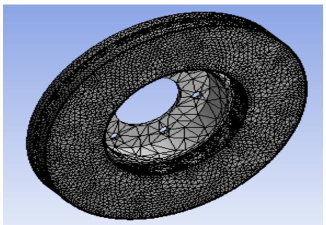


Fig.11. Meshing of a ventilated disc in ANSYS Multiphysics (154679 nodes- 94117 elements).

Three meshes have been tested automatically using an option called convergence in ANSYS Workbench Multiphysics. The number of elements forming each meshing is given in Table 3.

Table 3 Number of elements of the two considered meshs.

	Full disc	Ventilated disc
	Number of	Number of
Mesh 1	46025	77891
Mesh 2	114421	94117
Mesh 3	256613	369777

6.2 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]
- Materials: three types of Cast iron (FG 25 AL, FG 20, FG 15).

7 Results and discussions

The modeling of temperature in the disc brake will be carried out by taking account of the variation of a certain number of parameters 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 by the relation (1).

7.1 Influence of construction of the disc

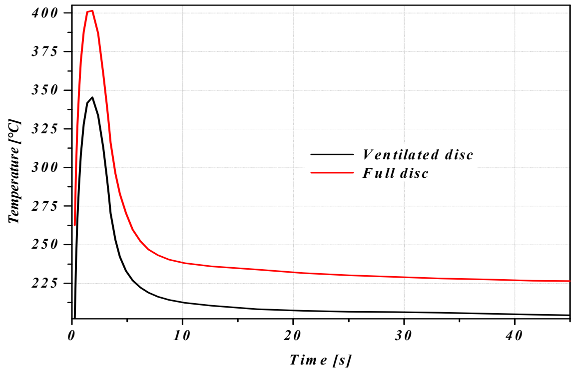


Fig.12. Temperature variation of a full and ventilated disc (FG 15) versus time.

(a)

(b)

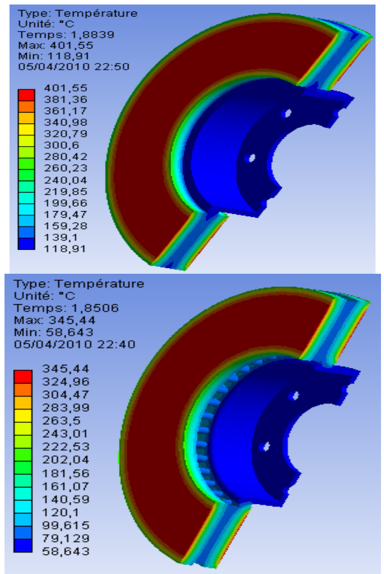


Fig.13.Temperature distribution of a full (a) and ventilated disc (b) of cast iron (FG 15).

Fig.12 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. For the two types of discs, one notices that starting from the first step of time one has a fast rise of the temperature of the disc followed by a fall of temperature after a certain time of braking.

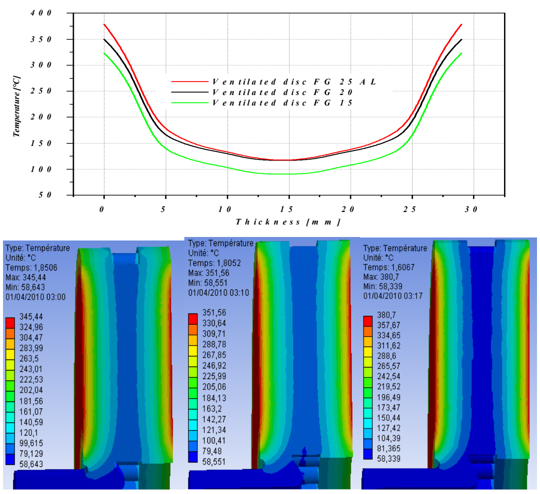
We quickly notices that for a ventilated disc out of cast iron FG15 ,the temperature increases until T_{max} = 345,44 °C at the moment t = 1,85 s, then it decreases rapidly in the course of time. The variation in temperature between a full and ventilated disc having same material is about 60 °C at the moment t = 1,8839 s. We can conclude that the geometric design of the disc is an essential factor in the improvement of the cooling process of the discs.

7.2 Influence of material of the disc

Fig.14 (a) shows for each type of the selected cast iron the temperature variation as a function of thickness at the moment when the temperature is maximum. The allure of the three curves is similar. We also note that the temperature decreases in the direction of median plane of the disc to reach its minimal value. In Fig.14(b) we see that there is inside the disc symmetry of colors. The part far away from the surface of blue contact color is not requested too much thermically. More the thermal conductivity of the material is low, more its temperature is high. The FG 15 is differentiated from the two other cast iron by smaller temperatures.

On Fig.15 the temperature variation versus radius for three materials (FG 25 Al, FG 20, FG 15) is presented. The shape of the temperature curves are the same one. The maximal temperature is in area of the mean disc radius.

According to Figs.14 and 15 the cast iron FG 15 has the best thermal behavior.



FG 15 FG 20 FG25 AL

Fig.14. Temperature variation through a disc thickness for three types of cast irons (FG 25 AL, FG 20 and FG 15).

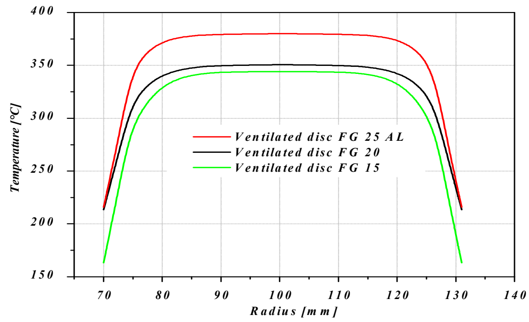
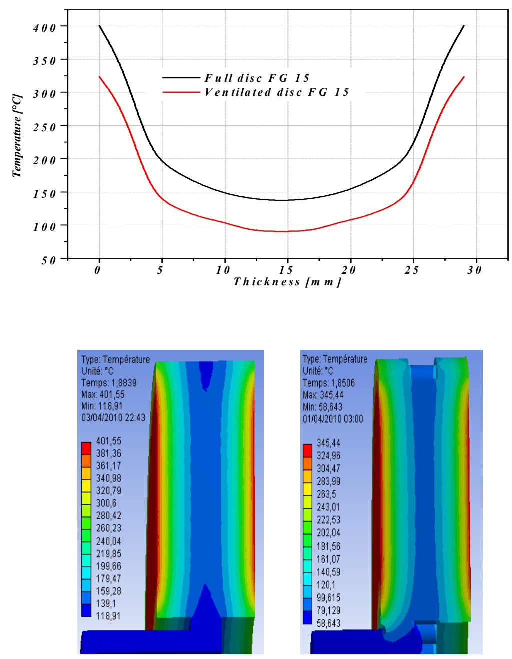


Fig.15. Temperature variation through a radius for three types of cast irons (FG 25 AL, FG 20 and FG 15).

Figs.16 and 17 respectively show the temperature variation according to the thickness and radius. It is noted that there is an appreciable variation of temperature between the two types of full and ventilated disc.

The influence of ventilation on the temperature field appears clearly at the end of the braking (t = 3,5 s).

Among the parameters having an influence on the thermal behavior of the discs brake there is the braking mode which depends on the driver and the circulation conditions. Certain modes of braking can involve the destruction of the disc and consequently to cause serious accidents of circulation. A braking mode is represented in the form of braking cycles, which describe the variation of vehicle speed versus time (v = f(t)).



Full disc Ventilated disc

Fig.16. Temperature variation through the thickness for both designs with same material (FG15).

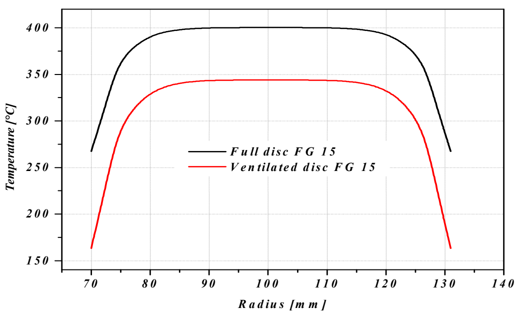


Fig. 17. Temperature variation through a radius or both designs with the same material (FG15).

These cycles may consist of a series of emergency brakings or cycles comprising of the braking phases followed by a downtime.

7.3 Influence of braking mode

The disc brake and the wheel are dimensioned according to the performance and economic requirements of the vehicle. They must support mechanical and thermal loads increasingly greater at mean velocities in permanent progression.

7.3.1 Repeated braking

During vehicle operating, the braking system is subjected to repeated actions of the driver. In this study, we considered two types of braking of which the total simulation time is estimated to be equal to 135 s.

Fig.18 shows a driving cycle of fourteen successive brakings, in the form of sawtooth.

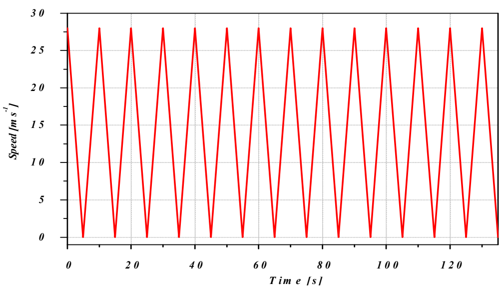


Fig. 18. Driving cycle with fourteen repeated braking (Mode 1).

Fig.19 shows another mode of braking where after each phase of braking one has an idle .

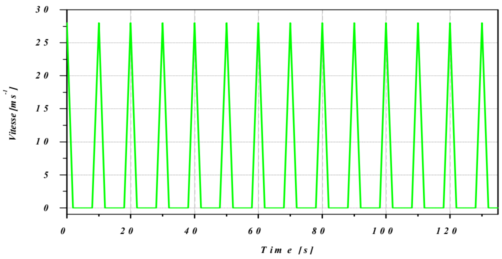


Fig.19. Cycle braking with phase of idles after each braking (mode 2).

Fig.20 shows the comparison of the change of temperature of the disc for a cyclic braking process between the first mode and the second mode. For two contours, we note that the temperatures in the disc rise firmly with each application of brake, then begin the exponential decline. The more the number of repetitions of braking increases, the more the maximum temperatures increase. The initial state of the disc changes after each cycle, the downtimes allow only one partial cooling. After each cooling phase, the disc begins to warm again. In fact, during successive brakings the capacity of cooling of the disc is insufficient to lower the surface temperature to near the initial temperature, which causes an accumulation of energy and therefore a higher surface temperature. These results show that the transient thermal behavior of a disc brake depends on the braking cycle imposed and it is dominating because it dictates the cooling time of the disc. According to Fig.20, we note that in the case of braking cycle mode 2, a reduction of the temperature of approximately 535°C is 45,19% compared to the first cycle. We conclude that the braking mode with a cooling phase influences very positively on the heat transfers in the disc what involves a reduction in the maximum temperature of interface which causes cracking and mechanical wear. In addition this tendency will enable us to ensure safety and fatigue life of the brake system component. Finally it would be interesting to carry out this calculation on brake test benches in order to validate these results of the numerical simulation.

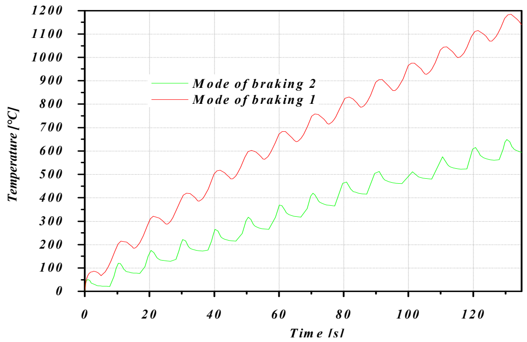
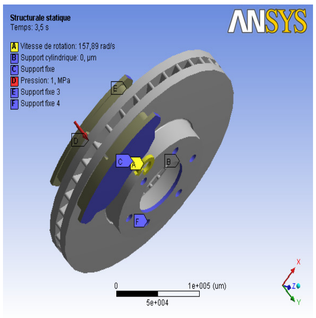


Fig.20 . Temperature variation of the two braking modes versus time

8 Coupled thermo-mechanical analysis

8.1 FE model and boudary conditions

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 [17]. Numerical simulations using the ANSYS finite element software package were performed in this study for a simplified version of a disc brake system which consists of the two main components contributing to squeal the disc and the pads. Various boundary conditions in embedded configurations imposed on the model (disc-pad), taking into account its environment direct, are respectively the simple case as shown in figure 21.The initial temperature of the disc and the pads is 20 °C and the surface convection condition is applied at all surfaces of the disc with the values of coefficient of exchange calculated previously and the convection coefficient (h) of 5 W/m^2.°C is applied at the surface of the two pads. The heat flux into the brake disc during braking can be calculated by the formula described in the first part. 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 (Fig.22). The thermal coupling will be carried out by thermal condition at a temperature nonuniform all takes the thermal environment of the model of it, For this reason, the order " thermal condition " will be used to deal with the thermomechanical coupled problem and to manage the transient state as illustrated in Fig.23.



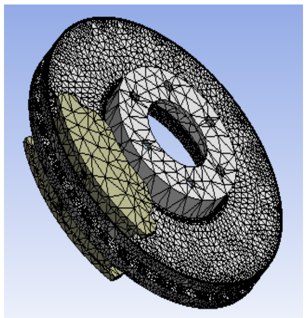


Fig.21Boundary conditions and loading imposed on the disc-pads

Fig.22 Refined mesh of model

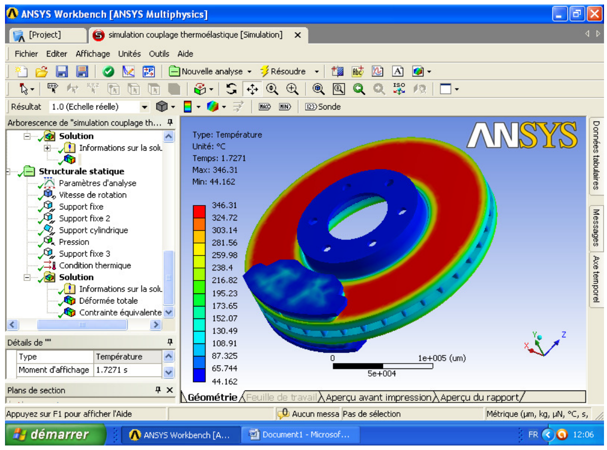
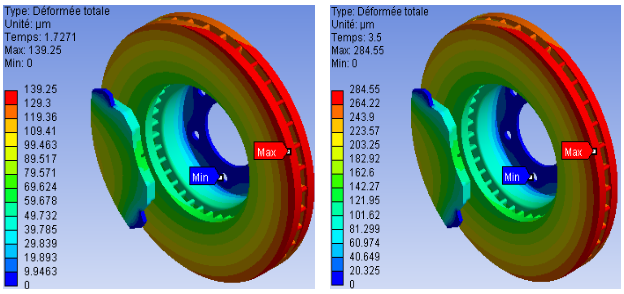


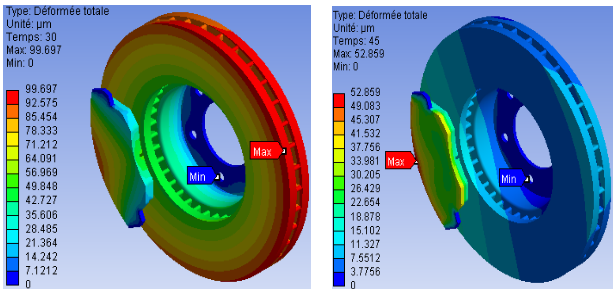
Fig.23 Analysis of the thermomechanical coupling in ANSYS Multiphysics

8.2 Thermal deformation

Fig.24 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 fin of the disc. Indeed, during a braking, the maximum temperature depends almost entirely on the storage capacity of heat of disc (in 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 umbrella.



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

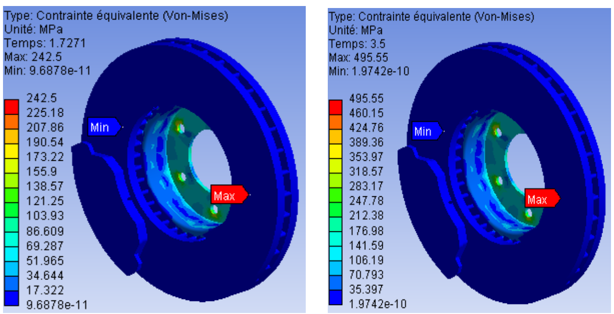


(c) t=30 s   (d) t=45 s

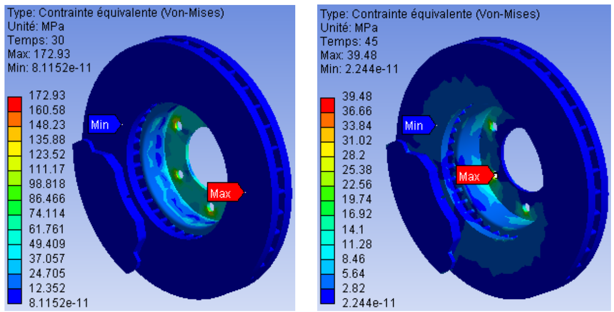
Fig.24 Total distortion distribution

8.3 Von Mises stress distribution

Fig.25 presents the distribution of the constraint equivalent of Von Mises to various moments of simulation, the scale of values varies from 0 MPa with 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 on the hub of the wheel by screws preventing its movement. And into 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 disc.



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



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

Fig.25 Von Mises stress distribution

Fig.26 gives us the comparison for the computation results of displacements; one observes the significant variation of the two graphs. It is noted that the effect of the temperature and the total deformations have a strong influence on the thermomechanical response of the model

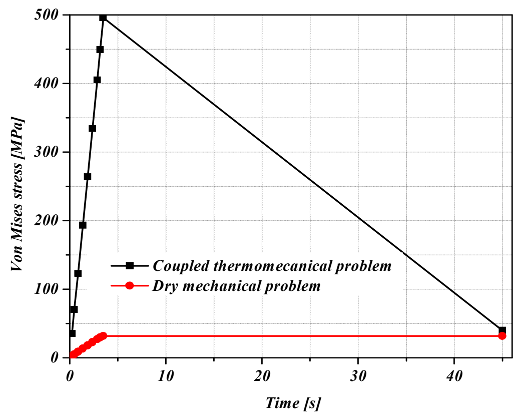
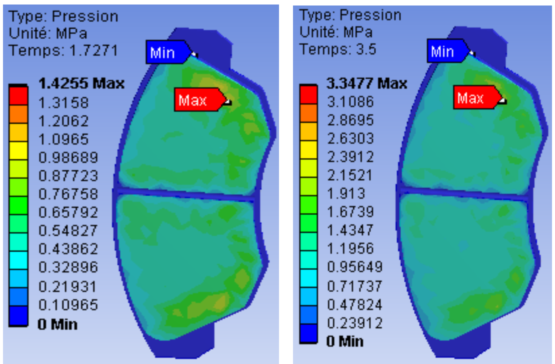


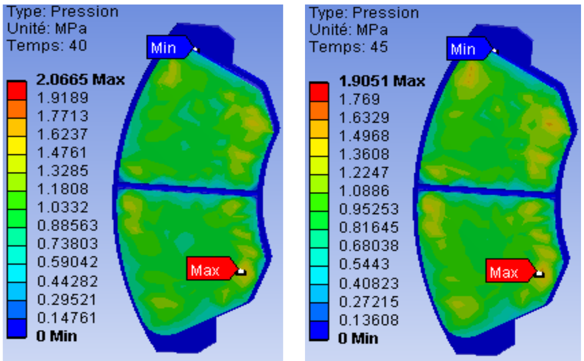
Fig.26 Comparison for the two cases of dealt with problems

8.3 Contact pressure

Fig.27 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 with 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 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 leads to the not-axisymmetric field of the temperature. This last affects thermal dilation and leads to variation of contact pressure distribution .



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



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

Fig.27 Contact pressures distribution in the inner pad

9 Conclusion

In this study, we presented a numerical simulation of the thermal behavior of a full and ventilated disc in transient state. By means the computer code ANSYS 11 we were able to study the thermal behavior of three types of cast iron (AL FG 25, FG 20, FG 15) for a determined braking mode.. In addition to the influence of the ventilation of the disc, we also studied the influence of the braking mode on the thermal behavior of the discs brake. The numerical simulation shows that radial ventilation plays a very significant role in cooling of the disc in the braking phase. The obtained results are very useful for the study of the thermomechanical behavior of the disc brake (stress, defomations, efficiency and wear).

Through the numerical simulation, we could note that the quality of the results concerning the temperature field is influenced by several parameters such as:

- Technological parameters illustrated by the design,
- Numerical parameters represented by the number of element and the step of time.
- Physical parameters expressed by the type of materials.
- Braking mode implemented.

With regard to the results of the coupling, we ended to the following resultants:

- The Von Mises stress and the total deformations of the disc and contact pressures of the brake pads increase in a notable way when the thermal and mechanical aspects are coupled.
- The various interactions between the thermomechanical phenomena generally correspond to damage mechanisms: deformations generate cracking by tiredness, rupture or wear.

An experimental study of the thermomechanical behavior of the brake discs and brake pads is essential to validate the numerical results.

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