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Investigation of temperature and thermal stress in ventilated disc brake based on 3D thermomechanical coupling model

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Abstract The objective of this study is to analyse the thermal behaviour of the full and ventilated brake discs of the vehicles using computing code ANSYS. The modelling of the temperature distribution in the disc brake is used to identify all the factors, and the entering parameters concerned at the time of the braking operation such as the type of braking, the geometric design of the disc, and the used material. 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 and the contact pressure on the pads. The results obtained by the simulation are satisfactory compared with those of the specialised literature.

1. Introduction

The modelling of the problems involved in the phenomena of energy transfer in general and the thermal case in particular is of primary importance, on the one hand, for the phase study or design of a product, and on the other hand, for the follow-up of the product in phase of operation. Parallel to technological progress, the significant theoretical developments were made in the field of the transfers of heat and of mass, and sciences related to thermodynamics in particular, and this discipline has developed for a few decades at intervals raised in many sectors: nuclear power, space, aeronautical, automobile, petrochemistry, etc. [1].

In 2002, Nakatsuji et al. [2] did a study on the initiation of hair-like cracks which formed around small holes in the flange of one-piece disc during overloading conditions. The study showed that thermally induced cyclic stress strongly affects the crack initiation in the brake discs. In order to show the crack initiation mechanism, the temperature distribution at the flange had to be measured. Using the finite element method, the temperature distribution under overloading was analysed. 3D unsteady heat transfer analyses were conducted

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Nomenclature

a	deceleration of the vehicle, ms^{-2}
A_d	disc surface swept by a brake pad, m^2
(C)	thermal capacity matrix, JK^{-1}
C_p	specific heat, $Jkg^{-1}K^{-1}$
E	Young modulus, GPa
g	gravitational acceleration, (9.81 m/s^2)
h	convective heat transfer coefficient, $Wm^{-2}\,K^{-1}$
k	thermal conductivity, $Wm^{-1}\,K^{-1}$
(K)	thermal conductivity matrix, WK^{-1}
$\{L\}$	vector operator
m	mass of the vehicle, kg
$\vec{n}$	unit normal
q_0	heat flux entering the disc, W/m^2
Q	heat quantity generated during the friction, W/m^2
Q^*	heat flux specified on a surface, W/m^2
S_T	surface temperature, m^2
S_Q	surface in heat flux, m^2
S_c	surface in convection, m^2
t	time, s
T	temperature, °C
T^*	temperature specified on a surface, °C
T_f	fluid temperature, °C
T_P	temperature imposed, °C
v	initial speed of the vehicle, ms^{-1}
$\{v\}$	vector speed of mass transport
z	braking effectiveness

Greek symbols

α	thermal expansion coefficient, 1/°C
ε_p	factor load distribution on the disc surface
ν	Poisson coefficient
ρ	mass density, kgm^{-3}
ν	kinematic viscosity, m^2/s
ϕ	rate distribution of the braking forces between the front and rear axle

using ANSYS. A 1/8 of the one-piece disc was divided into finite elements, and the model had a half thickness due to symmetry in the thickness direction. In 2000, Valvano and Lee [3] did a study on the technique to determine the thermal distortion of a brake rotor. The severe thermal distortion of a brake rotor can affect important brake system characteristics such as the system response and brake judder propensity. As such, the accurate prediction of thermal distortions can help in the designing of a brake disc. In 1997, Hudson and Ruhl [4] did a study on the air flow through the passage of a Chrysler LH platform ventilated brake rotor. Modifications to the production rotor's vent inlet geometry are prototyped and measured in addition to the production rotor. Vent passage air flow is compared to existing correlations. With the aid of Chrysler Corporation, investigation of ventilated brake rotor vane air flow is undertaken. The goal was to measure current vane air flow and to improve this vane flow to increase brake disc cooling. Temperature increases can strongly influence the properties of surface of materials in slip, support physicochemical and microstructural transformations and modify the rheology of the interfacial elements present in the contact [5]. Recent numerical models, presented to deal with rolling processes [6,7], have shown that the thermal gradients can attain important levels which depend on the heat dissipated by friction, the rolling speed, and the heat convection coefficient. Other authors [8,9] dealt with the evaluation of temperature in solids subjected to frictional heating. The temperature distribution due to friction process necessitates a good knowledge of the contact parameters. In fact, the interface is always imperfect – due to the roughness – from a mechanical and thermal point of view. Recent theoretical and experimental works [10,11] have been developed to characterise the thermal parameters which govern the heat transfer at the vicinity of a sliding interface. In certain industrial applications, the solids are provided with surface coating. A recent study has been carried out to analyse the effect of surface coating on the thermal behaviour of a solid subjected to friction process [12]. 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 [13,14]. This solution is applied to the problem of determining the transient temperatures reached at the friction surfaces of a disc brake when a constant deceleration is produced during braking [15].

2. Numerical modelling of the thermal problem

2.1. Equation of the problem

The first law of thermodynamics indicating the thermal conservation of energy gives:

$$C_p\left(\frac{\partial T}{\partial t}+\{v\}^T\{L\}T\right)+\{L\}^T\{Q\}=p \tag{1}$$

In our case, there is not an internal source ($p = 0$), and thus, Eq. (1) is written:

$$\rho C_p\left(\frac{\partial T}{\partial t}+\{v\}^T\{L\}T\right)+\{L\}^T\{Q\}=0 \tag{2}$$

With:

$$\{L\}=\begin{Bmatrix}\frac{\partial}{\partial x}\\ \frac{\partial}{\partial y}\\ \frac{\partial}{\partial z}\end{Bmatrix} \tag{3}$$

where $\{L\}$ is vector operator.

$$\{v\}=\begin{Bmatrix}v_x\\ v_y\\ v_z\end{Bmatrix} \tag{4}$$

where $\{v\}$ is vector speed of mass transport.The law of Fourier (2) can be written in the following matrix form:

$$\{Q\}=-[K]\{L\}T \tag{5}$$

With:

$$[K]=\begin{bmatrix}K_{xx} & 0 & 0\\ 0 & K_{yy} & 0\\ 0 & 0 & K_{zz}\end{bmatrix} \tag{6}$$

where [K] is matrix conductivity.

k_{xx}, k_{yy} and k_{zz} Represent the conditions along axes x, y and z, respectively. In our case, the material is isotropic thus $k_{xx} = k_{yy} = k_{zz}$.

By combining two Eqs. (2) and (5), we obtain:

$$\rho C_p\left(\frac{\partial T}{\partial t} + \{v\}^T\{L\}T\right) + \{L\}^T([K]\{L\}T) \tag{7}$$

By developing Eq. (7) we obtain:

$$\rho C_p\left(\frac{\partial T}{\partial t} + v_x\frac{\partial T}{\partial x} + v_y\frac{\partial T}{\partial y} + v_z\frac{\partial T}{\partial z}\right) = \frac{\partial}{\partial x}\left(k_x\frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k_y\frac{\partial T}{\partial y}\right) + \frac{\partial}{\partial z}\left(k_z\frac{\partial T}{\partial z}\right) \tag{8}$$

2.2. Initial Conditions

According to the experimental tests evoked in the literature, in our study, one considers that the initial temperature is equal:

$$T(x, y, z) = 60\ ^\circ\mathrm{C} \text{ at time } t = 0 \tag{9}$$

2.3. Boundary conditions

In general, in a thermal study, one finds three types of boundary conditions:

1. Temperature specified on a surface

$$S_T : T = T^* \tag{10}$$

2. Heat flux specified on a surface

$$S_Q : \{Q\}^T\{n\} = -Q^* \tag{11}$$

3. Convection specified on a surface

$$S_c : \{Q\}^T\{n\} = h(T_p - T_f) \tag{12}$$

3. Heat flux entering the disc

In a braking system, the mechanical energy is transformed into a calorific energy. This energy is characterised 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. Generally, the thermal conductivity of material of the brake pads is smaller than that 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 emitted by this surface is equal to the energy generated by friction. The initial heat flux q_0 entering the disc is calculated by the following formula [16]:

$$q_0 = \frac{1-\phi}{2}\frac{mgvz}{2A_d\varepsilon_p} \tag{13}$$

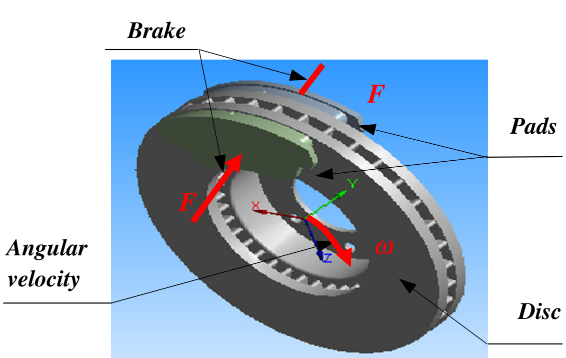


Figure 1 Disc-pads assembly with forces applied to the disc.

Table 1 Geometrical dimensions and application parameters of automotive braking.

Item	Values
Inner disc diameter, mm	66
Outer disc diameter, mm	262
Disc thickness (TH), mm	29
Disc height (H), mm	51
Vehicle mass m, kg	1385
Initial speed v_0, km/h	28
Deceleration a, m/s^2	8
Effective rotor radius R_{rotor}, mm	100.5
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	35993

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

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 grey cast iron (GFC) with high carbon content [17], with good thermophysical characteristics, and the brake pad has an isotropic elastic behaviour whose thermomechanical characteristics adopted in this simulation in the transient analysis of the two parts are recapitulated in Table 2.

4. Modelling in ANSYS CFX

The first stage is to create the CFD model which contains the fields to be studied in Ansys Workbench. In our case, we took only one quarter of the disc; then, we defined the field of the air surrounding this disc. ANSYS ICEM CFD will prepare various surfaces for the two fields in order to facilitate the mesh on which one will export the results towards CFX using the command 'Output to cfx' [18] (Fig. 2). After obtaining the

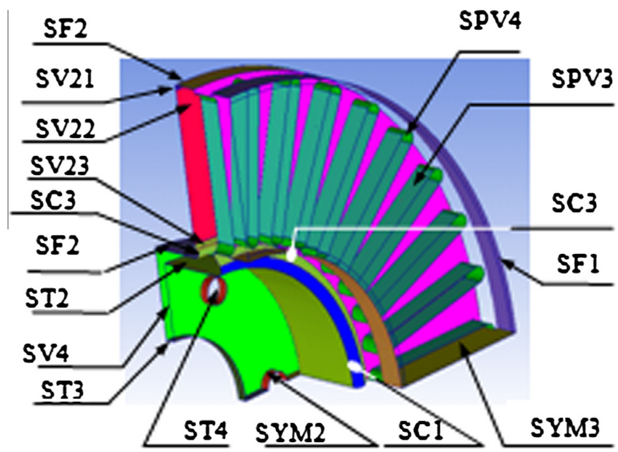


Figure 2 Definition of surfaces of the ventilated disc.

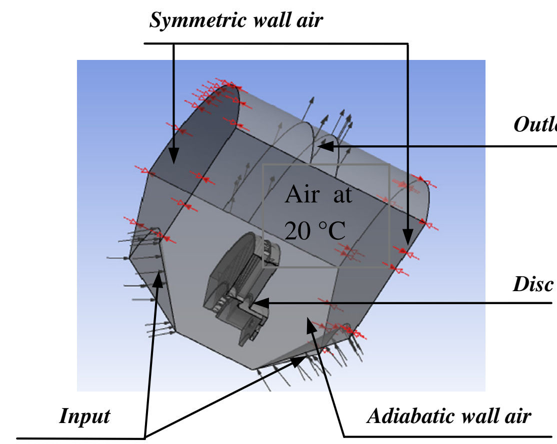


Figure 4 Brake disc CFD model.

model on CFX Pre and specifying 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 [19]. In order not to weigh down calculation, an irregular mesh is used in which the meshes are broader where the gradients are weaker (non-uniform mesh), (Fig. 3).

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

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 three cast iron materials (FG 25 Al, FG 20 and FG 15) with their thermal conductivity, respectively (43.7 W/m °C, 55 W/m °C and 57 W/m °C). Since the aim of this study is to determine the temperature field in a disc brake during the braking phase of a vehicle of average class, we take the following temporal conditions:

- Braking time = 3.5 (s).
- Increment time = 0.01 (s).
- Initial time = 0 (s).

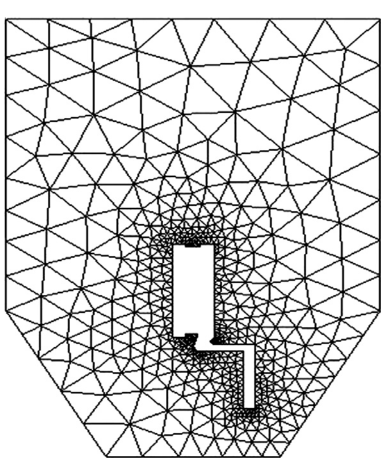


Figure 3 Irregular mesh in the wall.

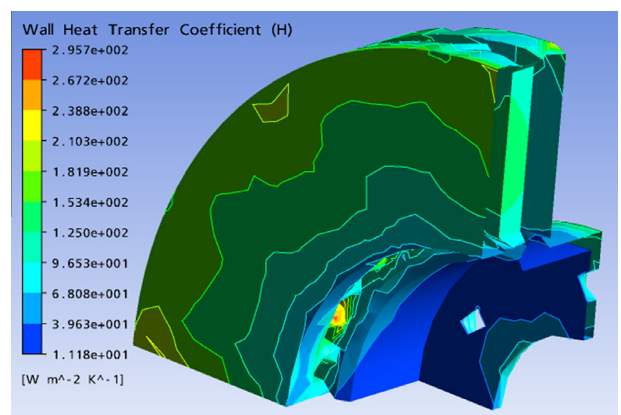


Figure 5 Distribution of heat transfer coefficient on a ventilated disc in the stationary case (FG 15).

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

After verification of the model and boundary conditions, we run the calculation by opening the menu 'File' and clicking on 'Write solver file'. The values of the coefficient of exchange will be taken average values calculated by the minimal and maximum values obtained using ANSYS CFX POST as it is indicated in Fig. 5.

Figs. 6 and 7 show the variation in 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.

5. 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. 8). 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

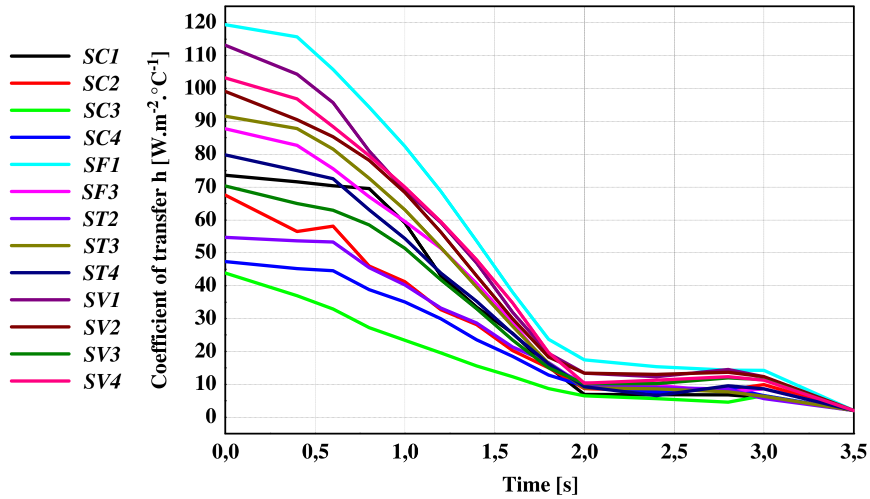


Figure 6 Variation in heat transfer coefficient (h) of various surfaces for a full disc in the non-stationary case (FG 15).

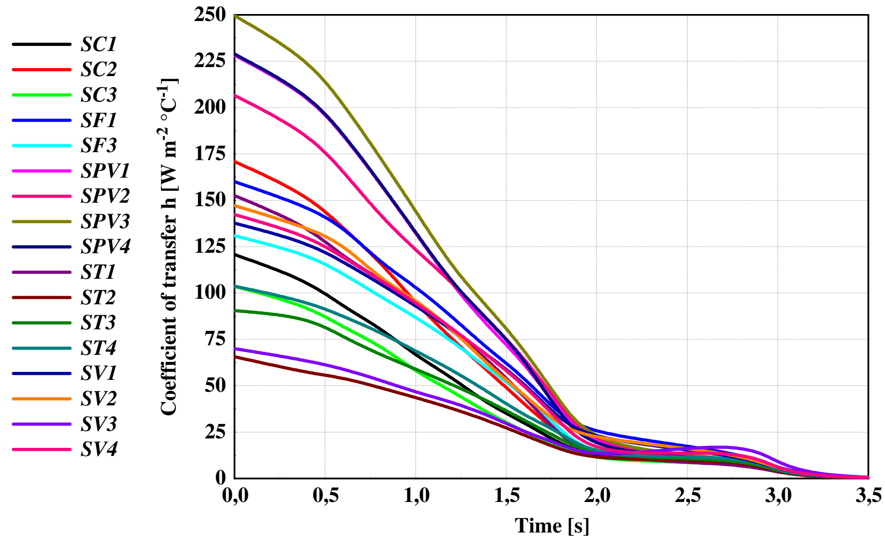


Figure 7 Variation in heat transfer coefficient (h) of various surfaces for a ventilated disc in transient case (FG 15).

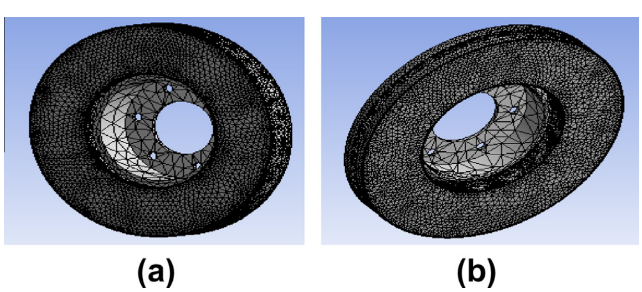


Figure 8 Meshing of the disc (a) full disc (172103 nodes-114421 elements) (b) ventilated disc (154679 nodes-94,117 elements).

very high. That is why the correct taking into account of the contact conditions involves the use of a refined mesh.

6. Loading and boundary conditions

The thermal loading is characterised 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 summarised 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 modelling of temperature in the disc brake will be carried out by taking account of the variation in 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 (13).

7.1. Influence of construction of the disc

Fig. 9 shows the variation in 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 notice that for a ventilated disc out of cast iron FG15, the temperature increases until T_{max} = 345 °C at the moment t = 1.85 s, and 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 = 1.88 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.

8. Coupled thermomechanical analysis

8.1. FE model and boundary 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 [20]. 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 Fig. 10. The initial temperature of the disc and the pads is 20 °C, the surface convection condition is applied at all surfaces of the disc with the values of the 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

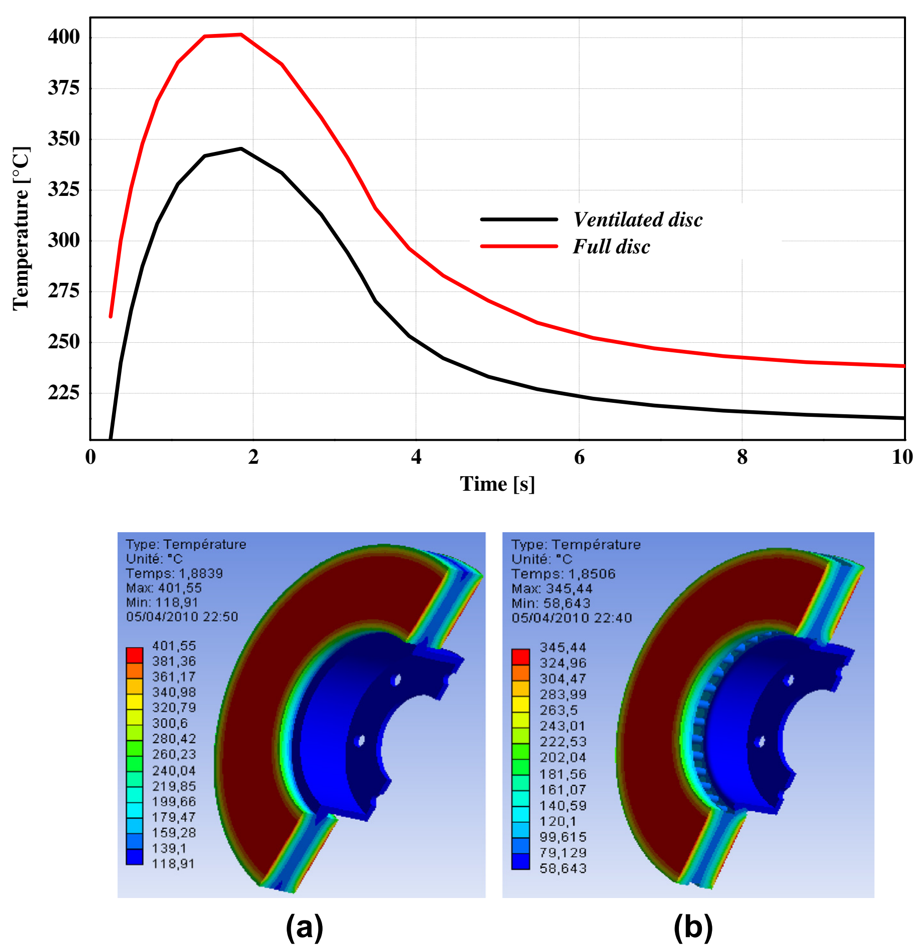


Figure 9 Temperature distribution of a full (a) and ventilated disc (b) of cast iron (FG 15).

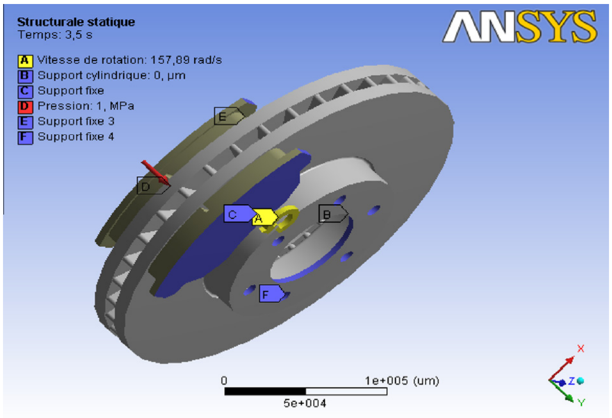


Figure 10 Boundary conditions and loading imposed on the disc-pads.

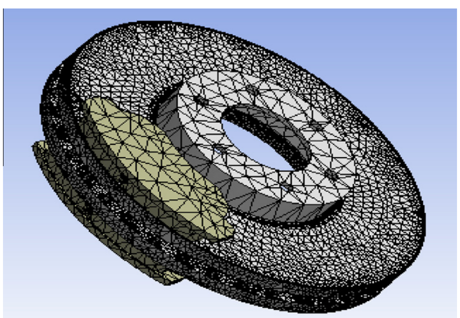


Figure 11 Refined mesh of the model.

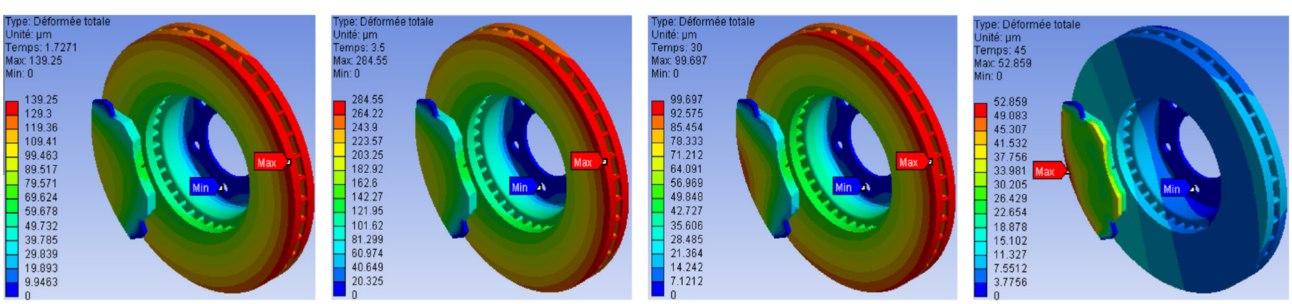
formula described in the first part. The FE mesh is generated using three-dimensional tetrahedral element with 10 nodes (solid 187) for the disc and pads. There are about 185901 nodes and 113367 elements are used (Fig. 11). The thermal coupling will be carried out by the thermal condition at a temperature non-uniform 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.

8.2. Thermal deformation

Fig. 12 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 fin of the disc. Indeed, during a braking, the maximum temperature depends almost entirely on the heat storage capacity of disc (on particular tracks of friction); this deformation will generate an asymmetry of the disc following the rise of temperature which will cause a deformation in the shape of an umbrella.

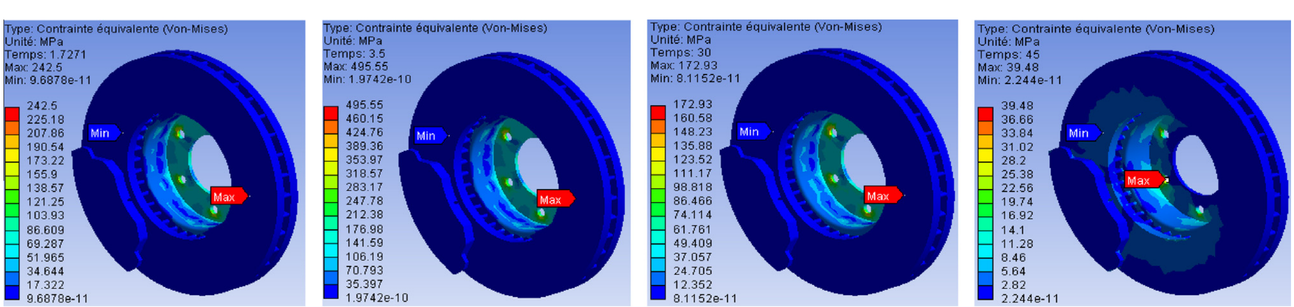
8.3. Von Mises stress distribution

Fig. 13 presents the distribution of the constraint equivalent of Von Mises stress to various moments of simulation, and the scale of values varies from 0 MPa to 495 MPa. The maximum



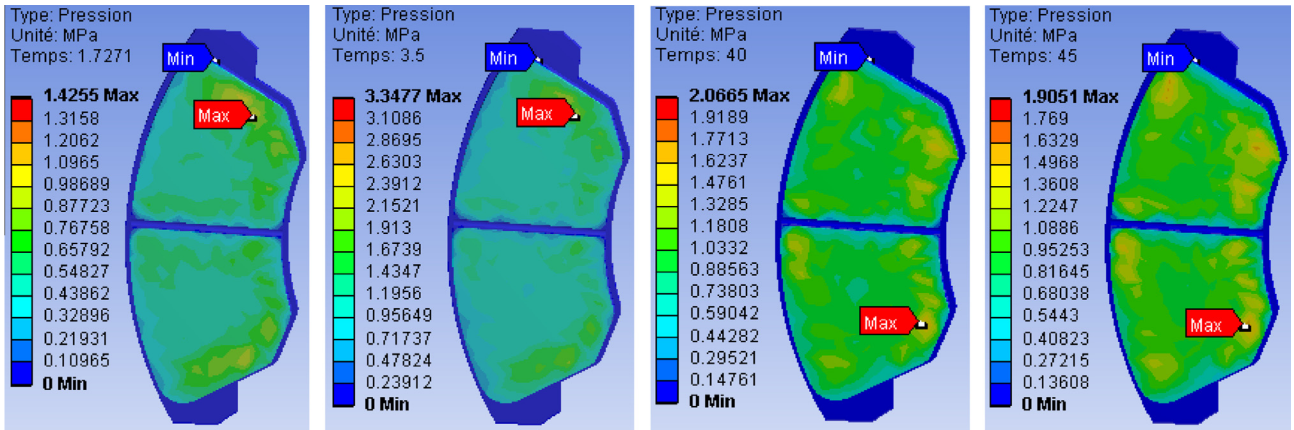
(a) t=1,72 s (b) t=3,5 s (c) t=30 s (d) t=45 s

Figure 12 Total distortion distribution.



(a) t=1,72 s (b) t=3,5 s (c) t=30 s (d) t=45 s

Figure 13 Von Mises stress distribution.



(a) t=1,72 s (b) t=3,5 s (c) t=40 s (d) t=45 s

Figure 14 Contact pressure distribution in the inner pad.

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, the effects of torsional stress and sheers generated at the level of the bowl which are 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.

8.4. Contact pressure

Fig. 14 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 3.34 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 in the contact pressure distribution.

9. Conclusion

In this study, we presented a numerical simulation of the thermal behaviour of a full and ventilated disc in transient state. By means of the computer code ANSYS 11, we were able to study the thermal behaviour of three types of cast iron (AL FG 25, FG 20 and 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 behaviour 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 behaviour of the disc brake (stress, deformations, 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 the following:

- Technological parameters illustrated by the design,
- Numerical parameters represented by the number of elements and the step of time.

With regard to the results of the coupling, we made the following conclusions:

- 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.

About the results obtained, in general, one can say that they are satisfactory in comparison with already carried out research tasks. Compared to the prospects, one finds interesting to also make an experimental study of the disc of brake, for example, on test benches in order to show a good agreement between the model and reality.

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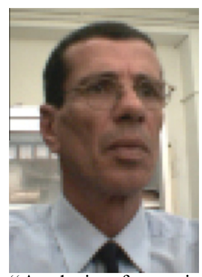

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