# STUDY OF THE THERMAL BEHAVIOUR OF DRY CONTACTS IN THE BRAKE DISCS « APPLICATION OF SOFTWARE ANSYS v11.0»


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***Abstract:*** *The braking operation is a process which converts the kinetic energy and the potential energy of the vehicle into other energies. The major part of the mechanical energy is transformed into heat. During the braking phase, the frictional heat generated at the interface disc - pads can lead to high temperatures. This phenomenon is even more important that the tangential stress as well as the relative sliding speeds in contact is important. The objective of this study is to analyze the thermal behavior of the full and ventilated brake discs of the vehicles using computing code ANSYS. The modeling 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 results obtained by the numerical simulation are satisfactory compared with those of the specialized literature.*



***Key words:*** *Dry contact - Friction- Brake Discs – Transient mode –Heat flux- Heat transfer coefficient –Thermal stress*


## I. INTRODUCTION

The thermal analysis is a primordial stage in the study of the brake systems, because the temperature determines the thermomechanical behavior of the structure. In the braking phase, the temperatures and the thermal gradients are very high. This generates stresses and deformations whose consequences are manifested by the appearance and the accentuation of cracks. 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.

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

## II. 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 [1]:

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

$z = a/g$: Braking effectiveness

$a$ : Deceleration of the vehicle [$ms^{-2}$]

$\phi$ : Rate distribution of the braking forces between the front and rear axle

$A_d$ : Disc surface swept by a brake pad [$m^2$]

$V$ : Initial speed of the vehicle [$ms^{-1}$]

$\varepsilon_p$ : Factor load distribution of the on the surface of the disc.

M : Mass of the vehicle [kg]

The figure 1 shows the ventilated disc – pads and the applied forces.

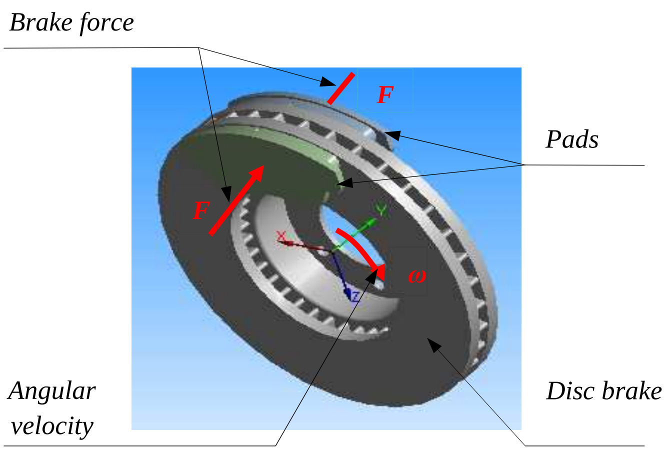



***Fig.1:*** *Disc-pads assembly with forces applied to the disc*

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.

| | |
|---|---|
| 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 forces– $\phi$- [%] | 20 |
| Factor of charge distribution on the disc $\varepsilon_p$ | 0.5 |
| Surface disc swept by the pad $A_d$ [$mm^2$] | 35993 |

***Table.1:*** *Geometrical Dimensions and application **p**arameters of automotive braking*

The disc material is gray cast iron with high carbon content FG, with good thermophysical characteristics.

## III. NUMERICAL MODELING OF THE THERMAL PROBLEM

### III.1 Finite Element Method

The finite element method is used in many applications to solve partial differential equations [2].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 a matrix and L a vector.

### III.2 Form Differential

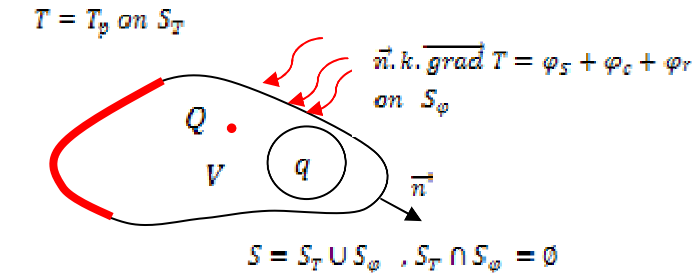



***Fig.2:*** *Thermal loads applied to a continuous medium*

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 $\varphi_S$ imposed on *a* $S_\varphi$ *surface* in [W/m$^2$]
- Heat transfer by convection $\varphi_c$ on *a* surface S$\varphi$
- Heat transfer by radiation $\varphi_r$ on *a* surface *S$\varphi$*

The resolution of a thermal problem consists to find the temperature field $T(x, y, z, t)$ at any point of the solid so that [3] :

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

- With the boundary conditions

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

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

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

Where

$\rho$ : 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 equation is written in weak formulation as follows [4], [5], [6]:

$$\int_V T^*\rho C_p\ dV + \int_V \overrightarrow{grad}\ T^*.\left(k.\overrightarrow{grad}\ T\right)dV$$
$$-\int\ T^*\left(\varphi_S + h(T_f - T) + \varepsilon\sigma\ (T_\infty^4 - T^4)\right) -$$

$$\int_V T^*\ q\ dV = 0\ \forall\ T^* \qquad (5)$$

$T^*$ 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* [3]:

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

$[N(x,y,z)]$ : The matrix of interpolation.

$\{T(t)\}$ : Vector of the nodal temperatures.

By carrying the following relations in the equation (5):

$$T = [N]\{T\} \qquad (8)$$
$$\dot{T} = [N]\{\dot{T}\} \qquad (9)$$
$$\{gradT\} = [B]\{T\}\text{avec}[B] = [\{B_1\}\cdots\{B_i\}\cdots\{B_n\}] \qquad (10)$$

$$\dot{T} = [N]\{\dot{T}\} = \{T^*\}^T[N]^T, \{grad\ T^*\} = [B]\{T^*\},$$
$$\{grad\ T^*\}^T = \{T^*\}^T[B]^T$$

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).

### III.3. Initial Conditions

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

$$T(x,y,z,t) = \ 60\ {}^\circ C \qquad at\ time\ \ t = 0 \qquad (4)$$

### III.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 the 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.

## IV.PRESENTATION OF THE COMPUTING CODE ANSYS

ANSYS is software program, created into 1970 in the United States; its modules are software programs that implement the finite element method to solve models previously discretized. The modules used for this study are:

- **ANSYS Workbench:** This platform offers a different approach in the construction a model using the original computer code ANSYS [4].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 resolution, 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.

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

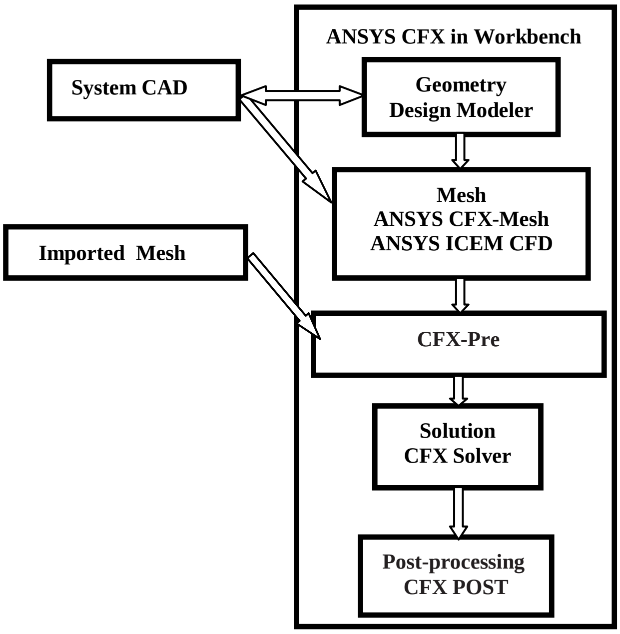



***Fig.3:*** *Simulation steps with CFX* [4]

## V. DETERMINATION OF THE COEFFICIENT OF EXCHANGE BY CONVECTION (h)

### V.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.

When a vehicle is braked, 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.

### V.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. An unsteady-state analysis is necessary.

The figure 4 shows the elaborate model CFD which will be used in **ANSYS CFX Pre.**

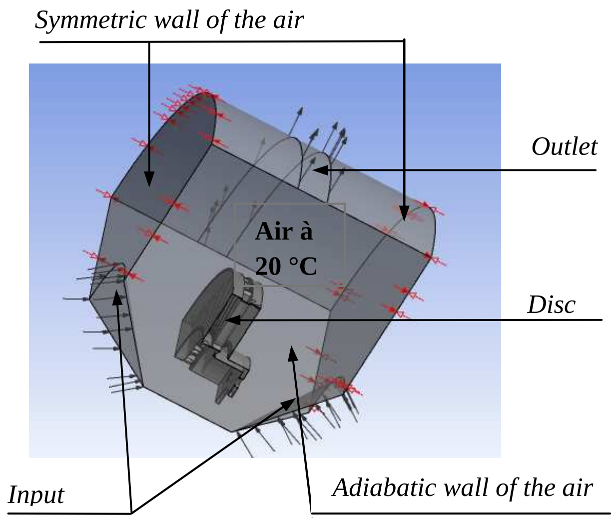



***Fig.4:*** *Brake disc CFD model*

**a) Physical model**

In this step, one declares all of the physical characteristics of the fluid and the solid. After the meshing, on define 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 library 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 select 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, ensures 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"**.

### h) Results of the calculation of the coefficient h

The figures 5 and 6 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.

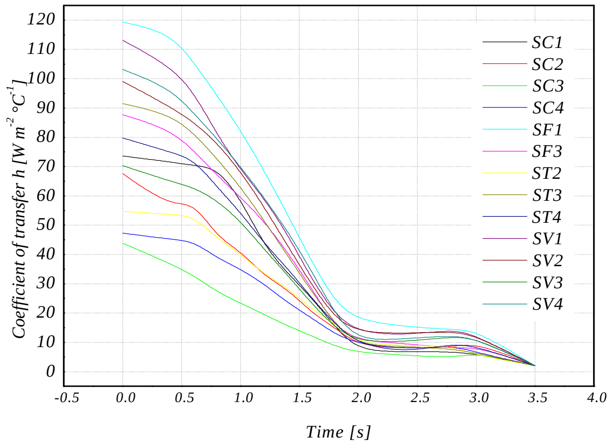



***Fig.5:*** *Variation of heat transfer coefficient (h) various surfaces for a full disc in the non stationary case (FG 15)*

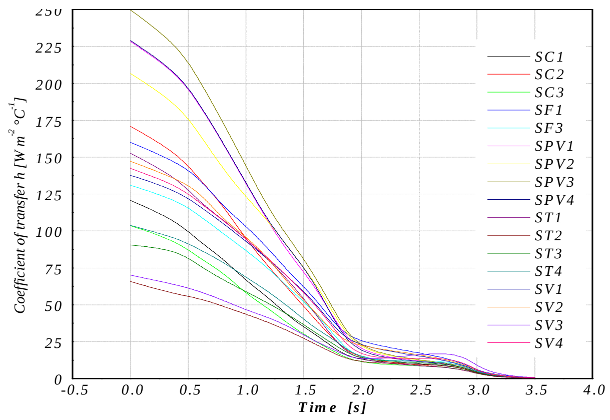



***Fig.6****: Variation of heat transfer coefficient (h) various surfaces for a ventilated disc in transient case (FG 15)*

## VI. 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 7. The variation of the heat flux during the simulation time is represented on the figure 8.

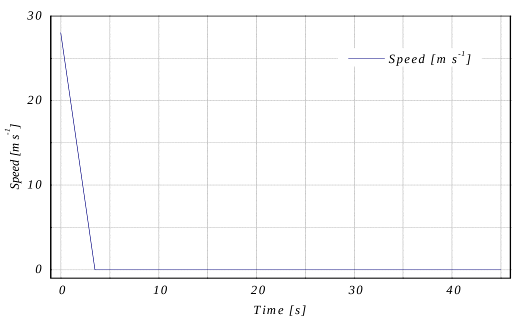



***Fig.7:*** *Speed of braking versus time (braking of type 0)*

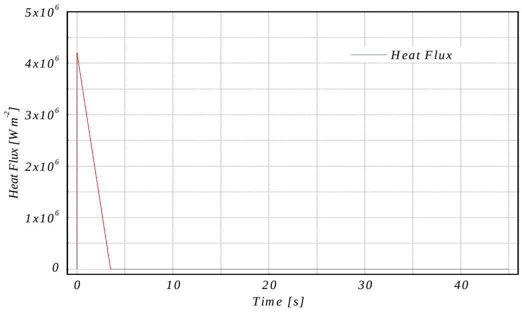



***Fig. 8:*** *Heat Flux versus time*

**VI.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.9 and 10).In this simulation, the meshing was refined in the contact zone (disc-pad).This is important because in this zone the temperature varies significantly.

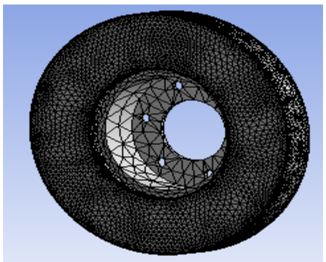


***Fig.9****: Meshing of a full disc in ANSYS Multiphysics (172103 nodes – 114421 elements)*

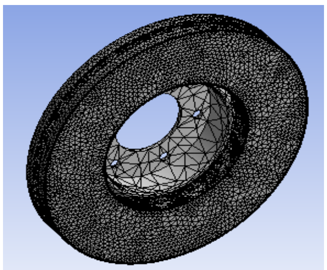


***Fig.10:*** *Meshing of a ventilated disc in ANSYS Multiphysics (154679 nodes- 94117 elements)*

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 2.

| | *Full disc* | *Ventilated disc* |
|---|---|---|
| | *Number of elements* | *Number of elements* |
| *Mesh 1* | 46025 | 77891 |
| *Mesh 2* | *114421* | *94117* |
| *Mesh 3* | 256613 | 369777 |

***Table 2:*** *Number of elements of the two considered meshs*

**VI. 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 the physical properties of 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).

**VII. RESULTS AND DISCUSSIONS**

The modeling of the 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).

**VII. 1 Influence of construction of the disc**

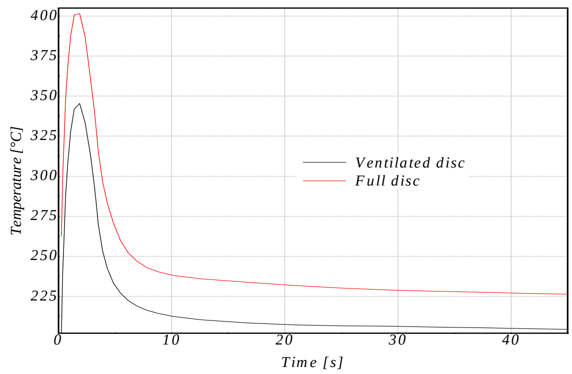
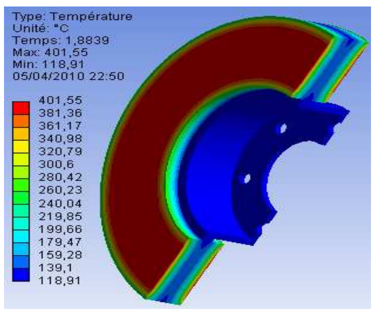



***Fig.11:*** *Temperature variation of a full and ventilated disc (FG 15) versus time*

***(a)***




***(b)***

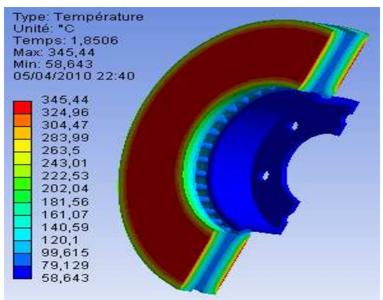



***Fig.12 :*** *Temperature distribution of a full (a) and ventilated disc (b) of cast iron (FG 15)*

Figure 11 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.

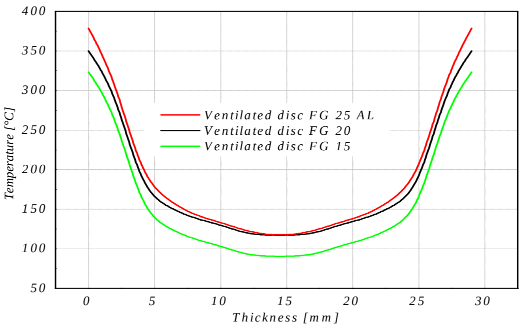
We quickly notices that for a ventilated disc out of cast iron FG15 ,the temperature increases until $T_{max}$ = 345,44 °C at the moment t = 1,85 s, then it decrease rapidly in the course of time. The variation in temperature between a full and ventilated disc having same material is about 60 °C at the 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.

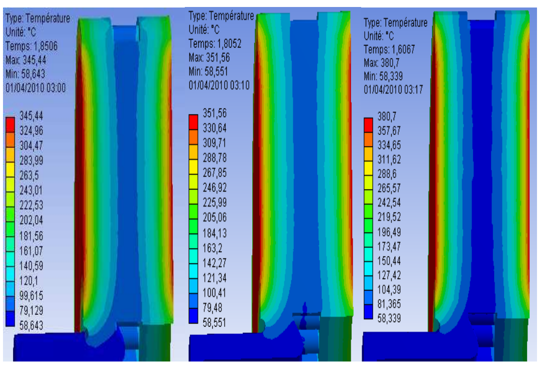
## VII. 2. Infuence of material of the disc

Figure 13 (a) shows for each type of 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 direction of the median plane of the disc to reach its minimal value. In figure 13(b) we see that there is inside the disc symmetry of colors. The part far away from surface of blue contact color is not requested too much thermically. More the thermal conductivity of material is low, more its temperature is high. The FG 15 is differentiated from the two other cast iron by smaller temperatures.

On figure 14 is represented the temperature variation versus radius for three materials (FG 25 Al, FG 20, FG 15).The shape of the temperature curves are the same one. The maximal temperature is in area of the mean disc radius.

According to figures 13 and 14 the cast iron FG 15 has the best thermal behavior.



FG 15 FG 20 FG25 AL

*Fig.13:* *Temperature variation through a disc thickness for three type of cast irons (FG 25 AL, FG 20 and FG 15)*

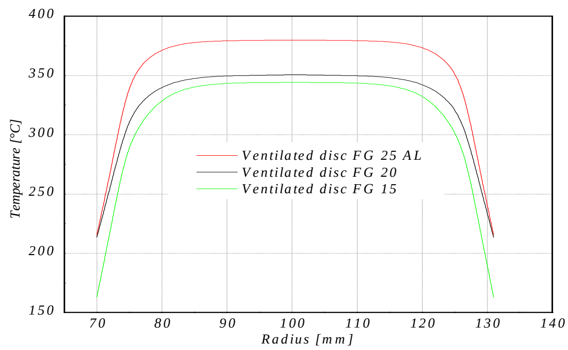
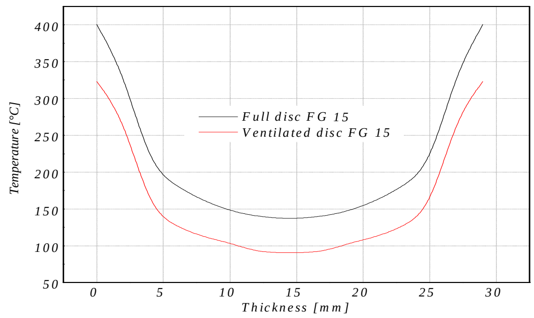


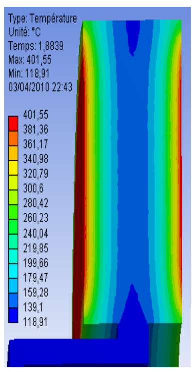

***Fig.14:*** *Temperature variation through a radius for three type of cast irons (FG 25 AL, FG 20 and FG 15)*

The figures 15 and 16 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.

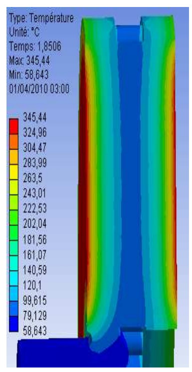
The influence of ventilation on the temperature field appears clearly at the end of the braking (t = 3,5 s).

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.15:*** *Temperature variation through the thickness for both designs with same material (FG15)*

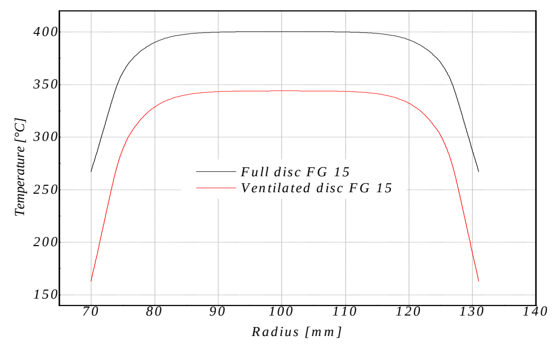



***Fig. 16:*** *Temperature variation through a radius or both designs with the same material (FG15)*

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

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### VII. 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.

### VII. 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.

Figure 17 shows a driving cycle of fourteen successive brakings, in the form of sawtooth.

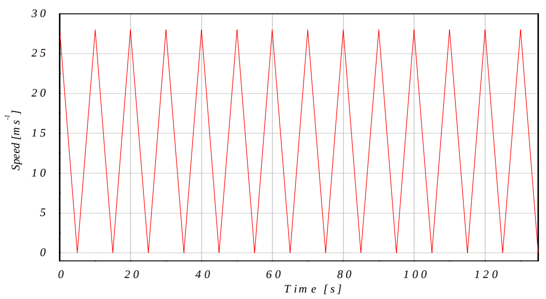


*Fig. 17: Driving cycle with fourteen repeated braking (Mode 1)*

each phase of braking one has an idle .

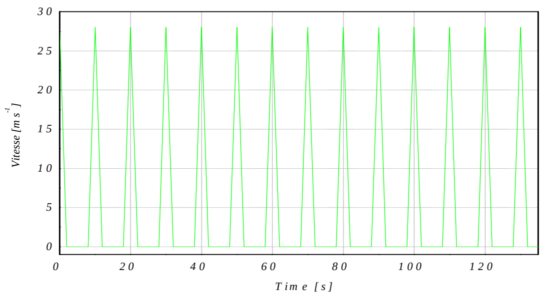


***Fig.18:*** *Cycle braking with phase of idles after each braking (mode 2)*

Figure 19 shows the comparison of the change of the temperature of the disc for a cyclic braking process between the first mode and the second mode. For two contours, we notes 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 figure 19, 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 the safety and the 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.

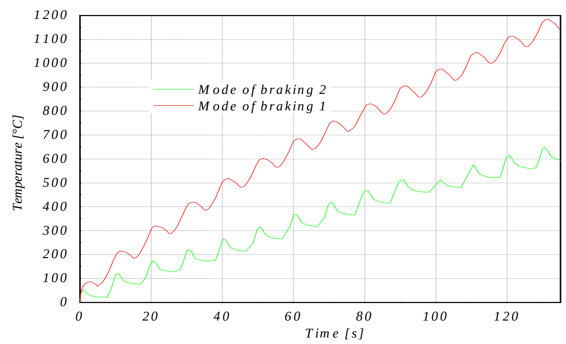


***Fig.19 :*** *Température variation of the two braking modes versus time*

## VIII. 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 was 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.

The modeling of the disc material must take account of the anelastic behavior of the cast iron according to the real temperature of the disc. To be predictive, the calculation of the disc must respect the complexity of the system (three-dimensional problem, rotation of the disc, anelastic material behavior, and anisothermes stress…).

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

## BIBLIOGRAPHICAL REFERENCES


[1] Reimpel J.: Technologie de freinage. Vogel Verlag, Würzburg, 1998.

[2] Dhatt G, Touzot G. : Une présentation de la méthode des éléments finis. Malouine S.A., Paris, 1984

[3] Debard Y. : Méthode des éléments finis : Modélisation Numérique et Réalité Virtuelle. Master Université du Maine mars *2006*

[4] Bathe K.-J. : Finite element procedures in engineering analysis. Prentice Hall, 1996.

[5] Huebner K. H., Thornton E. A. et Byron T. G.: The finite element method for engineers. Wiley, 1995.

[6] Lewis R. W., Morgan K., Thomas H. et Seetharamu K. N.: The finite element method in heat transfer analysis. Wiley, 1996.

[7] www.ansys.com