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**Key words:** brake discs, pads, heat flux, heat-transfer coefficient, Von Mises stress, contact pressure

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# STRUCTURAL AND THERMAL ANALYSIS OF AUTOMOTIVE DISC BRAKE ROTOR

The main purpose of this study is to analyze the thermomechanical behavior of the dry contact between the brake disc and pads during the braking phase. The simulation strategy is based on softawe ANSYS11. The modeling of transient temperature in the disc brake is actually used to identify the factor of geometric design of the disc to install the ventilation system in vehicles. The thermal-structural analysis is then used with coupling to determine the deformation established and the Von Mises stress in the disc, the contact pressure distribution in pads. The results are satisfactory compared to those of the specialized literature.

## 1. Introduction

Brakes are most important safety parts in the vehicles. Generally all of the vehicles have their own safety devices to stop their car. Brakes function is to slow down and stop the rotation of the wheel. To stop the wheel, braking pads are forced mechanically against the rotor disc on both surfaces. They are compulsory for all of the modern vehicles and the safe operation of vehicles. In short, brakes transform the kinetic energy of the car into heat energy, thus slowing its speed.

Ventilated disc-pad brakes (Fig. 1) are widely used for reducing velocity due to their braking stability, controllability and ability to provide a wide-ranging brake torque. The brake disc with vanes rotates through the caliper. A ventilated disc with straight vanes is most popular, easy and straightforward to make. The pressure on the pistons pushes the pads with three-dimensional geometry against the brake disc and produces brake torque.

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Most of the mechanical energy of a moving vehicle is converted into heat through the friction between the brake disc and pads in the braking process and 99% of heat energy is dissipated through the brake disc and pad. The braking processes in the friction units of a brake are very complicated. In the course of braking, all parameters of the processes (velocity, load, temperature and the conditions of contact) vary with time.

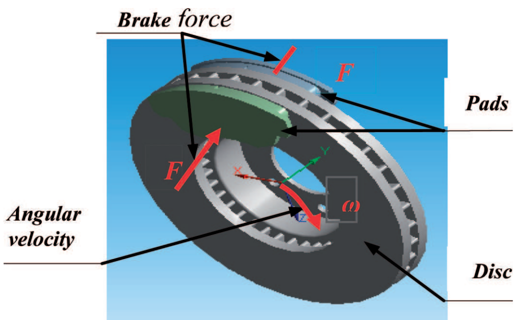



Fig. 1. Disc-pads assembly with forces applied to the disc

Normally, thermal stress analysis has been performed to any of material related to thermal process in order to oversee the behavior and character of material. Any abnormality associated with thermal input will give the high values on the stress magnitude of the studied materials.

The high values of stress magnitude will shows deformation on certain areas which load has been applied on it. Design and analysis of certain parts or component will took much time and it is costly. Therefore, without any analysis or design tools, it would be limitations on repeated analysis. For decades, finite element analysis (FEA) has been a preferred method to address some of the above concerns. It can be used to compare the design alternatives and hence, optimize the brake rotor design prior to production of prototype components [1].

Choi and Lee, [2] presented a paper on finite element analysis of transient thermoelastic behaviors in disc brakes. A transient analysis for thermoelastic contact problem of disc brakes with frictional heat generation is performed using the finite element method. To analyze the thermoelastic phenomenon occurring in disc brakes, the coupled heat conduction and elastic equations are solved with contact problems. The numerical simulation for the thermoelastic behavior of disc brake is obtained in the repeated brake condition. The computational results are presented for the distributions of pressure and temperature on each friction surface between the contacting bodies.

Talati and Jalalifar [3] presented a paper on analysis of heat conduction in a disc brake system. In this paper, the governing heat equations for the disc and the pad are extracted in the form of transient heat equations with heat generation that is dependant on time and space. In the derivation of the heat equations, parameters such as the duration of braking, vehicle velocity, geometries and the dimensions of the brake components, materials of the disc brake rotor and the pad and contact pressure distribution have been taken into account. The problem is solved analytically using Green's function approach. It is concluded that the heat generated due to friction between the disc and the pad should be ideally dissipated in the environment to avoid decreasing the friction coefficient between the disc and the pad and to avoid the temperature rise of various brake components and brake fluid vaporization due to excessive heating.

Naji et al. [4] presented a mathematical model to describe the thermal behavior of a brake system which consists of the shoe and the drum. The model is solved analytically using Green's function method for any type of the stopping braking action. The thermal behavior is investigated for three specified braking actions which are the impulse, the unit step and trigonometric stopping actions. Thermal response of disc brake systems to different materials used for the disc–pad couple has been studied by many researches [5-12]. Aerodynamic cooling of high performance disc brake systems is investigated by many researchers, as well [13-15].

Kang and Cho [16] analyzed the geometry of vents in motorcycle disc brakes which affects the surface of the disc and the thermal characteristics of disc brakes. Thermal deformation analysis and thermal stress analysis due to heat transfer were carried out through the finite element analysis for ventilated disc and solid disc. For 3-dimensional modeling and finite element analysis of the discs, the commercial code ANSYS Workbench was used.

Thilak VMM et al. [17] conducted a transient thermal and structural analysis of a brake disc to evaluate its performance under severe braking conditions and then to assist in disc rotor design and analysis. The usage of new materials was investigated which aims at improving the braking efficiency and providing greater stability to the vehicle. This study was done using ANSYS 11 software to analyze the temperature distribution, variation of the stresses and deformation across the disc brake profile. The new materials under study were aluminum base metal matrix composite and High Strength Glass Fiber composites. These materials have a promising friction and wear behavior as a brake disc. The transient thermo elastic analysis of discs in repeated brake applications was performed and the results were compared to that of cast iron disc.

Yildiz and Duzgun [18] analyzed the stress of ventilated brake discs by the finite element method, three different ventilated brake discs were designed and manufactured and their performances of braking force were investigated experimentally in addition to those of a full disc. Afterwards, stress analyses were performed by FEA. In these analyses, a different approach, the variable loading on brake pads, was also introduced. It was found that this approach remarkably reduced the stresses on ventilated discs. Hence, this unique approach could be an improvement for eliminating crack formations in ventilated brake discs. In the study developed by Moses and John [19], fully coupled thermoelastic instability problem for a disc brake system was analyzed. Thermal and mechanical model for the disc brake would be generated and solved using Comsol Multiphysics developed base on the finite element principle. The work models the heat generation and dissipation in a disc brake during braking and the following release period. The model simulates the braking action by investigating both the thermal and elastic actions occurring during the friction between the two sliding surfaces, represented by the maximum temperature on the contact surface. Brake pad and disc were selected, and parameters set to certain values from existing literatures. In the study carried out by Chung et al. [20], a transient FE analysis method was used to analyze the full coupled thermoelastic instability problem for a disc brake system. The three dimensional mechanical and thermal models for the disc brake were generated separately and solved iteratively using the staggered approach The simulation results, such as the maximum disc temperature, BTV are compared to the data from the dynamo test and the reliabilities of the analysis technique and simulation model have beenverified.

Hassan et al. [21] and Li et al. [22] have predicted the temperature distribution for disc brakes by carrying out a coupled thermal–mechanical analysis as a part of their works. They have applied angular displacements to the rotor to maintain a specific velocity and calculated the heat generation at the interface from contact pressure. However, in reality, the wheel is in deceleration during braking.

In this study, we will perform a modeling of the thermomechanical behavior of the dry contact between the discs of brake pads at the time of braking phase; the strategy of calculation is based on the software Ansys 11. The latter is comprehensive mainly for resolution of the complex physical problems. The numerical simulation of the coupled transient thermal and stress field is carried out by sequentially thermal-structural coupled method based on Ansys.

## 2. Theoretical Background

### 2.1 Finite element formulation for heat conduction

The unsteady heat conduction equation of each body for an axis-symmetric problem described in the cylindrical coordinate system is given as follows:

$$\rho c \frac{\partial T}{\partial t} = \frac{1}{r}\frac{\partial}{\partial r}\left(r k_r \frac{\partial T}{\partial r}\right) + \frac{\partial}{\partial z}\left(k_z \frac{\partial T}{\partial z}\right) \tag{1}$$

With the boundary conditions and initial condition

$$T = T^* \text{ on } \Gamma_0 \tag{2}$$

$$q_n = h(T - T_\infty) \text{ on } \Gamma_1 \tag{3}$$

$$q_n = q_n^* \text{ on } \Gamma_2 \tag{4}$$

$$T = T_0 \text{ at time} = 0 \tag{5}$$

Where $\rho$, $c$, $k_r$ and $k_z$ are the density, specific heat and thermal conductivities in $r$ and $z$ direction of the material, respectively. Also, $T^*$ is the prescribed temperature, is the heat-transfer coefficient, $q_n^*$ is the heat flux at each contact interface due to friction, $T_\infty$ is the ambient temperature, $T_0$ is the initial temperature and $\Gamma_0$, $\Gamma_1$ and $\Gamma_2$ are the boundaries on which temperature, convection and heat flux are imposed, respectively.

Using Galerkin's approach, a finite element formulation of unsteady heat Eq.(1) can be written in the following matrix form as

$$C_T \dot{T} + KH_T T = R \tag{6}$$

Where $C_T$ is the capacity matrix, $KH_T$ is the conductivity matrix. $T$ and $R$ and are the nodal temperature and heat source vector, respectively.

In order to solve Eq.(6), generalized trapezoidal rule was used. Based on the assumption that temperature $T_t$ at time $t$ and temperature $T_{t+\Delta t}$ at time $t + \Delta t$ have the following relation:

$$T_{t+\Delta t} = T_t + \left[(1-\beta)\dot{T}_t + \beta \dot{T}_{t+\Delta t}\right]\Delta t \tag{7}$$

Eq.(7) can be used to reduce the ordinary differential Eq.(6) to the following implicit algebraic equation:

$$(C_T + b_1 KH_T)T_{t+\Delta t} = (C_T - b_2 KH_T)T_t + b_2 R_t + b_1 R_{t+\Delta t} \tag{8}$$

Where $C_T$ is the capacity matrix, $KH_T$ is the conductivity matrix. $T$ and $R$ and are the nodal temperature and heat source vector, respectively.

Where the variable $b_1$ and $b_2$ are given by

$$b_1 = \beta\Delta t,\ b_2 = (1-\beta)\Delta t \tag{9}$$

For different values of $\beta$, the well-known numerical integration scheme can be obtained [23]. In this study, $0.5 \leq \beta \leq 1.0$ was used, which is an unconditionally stable scheme.

## 3. CFD Modeling and thermal analysis of the problem

### 3.1. Heat flux entering the disc

The brake disc consumes the major part of the heat, usually greater than 90% [24], by means of the the effective contact surface of the friction coupling. Considering the complexity of the problem and the limitation in the average data processing, one identifies the pads by their effect, represented by an entering heat flux (Fig. 2).

The initial heat flux $q_0$ $q_0$ entering the disc is calculated by the following formula [25]

$$q_0 = \frac{1-\phi}{2}\frac{mgv_0 z}{2A_d\varepsilon_p} \tag{10}$$

where $z = a/g$ is the braking effectiveness, $a$ is the deceleration of the vehicle [$\mathrm{ms}^{-2}$], $\phi$ is the rate distribution of the braking forces between the front and rear axle, $A_d$ $A_d$: disc surface swept by a brake pad [$\mathrm{m}^2$], $v_0$ is the initial speed of the vehicle [$\mathrm{ms}^{-1}$], $\varepsilon_p$: is the factor load distribution on the surface of the disc, $m$: is the mass of the vehicle [kg], $g$: is the acceleration of gravity (9.81) [$\mathrm{ms}^{-2}$].

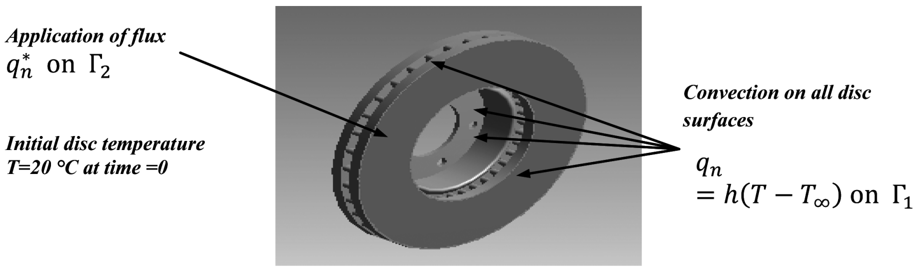



Fig. 2. Bondary conditions of the thermal analysis

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 (GF) with high carbon content [26], with a good thermophysical characteristics and the brake pad has an isotropic

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 $\phi$, | 20 |
| Factor of charge distribution of the disc $\varepsilon_p$ | 0.5 |
| Surface disc swept by the pad $A_d$, mm$^2$ | 35993 |

elastic behavior, thermomechanical characteristics of which adopted in this simulation of the two parts are recapitulated in Table 2.

Rotors are made of cast iron for three reasons [27]:

- It is relatively hard and resists wear.
- It is cheaper than steel or aluminum.
- It absorbs and dissipates heat well to cool the brakes.

Table 2.

Thermoelastic properties used in simulation

| Material Properties | Pad | Disc |
|---|---|---|
| Thermal conductivity, $k$(W/m°C) | 5 | 57 |
| Density, $\rho$ (kg/m$^3$) | 1400 | 7250 |
| Specific heat, $c$ (J/Kg. °C) | 1000 | 460 |
| Poisson's ratio, $\upsilon$ | 0.25 | 0.28 |
| Thermal expansion, $\alpha$ ($10^{-6}$ / °C) | 10 | 10,85 |
| Elastic modulus, $E$(GPa) | 1 | 138 |
| Coefficient of friction, $\mu$ | 0.2 | 0.2 |
| **Operation Conditions** | | |
| Angular velocity, $\omega$ | | 157.89 |
| Hydraulic pressure, $P$(MPa) | | 1 |

It is very difficult to exactly model the brake disc. In order to do so, the researchers are still going on to find out transient thermal behavior of disc brake during braking applications. There is always a need of some assumptions to model any complex geometry. One makes these assumptions keeping in mind the difficulties involved in the theoretical calculation and the importance of the parameters that are taken and those which are ignored. In modeling we always ignore the things that are of less importance and have little impact on the analysis. The assumptions are always made depending upon the details and accuracy required in modeling.

Due to the application of brakes on the car disc brake rotor, heat generation takes place due to friction and this thermal flux has to be conducted and dispersed across the disc rotor cross section. The condition of braking is very much severe and thus the thermal analysis has to be carried out. The thermal loading as well as structure is axis-symmetric. Hence, axis-symmetric analysis can be performed, but in this study we performed 3-D analysis, which is an exact representation for this thermal analysis. Thermal analysis is carried out and with the above load structural analysis is also performed for analyzing the stability of the structure.

To simplify the analysis, several assumptions have also been made as follows [28]:

- All kinetic energy at disc brake rotor surface is converted into frictional heat or heat flux.
- The heat transfer involved for this analysis only conduction and convection process. The heat transfer radiation can be neglected in this analysis because of small amount which is 5% to 10% [29].
- The disc material is considered as homogeneous and isotropic.
- The domain is considered as axis-symmetric.
- Inertia and body force effects are negligible during the analysis.
- The disc is stress-free before the application of brake.
- In this analysis, the ambient temperature and initial temperature has been set to 20°C
- All other possible disc brake loads are neglected.
- Only certain parts of disc brake rotor will apply with convection heat transfer such as cooling vanes area, outer ring diameter area and disc brake surface
- Uniform pressure distribution by the brake pad onto the disc brake surface.

The thermal conductivity and specific heat are a function of temperature, Figures 3 and 4.

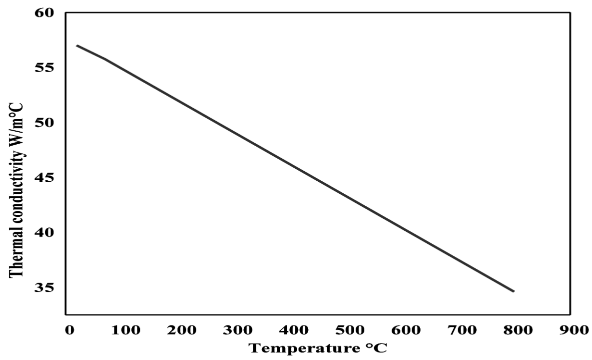



Fig. 3. Thermal conductivity versus temperature

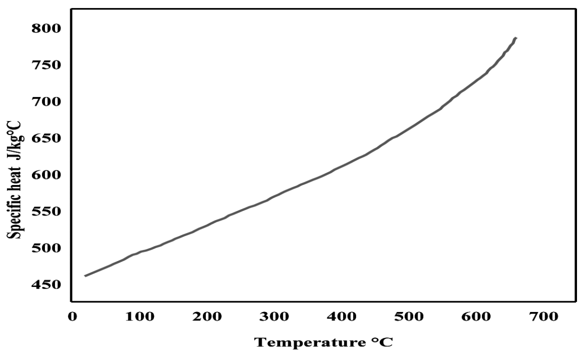



Fig. 4. Specific heat versus temperature

### 3.2. Mesh generation

For illustration purposes, Figure 5 shows the fluid mesh used for CFD analysis. In our case, we used a linear tetrahedral element with 30717 nodes and 179798 elements.

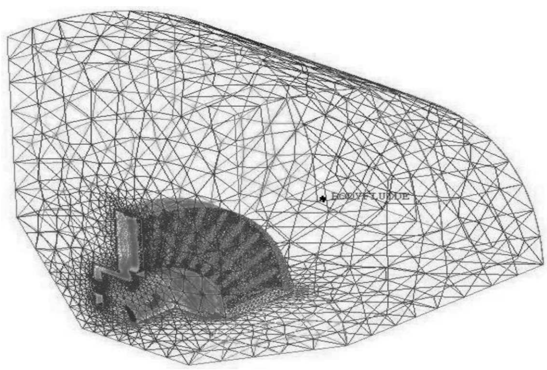


Fig. 5. Tetrahedral fluid mesh

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

### 3.3. Modeling in ANSYS CFX

The solution scheme employees the $\kappa$-$\varepsilon$ model with scalable wall function and sequential load steps. For the preparation of the mesh of CFD model, one defines initially various surfaces of the disc in ICEM CFD, as shown in Fig. 6 and Fig. 7; we used a linear tetrahedral element with 30717 nodes

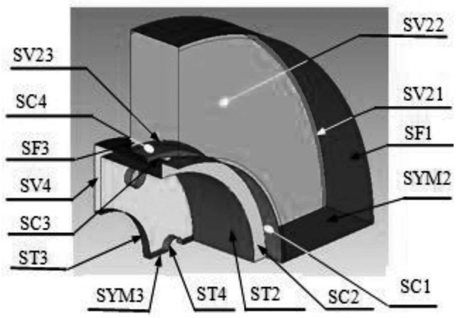



Fig. 6. Definition of surfaces of the full disc

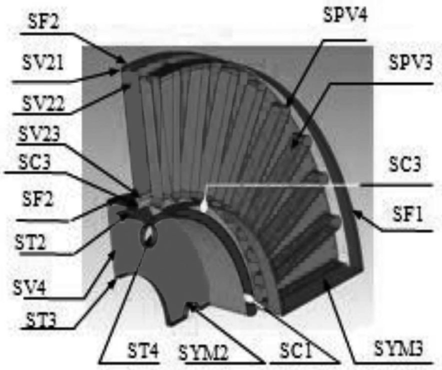



Fig. 7. Definition of surfaces of the ventilated disc

and 179798 elements. In order not to weigh down calculation, an irregular mesh is used in which the meshs are broader where the gradients are weaker (nonuniform mesh) (Fig. 8).

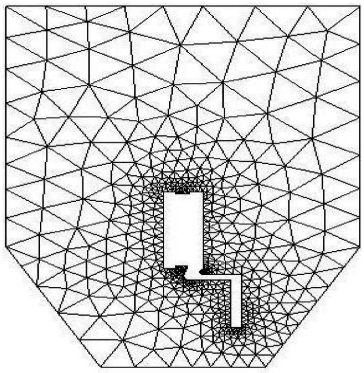


Fig. 8. Irregular mesh in the wall

The CFD models were constructed and were solved using ANSYS-CFX software package. The model applies periodic boundary conditions on the section sides and the radial and axial lengths of the air domain surrounding the disc. The disc is modeled attached to an adiabatic shaft whose axial length spans that of the domain. The air around the disc is considered at T∞ =20°C and open boundaries with zero relative pressure were used for the upper, lower and radial ends of the domain (Fig. 9)

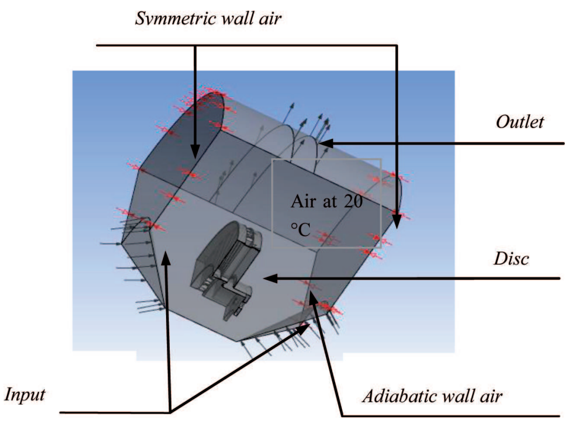



Fig. 9. Brake disc CFD model

The airflow through and around the brake disc was analyzed using the ANSYS CFX software package. The ANSYS-CFX solver automatically calculates heat transfer coefficient at the wall boundary. Afterwards, the heat transfer coefficients considering convection were calculated and organized in such a way that they could be used as a boundary condition in thermal analysis. Averaged heat transfer coefficient had to be calculated for all disc using ANSYS CFX-Post as it is indicated in Figs. 10 and 11.

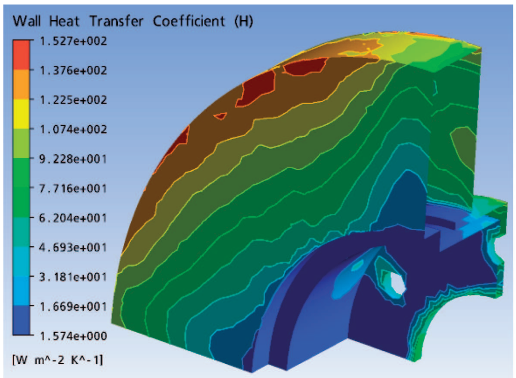



Fig. 10. Distribution of heat transfer coefficient on a full disc in the steady state case (FG 15)

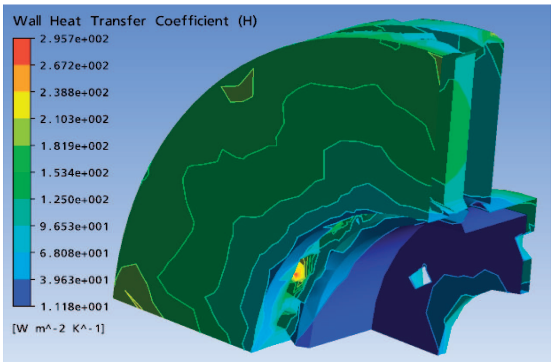



Fig. 11. Distribution of heat-transfer coefficient on a ventilated disc in the stationary case (FG 15)

a) *Results of the calculation of the heat-transfer coefficient (h)*

The heat-transfer coefficient is a parameter related with the velocity of air and the shape of brake disc and many other factors. In different velocity of air, the heat transfer coefficient in different parts of brake disc changes with time [30]. Heat transfer coefficient will depend on air flow in the region of brake rotor and vehicle speed, but it does not depend on material.

From the comparison between Figs. 12 and 13 concerning the variation of heat-transfer coefficients in the nonstationary mode for the two types of design full and ventilated disc; one notes that the introduction of the system of ventilation directly influences the value of this coefficient for same surface, which is logically significant because this mode of ventilation results in the reduction in the differences in wall-fluid temperatures.

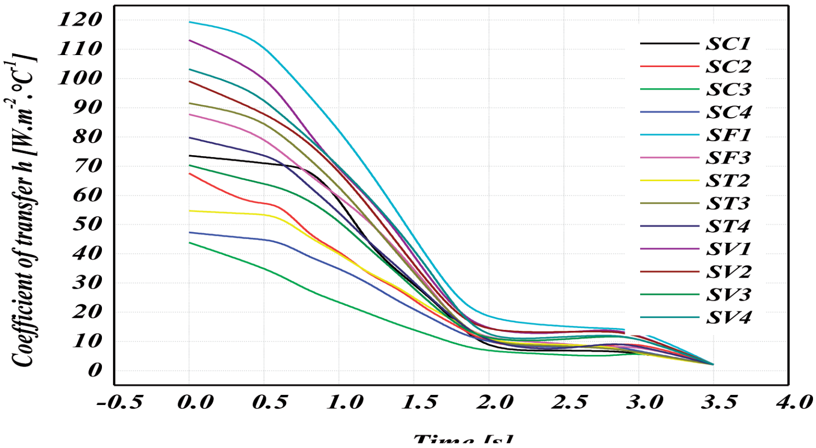



Fig. 12. Variation of heat-transfer coefficient ($h$) of various surfaces for a full disc in the nonstationary case (FG 15)

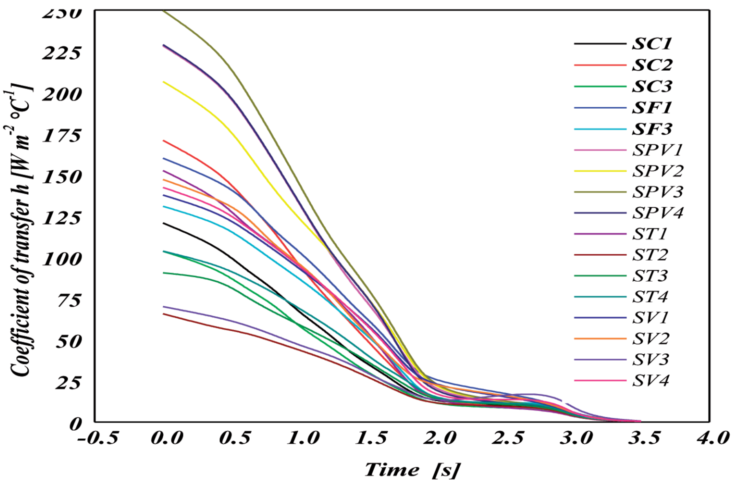



Fig. 13. Variation of heat-transfer coefficient ($h$) of various surfaces for a ventilated disc in transient case (FG 15)

### 3.4. 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 Fig. 14. The variation of the heat flux during the simulation time is represented in Fig. 15.

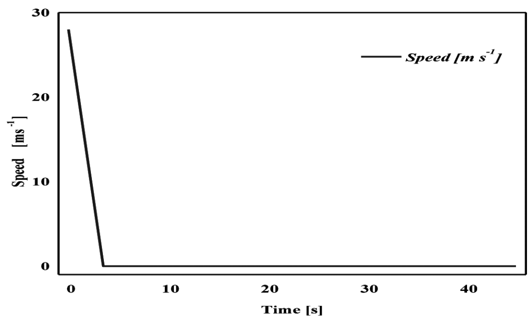



Fig. 14. Speed of braking versus time (braking of type 0)

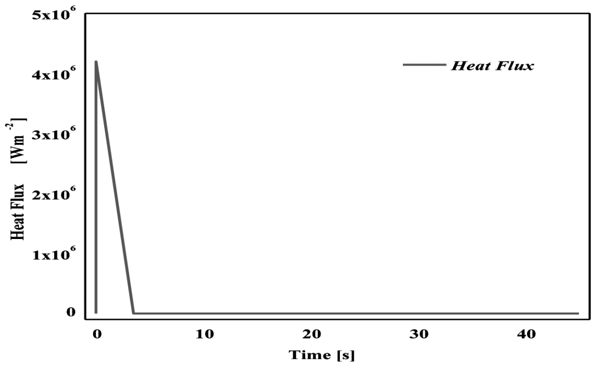



Fig. 15. Heat Flux versus time

### 3.5. Meshing of the disc

The elements used for the mesh of the full and ventilated disc are tetrahedral 3D elements with 10 nodes (isoparametric) (Fig. 16).

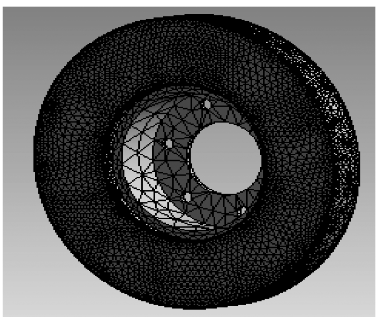
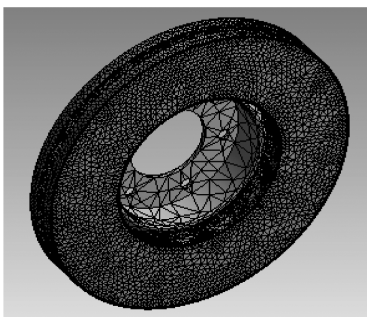
(a)  (b) 

Fig. 16. Meshing of the disc (a) full disc (172103 nodes -114421 elements (b) ventilated disc (154679 nodes- 94117 elements)

### 3.6. Initial and boundary conditions

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

- Total time of simulation = 45 [s]
- Increment of initial time = 0.25 [s]
- Increment of minimal initial time = 0.125 [s]
- Increment of maximal initial time = 0.5 [s]
- Initial temperature of the disc = 60 [°C]
- Materials: Grey Cast iron FG 15.
- Convection: One introduces the values of heat-transfer coefficient (*h*) obtained for each surface in the shape of a curve (Figs. 12, 13)
- Flow: One introduces the values obtained of flux entering by means of the code CFX.

## 4. Results and discussions

### 4.1. Influence of construction of the disc

Fig. 17 shows the variation in the temperature according to time during the simulation. From the first step, the variation in the temperature shows a great growth which is due to the speed of the physical course of phenomenon during braking, namely friction, plastic microdistortion of contact surfaces.

For the full disc, the temperature reaches its maximum value of 401.55°C at the moment $t$ = 1.8839 s and then it falls rapidly to 4.9293 s – as of this moment and to the end (t = 45 s) of simulation, the variation in the temperature becomes slow. It is noted that the interval [0-3.5] s represents the phase of forced convection. During this phase, one observes the case of the free convection until the end of the simulation. In the case of the ventilated disc, one observes that the temperature of the disc falls approximately by 60°C compared with the first case. It is noted that the ventilation in the design of the discs of brake plays an important role in producing a better system of cooling.

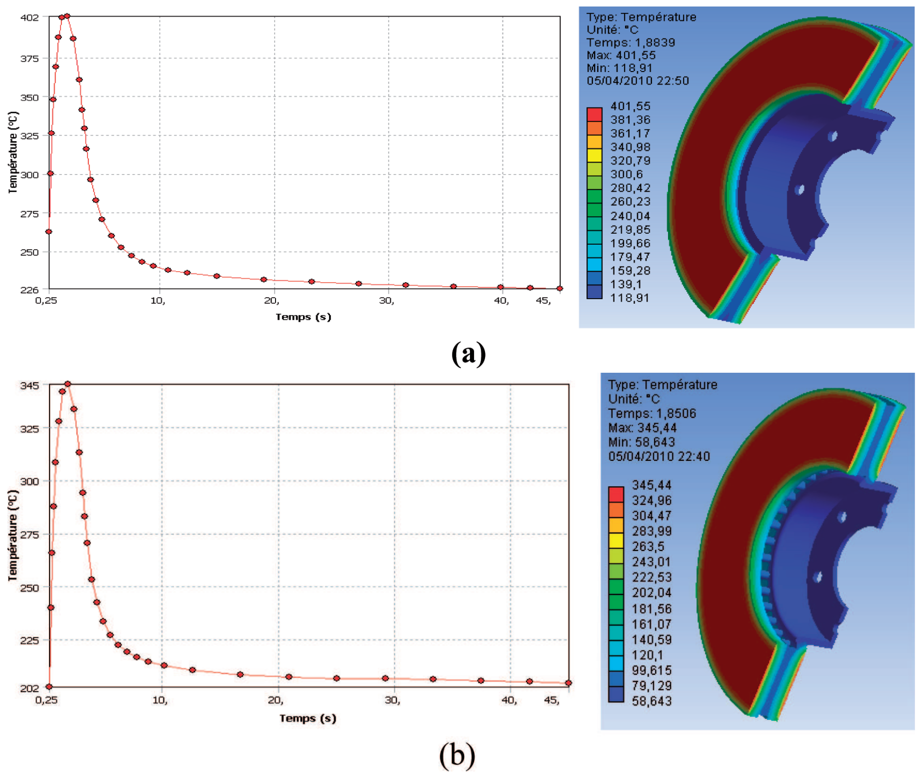



Fig. 17. Temperature distribution of a full (a) and ventilated disc (b) of cast iron(FG 15)

## 5. Coupled Thermomechanical Analysis

Disc brakes operate by pressing a set of brake pads against a rotating brake disc; the frictional forces cause deceleration. The dissipation of the frictional heat generated is critical for effective braking performance. Temperature changes of the brake cause axial and radial deformation; and then change in shape, in turn, affects the contact between the pads and the disc. Thus, the system should be analyzed as a fully coupled thermomechanical system.

### 5.1. Coupled Field Analysis

In the present study, the structural and thermal analyses are coupled.The mathematical representation [31] of coupling between the finite element equations of two analyses is

$$\begin{bmatrix} M & 0 \\ 0 & 0 \end{bmatrix} \begin{Bmatrix} \{\ddot{u}\} \\ \{\ddot{T}\} \end{Bmatrix} + \begin{bmatrix} C & 0 \\ 0 & C^t \end{bmatrix} \begin{Bmatrix} \{\dot{u}\} \\ \{\dot{T}\} \end{Bmatrix} + \begin{bmatrix} K & 0 \\ 0 & K^t \end{bmatrix} \begin{Bmatrix} \{u\} \\ \{T\} \end{Bmatrix} = \begin{Bmatrix} \{F\} + \left\{F^{th}\right\} \\ \{Q\} \end{Bmatrix} \tag{11}$$

where

$$K^t = K^{tb} + K^{tc}c$$

$$\{F\} = \left\{F^{nd}\right\} + \{F^{pr}\} + \{F^{ac}\}$$

$$\{Q\} = \left\{Q^{nd}\right\} + \{Q^g\} + \{Q^c\}$$

This is called a weak coupling and at least two iterations are required to achieve a coupled response. In the present analysis, the matrices $M$ and $C$ are null matrices and the force vector due to acceleration effects $\{F^{ac}\}$ is zero because the transient effects on structural degrees of freedom are not considered. The thermal distortions are included in the analysis by thermal strain load vector $\left\{F^{th}\right\}$ in equation (11).

### 5.2. FE model and boundary conditions

The 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.

To express the heat transfer in the disc brake model, several thermal boundary conditions and initial condition need to be defined. As shown in Figure 18, at the interface between the disc and brake pads, heat is generated due to sliding friction, which is shown in dashed lines. For the exposed region of the disc and brake pads, it is assumed that heat is exchanged with the environment through convection. Therefore, convection surface boundary condition is applied there. At the surface of the back plate, adiabatic or insulated surface boundary condition is used and (Figure 18).

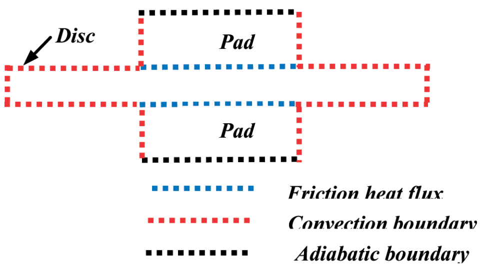



Fig. 18. Thermal boundary conditions applied on disc brake

The various boundary conditions in embedded configurations are imposed on the model (disc-pad), taking into account its environment direct, are respectively, the simple case as shown in Fig. 19.

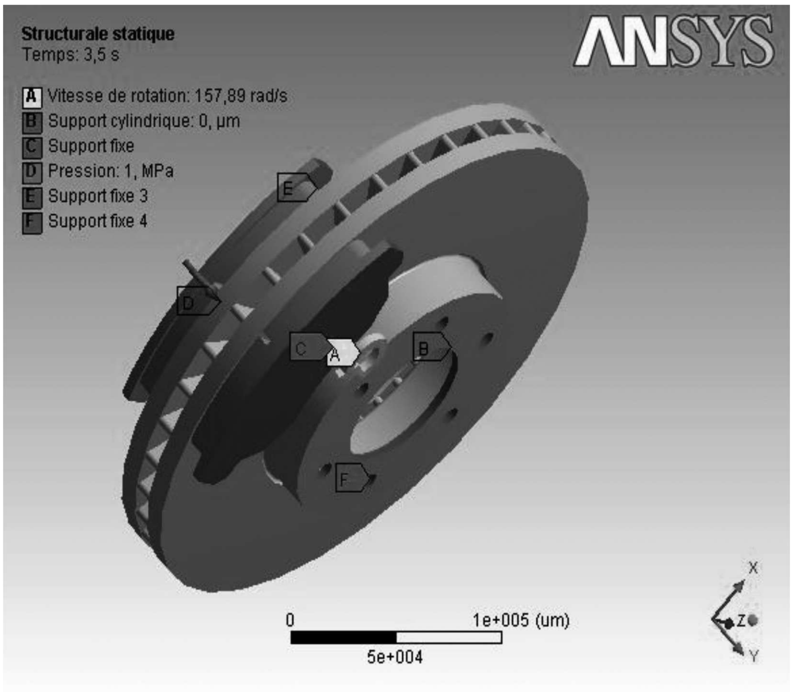



Fig. 19. Boundary conditions and loading imposed on the disc-pads

A fully coupled thermomechanical model was set up to predict the temperature changes of the brake disc shape caused by axial and radial deformation. Thermal conduction and convective heat transfer were the two modes of heat transfer considered. The convection heat transfer coefficient values is represented by the curves of Figure 13 over all exposed surfaces of the disc, and radiative heat transfer is considered negligible [32].

The initial air temperature of the disc and pads is 20°C, and the surface convection condition is applied at all surfaces of the disc, with the values of the exchange coefficient calculated previously and the convection coefficient ($h$) of 5 W/m$^2$°C is applied to the surfaces of the two pads. The heat flux into the brake disc during a braking process can be calculated by the formula described in the first part. The FE mesh is generated using 3D tetrahedral element with 10 nodes (solid 187) for the disc and pads. Overall, 185901 nodes and 113367 elements are used (Fig. 20).

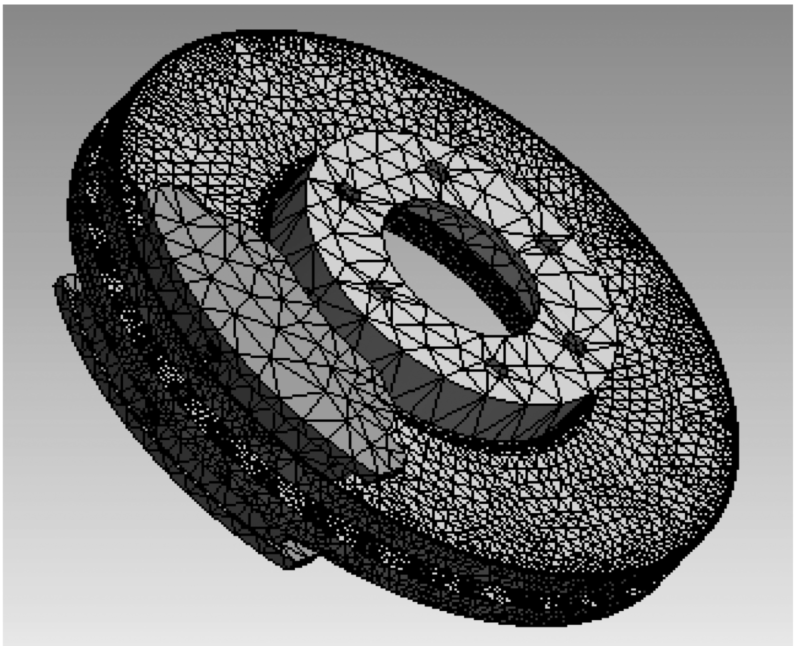


Fig. 20. Refined mesh of the model

A frictional contact pair was defined between disc-pad interfaces. The element types used were Quadratic Triangular Contact (Conta174) and Quadratic Triangular Target (Targe 170). The mechanical properties of the most common disc brakes (grey cast iron) and pad materials used in the analysis are given in Table 2.

The thermal coupling will be carried out by a thermal condition with nonuniform temperature by holding the thermal environment of our model. For this reason, the order"thermal condition,,will be used to deal with the thermomechanical coupled problem and to manage the transient state.

In this study, a transient thermal analysis will be carried out to investigate the temperature variation across the disc using ANSYS software. Further structural analysis will also be carried out by coupling thermal analysis.

### 5.3. Structural boundary conditions applied on disc

The structural conditions for the FE model consist of a fixed support, cylindrical. The fixed support is placed on the bolt holes disc where the rotational mouvment is allowed in the y-axis, its angular velocity is constant $\omega$ = 157.89 rad/s. The cylindrical support is placed in the model where the bearings would normally sit and constrain the assembly to rotation only of which the degrees of freedom are:

| | |
|---|---|
| Radial | Free |
| Axial | Fixed |
| Tangential | Fixed |

### 5.4. Structural boundary conditions and loading applied on pads

The boundary conditions applied on the pads are defined according to the movements authorized by the caliper. Indeed, one of the roles of the caliper is to retain the pads which have the natural tendency to follow the movement of the disc when the two structures are in contact. The caliper maintains also the pads in direction Z.

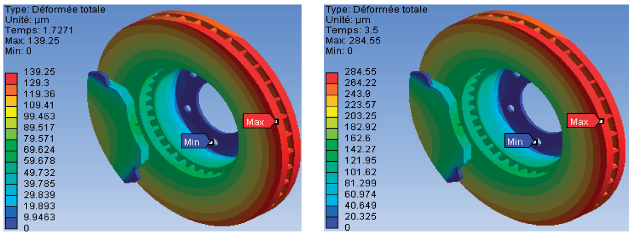
Thus, the conditions imposed on the pads are:

- The pad is embedded on its edges in the orthogonal plan on the contac surface, thus authorizing, a movement of rigid body in normal direction with the contact such as, one can find it, in automobile brake assembly [33].
- A fixed support in the finger pad.
- A pressure $P$ of 1 MPa applied on the piston pad.
- The friction contact conditions are applied between the surfaces of the disc and pads with the friction coefficient ($\mu$) of 0.2.

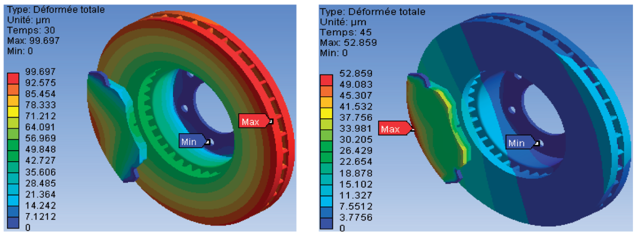
## 6. Thermo-mechanical analysis results

### 6.1. Thermal deformation

Fig. 21 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 to 284.55 $\mu$m. The value of maximum displacement recorded during this simulation is at the moment $t$ =3.5 s, which corresponds to the time of braking. One observes a strong distribution which increases with time on the friction tracks and the external crown and the cooling fins of the disc. Indeed, during a braking moment, 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.




(a) t=1.7271 s (b) t=3.5 s

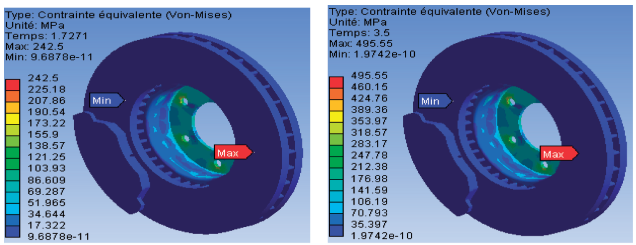



(c) t=30 s (d) t=45 s

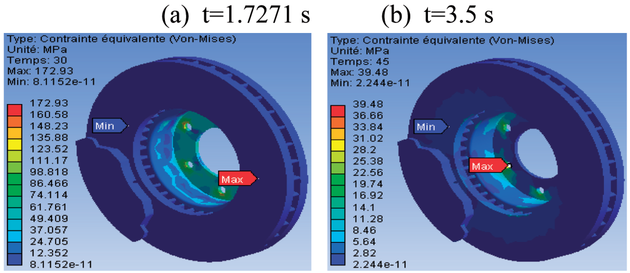
Fig. 21. Total distortion distributio

## 6.2. Von Mises stress distribution

Fig. 22 presents the distribution of the stress equivalent of Von Mises for various moments of simulation, the scale of values varying from 0 to




(a) t=1.7271 s (b) t=3.5 s



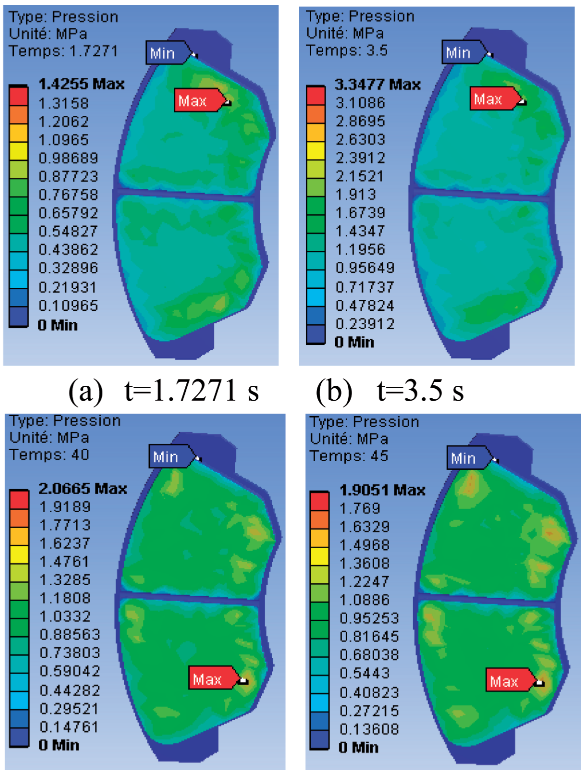

(c) t=30 s (d) t=45 s

Fig. 22. Von Mises stress distribution

495.56 MPa. The maximum value recorded during this simulation of the thermomechanical coupling is very significant compared to that obtained with the assistance in the mechanical analysis under the same conditions. One observes a strong constraint on the level of the bowl of the disc. Indeed, the disc is fixed to the hub of the wheel by screws, thus preventing its movement. In the presence of the rotation of the disc and the requests of torsional stress and sheers generated at the level of the bowl which are able to create the stress concentrations. The repetition of these effects will lead to risks of rupture on the level of the bowl of the disc.

### 6.3. Contact pressure

Fig.23 shows the contact pressure distribution in the friction interface of the inner pad taken at various times of simulation. For this distribution, the scale varies from 0 to 3.3477 MPa and reached a value of pressure at the moment $t$ =3.5 s, which corresponds to the null rotational speed. It is also




(a) t=1.7271 s (b) t=3.5 s

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

Fig. 23. Contact pressure distribution in the inner pad

noticed that the maximum contact pressure is located on the edges of the pad decreasing from the leading edge toward the trailing adge from friction. This pressure distribution is almost symmetrical compared with 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 of 3D, the pressure produces the symmetric field of the temperature. This last one affects thermal dilation and leads to a variation of the contact pressure distribution.

## 7. Conclusion

In this article, we have presented the analysis of thermomechanical behavior of the dry contact between the disc and pads during the braking process; the modeling is based on the ANSYS 11.0. We have shown that the ventilation system plays an important role in cooling the discs and provides a good high-temperature resistance.

The analysis results showed that temperature field and stress field in the process of braking phase were fully coupled. The temperature, the Von Mises stress and the total deformations of the disc and contact pressures of the pads increases as the thermal stresses are apart from the mechanical stress which causes the crack propagation and fracture of the bowl and wear of the disc and pads.

Regarding the calculation results, we can say that they are satisfactorily in agreement with those commonly found in the literature investigations. It would be interesting to solve the problem in thermomechanical disc brakes with an experimental study to validate the numerical results, for example, on test benches, in order to demonstrate a good agreement between the model and reality.

Regarding the outlook, there are three recommendations for the expansion of future work related to disc brake that can be done to better understand the effects of thermomechanical contact between the disc and pads. tThe recommendations are as follows:

- Experimental study to verify the accuracy of the numerical model developed.
- Tribological and vibratory study of the contact disc – pads;
- Study of dry contact sliding under the macroscopic aspect (macroscopic state of the surfaces of the disc and pads).

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REFERENCES


[1] Valvano T., Lee K.: An Analytical Method to Predict Thermal Distortion of a Brake Rotor. *Society of Automotive Engineers*, Inc 2000-01-0445, 2000.

[2] Choi Ji-Hoon and In Lee: "Finite element analysis of transient thermoelastic behaviors in disk brakes", Wear, V 257, pp. 47-58 (2004).

[3] Talatia Faramarz and Jalalifar S.: Analysis of heat conduction in a disk brake system. J *Heat Mass Transfer* 45: 1047-1059 (2000).

[4] Naji M., Al-Nimr M., Masoud S.: Transient 5thermal behavior of a cylindrical brake system. *J Heat Mass Transf* 36:45-49 (2000).

[5] Mosleh M., Blau P.J., Dumitrescu D.: Characteristics and morphology of wear particles from laboratory testing of disk brake materials. J *Wear* 256:1128-1134 (2004).

[6] Mutlu I., Alma M.H., Basturk M.A.: Preparation and characterization of brake linings from modified tannin-phenol formaldehyde resin and asbestos-free fillers. *J Mat Sci* 40(11): 3003-3005 (2005).

[7] Hecht R.L., Dinwiddie R.B., Wang H.: The effect of graphite flake morphology on the thermal diffusivity of gray cast irons used for automotive brake discs. *J Mat Sci* 34(19):4775-4781 (1999).

[8] Gudmand-Høyer L., Bach A., Nielsen G.T., Morgen P.: Tribological properties of automotive disc brakes with solid lubricants. J *Wear* 232(2):168-175 (1999).

[9] Uyyuru R.K., Surappa M.K., Brusethaug S.: Tribological behavior of Al-Si-SiCp composites/automobile brake pad system under dry sliding conditions. *J Tribol Int* 40(2):365-373 (2007).

[10] Cho M.H., Cho K.H., Kim S.J., Kim D.H., Jang H.: The role of transfer layers on friction characteristics in the sliding interface between friction materials against gray iron brake disks. *Trib Lett* 20(2):101-108 (2005).

[11] Boz M., Kurt A.: The effect of Al2O3 on the friction performance of automotive brake friction materials. *J Tribo Int* 40(7):1161-1169 (2007).

[12] Blau P.J., McLaughlin J.C.: Effects of water films and sliding speed on the frictional behavior of truck disc brake material. *Trib Int* 36(10):709-715 (2003).

[13] McPhee A.D., Johnson D.A.: Experimental heat transfer and flow analysis of a vented brake rotor. *Int J Thermal Sci* 47(4):458-467 (2007).

[14] Wallis L., Leonardi E., Milton B., Joseph P.: Air flow and heat transfer in ventilated disk brake rotors with diamond and tear-drop pillars. *Numer Heat Transf Part A* 41:643-655 (2002). URL:

[15] Johnson D.A., Sperandei B.A., Gilbert R.: Analysis of the flow through a vented automotive brake rotor. *J Fluids Eng* 125:979-986 (2003).

[16] Kang S.S., Cho S.K.: Thermal deformation and stress analysis of disk brakes by finite element method *Journal of Mechanical Science and Technology* 26 (7) (2012) 2133-2137 (2012).

[17] Thilak V.M.M., Krishnaraj R.: Transient thermal and structural analysis of the rotor disc of disc brake, *International Journal of Scientific & Engineering Research* Volume 2, Issue 8, August-2011, ISSN 2229-5518, (2011).

[18] Yildiz Y., Duzgun M.: Stress analysis of ventilated brake discs using the finite element method.International Journal of Automotive Technology, Vol. 11, No. 1, pp. 133-138 (2010).

[19] Moses Omolayo Petinrin and John Ogheneortega Oji: "Numerical Simulation of Thermoelastic Contact Problem of Disc Brake with Frictional Heat Generation„New York Science Journal 2012; 5(10), 39-43.

[20] Park S.P., Chai T.W., J.B. and Chung W.S.: "Thermo-mechanical finite element analysis of hot judder phenomenon of a ventilated disc brake system„Int. J.Precis.Eng.Manuf., Vol. 12, No.5, pp. 821-828, 2011.

[21] Hassan M.Z., Brooks P.C., and Barton D.C.: Fully coupled thermal-mechanical analysis of automotive disc brake squeal. In Proceedings of the Automotive Research Conference, Huddersfield, UK, January 2008, paper 05AARC2008.
[22] Li L., Ouyang H., and Abu Baker A.R.: Transient analysis of car disc brake squeal with temperature effects.In Proceedings of the Automotive Research Conference, Huddersfield, UK, January 2008, paper 09AARC2008.
[23] Beeker A.A.: The Boundary Element Method in Engineering, McGraw-Hill, New York, (1992).
[24] Cruceanu C.: Frâne pentru vehicule feroviare (Brakes for railway vehicles), Ed. MATRIXROM, Bucureşti, 2007, ISBN 978-973-755-200-6, 388 pag.
[25] Reimpel J.: Technologie de freinage. Vogel Verlag, Würzburg, 1998.
[26] Gotowicki, Pier Francesco; Nigrelli, Vinzenco; Mariotti, Gabriele Virzi: Numerical and experimental analysis of a pegs- wing ventilated disk brake rotor, with pads and cylinders, 10 th EAEC *European Automotive Congress* – Paper EAEC05YUAS04– P 5, June 2005.
[27] Ameer Fareed Basha Shaik, Ch. Lakshmi Srinivas: Structural and thermal analysis of disc brake with and without crossdrilled rotar of race car. International Journal of Advanced Engineering Research and Studies, IJAERS/Vol. 1 (4) July-Sept., 2012 pp 39-
[28] Khalid M.K., Mansor M.R., Abdul Kudus S.I., Tahir M.M., and Hassan M.Z. Performance Investigation of the UTeM Eco- Car Disc Brake System International Journal of Engineering & Technology IJET-IJENS Vol. 11, No. 06, pp. 1-6.
[29] Limpert R.: *Brake Design and Safety*. $2^{nd}$ Edition, Warrendale, Pennsylvania: Society of Automotive Engineering Inc., 1999, pp. 137-144.
[30] Zhang J., Xia C.: Research of the TransientTemperatureField and Friction Properties on Disc Brakes, Proceedings of the 2012 $2^{nd}$ International Conference on Computer and Information Application (ICCIA 2012) 201-204 (2012).
[31] *ANSYS user's manual,* (ANSYS Inc. Canonsburg, USA).
[32] Sarip S.B.: "Lightweight friction brakes for a road vehicle with regenerative braking„Ph.D. Thesis, Engineering, Design and Technology, Bradford University, 2011.
[33] Nicolas Coudeyras: „Non-linear analysis of multiple instabilities to the rubbing interfaces:application to the squealing of brake"„PhD Thesis, Central school of Lyon-speciality: Mechanics, December 2009.

## Analiza strukturalna i termiczna wirnika samochodowego hamulca tarczowego

S t r e s z c z e n i e

Głównym celem pracy jest analiza zjawisk termomechanicznych w obszarze suchego styku pomiędzy tarczą hamulca a klockiem ciernym występujących w fazie hamowania. Strategia symulacji jest oparta na oprogramowaniu ANSYS11. Wyniki modelowania stanów przejściowych temperatury w tarczy hamulca wykorzystano do identyfikacji geometrycznych współczynników projektowych, użytych do instalacji systemu wentylacji. Sprzężona analiza strukturalno-termiczna jest następnie użyta do wyznaczenia deformacji tarczy, naprężenia Von Misses'a, oraz rozkładu ciśnienia kontaktowego w klockach ciernych. Wyniki analizy wykazują zadowalającą zgodność z podobnymi podawanymi w literaturze.