Available online at www.elixirpublishers.com (Elixir International Journal)

Mechanical Engineering

Elixir Mech. Engg. 55A (2013) 13244-13250

ISSN: 2229-712X

Thermal- mechanical coupled analysis of a brake disc rotor

Ali Belhocine and Mostefa Bouchetara

Faculty of Mechanical Engineering, University of Sciences and the Technology of Oran, L.P 1505 El -MNAOUER, USTO 31000 ORAN (Algeria)

ARTICLE INFO

Article history:

Received: 11 September 2012;

Received in revised form:

1 February 2013;

Accepted: 19 February 2013;

Keywords

Brake discs,

Heat flux,

Heat transfer coefficient,

Von Mises stress,

Contact pressure.

ABSTRACT

The main purpose of this study is to analysis the thermomechanical behavior of the dry contact between the brake disc and pads during the braking phase. The thermal-structural analysis is then used coupling to determine the deformation and the Von Mises stress established in the disc, the contact pressure distribution in pads. The results are satisfactory when compared with those of the specialized literature.

Introduction

The braking process is in fact the matter of energy balance. The aim of braking system is to transform mechanical energy of moving vehicle into the some other form, which results by decreasing of the vehicle speed. The kinetic energy is transformed into the thermal energy, by using the dry friction effects and, after that, dissipated into the surroundings [1]. Have developed for a few decades at intervals raised in many sectors :nuclear power, space, aeronautics, automotive, petro chemistry, 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 discs 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 analyzed. 3D unsteady heat transfer analyses were conducted 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 & Lee [3] did a study of 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 & Ruhl [4] did a study of the air flow through the passage of a Chrysler LH platform ventilated brake rotor. Modifications to the production rotor's vent inlet geometry were prototyped and measured in addition to the production rotor. Vent passage air flow was compared to existing correlations. With the aid of Chrysler Corporation, investigation of ventilated brake rotor vane air flow was 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 the surface of materials in slip, support physicochemical and microstructural transformations and modify the rheology of the interface 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 .Many other works [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 characterize the thermal parameters which govern the heat transfer at the vicinity of a sliding interface. In certain industrial applications, the solids are provided with a surface coating. A recent study has been carried out to analyze the effect of surface coating on the thermal behavior of a solid subjected to the friction process [12]. Increased thermal efficiency and the integrity of materials in high-temperature environments is an essential requirement in modem engineering structures in, automotive, aerospace, nuclear, offshore, environmental and other industries. Nowadays, the finite element method is used regularly to obtain numerical solutions for heat transfer problems. The most common choice when using finite elements is a standard Galerkin formulation [13]

In this work, we will make a modeling of the thermomechanical behavior of the dry contact between the disc of brake pads at the time of braking phase, the strategy of calculation is base on the software Ansys 11 [14]. This last is elaborate mainly for the resolution of the complex physical problems. The numerical simulation of the coupled transient thermal field and stress field is carried out by sequentially thermal-structurally coupled method based on Ansys.

Tele:

E-mail addresses: al.belhocine@yahoo.fr

Heat flux entering the disc

In the case of disc brake, the effective friction processes between the pads and the disc are extremely complex due to the fact that the present time brake pads, due to their composite structure [15], do not have constant chemical-physic proprieties, the organic contained elements being subject of a series of transformations under the influence of temperature increase. The heat distribution between the brake disc and the friction pads is mostly dependent on material characteristics, among whom a major influence is due to the density $\rho_{d,p}$ [kg/m^3], the thermal conductivity $k_{d,p}$ [W/m.K] and the specific heat $C_{d,p}$ [J/kg.K] of the disc's (index *d*) and braking pad's materials respectively (index *p*). Denoting Q_d and Q_p [J] the heat quantities assumed by the disc and the braking pads respectively, one could be expressed in the following manner [16]

$$\frac{Q_d}{Q_p} = \frac{\sqrt{\rho_d k_d C_d}}{\sqrt{\rho_p k_p C_p}} \qquad (1)$$

Because the braking disc is not entirely covered by the friction pads, within computing we have to consider the ratio between the disc surface S_d and the pad surface S_p. Denoting the ratio of heat's division between the disk and pads with:

$$\varphi_c = \frac{Q_d S_d}{Q_p S_p} = \frac{\sqrt{\rho_d k_d C_d}}{\sqrt{\rho_p k_p C_p}} \cdot \frac{S_d}{S_p} \qquad (2)$$

and considering *Q* [J] the heat quantity generated during the friction process, the heat quantities assumed by the pads and by the disc are:

$$\varphi_d = Q.\frac{\varphi_C}{1+\varphi_C} \qquad (3)$$

$$\varphi_p = Q.\frac{1}{1+\varphi_C} \qquad (4)$$

The brake disc assumes the most part of the heat, usually more than 90% [17], through the effective contact surface of the friction coupling. Considering the complexity of the problem and average data processing limited, one replaced the pads by their effect, represented by an entering heat flux (Fig.1).

The initial heat flux q_0 entering the disc is calculated by the following formula [18]:

$$q_0 = \frac{1-\emptyset}{2} \cdot \frac{m\, g\, v_0\, z}{2\, A_d\, \varepsilon_p} \qquad (5)$$

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

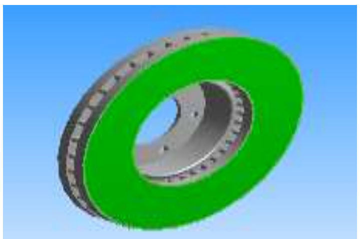


Fig.1. Application of flux

The loading corresponds to the heat flux on the disc surface. The dimensions and the parameters used in the thermal calculation are recapitulated in Table 1.

Table 1. Parameters of automotive brake application

Inner disc diameter, mm	66
Outer disc diameter, mm	262
Disc thickness (TH) ,mm	29
Disc height (H) ,mm	51
Vehicule mass m , kg	1385
Initial speed v_0 , km/h	28
Deceleration a , m/s^2	8
Effective rotor radius R_{rotor} ,mm	100.5
Rate distribution of the braking forces Φ , %	20
Factor of charge distribution of the disc ε_p	0.5
Surface disc swept by the pad A_d , mm^2	35993

The disc material is gray cast iron (GFC) with high carbon content [19], with good thermophysical characteristics and the brake pad has an isotropic elastic behavior whose thermo-mechanical characteristics adopted in this simulation in the of the two parts are recapitulated in Tab 2.

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

Thermal analysis of the problem

Transient heat conduction in three dimensional heat transfer problem is governed by the following differential equation [20]

$$-\left(\frac{\partial q_x}{\partial x} + \frac{\partial q_y}{\partial y} + \frac{\partial q_z}{\partial z}\right) + Q = \rho C_p \frac{\partial T}{\partial t} \qquad (6)$$

Where q_x, q_y and q_z are conduction heat fluxes in x, y and z-directions, respectively, C_p is the specific heat, ρ is the specific mass, *Q* is internal heat generation rate per unit volume and *T* is the temperature that varies with the coordinates as well as the time *t*. The conduction heat fluxes can be written in the form of temperature using Fourier's law. Assuming constant and uniform thermal properties, the relations are:

$$q_x = -k_x \frac{\partial T}{\partial x}, q_y = -k_y \frac{\partial T}{\partial y}, \qquad q_z = -k_z \frac{\partial T}{\partial z} \qquad (7)$$

Where k_x, k_y and k_z are thermal conductivity in x, y and z-directions, respectively. Heat transfer boundary conditions consist of several heat transfer modes that can be written in different forms. The boundary conditions frequently encountered are as follows [21, 22]:

$$T_s = T_1(x, y, z, t) \qquad (8)$$
$$-q_s = h(T_s - T_\infty) \qquad (9)$$

Where T_1 is the specified surface temperature; q_s the specified surface heat flux (positive into a surface); h the convective heat transfer coefficient; T_s the unknown surface temperature, and T_∞ the convective exchange temperature.

Modeling in ANSYS CFX

The finite volume method consists of three stages; the formal integration of the governing equations of the fluid flow over all the (finite) control volumes of the solution domain. Then discretisation, involving the substitution of a variety of finite-difference-type approximations for the terms in the integrated equation representing flow processes such as convection, diffusion and sources. This converts the integral equation into a system of algebraic equations, which can then be solved using iterative methods [23]. The first stage of the process, the control volume integration, is the step that distinguishes the finite volume method from other CFD methods. The statements resulting from this step express the ‘exact’ conservation of the relevant properties for each finite cell volume. This gives a clear relationship between the numerical analogue and the principle governing the flow.To enable the modeling of a rotating body (in this case the disc) the code employs the rotating reference frame technique. For the preparation of the mesh of CFD model, one defines initially, various surfaces of the disc in ICEM CFD as the Fig.2 shows it, we used a linear tetrahedral element with 30717 nodes and 179798 elements. In order not to weigh down calculation, an irregular mesh is used in which the mesh is broader where the gradients are weaker (non-uniform mesh), (Fig. 3).

The CFD models were constructed and were solved using ANSYS-CFX software package [24]. The model applies periodic boundary conditions on the section sides. As the brake disc is made from sand casted grey cast iron, The disc model is attached to an adiabatic shaft whose axial length spans that of the domain. Air around the disc is considered to be 20 °C, and open boundaries with zero relative pressure were used for the upper, lower and radial ends of the domain. Material data were taken from Ansys material data library for air at 20 °C.Reference pressure was set to be 1 atm, turbulence intensity low and turbulent model used was k-ε. (Fig.4)

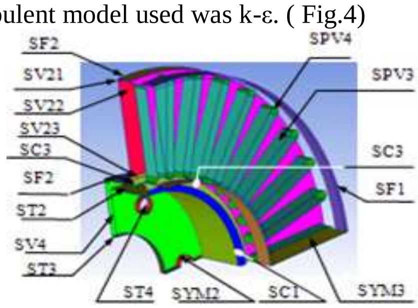


Fig 2. Definition of surfaces of the ventilated disc

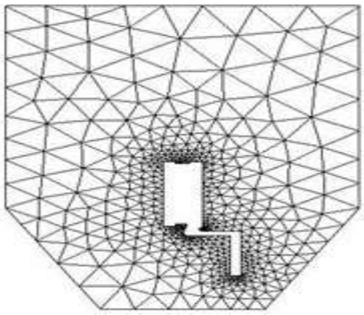


Fig.3. Irregular mesh in the wall

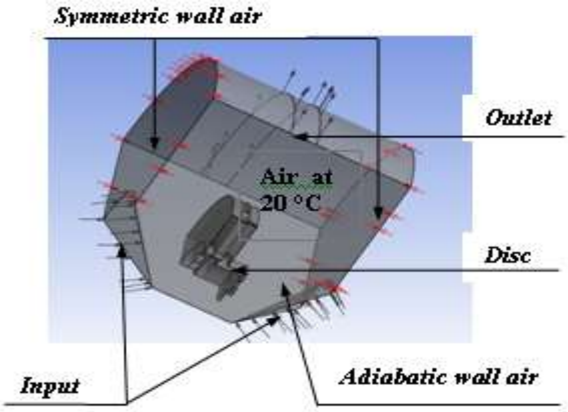


Fig.4 Brake disc CFD model

The airflow through and around the brake disc was analysed using the ANSYS CFX software package. The Ansys CFX solver automatically calculates heat transfer coefficient at the wall boundary .Afterwards the heat transfer coefficients considering convection were calculated and organized in such a way, that they could be used as a boundary condition in thermal analysis. Averaged heat transfer coefficient had to be calculate for all disc using Ansys CFX Post as it is indicated in Fig.5.

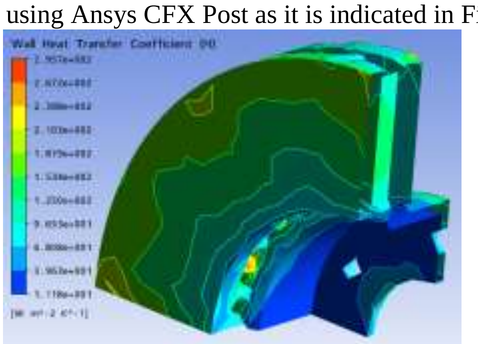


Fig.5 Distribution of heat transfer coefficient on a ventilated disc in the stationary case (FG 15).

a)Results of the calculation of the coefficient h

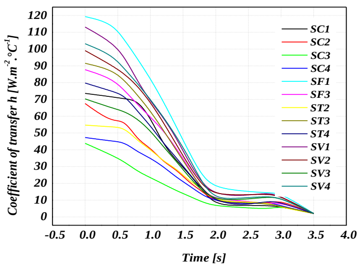


Fig.6 Variation of heat transfer coefficient (h) of various surfaces for a full disc in the non stationary case (FG 15)

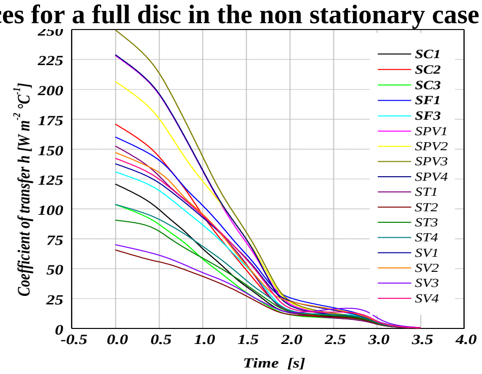
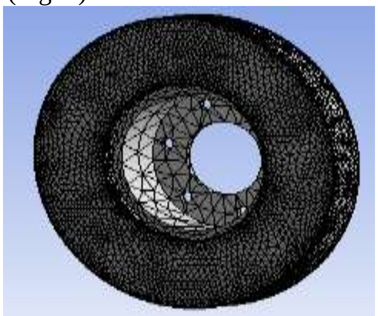


Fig. 7 Variation of heat transfer coefficient (h) of various surfaces for a ventilated disc in transient case (FG 15)

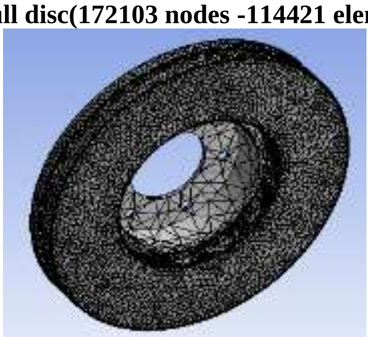
The comparison between Figs.6 and 7 concerning the variation of heat transfer coefficient in the non stationary mode for the two types of design full and ventilated, one notes that the introduction of the system of ventilation influences directly the value of this coefficient for same surface what is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

Meshing of the disc

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



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



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

Fig.8 Meshing of the disc

Initial and boundary conditions

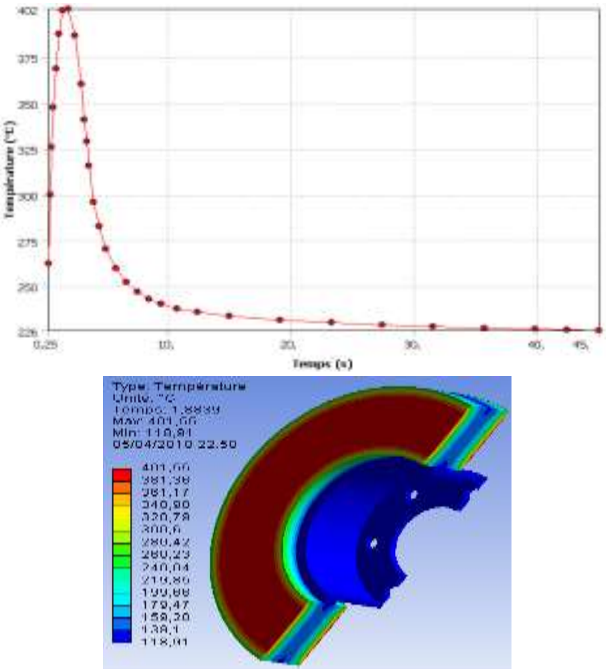
The boundary conditions are introduced into module ANSYS Workbench [Multiphysics], by choosing the mode of first simulation of the all (permanent or transitory), and by defining the physical properties of materials.These conditions constitute the initial conditions of our simulation.After having fixed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

- Total time of simulation = 45 [s]
- Increment of initial time = 0.25 [s]
- Increment of minimal initial time = 0.125 [s]
- Increment of maximal initial time = 0.5 [s]
- Initial Temperature of the disc = 60 [°C]
- Materials: Grey Cast iron FG 15.
- Convection:One introduces the values of the heat transfer coefficient (h) obtained for each surface in the shape of a curve (Figs. 6, 7),
- Flux:One introduces the values obtained by flux entering by code CFX.

Results and discussions

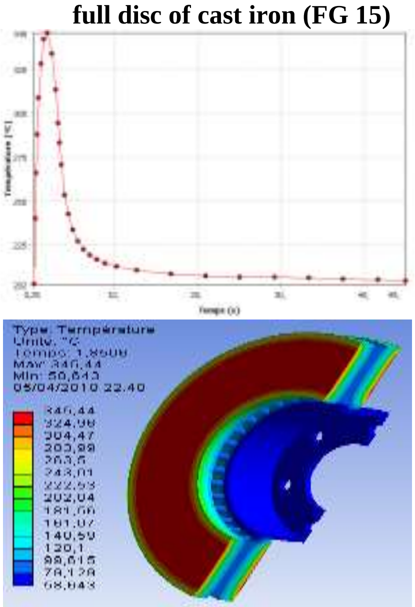
Influence of construction of the disc

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



t= 1.8839 s , T_{max}=401.66 °C

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



t=1.8506 s t, T_{max}=346.44 °C

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

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

Coupled Thermo-Mechanical Analysis

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 [25]. Fig.11 shows the finite element model and boundary conditions embedded configurations of the model composed of a disc and two pads. The initial temperature of the disc and pads is 20°C, and the surface convection condition is applied at all surfaces of the disc and the convection coefficient (h) of 5 W/m^2.°C is applied at the surface of the two pads. The FE mesh is generated using three-dimensional tetrahedral element with 10 nodes (solid 187) for the disc and pads. There are about 185901 nodes and 113367 elements are used (Fig.12).

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

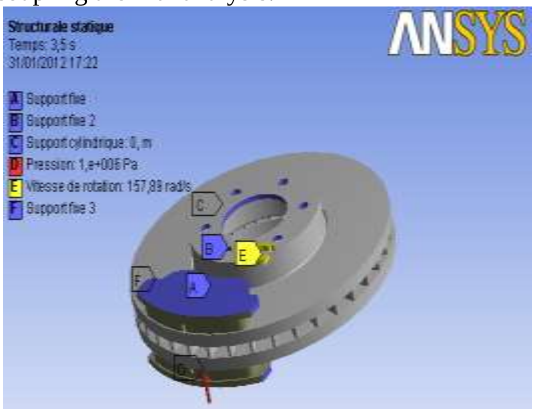


Fig.11. Boundary conditions and loading imposed on the disc-pads

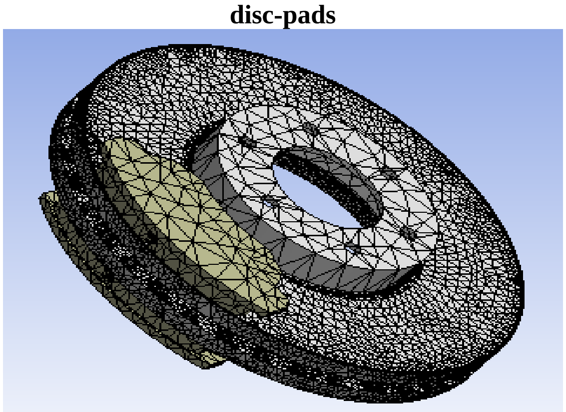
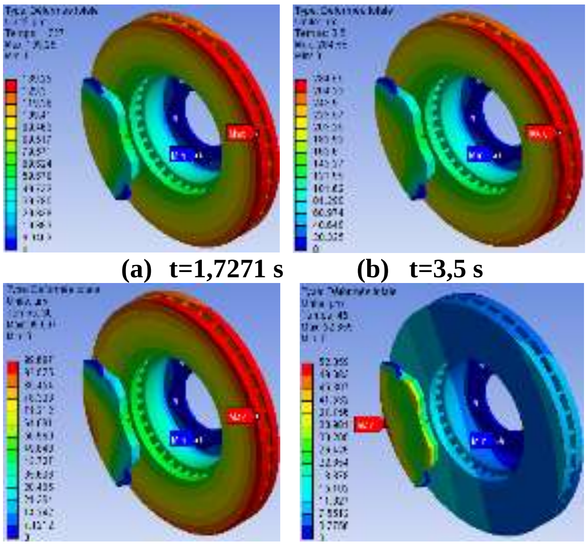


Fig.12. Refined mesh of the model

Thermal deformation

Fig.13. gives the distribution of the total distortion in the whole (disc-pads) for various moments of simulation. For this figure, the scale of values of the deformation varies from 0 μm to 284,55 μm. The value of the maximum displacement recorded during this simulation is at the moment t=3,5 s which corresponds to the time of braking. One observes a strong distribution which increases with time on the friction tracks and the external crown and the cooling fins 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 a asymmetry of the disc following the rise of temperature which will cause a deformation in the shape of an umbrella.



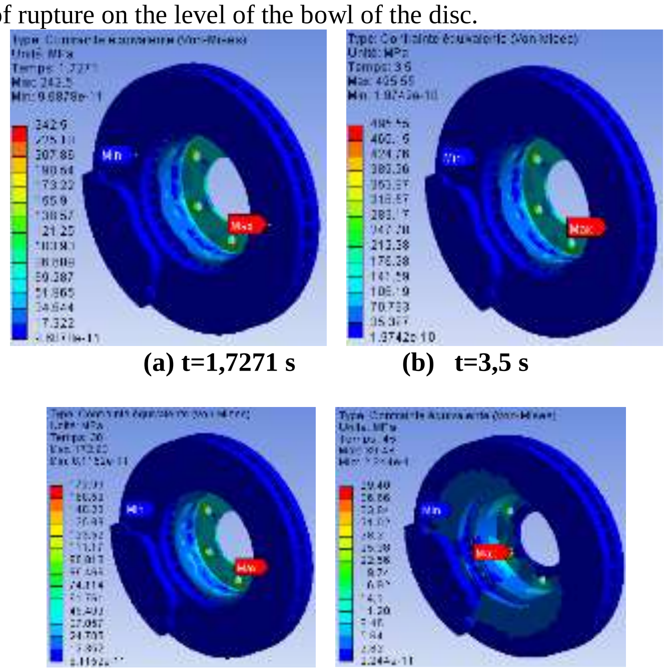
(a) t=1,7271 s (b) t=3,5 s

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

Fig13. Total distortion distribution

Von Mises stress distribution

Fig.14. presents the distribution of the constraint equivalent of Von Mises to various moments of simulation, the scale of values varies from 0 MPa to 495,56 MPa.The maximum value recorded during this simulation of the thermomechanical coupling is very significant 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 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 involve risks of rupture on the level of the bowl of the disc.



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

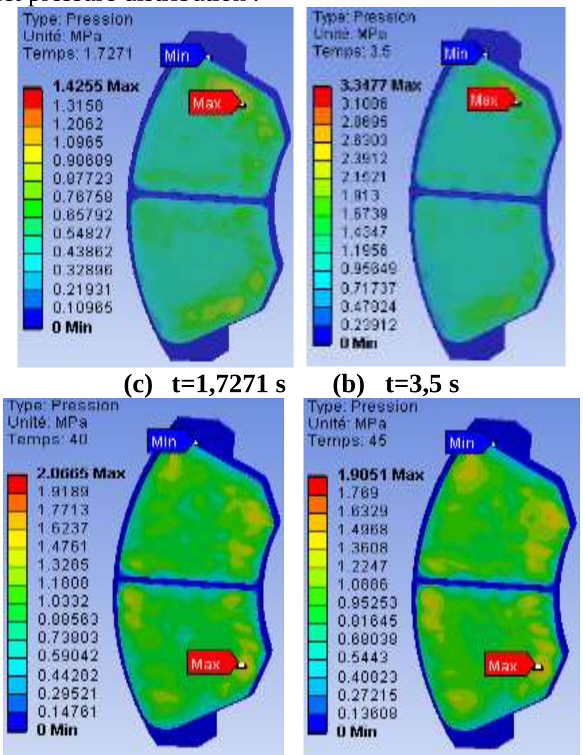
(c) t=30 s (d) t=45 s

Fig. 14. Von Mises stress distribution.

Contact pressure

Fig.15. shows the contact pressure distribution in the friction interface of the inner pad taken for at various times of simulation.For this distribution the scale varies from 0 MPa to 3,3477 MPa and reached a value of pressure at the moment t=3,5 s which corresponds to the null rotational speed.It is also noticed that the maximum contact pressure is located on the

edges of the pad 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 of the contact pressure distribution .



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

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

Fig.15. Contact pressure distribution in the inner pad

Conclusion

In this publication, we presented the analysis of thermomechanical behavior of the dry contact between the brake disc and pads during the braking process; the modeling is based on the ANSYS 11.0. We have shown that the ventilation system plays an important role in cooling disks and provides a good high temperature resistance.The analysis results showed that, temperature field and stress field in the process of braking phase were fully coupled. The temperature, Von Mises stress and the total deformations of the disc and contact pressures of the pads increases as the thermal stresses are additional to mechanical stress which causes the crack propagation and fracture of the bowl and wear of the disc and pads. Regarding the calculation results, we can say that they are satisfactory commonly found in the literature investigations. It would be interesting to solve the problem in thermo-mechanical disc brakes with an experimental study to validate the numerical results, for example on test benches, in order to show a good agreement between the model and reality.

References

[1] Milenković, P. D. et .al. , Milenković, P. D.et.al.: The influence of brake pads thermal conductivity on passanger car brake system efficiency , *Thermal Science*: Year 2010, Vol. 14, Suppl., pp. S221-S230

[1]H.Belghazi, Analytical solution of unsteady heat Conduction in a two - layered material in imperfect contact subjected to a moving heat source, Phd Thesis, university of Limoges, 2010.

[2] T Nakatsuji, K Okubo, T Fujii , M Sasada, Y Noguchi (2002) Study on Crack Initiation at Small Holes of One - piece Brake Discs. *Society of Automotive Engineers* , Inc 2002-01-0926

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

[4] M D Hudson & R L Ruhl (1997) Ventilated Brake Rotor Air Flow Investigation. *Society of Automotive Engineers*, Inc 1997-01-033

[5] Denape. J, Laraqi. N, Aspect thermique du frottement: mise en évidence expérimentale et éléments de modélisation, *Mecanique & Industries*, N°1, pp. 563-579, (2000).

[6] M. Hamraoui, Thermal behaviour of rollers during the rolling process, *Applied Thermal Engineering*, 29 (11-12) (2009) 2386-2390

[7] M. Hamraoui, Z Zouaoui, Modelling of heat transfer between two rollers in dry friction, *International Journal of Thermal Sciences* , 48 (6) (2009) 1243-1246

[8] N. Laraqi, Velocity and relative contact size effect on the thermal constriction resistance in sliding solids , *ASME J. Heat Transfer*, 119 (1997) 173-177.

[9] H. Yapıcı, M. S. Genç, G. Özısık , Transient temperature and thermal stress distributions in a hollow disk subjected to a moving uniform heat source , *Journal of thermal stress* 31 (2008) 476-493.

[10] N. Laraqi, N. Alilat , J.M. Garcia-de-Maria, A. Baïri, Temperature and division of heat in a pin - on-disc frictional device – Exact analytical solution, *Wear*, 266 (7- 8) (2009) 765-770.

[11] J.G. Bauzin, N. Laraqi, Simultaneous estimation f frictional heat flux and two thermal contact parameters for sliding solids, *Numerical Heat Transfer* 45 (4) (2004) 313-328.

[12] A. Baïri, J.M. Garcia-de-Maria, N. Laraqi, Effect of thickness and thermal properties of film on the thermal behavior of moving rough interfaces, European Physical *Journal – Applied Physics*, 26 (1) (2004) pp. 29-34.

[13] Mijuca, D. M., iberna A. M., Medjo B. I : a new multifield finite element method in steady state heat analysis . *Thermal Science*: Vol. 9 (2005), No. 1, pp. 111-130

[14] Lele Zhang, Qiang Yang, D. Weichert , Nanlin Tan. Simulation and Analysis of Thermal atigue Based on Imperfection Model of Brake Discs, Beijing Jiaotong University , PAMM · *Proc. Appl. Math.* Mech. **9**, 533 – 534 (2009)

[15] Fiche U.I.C . 541-3 : FREIN – Frein à disques et garnitures de frein à disques, 4e édition, 01.07.1993.

[16] Saumweber, E., Temperatur berechnung in Bremsscheiben fürein beliebiges Fahrprogramm, Leichtbau der Verkehrsfahrzeuge, Heft 3 / 1969, Augsburg.

[17] Cruceanu , C., Frâne pentru vehicule feroviare (Brakes for railway vehicles) , Ed. MATRIXROM, Bucureşti, 2007, ISBN 978- 973-755-200-6, 388 pag.

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

[19] Gotowicki, Pier Francesco ; Nigrelli , Vinzenco ; Mariotti, Gabriele Virzi. 2005. Numerical and experimental analysis of a pegs- wing ventilated disk brake rotor , with pads and cylinders, 10th EAEC *European Automotive Congress* –Paper EAEC05YUAS04– P 5, June 2005.

[20**]** Yu, H., et al.: study on temperature distribution due to freezing and thawing at the fengman concrete gravity dam , *thermal science*, year 2011, vol. 15, suppl. 1, pp. s27-s32
[21] Sergerlind, L. J., Applied Finite Element Analysis, John Wiley and Sons, New York, USA, 1984
[22] Hinton, E., Owen, D. R. J., An Introduction to Finite Element Computations, Pineridge Press Limited, UK, 1981
[23] H K Versteeg & W Malalasekera – An Introduction to Computational Fluid Mechanics, The Finite Volume Method
[24] Ansys v.11 user'Manual guide .
[25] M. Nouby and K. Srinivasan .Parametric Studies of Disc Brake Squeal using Finite Element Approach "Anna University Chennai- 600025 *ndia.Jurnal Mekanikal*. December 2009, No.29, 52-66.