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NUMERICAL SIMULATION OF THE THERMO MECHANICAL BEHAVIOR OF THE AUTOMOTIVE BRAKE DISC IN DRY SLIDING CONTACT WITH THE PADS

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Abstract: *During braking and when the disk brought into contact with the brake pads which represent the friction body, mechanical stresses are imposed at the contact zone. All physical parameters (temperature, pressure speed and mechanical characteristics, and tribological conditions change over time), heat from friction generated at the interface, and temperature may exceed the critical value. All these problems that allowed us to do this study which concerns the numerical simulation by finite elements of a mechanical torque in dry sliding contact with motor vehicle disk/brake pads at the moment of stop braking using the ANSYS calculation code 14.5 which is based on the finite element method with its friction contact management algorithms. This behavior was analyzed in the transient case in terms of equivalent stresses and deformations (Von Mises) as a function of the braking conditions (the type of loading, the speed of rotation of a disk, the pressure force applied to the brake pads, the coefficient of friction between the disk and the pads), and the thermal conditions (the temperature of the disk, and the heat flux in the disk, and the heat exchange by convection over the entire surface of the disk), the geometrical characteristics of the disk pads assembly and the position of the pads with respect to the brake disk and the mechanical parameters assembly and the position of the pads with respect to the brake disk and the mechanical parameters (Young's modulus, density, Poisson coefficient). This analysis allows us to see the behavior of the disk and the pads in contact and to recognize these damages in order to find the optimal technological solutions that will meet the needs of the engineer responsible for the design of the braking system, in particular the disk-pads torque, and to improve this system and make it more reliable and for an optimal and economical choice of the disk and pads well resist heat.*

Key words: *Brake pads, brake disc, contact, dry sliding contact, ANSYS, mechanical behavior.*

1. INTRODUCTION

In the transportation field, today's vehicles are more powerful and faster. Therefore, braking systems must ensure efficiency, reliability and comfort with new technologies. The braking system, which is a major safety component, is a very current research topic for automotive engineers and researchers. The phenomenon of friction between two surfaces that slide on each other when there is contact between two solids leads to a loss of mechanical energy that is transformed into heat. Moreover, friction and wear are independent phenomena. It is indeed possible to design systems with low wear and high friction (brakes) or high wear and low friction (machining) [1], [2]. The operation of the braking system to slow down or stop the moving vehicle is based on the dissipation of the kinetic energy of the vehicle into thermal energy resulting from the disk-pad friction. The brake is therefore a heat absorption system. Its efficiency depends on the capacity of its components to absorb and resist heat, and also on the friction coefficient. In friction, [3] presented a thermal study on sliding contacts with application to braking. These authors used a numerical model and an experimental device was developed on the principle of three- body contact. The

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experimental and numerical results obtained are coherent and show the interest and the representativeness of a model with three volumes, homogeneous and continuous bodies. We can also mention the study of [4], and more recently that of [5] on the thermal behavior in brake discs. The study carried out by [4] allowed to model numerically in 3D the thermal behavior of the brake for a solid and ventilated disk. The authors in [5] worked on the heat transfer from the high temperature zone to the low temperature zone by incorporating heat pipes on the surfaces of ventilated brake discs. Experimental and numerical results showed a decrease in the highest temperatures and a greater uniformity of the temperature during braking when heat pipes were inserted on the surfaces of ventilated brake discs. More recently, the same authors in [6] studied the transient phenomenon of the temperature field on a ventilated disk during its sudden braking phase. The experimentation with thermocouples placed on points on the surface of the disc allowed to verify the numerical results and to draw the temperature curves in the circumferential and radial direction of the disc.

In the case of dry contact between the brake disc and brake pads during braking, the work of [7] , based on numerical simulations, allowed the determination of deformations and Von Mises stresses as well as the distribution of contact pressure in the brake pads for the case of a solid disc and a ventilated disc.

The main objective of the present study is the modeling and numerical simulation of the thermoelastic behavior of the dry sliding contacts of the brake disc-pad assembly in the presence of friction between the two contacting solids. It is important to underline the thermal influence on the elastic behavior due to the dissipation of energy and therefore heat produced at the contact area between the disc and the pads. This influence has been taken into account with the pressure exerted by the pads on the disc during braking. The numerical simulation was carried out by the element method under the ANSYS Workbench code. The results obtained in terms of equivalent deformations and stresses (Von Mises) allowed to analyze the influence of various parameters such as the mechanical characteristics of the parts in contact, the friction coefficient, the rotation speed of the disc, the pressure applied on the pads while taking into account the braking time.

2. THERMOMECHANICAL ANALYSIS OF THE BEHAVIOR OF THE BRAKE DISC IN DRY SLIDING CONTACT WITH THE PADS

In this study, a numerical approach is proposed for the simulation of the thermo mechanical behavior of the disc-pad braking torque with dry sliding contacts as a function of the thermal and mechanical boundary conditions. To this end, the ANSYS computer code is used to develop the geometric model of the disc-pad couple and the numerical model of the simulation which is based on the finite element method. The computer code has friction contact management algorithms based on Lagrange multipliers or the penalty method [8]. The analysis of the thermo mechanical behavior of the disc-pad couple is carried out according to the braking conditions (type of load, vehicle speed, and number of braking cycles) and the geometric characteristics of the disc-lining assembly as well as the mechanical parameters. And thermal (Young's modulus, density, Poisson's ratio and friction, coefficients of thermal convection and heat flux, and initial temperature)

It will be assumed that the brake pads are bodies made of friction materials, flexible while the ventilated disc in FG15 is rigid. The contact pressure and the disc rotation speed are considered as input data for the numerical simulation.

2.1 Vehicle specifications

Table 1

Vehicle characteristics [9]

Parameters	Designation	Values
m	Vehicle mass (kg)	1700
V_0	Initial speed (m/s)	40
V_f	Vehicle speed at the end of braking (m/s)	0
b	Vehicle deceleration (m/s^2)	-20

R_d	Brake disc radius (m)	0.144
A_d	Disc friction surface (m^2)	0.44772
A_p	Wafer surface (m^2)	0.27085
μ	Coefficient of friction disc - pads	0.2
R_p	Tire radius (m)	0.2516
t_s	Break time (s)	20

The materials most commonly used for making discs are graphite cast irons. In this study, gray cast iron with high carbon content FG15 which has the best thermal performance [7], and was chosen which has good conductivity, fairly good mechanical strength and low wear [10], [11]. The material chosen for the linings, we used an organic matrix friction material characterized by a good coefficient of friction (as high and constant as possible, whatever the variation in temperature, contact pressure or rotation speed of the discs) [12]. The insert holders are made of mild steel; they strive to distribute the force exerted by the hydraulic piston over the entire surface of the pads in order to guarantee the widest and most homogeneous disc-pad contact surface possible. Table 2 summarizes the properties of the organic matrix composite material for the brake pads and the gray cast iron material for the brake disc.

Table 2

Mechanical properties of gray cast iron brake disc (a) and organic matrix composite brake pads (b) [9]

Properties	(a)	(b)
Density (kg /m^3)	7250	2500
Young's modulus(Pa)	1.38E+11	3E+09
Poisson coefficient	0.28	0.25
Compressibility modulus(Pa)	1,15E+11	2E+09
Shear modulus (Pa)	5,3077E+10	1.2E+09
Thermal conductivity, k (W/m°C)	57	5
Specific heat capacity, c (J/Kg. °C)	460	1000

2.2 Disc and pad geometry models by Solid work

Figures fig.1, fig.2, fig.3, shows the geometric models of the disk and pads developed using SolidWorks software.

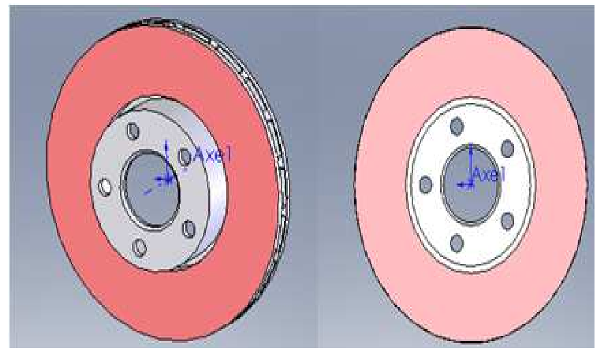


Fig.1. Ventilated disc

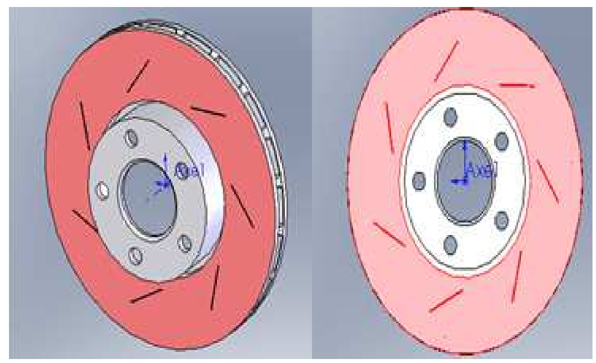


Fig.2. Grooved ventilated disc

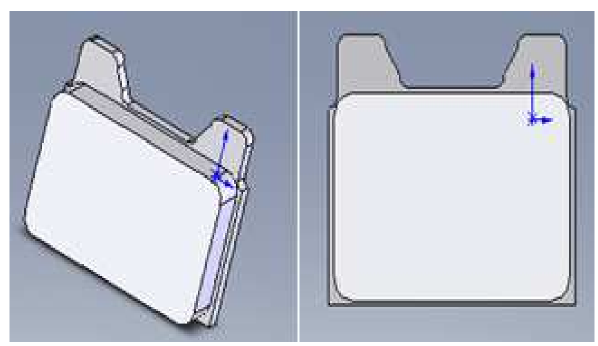


Fig.3. brake pad

3. THERMAL ANALYSIS

In the case of stopping braking, the kinetic energy of the vehicle converted into thermal energy is equal to:

$$E_c = \frac{1}{2} M.V_0{}^2 \quad (1)$$

Where M is the total mass of the vehicle, V_0 the initial speed.

To obtain the amount of heat dissipated by each of the brake discs, we need to know the weight distribution of the vehicle, expressed by the coefficient β. Thus, the amount of heat dissipated by each of the front brake discs will be [13]:

$$E_{cf} = \frac{1}{2}.\beta.M.{V_0}^2 \qquad (2)$$

We will take β equal to 30% the mass of the vehicle.

The braking force applied to each front wheel is equal to [13]:

$$F_p = \frac{E_{cf}}{2.x_f} \qquad (3)$$

$$x_f = V_0.t - \frac{1}{2}.\left(\frac{V_0}{t}\right)t_f^2 \qquad (4)$$

3.1 Thermal convection

The energy transferred between the solid surface of the disc and the moving air by the mode of convection, this energy is the movement of the air which removes the heated air near the surface and replaces it with more air costs.

During and when the disc is in motion 90% of the heat generated is transferred by convection to the ambient air [14].

Thermal convection is expressed by the following equation:

$$Q = hA_S\,(T_S - T_\infty) \qquad (5)$$

As: the surface area of the rotor (m^2).

Ts and T∞ are the surface temperatures of the brake rotor and ambient air temperature respectively.

3.2 Air flow

Calculating an airflow in a vented disc and through the fins in a car is very difficult, so it is done by the following empirical equation: [15], [16]

$$V_{air=}\omega D_0\sqrt{-0.0201 + (0.2769 \times D_I) - \left(0.0188 \times {D_I}^2\right)} \quad (6)$$

3.3.Numerical Validation of our work against Stephens experimental analysis [17]

The validation of our thermal analysis by the finite element method (transient thermal ANSYS) of a ventilated disc in FG15 was validated by the results of the experimental analysis of stephens [17]. Stephens' experimental analysis was carried out by the measured values of the temperature distributed on the surfaces of the ventilated disc of a racing car by the use of the friction thermocouple [17].

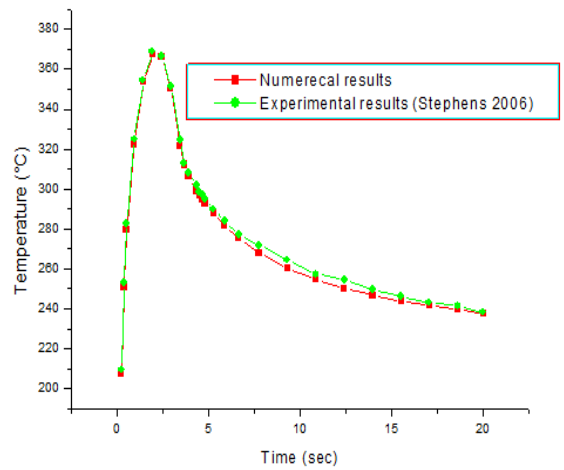


Fig.5. Validation of the numerical analysis of our work against Stephens experimental analysis [17]

4. COUPLED THERMO-MECHANICAL NUMERICAL ANALYSIS

4.1 Thermal analysis

4.1.1 Comparison between ventilated disc with grooves and non-grooved ventilated disc

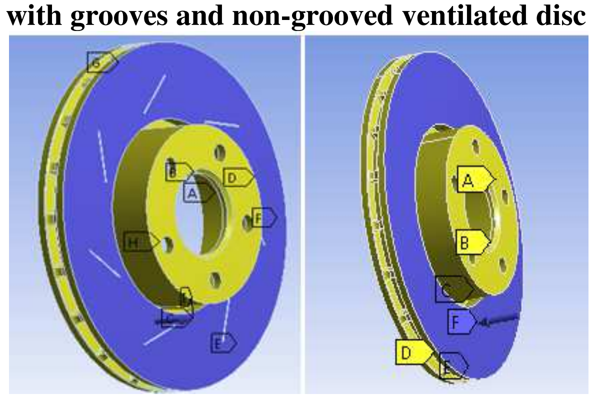
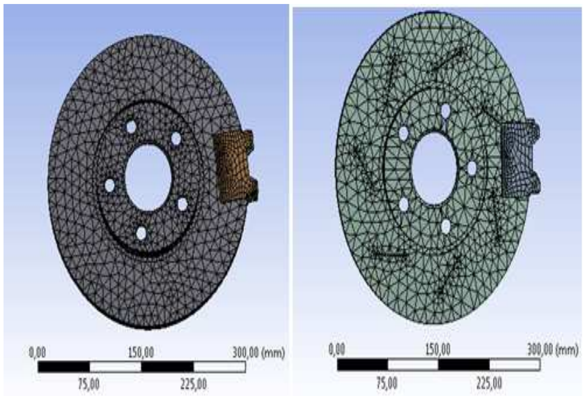


Fig.6. Thermal conditions



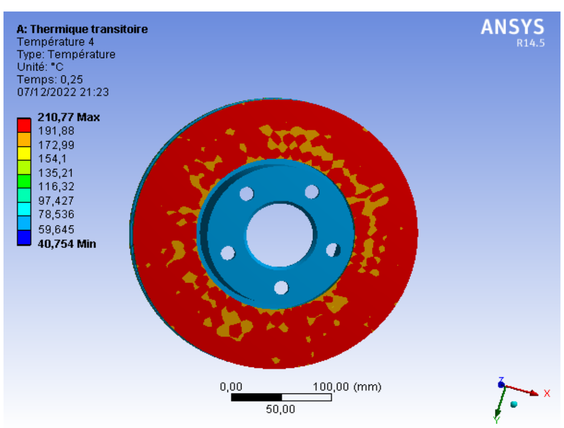
a) Ungrooved ventilated disc b) Ventilated disc

Fig.7. Mesh model

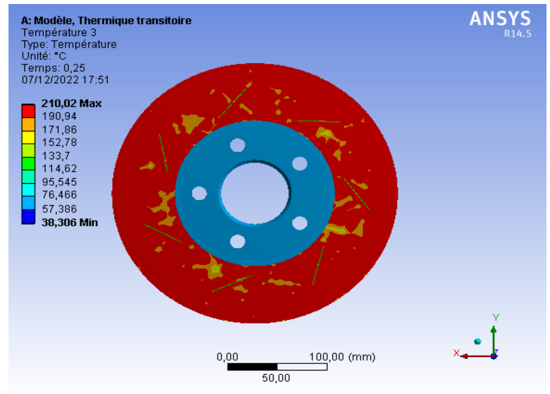
4.1.2 Thermal comparison results

Figures fig.8, fig.9, fig.10 show that the temperature distribution at t=0.25s, t=1.5s, t=20s with maximum values appears in the ungrooved ventilated disc while the lower values is that of the ventilated disc with grooves.

The ventilated disc with grooves supports better than the ventilated non-grooved disc because of the grooves which allow the disc to cool down faster and their ability to dissipate heat. This proves to us that the geometry design plays a main role in the temperature distribution in the braking system.

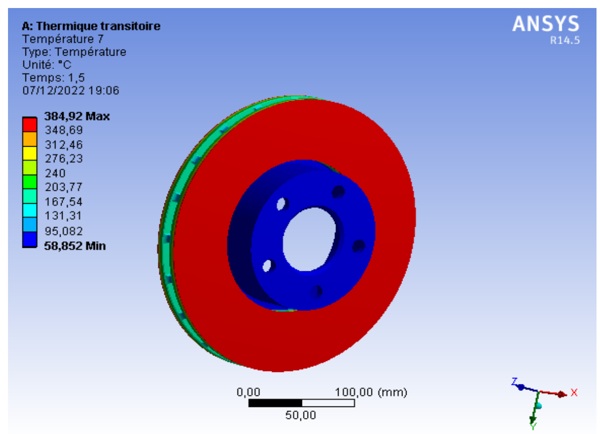


a)Ungrooved ventilated disc (Tmax=210.77°C, Tmin= 40.754°C)

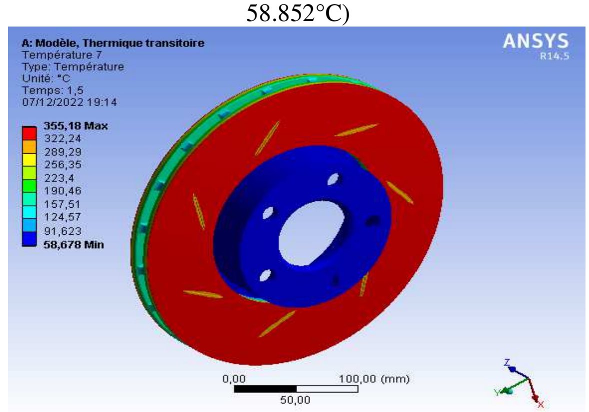


b) Ventilated disc with grooves (Tmax=210.02°C,Tmin=38.786°C)

Fig.8. Temperature distribution on a ventilated disk with grooves at t=0.25s

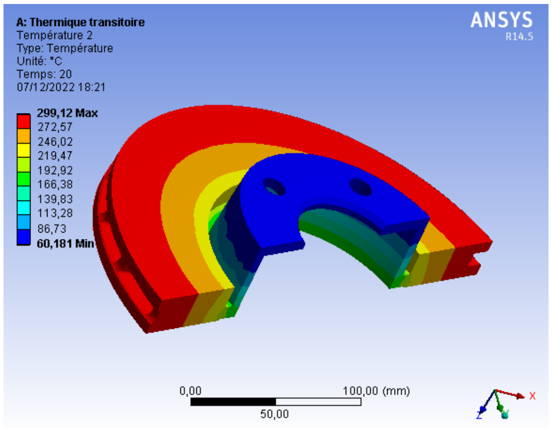


a)Ungrooved ventilated disc (Tmax=384.92°C, Tmin= 58.852°C)

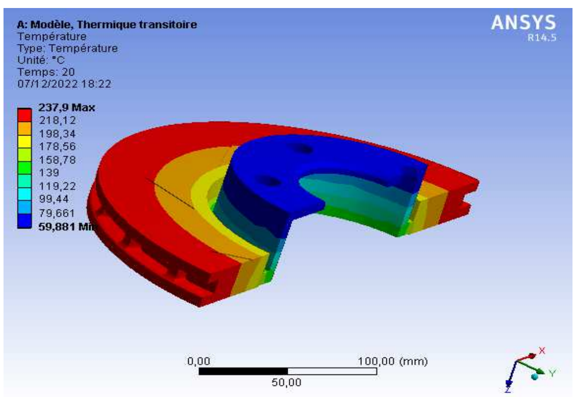


b) Ventilated disc with grooves (Tmax=355.18°C, Tmin= 58.678°C)

Fig.9. Temperature distribution on a ventilated disk with grooves at t=1.5s



a) Ungrooved ventilated disc (Tmax=299.12°C, Tmin= 60.181°C)



a) Ventilated disc with grooves (Tmax=237.9°C,Tmin=59.881°C)

Fig.10. Temperature distribution at t=20s

4.2 Structural analysis

It will be assumed that the brake pads are bodies made of flexible friction materials, whereas the grooved ventilated disc and the non-grooved ventilated disc in FG15 are rigid. The contact pressure and the rotational speed of the disc are considered as input data for the numerical simulation as shown in the table above.

The braking force applied to each front wheel disc is equal to:

$$F_d = \frac{E_{cf}}{2.\frac{R_d}{.R_p}.\left(V_0 t - \frac{1}{2}\left[\frac{V_0}{t}\right]t_f^2\right)} \qquad (5)$$

The braking speed is equal to:

$$V_f = V_0\left(1 - \frac{t}{t_f}\right) \qquad (6)$$

For the case of stopping braking, we have V_f =0.

The initial rotation speed of the disk is given by the following relation:

$$\omega_d = \frac{V_0}{R_d} \qquad (7)$$

The pressure exerted on the disc by the pads is calculated according to [18]

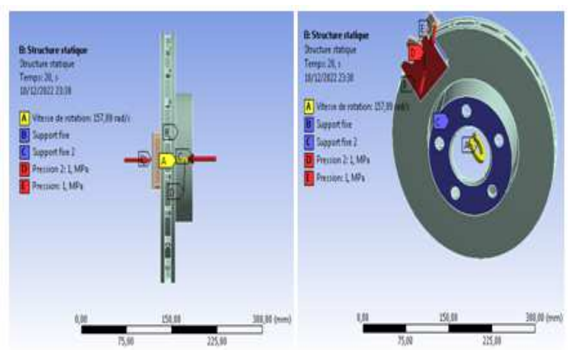
$$p = \frac{F_d}{A_{C\,p}.\mu} \qquad (8)$$

For the chosen vehicle, we have:Fd = 541.7[N], ωd = 159tr/min, p = 1MPa.

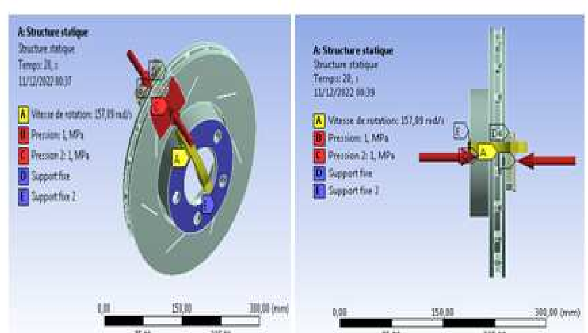
To perform the digital simulation during the braking phase, the following temporal conditions are considered:

- Braking time = 20 [s]
- Step of the initial time = 0.25 [s]
- Minimum initial time step = 0.125 [s]
- Maximum initial time step = 0.5 [s]

4.2.1 Conditions to the limits



a) Ungrooved vented disc



b) Grooved ventilated disc

Fig.11. Conditions to the limits

5. Results and interpretations

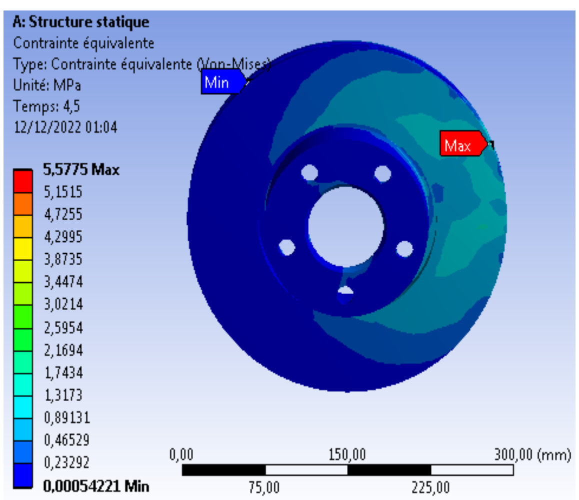
5.1 Variation of the equivalent stress as a function of time

To make a comparative study of the distribution of the equivalent Von Mises stress, two types of disk were chosen in the braking torque; slotted vented disc/pads, and non-slotted vented disc/pads.

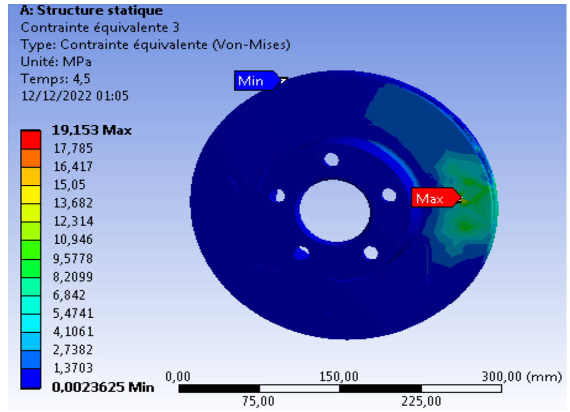
The figures fig.12, fig.13, fig.14, show the state of the Von Mises stresses for the two types of disk at t=0.45s, t=10s, t=20s.

In each of these figures, it can be seen that the stress concentration is localized in the contact zone (disc/pads) and at the level of the outwardly oriented groove and it propagates towards the connection zone between the tracks and the boles and towards the outer part of the pads exert the tightening.

The equivalent stress varies nonlinearly with time, and the distribution with maximum values appears in the slotted ventilated disc while the lower values are that of the non-grooved ventilated disc.

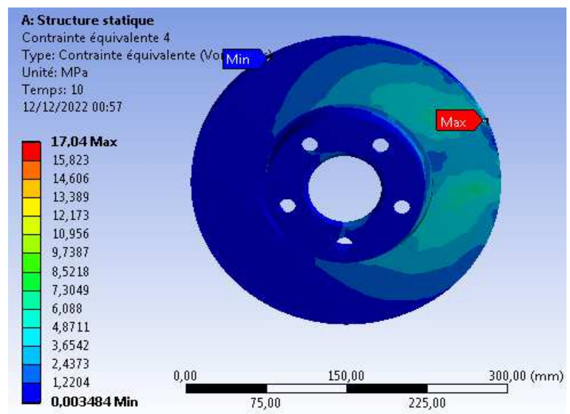


a) Ungrooved vented disc σmax = 5,5775MPa

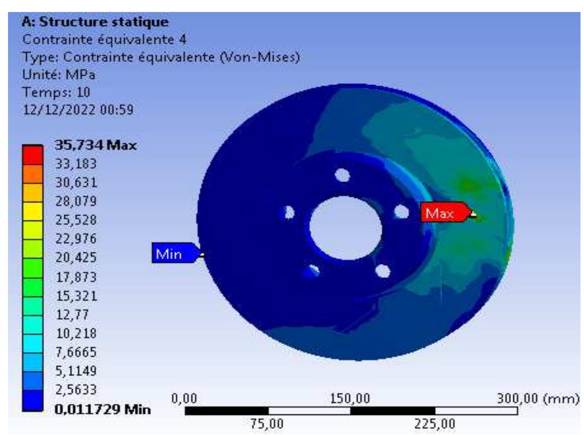


b) Grooved ventilated disc σmax=19.153 MPa

Fig.12. Distribution of the stress equivalent to t=4.5 s

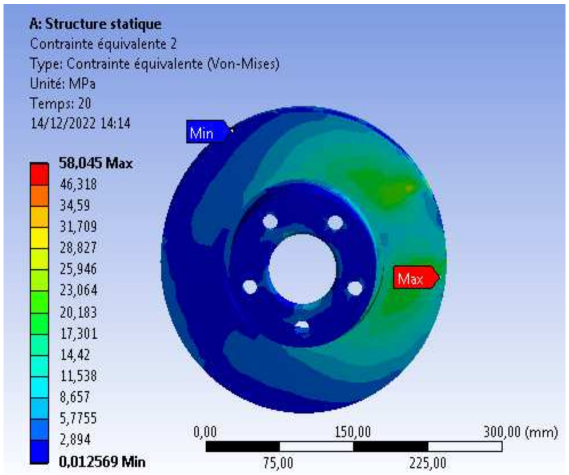


a) Ungrooved vented disc σmax=17.04Mpa

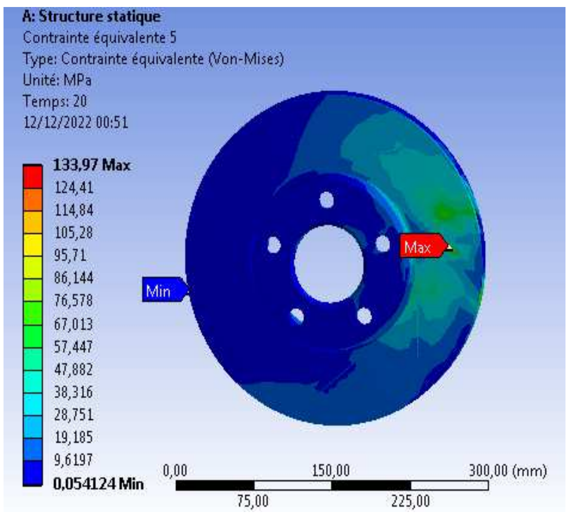


b) Grooved ventilated disc σmax=35.734 MPa

Fig.13. Distribution of the stress equivalent to t=10 s



a) Ungrooved vented disc σmax=58.045 Mpa



b)Grooved ventilated disc σmax=133.97 MPa

Fig.14. Distribution of the stress equivalent to t=20 s

6. CONCLUSION

In this study, the numerical analysis of the thermo mechanical behavior of dry sliding contacts of the braking torque (disc/pads) was carried out using an ANSYS 14.5 calculation code which is based on the finite element method.

It can be seen that the stress field depends not only on the coefficient of friction, but on other parameters such as the speed of rotation of the disc, the temperature, the choice of material of the braking torque, as well as the variation of the braking time. The type of loading applied to the brake pads also has a role on the thermo mechanical behavior of the dry sliding contact between the disc and the pads.

When the disc is brought into contact with the pads, the heat resulting from the friction produced on the interface of the surface of the disc and the linings can cause a very great heating of the latter which leads to undesirable effects such as phenomena of deterioration and thermal cracking, thermo-elastic instability.

From the results of the thermal and mechanical analysis, it can be concluded that the geometry design plays a major role in the thermo-mechanical behavior.

The presence of the grooves in the contact area and its positions relative to the pads when in contact with the disc during braking have an influence on the distribution of temperature and equivalent stress.

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SIMULARE NUMERICĂ A COMPORTAMENTULUI TERMOMECANIC AL DISCULUI DE FRANĂ AUTO ÎN CONTACT DE ALUNECARE USCATA CU PLACUȚELE

Rezumat: În timpul frânării și când discul este adus în contact cu plăcuțele de frână care reprezintă corpul de frecare, la zona de contact sunt impuse solicitări mecanice. Toți parametrii fizici

(temperatura, viteza presiunii şi caracteristicile mecanice şi condițiile tribologice se modifică în timp), căldura de la frecare generată la interfață și temperatura pot depăşi valoarea critică. Toate aceste probleme care ne-au permis să facem acest studiu care se referă la simularea numerică prin elemente finite a unui cuplu mecanic în contact uscat de alunecare cu discul autovehiculului/placuțele de frână în momentul opririi frânării folosind codul de calcul ANSYS 14.5 care se bazează pe metoda elementelor finite cu algoritmii săi de gestionare a contactelor de frecare. Acest comportament a fost analizat în cazul tranzitoriu în termeni de solicitări și deformații echivalente (Von Mises) în funcție de condițiile de frânare (tipul de încărcare, viteza de rotație a unui disc, forța de presiune aplicată plăcuțelor de frână, coeficientul de frecare dintre disc și plăcuțe) și condițiile termice (temperatura discului și fluxul de căldură în disc și schimbul de căldură prin convecție pe întreaga suprafață a discului), caracteristicile geometrice ale discului. ansamblu plăcuțe de disc și poziția plăcuțelor față de discul de frână și ansamblul parametrilor mecanici și poziția plăcuțelor față de discul de frână și parametrii mecanici (modulul Young, densitatea, coeficientul Poisson). Această analiză ne permite să vedem comportamentul discului și al plăcuțelor în contact și să recunoaştem aceste avarii pentru a găsi soluțiile tehnologice optime care să răspundă nevoilor inginerului responsabil cu proiectarea sistemului de frânare, în special discul. -cuplul plăcuțelor, iar pentru a îmbunătăți acest sistem și a-l face mai fiabil și pentru o alegere optimă și economică a discului și a plăcuțelor rezistă bine la căldură.

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