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NUMERICAL SIMULATION OF THERMOMECHANICAL BEHAVIOUR OF AUTOMOTIVE BRAKE DISC IN DRY SLIDING CONTACT WITH PADS

NUMERIČKA SIMULACIJA TERMOMEHANIČKOG PONAŠANJA AUTOMOBILSKOG KOČIONOG DISKA U SUVOM KLIZNOM KONTAKTU SA PLOČAMA

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Ključne reči
- kočione ploče
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- ANSYS®
- mehaničko ponašanje

Abstract

During braking and when the disk comes into contact with brake pads that represent the friction body, mechanical stresses are imposed at the contact zone. All physical parameters (temperature, pressure speed, mechanical characteristics, and tribological conditions change over time), heat from friction generated at the interface, and temperature may exceed critical value. These problems allow us to do this study concerning the numerical simulation by finite elements of a mechanical torque in dry sliding contact with vehicle disk/brake pads at the moment of stop braking using ANSYS® code 14.5 based on the finite element method with friction contact management algorithms. This behaviour is analysed in the transient case in terms of equivalent stresses and deformations (von Mises) as a function of braking conditions (type of loading; speed of disk rotation; pressure force applied to the brake pads; coefficient of friction between disk and pads), and thermal conditions (disk temperature, heat flux in disk; heat exchange by convection over the entire disk surface); geometrical characteristics of the disk pad assembly and position of pads with respect to brake disk and mechanical parameters assembly and position of pads with respect to brake disk and mechanical parameters (Young's modulus, density, Poisson coefficient). The analysis allows us to see the disk behaviour and pads in contact, and to recognise the damages in order to find 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, to improve the system and make it more reliable, and for an optimal and economical selection of disk and pads with heat resistance.

Izvod

Pri kočenju, kada disk dolazi u kontakt sa kočionim pločicama koje čine površine trenja, nastaju mehanička naprezanja u zoni kontakta. Svi fizički parametri (temperatura, brzina pritiska, mehaničke karakteristike, tribološki uslovi se menjaju tokom vremena), toplota od trenja koja se stvara na vezi i temperatura mogu da pređu kritičnu vrednost. Svi ovi problemi omogućili su nam da uradimo studiju koja sadrži numeričku simulaciju konačnim elementima mehaničkog momenta u suvom kliznom kontaktu sa diskom/kočionim pločicama vozila u trenutku zaustavljanja kočenjem, koristeći računarski program ANSYS® 14.5 koji se zasniva na metoda konačnih elemenata sa algoritmima za upravljanje kontaktnim trenjem. Ovo ponašanje se analizira u prelaznom režimu u smislu ekvivalentnih napona i deformacija (fon Mizes) u funkciji uslova kočenja (vrsta opterećenja; brzina rotacije diska; sila pritiska na kočione pločice; koeficijent trenja između diska i pločica), kao i termičke uslove (temperatura diska, provođenje toplote u disku, razmena toplote konvekcijom preko cele površine diska); geometrijske karakteristike sklopa diska i pločica; položaja pločica u odnosu na kočioni disk; skup mehaničkih parametara i položaj pločica u odnosu na kočioni disk i mehaničke parametre (modul elastičnosti, gustina, Poasonov koeficijent). Ova analiza nam omogućava da sagledamo ponašanje diska i pločica u kontaktu i da prepoznamo oštećenja kako bismo pronašli optimalna tehnološka rešenja koja će zadovoljiti potrebe inženjera odgovornog za projektovanje kočionog sistema, a posebno obrtnog momenta na pločicama, radi poboljšanja sistema, čineći ga pouzdanijim, za optimalan i ekonomičan izbor diska i pločica, otpornih prema toploti.

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, a major safety component, is a 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 kinetic energy of the vehicle into thermal energy resulting from 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, presented is a thermal study on sliding contacts with application to braking, /3/. The authors used a numerical model and developed an experimental device on the principle of three-body contact. Experimental and numerical results obtained are coherent and show interest and representativeness of a model with three volumes, homogeneous, and continuous bodies. We can also mention the study in /4/, and more recently that in /5/ on the thermal behaviour of brake discs. The study carried out by /4/ allowed to model numerically in 3D the thermal behaviour of the brake for a solid and ventilated disk. The authors in /5/ worked on the heat transfer from the high-temperature to the low-temperature zone by incorporating heat pipes on the surfaces of ventilated brake discs. Experimental and numerical results show a decrease in the highest temperatures and a greater uniformity of temperature during braking when heat pipes are 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 draw temperature curves in circumferential and radial directions 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, allows 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- and a ventilated disc.

The main objective of the present study is the modelling and numerical simulation of thermoelastic behaviour of the dry sliding contacts in 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 behaviour due to the dissipation of energy and, therefore, heat produced at the contact area between the disc and pads. This influence has been taken into account with the pressure exerted from the pads onto the disc during braking. The numerical simulation is carried out by finite element method under the ANSYS Workbench code. Results obtained in terms of equivalent deformations and stresses (von Mises) allowed to analyse the influence of various parameters such as: mechanical characteristics of parts in contact, friction coefficient, disc rotation speed, pressure applied on pads, while taking into account the braking time.

THERMOMECHANICAL ANALYSIS OF BRAKE DISC BEHAVIOUR IN DRY SLIDING CONTACT WITH PADS

In this study, a numerical approach is proposed for simulating the thermomechanical behaviour of the disc-pad braking torque with dry sliding contacts as a function of thermal and mechanical boundary conditions. To this end, the ANSYS® code is used to develop the geometric model of the disc-pad couple and the numerical model of the simulation based on the finite element method. The computer code has friction contact management algorithms based on Lagrange multipliers, or the penalty method, /8/. Analysis of the disc-pad couple thermomechanical behaviour 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 mechanical and thermal parameters (Young's modulus, density, Poisson's ratio, coefficients of friction, thermal convection and heat flux, and initial temperature).

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

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, has the best thermal performance, /7/, and is chosen with good conductivity, fairly good mechanical strength, and low wear, /10, 11/. As 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 brake pads and 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

Disc and pad geometry models by SolidWorks®

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

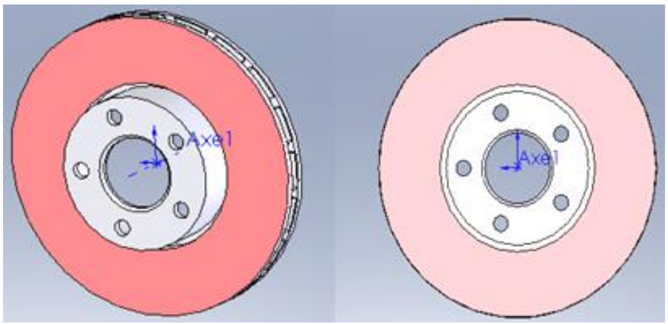


Figure 1. Ventilated disc.

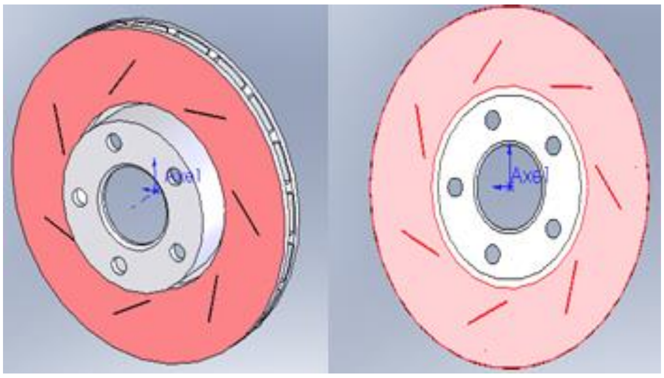


Figure 2. Grooved ventilated disc.

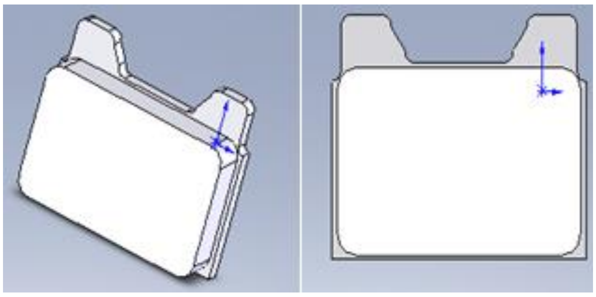


Figure 3. Brake pad.

THERMAL ANALYSIS

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

$$E_c = \frac{1}{2} M V_0^2 , \tag{1}$$

where: M is total mass of the vehicle; V_0 is 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/:

$$E_{cf} = \frac{1}{2} \beta M V_0^2 . \tag{2}$$

We will take β equal to 30 % of the vehicle mass. The braking force applied to each front wheel is equal to, /13/:

$$F_p = \frac{E_{cf}}{2x_f} , \tag{3}$$

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

where: t_f is braking time.

Thermal convection

The energy transferred between the solid surface of the disc and the moving air by mode of convection, is the movement of the air which removes 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 equation:

$$Q = h A_s (T_s - T_\infty) , \tag{5}$$

where: A_s is surface area of the rotor (m^2); T_s and T_∞ are surface temperature of the brake rotor and ambient air temperature, respectively.

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/:

$$V_{air} = \omega D_0 \sqrt{-0.0201 + 0.2769 D_I - 0.0188 D_I^2} . \tag{6}$$

Numerical validation against Stephens experimental analysis

Our thermal analysis by finite element method (Transient Thermal ANSYS®) of a ventilated disc in FG15 is validated by the results of experimental analysis in /17/. Stephens' experimental analysis is carried out by the measured values of temperature distributed on the surfaces of the ventilated disc of a racing car, by the use of a friction thermocouple, /17/. Figure 4 shows the validation of thermal results obtained in this study, as shown in Figs. 7, 9, 11, with the results of Stephens' experimental study under the same conditions.

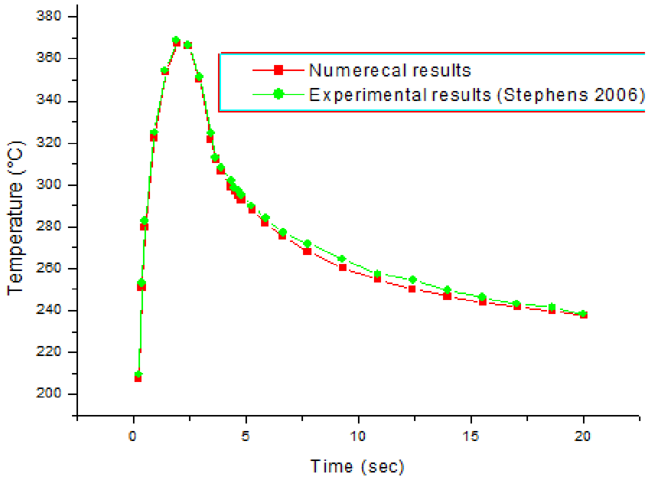


Figure 4. Validation of the numerical analysis of our work against Stephens experimental analysis, /17/.

COUPLED THERMOMECH. NUMERICAL ANALYSIS

Thermal analysis

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

Thermal comparison results

Figure 5 shows thermal conditions: heat flux: 4.197×10^6 (in blue); and temperature t = 40 °C, heat exchange coefficient h (yellow). Figure 6 shows the mesh of the two braking couples: a) ungrooved ventilated disc; b) ventilated disc. The entire braking system is modelled by finite elements. The ANSYS Workbench® performs a volumetric mesh of the brake disc and pad.

Figures 7 to 12 show temperature distribution at t = 0.25; 1.5; 20 s with maximal values at the ungrooved ventilated disc, while lower values are at ventilated disc with grooves.

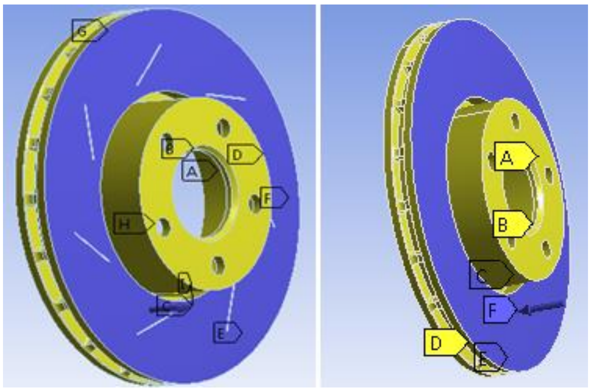
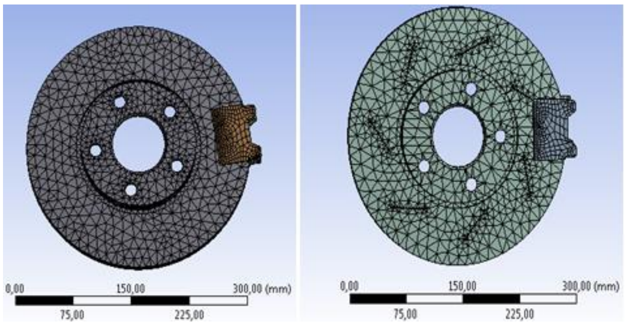


Figure 5. Thermal conditions.



a) ungrooved ventilated disc b) ventilated disc

Figure 6. Mesh model.

Because of grooves allowing the disc to cool down faster and the ability to dissipate heat, the ventilated disc with grooves supports better heat dissipation than ventilated non-grooved disc. This proves that the design geometry plays a main role in temperature distribution of the braking system.

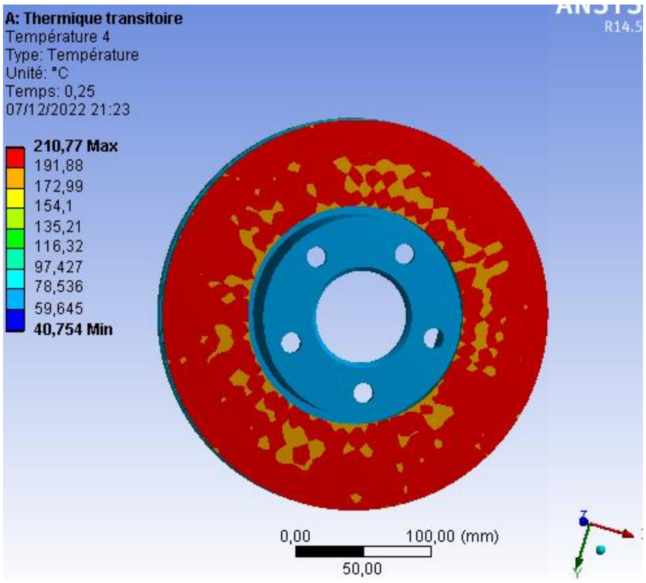


Figure 7. Temperature distribution on an ungrooved ventilated disc at t = 0.25 s (T_{max} = 210.77 °C, T_{min} = 40.754 °C).

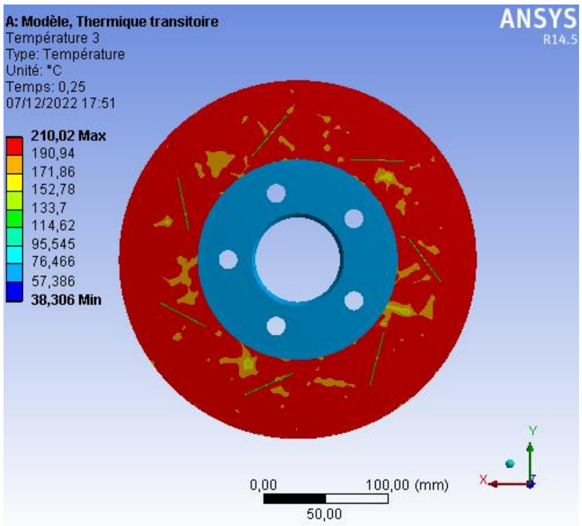


Figure 8. Temperature distribution on a ventilated disc with grooves at t = 0.25 s (T_{max} = 210.02 °C, T_{min} = 38.786 °C).

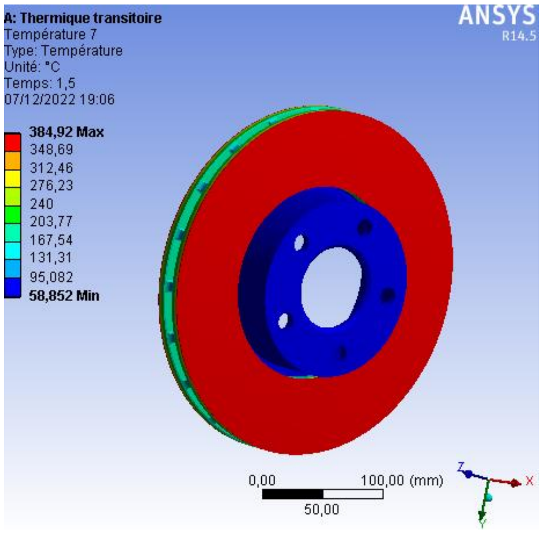


Figure 9. Temperature distribution on an ungrooved ventilated disc at t = 1.5 s (T_{max} = 384.92 °C, T_{min} = 58.852 °C).

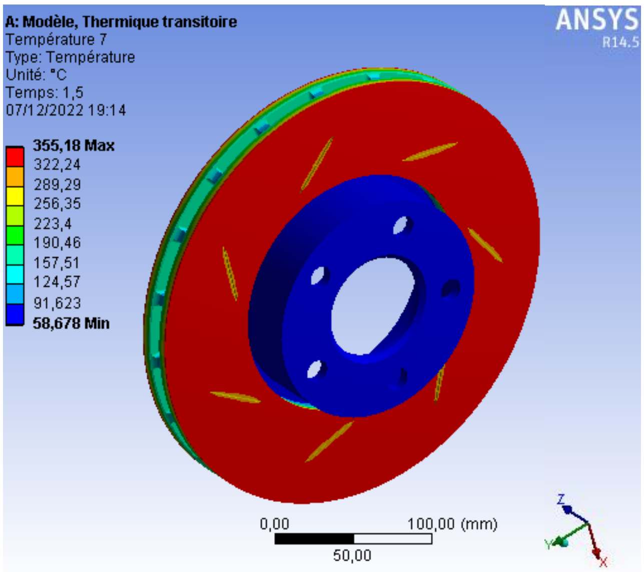


Figure 10. Temperature distribution on a ventilated disc with grooves at t = 1.5 s (T_{max} = 355.18 °C, T_{min} = 58.678 °C).

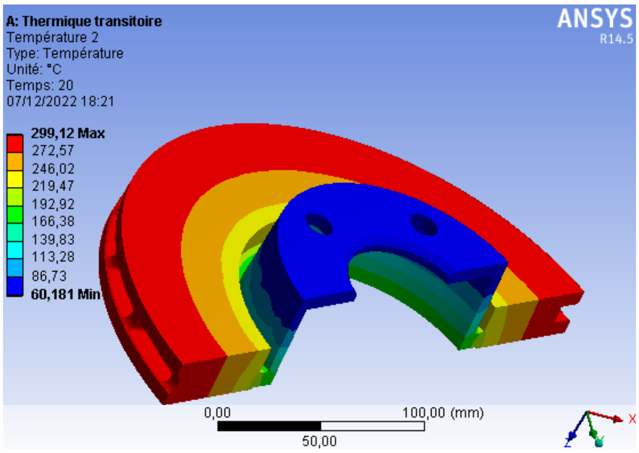


Figure 11. Temperature distribution on an ungrooved ventilated disc at t = 20 s (T_{max} = 299.12 °C, T_{min} = 60.181 °C).

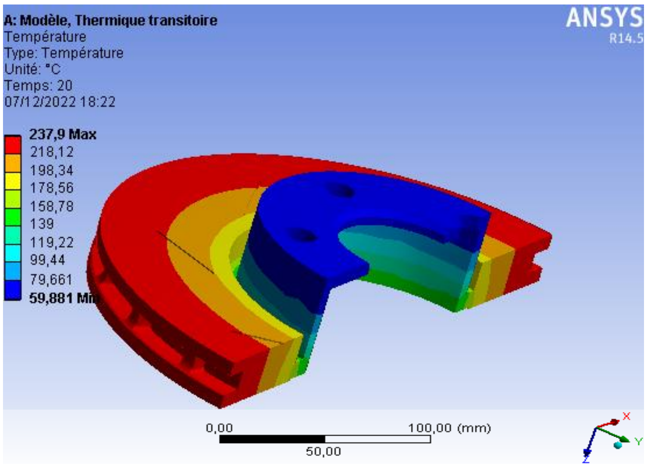


Figure 12. Temperature distribution on a ventilated disk with grooves at t = 20 s (T_{max} = 237.9 °C, T_{min} = 59.881 °C).

STRUCTURAL ANALYSIS

It is 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 rotational speed of the disc are considered as input data for the numerical simulation, as shown in Figs. 13 and 14.

The braking force applied to each front wheel disc is:

$$F_d = \frac{E_{cf}}{2\frac{R_d}{R_p}\left(V_0 t - \frac{1}{2}\frac{v_0}{t}t_f^2\right)}. \tag{7}$$

where: v_0 is initial speed (m/s).

The braking speed is equal to:

$$V_f = V_0\left(1 - \frac{t}{t_f}\right). \tag{8}$$

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

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

$$\omega_d = \frac{V_0}{R_d}. \tag{9}$$

The pressure exerted on the disc by the pads is calculated according to /18/,

$$p = \frac{F_d}{A_{cp}\mu}. \tag{10}$$

For the chosen vehicle: F_d = 541.7 N; ω_d = 158.98 rad/s; p = 1 MPa; A_{cp} pad surface in contact with disc.

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.

Boundary conditions

Figure 13 shows boundary conditions applied to the braking torque (full ventilated disc/pads). Figure 14 shows the same boundary conditions applied to the braking torque (ventilated grooved disc/pads). Boundary conditions applied to the braking torque used in this study: pads are embedded on their edges in the plane orthogonal to the contact surface; a fixed support in the outer pad; pads that press the disc are represented by pressure P = 1 MPa applied to the two pads; disc rotation speed ω = 158.98 rad/s; coefficient of friction μ = 0.2; rigid disc in gray cast iron, and the pad is flexible in organic matrix composite material.

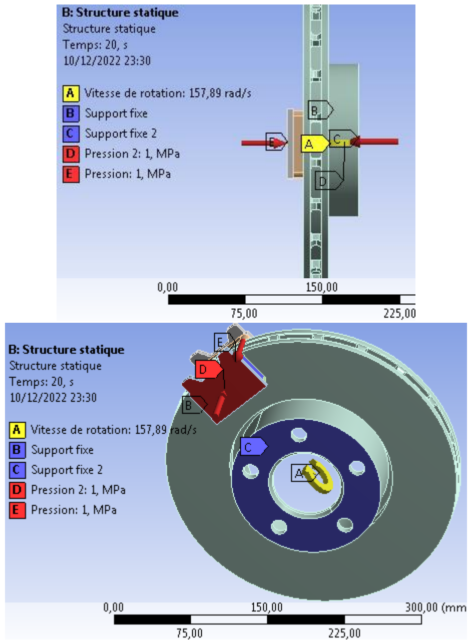


Figure 13. Boundary conditions of ungrooved ventilated disc.

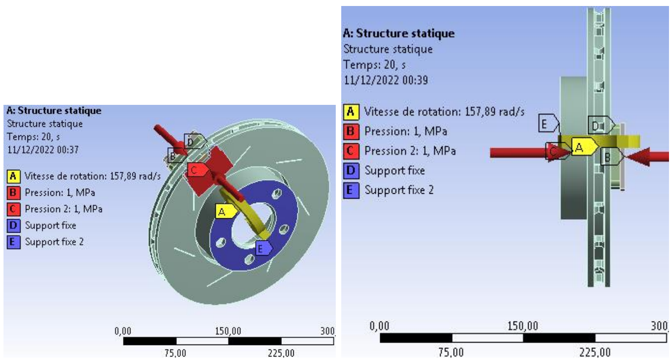


Figure 14. Boundary conditions of grooved ventilated disc.

RESULTS AND INTERPRETATIONS

Variation of equivalent stress as a function of time

To make a comparative study of the distribution of equivalent von Mises stresses, two types of discs are chosen in the braking torque: slotted vented disc/pads; and non-slotted vented disc/pads.

Figures 15 to 20 show the state of von Mises stresses for the two types of discs at t = 0.45, 10, and 20 s.

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 boles, and towards the outer part of the pads exerting the tightening.

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

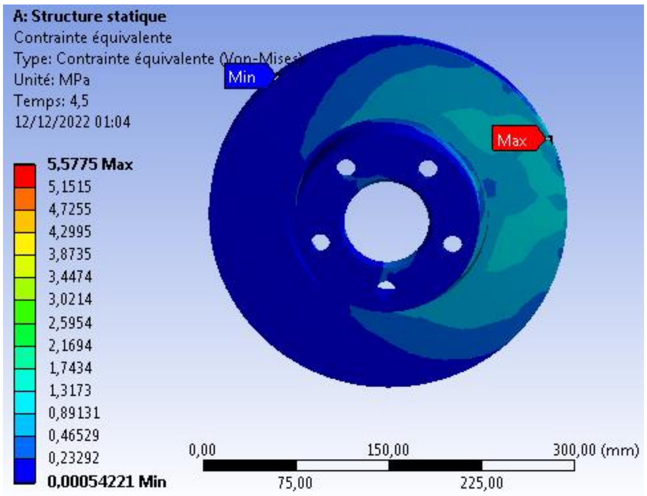


Figure 15. Equivalent stress distribution (von Mises) of the ungrooved ventilated disc at t = 4.5 s (σ_{max} = 5.5775 MPa).

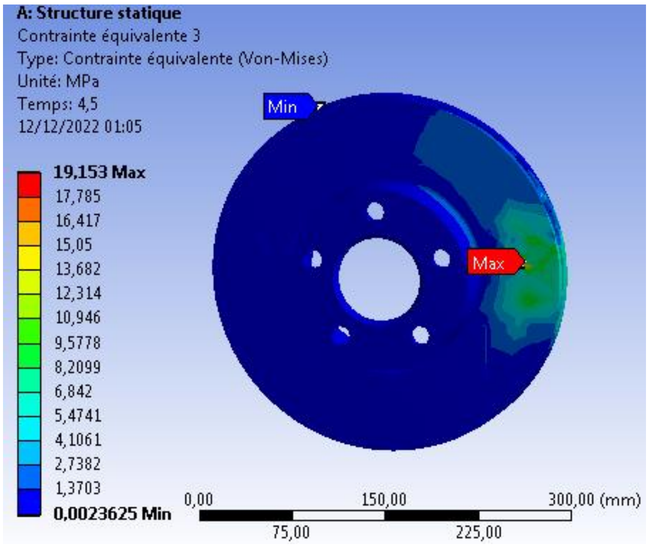


Figure 16. Equivalent stress distribution (von Mises) of the grooved ventilated disc at t = 4.5 s (σ_{max} = 19.153 MPa).

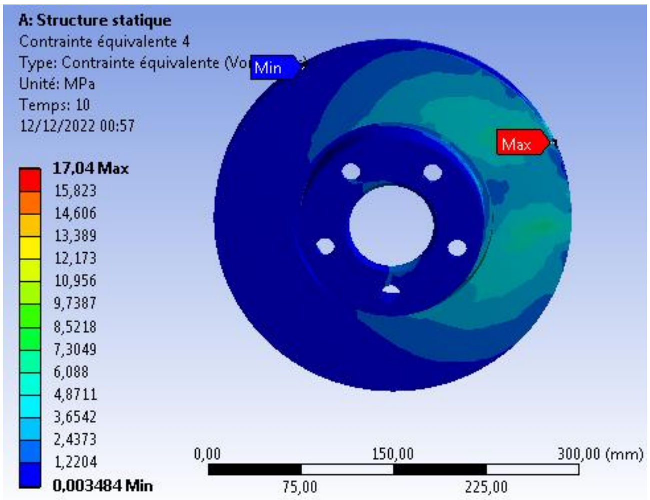


Figure 17. Equivalent stress distribution (von Mises) of the ungrooved ventilated disc at t = 10 s (σ_{max} = 17.04 MPa).

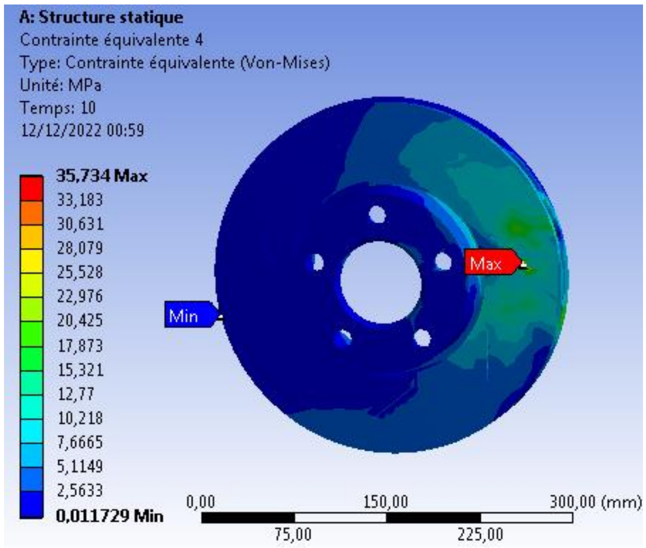


Figure 18. Equivalent stress distribution (von Mises) of the grooved ventilated disc at t = 10 s (σ_{max} = 35.734 MPa).

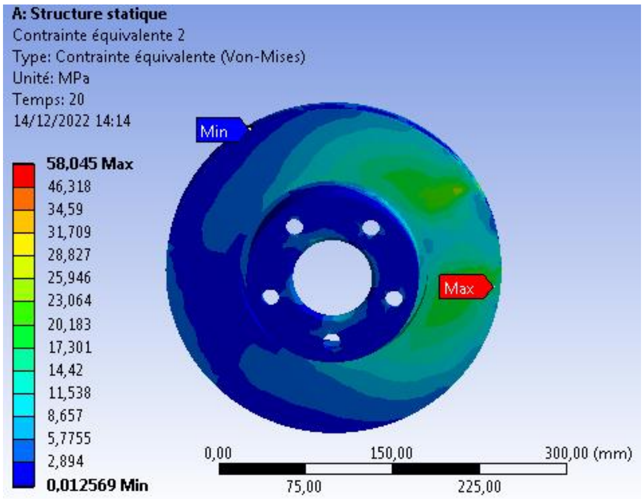


Figure 19. Equivalent stress distribution (von Mises) of the ungrooved ventilated disc at t = 20 s (σ_{max} = 58.045 MPa).

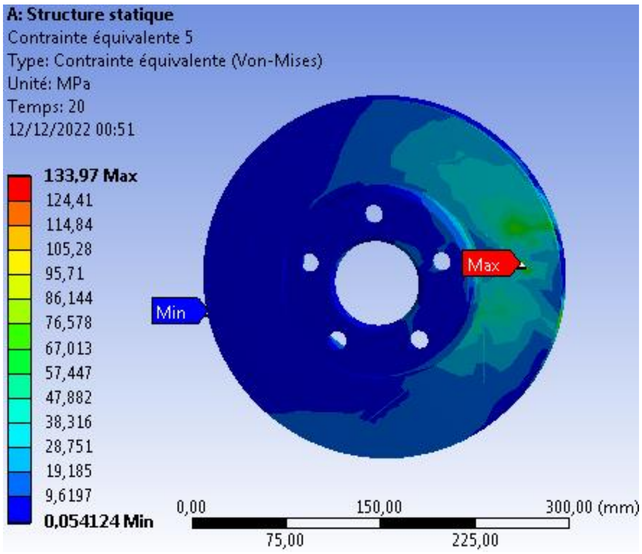


Figure 20. Equivalent stress distribution (von Mises) of the grooved ventilated disc at t = 20 s (σ_{max} = 133.97 MPa).

CONCLUSION

In this study, the numerical analysis of the thermomechanical behaviour of dry sliding contacts of the braking torque (disc/pads) is carried out using ANSYS 14.5 code, 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 disc rotation speed, temperature, 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 thermomechanical behaviour of the dry sliding contact between the disc and 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 very large heating of the latter which leads to undesirable effects such as phenomena of deterioration, thermal cracking, thermoelastic instability.

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

The presence of grooves in the contact area and their 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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