Numerical simulation of the elastic 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: - dry sliding contact - friction - wear - disc – pads.

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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 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.

1. Analysis of the elastic behavior of the dry sliding contacts of the disk - pad pair

In this study, a numerical approach is proposed for the simulation of the static elastic behavior of the disk-pad pair with dry sliding contactsas a function of the mechanical boundary conditions. For this purpose, the Ansys finite element code is used to elaborate the geometrical model of the disk-pad pair and the numerical model of the finite element simulation. The computational code has algorithms for handling frictional contacts based on Lagrange multipliers or the penalty method [8]. The analysis of the static elastic behavior of the disk-pad pair is carried out as a function of the braking conditions (type of loading, vehicle speed, number of braking cycles) and of the geometric characteristics of the disk-pad assembly as well as of the mechanical parameters (Young's modulus, density, the fish coefficient and friction).

We will assume that the brake pads are bodies made of friction materials, flexible while the disc is ventilated with grooves and rigid. The contact pressure and the rotation speed of the disc are considered as input data for the numerical simulation.

1.1. Vehicle specifications

From the vehicle data, the braking force on the tires and the stopping time can be determined as a function of the initial speed, the vehicle load and the road profile (level or slope) as presented in Table 1.

Table 1 shows the characteristics of the selected vehicle.

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	Vehicledeceleration (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)	2

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

$$E_c = \frac{1}{2} M . {V_0}^2 \qquad (1.1)$$

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 [9]:

$$E_{cf} = \frac{1}{2} . \beta . M . {V_0}^2 \qquad (1.2)$$

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

The braking force applied to each front wheel is equal to [9]:

$$F_p = \frac{E_{cf}}{2 . x_f} \qquad (1.3)$$

$$x_f = \mathrm{V}_0 . t - \frac{1}{2} . \left(\frac{V_0}{t}\right) \mathrm{t}_\mathrm{f}^2 \qquad (1.4)$$

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

$$F_d = \frac{E_{cf}}{2 . \frac{\mathrm{R_d}}{.\mathrm{R_p}} . \left(\mathrm{V}_0 t - \frac{1}{2}\left[\frac{\mathrm{V}_0}{t}\right] \mathrm{t}_\mathrm{f}^2\right)} \qquad (1.5)$$

The braking speed is equal to:

$$V_f = V_0 (1 - \frac{t}{t_f}) \qquad (1.6)$$

For the case of stopping braking, we have $V_f = 0$.
The initial rotation speed of the disk is given by the following relation:

$$\omega_\mathrm{d} = \frac{\mathrm{V}_0}{\mathrm{R_d}} \qquad (1.7)$$

The pressure exerted on the disc by the pads is calculated accordingto [10]

$$p = \frac{F_d}{A_{C\,p} . \mu} \qquad (1.8)$$

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 = 4.5 [s]
- Step of the initial time = 0.25 [s]
- Minimum initial time step = 0.125 [s]
- Maximum initial time step = 0.5 [s]

1.2. Disc and pad materials

The materials commonly used for the manufac-ture of discs are graphite cast irons. In this study, high carbon gray cast iron FG25 was chosen which has good con-ductivity, fairly good mechanical strength and low wear [11], [12]. For the linings, we opted for an organic matrix material characterized by a good friction coefficient (as high and constant as possible, regardless of the variation in temperature, contact pressure or disc rotation speed) [13]. The pad holders are made of mild steel; they sever to distribute the force exerted by the hydraulic piston over the entire surface of the pads in order to guarantee the largest and most homogeneous disk-pad contact area 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) [14].

Properties	(a)	(b)
Density (kg /m^3)	7200	2500
Young's modulus(Pa)	1.1E+11	3E+09
Poisson coefficient	0.28	0.25
Compressibility modulus(Pa)	8.3333E+10	2E+09
Shear modulus (Pa)	4.2969E+10	1.2E+09

1.3. Disc and pad geometry models

Figure 1, Figure 2, Figure 3, and Figure 4 show the geometric models of the disk and pads developed using SolidWorks software.

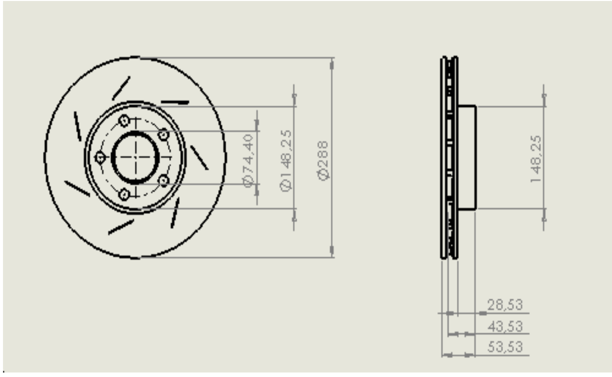

Figure 1. Left view of the disc

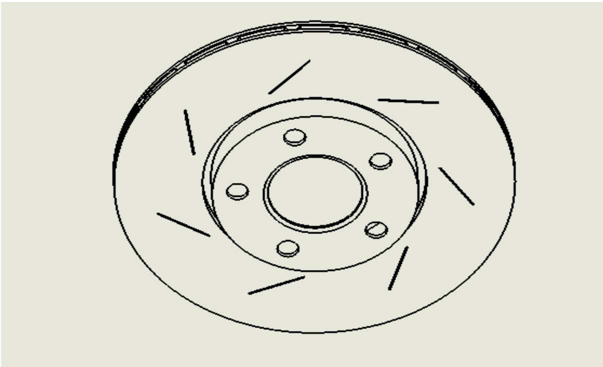

Figure 2. Profile view of the disc

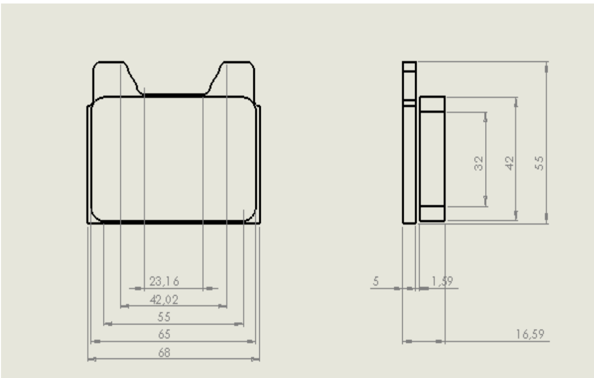

Figure 3. Left view of the brake pad

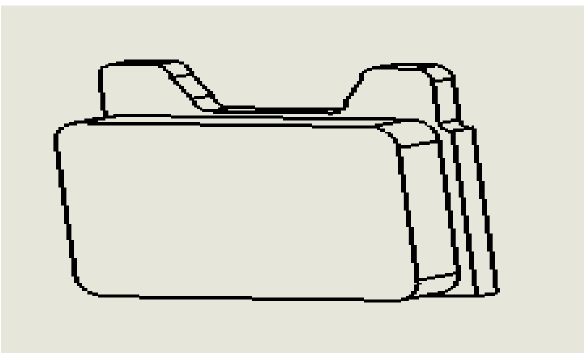

Figure 4. Profile view of the brake pad

1.4. Geometrical models of the disc and pad pair

Figure 5, and Figure 6 respectively show the brake model (disc – pads) and the different pad position configurations that will be exported to the ANSYS computer code for numerical simulation.

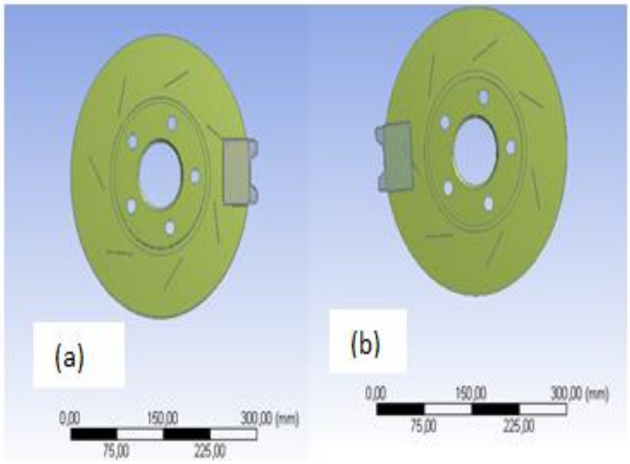

Figure 5. (a) Right brake pads , (b) Left brake pads

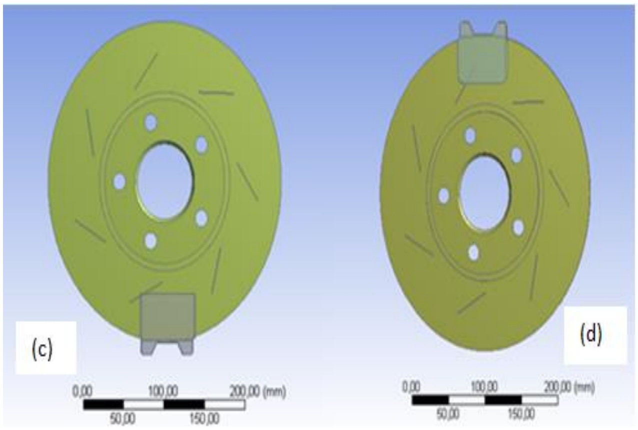

Figure 6.(c) Brakepads at the bottom , (d) Brake pads on top

1.5. Finite element mesh model of disc brake components (disc, pads)

The finite element discretization of the disk assembly and wafers was performed by Ansys Workbench software. Figure 4 gives respectively the volume mesh by default of the disc and of the brake pad. The mesh of the break pad for different knots are presented in Figure 7, Figure 8 and Figure 9.

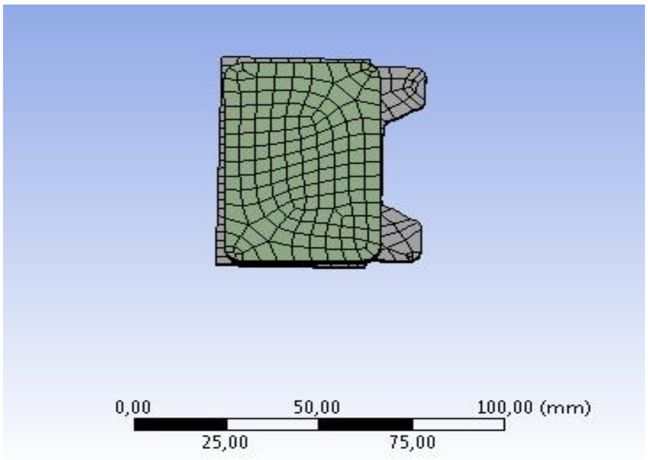

Figure 7. Mesh of a brake pad (Knots 829, Items 1032)

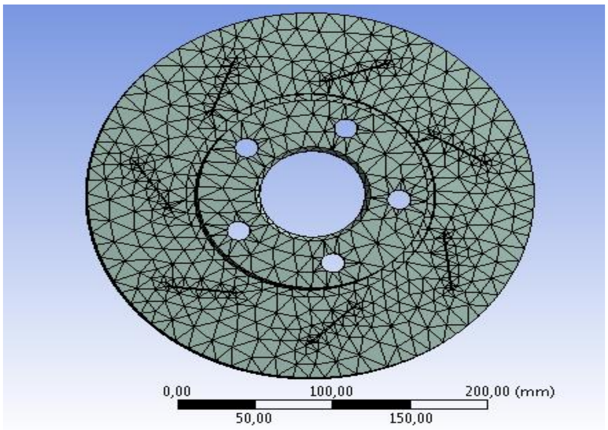

Figure 8. Mesh models of the disc (Knots 27064, Items 16048)

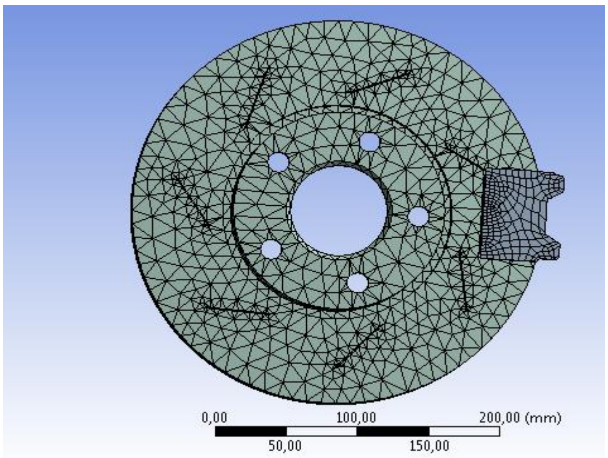

Figure 9. (Grooved disc / pads) (Knots 30786 , Items 16558)

1.6. Boundary conditions

The boundary conditions applied to the disc-pad torque are illustrated in Figure 10, Figure 11.

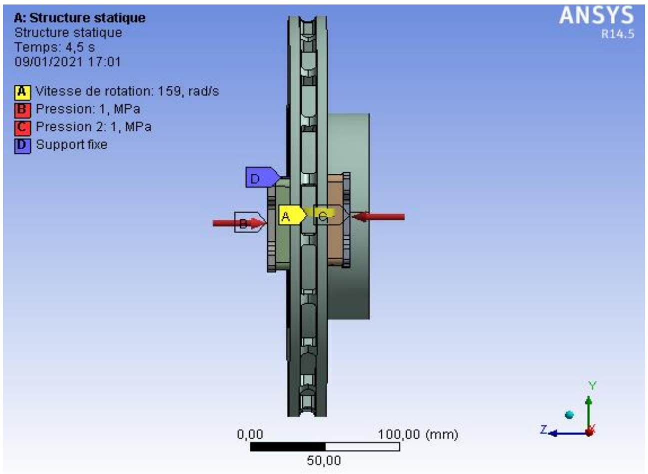

Figure 10. Left view of the boundary conditions and the load applied to the grooved disc / brake pad

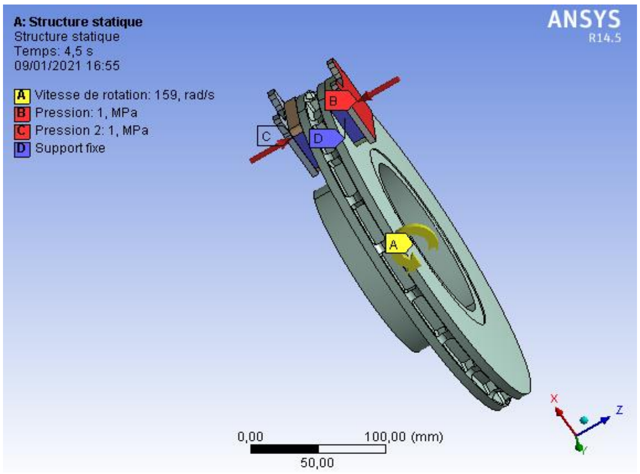

Figure 11. Profile view of the boundary conditions and the load applied to the grooved brake disc / pads

1.7. Thermal conditions applied to the torque (brake disc / pads)

The study is assumed to be thermoelastic considering the transient problem with the following thermal conditions:

• The temperature of the disc is: T = 400C at time t = 0 s,

• A flow of heat in the two contact areas between the disc and the pads,

• Heat exchange by convection over the entire surface of the disc.

2. Results and interpretations

2.1. Distribution of the equivalent elastic strains as a function of the braking time for the four positions

In this comparative study of the mechanical stresses, we chose four configurations of the disc-pads pair having a different position of the pads while keeping the same boundary conditions. Figure 12, Figure 13, Figure 14, and Figure 15 give an overview of the deformation zones on the disc and the pads. It can be seen that the disc has the lowest deformation due to its rigidity and the hardness of the material used (gray cast iron). Gray cast iron is most commonly used in brake discs because it is more resistant to wear and heat. While the deformations at the level of the pads are more important because of the properties of the material and their flexibility.

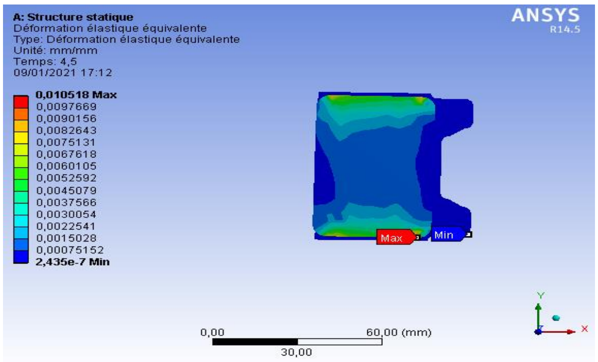

Figure 12. Right brake pads (max value = 0.01051)

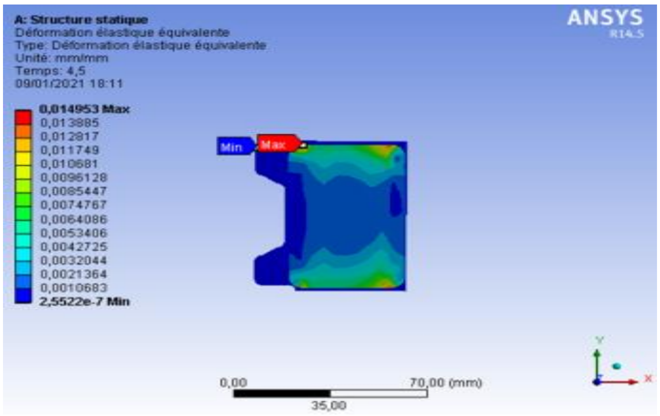

Figure 13. Left brake pads (max value = 0, 01495)

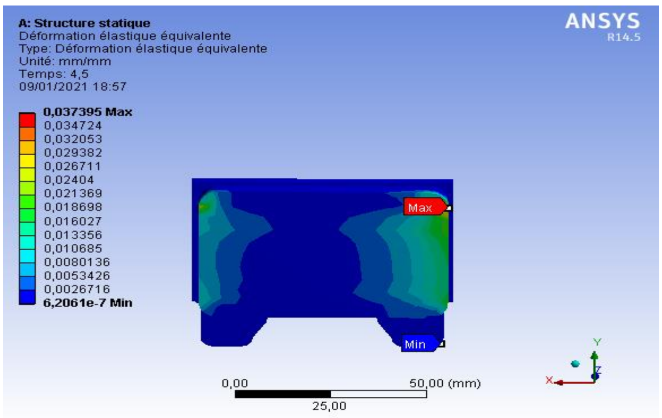

Figure 14. Brake pads at the bottom (max value = 0.037395)

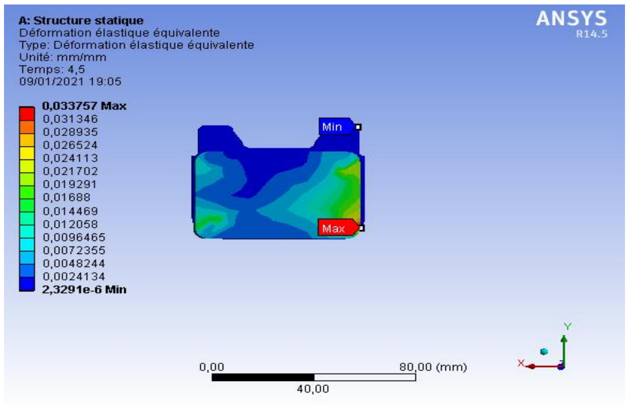

Figure 15. Brake pads on top (max value =0.03375)

2.1.1. Variation of the equivalent elastic strain in the disc as a function of time for the four configurations of the disc/brake pad pair

Figure 16 shows that regardless of the position of the pads, the equivalent elastic strain increases in a non-linear fashion with time. For cases where the brake pads are placed at the bottom and at the top, the equivalent elastic deformation is the same. At the end of braking at t = 4.5s, the maximum equivalent elastic deformation reaches 0.037 mm / mm at the outer edges of the two linings. The time-dependent deformation of the inserts placed on the right is smaller than that of the inserts placed on the left. At the end of braking, it is 0.01051mm / mm for the case of the pads on the left and 0.01495 mm / mm for the pads on the right. This can be explained through certain influencing factors such as the phase and direction of attack of the pads, the mounting positions, pollution of the friction surface (metal encrustation on the friction surface), the effect thermal (overheating of the pads and the disc at the end of braking). All of these factors lead to accelerated wear and deformation of the pads.

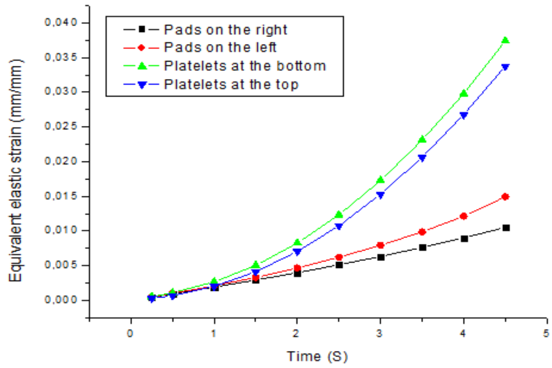

Figure 16. Variation of the equivalent elastic strain in the disc as a function of time

2.2 Equivalent stresses of Von Mises

Figure 17, Figure 18, Figure 19, and Figure 20 show the state of the equivalent Von Mises stresses for the four configurations. In each of the four variants, it can be seen that the equivalent stresses are concentrated above all at the level of the grooved surface of the contact zone between the pad and the disc and decrease in the direction of the bowl of the disc and the ventilation fins. The maximum value obtained at the level of the groove can cause cracks in the groove and even its rupture.

From the results obtained, we note that for the pads located at the top and bottom of the disc the equivalent stresses are higher than those of the pads placed to the right and left of the disc. The stresses in the brake mounts are smaller than that in the disc. It can be concluded that the presence of the groove in the contact zone and its position relative to the pads during contact with the disc have an influence on the equivalent stresses.

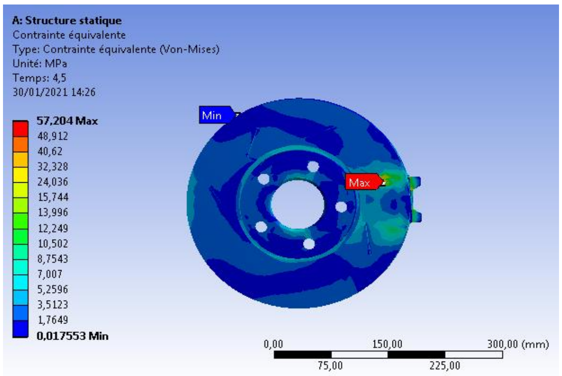

Figure 17. Right brake pads (σ_{eqmax}= 57,204MPa)

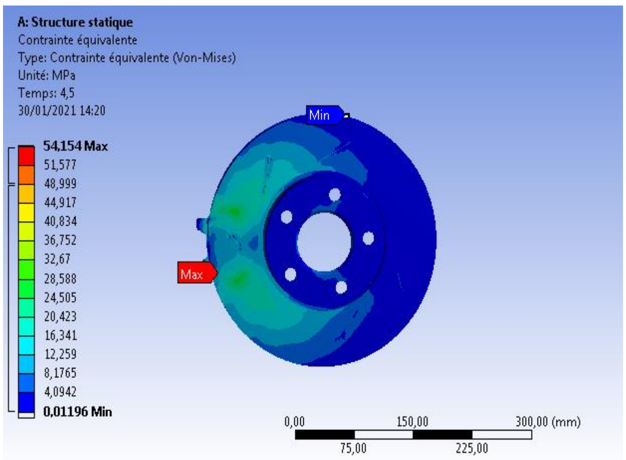

Figure 18. Left brake pads (σ_{eqmax}= 54,154MPa)

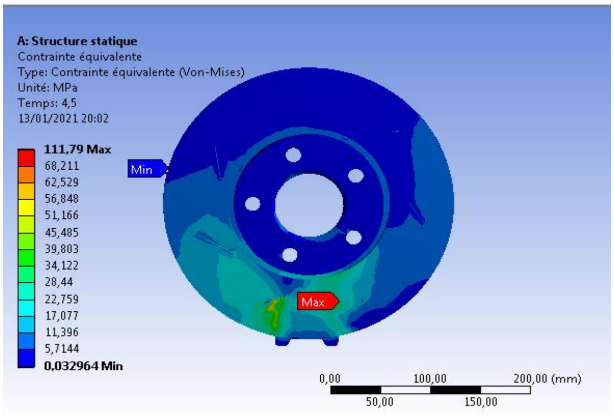

Figure 19. Brake pads at thebottom ($\sigma_{eq\ max}$ = 111,79MPa)

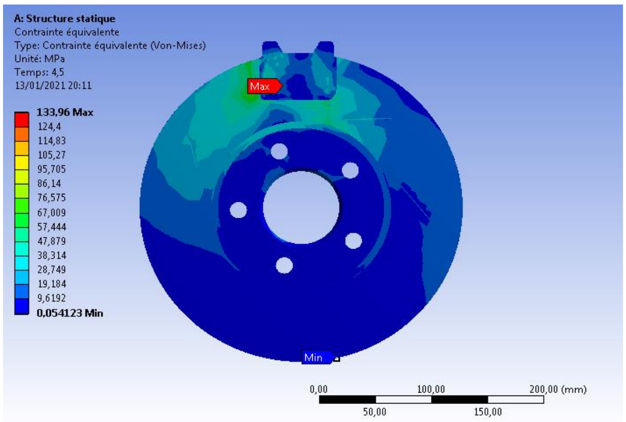

Figure 20. Brake pads on top (σ_{eqmax} = 133,96MPa)

2.2.1. Variation of the equivalent stress as a function of time for the four configurations of the disc / pad pair

The graph in Figure 21 shows that the equivalent stress in the disk varies in a non-linear fashion with time. For the pads located at the top of the disc, the equivalent stress is greater than that of the pads placed at the bottom of the disc. At times t = 4.5 s, it is of the order of 133.96 MPa for the pads located at the top of the disc and 111.79 MPa for the pads located at the bottom of the disc.

In the case of the pads placed to the right and to the left of the disc, the equivalent stresses are almost identical, but less important than those of the previous variants. At time t = 4.5s, it is of the order of 57.204 MPa for the pads on the right and 54.154MPa for the pads on the left. They are half of the previous variants. Note that the equivalent stress is greater in the grooved contact zone between the disc and the pads than the other parts of the disc.

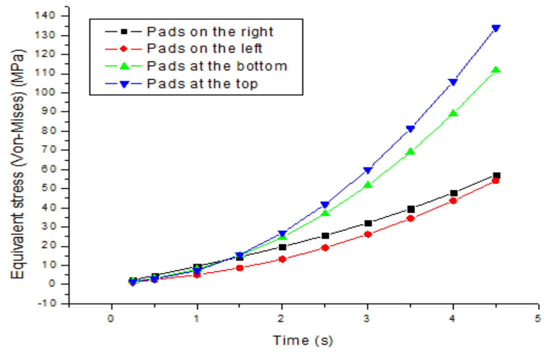

Figure 21. Variation of the equivalent stress $\sigma_{eq\ max}$ as a function of time

2.3. Effect of the parameters on the equivalent stresses of the disc-pad couples

This part of the study is devoted to the influence of a certain number of parameters considered significant, such as the speed of rotation of the disc, the coefficient of friction, the pressure applied on the brake pads and the choice of the material of the disc on the Von Mises stress distribution in the disc and pads when braking.

2.3.1. Influence of the rotational speed of the brake disc

Figure 22, Figure 23, Figure 24, Figure 25, and Figure 26 show the evolution of Von Mises stresses in the disc-pad torque by increasing the initial rotational speed of the brake disc.

These figures show that during the braking phase, the equivalent stress increases significantly with the increase in the initial rotational speed of the disc, especially in the contact area and in the ends of the pads. Conversely, it decreases in the direction of the ventilation fins and the perforated bowl of the disc.

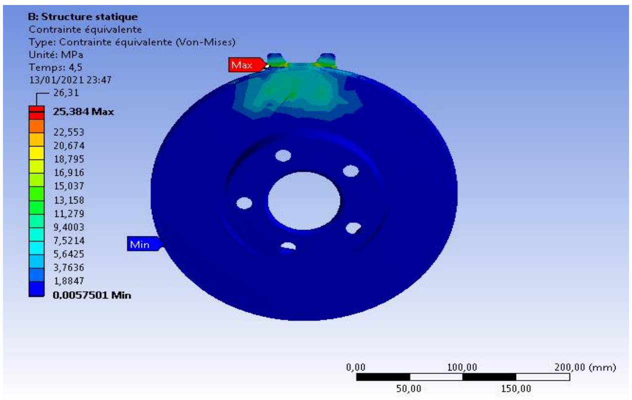

Figure 22. ω =30 rd/s (σmax =25,384MPa)

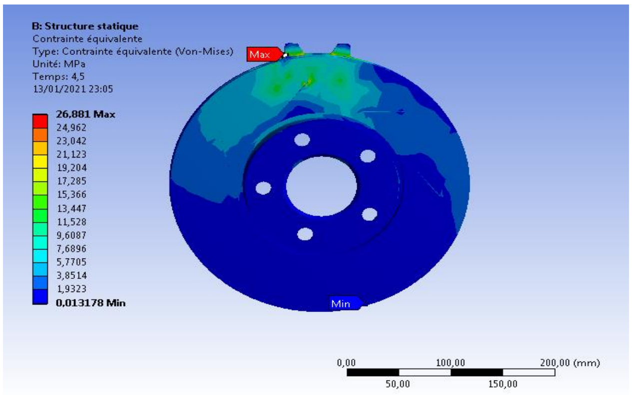

Figure 23. ω=60rd/s(σmax=26,881MPa)

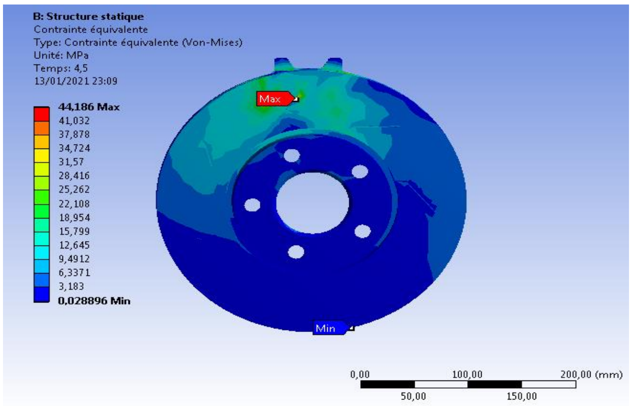

Figure 24. ω=90 rd/s(σmax=44,186MPa)

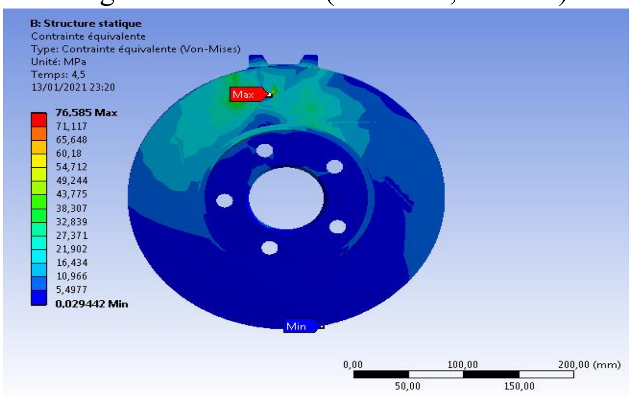

Figure 25. ω=120 rd/s(σmax=76,585MPa)

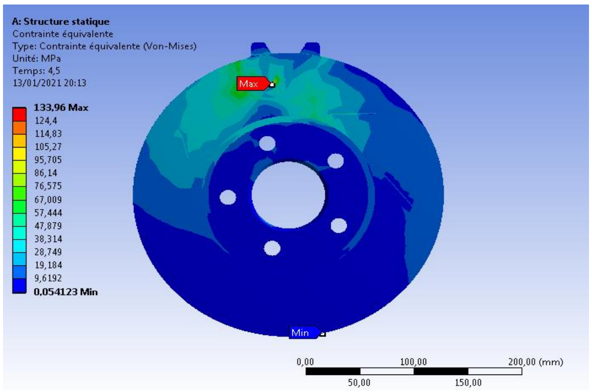

Figure 26. ω =159 rd/s(σmax=26,31Mpa)

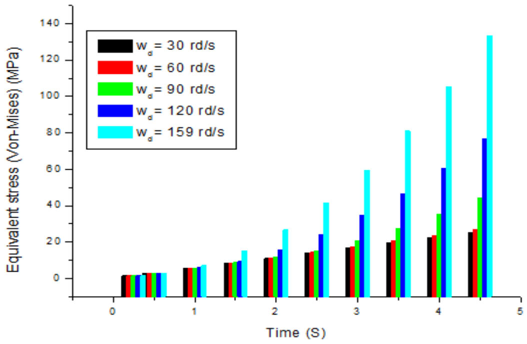

Figure 26. Variation of the equivalent stress as a function of the speed of rotation of the disc

2.3.2. Influence of the coefficient of friction

The influence of the coefficient of friction on the equivalent stress of the disc-pad torque is shown in Figure 27. We see that the equivalent stress increases with the time of braking, but it decreases slightly with the increase of the coefficient of friction. The maximum value of the stress is located at the level of the contact zone at the end of braking.

The influence of the coefficient of friction on the equivalent stress of the disc-pad torque is shown in Figure 27. We see that the equivalent stress increases with the time of braking, but it decreases slightly with the increase of the coefficient of friction. The maximum value of the stress is located at the level

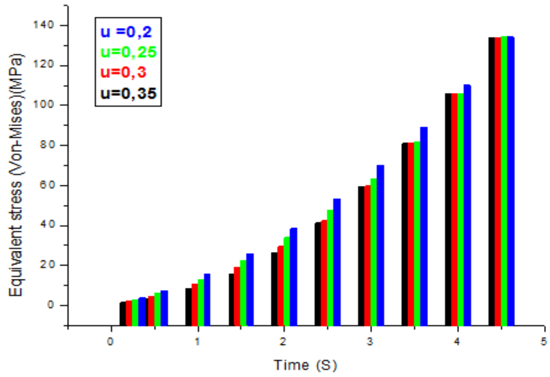

Figure 27. Variation of the equivalent Von-Mises stress as a function of the friction coefficient

2.3.3. Influence of the disc material

Table 3. Mechanical characteristics of the brake discs (gray cast iron and stainless steel) [14], [15].

Properties	**stainless steel**	**Grey cast iron**
Density (kg/m^3)	7750	7200
Young's modulus (Pa)	1.93E+11	1.1E +11
Poisson coefficient	0.31	0.28
Compressibility modulus (Pa)	1.693E+11	8.3333E+10
Shear modulus (Pa)	7.3664E+10	4.2969E+10

Figure 28 shows the evolution of the equivalent stress for the two materials studied as a function of time. We first notice that the maximum value of the equivalent stress of the stainless steel disc is higher than that of gray cast iron. The difference in equivalent stress for these two materials is all the greater with increasing time. At the end of braking, this difference is of the order of 39.14 MPa. As mentioned previously, although the equivalent stress of the gray cast iron disc is smaller than that of the stainless steel disc, the gray cast iron disc is however distinguished by very good resistance to corrosion and deformation at hot. The mechanical characteristics are presented in Table 3.

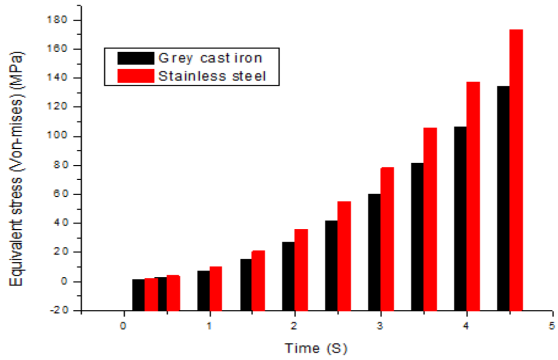

Figure 28. Von Mises stress as a function of time for the two disc materials (stainless steel / gray cast iron)

2.3.4. Influence of the pressure applied to the brake pads

In the braking system, the pads press the disc and generate friction during braking. Mechanical loading is represented by the pressure of a hydraulic piston applied to the pads.

The variation of the equivalent stresses under the influence of the applied pressure is presented in

Figure 29. It is noted that the increase in the value of pressure also has a role on the elastic behavior of the couple disc / pads since the equivalent stresses of Von Mises increases with the pressure exerted.

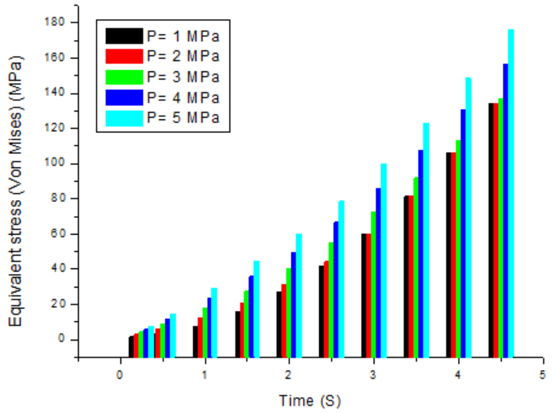

Figure 29. Von Mises equivalent stress as a function of the pressure applied to the brake pads.

3. General conclusion

1. The numerical analysis of the transient thermoelastic behavior of the dry sliding contacts of the disc-wafer pair was carried out using the ANSYS 14.5 calculation code based on the finite element method. We have been able to show the influence of the position of the brake pads on the disc and of certain significant braking parameters such as the coefficient of friction, the choice of material and the initial speed of rotation of the disc, the pressure applied to the pads. Brake pressure, and thermal conditions (disc temperature, heat flow through the disc and heat exchange by convection over the entire surface of the disc). The stresses and strains of each configuration of the disc-pad couple are expressed as a function of the braking time.

2. The comparative study of the four variants of the disc-pad pair shows that the maximum stresses are concentrated at the level of the grooved surface of the disk (contact zone between the pad and the disk), then they decrease in the direction of the base of the bowl of the disk, and ventilation fins. The pads located at the top and bottom of the disc give greater stresses than those of the pads placed to the right and left of the disc; this is due to the position of the groove.

3. Numerical analysis of the thermoelastic behavior of dry sliding contacts shows that the coefficient of friction has a low influence on the equivalent stress of a disc regardless of the braking time. The equivalent stress decreases with the increase in the coefficient of friction.

4. The results show that the stress field and the strain field depend not only on the coefficient of friction, but on other parameters such as; the initial speed of rotation, the type of loading applied to the pads (the pressure), the temperature of the disc, the choice of the material of the disc as well as the variation of the braking time.

5. The equivalent stress in the contact area increases with increasing initial rotational speed of the disc. The maximum equivalent stress that a stainless steel disc can withstand is higher than that of gray cast iron. Despite the fragility of gray cast iron, the latter has a technological advantage such as good resistance to corrosion and hot deformation.

6. The influence of the pressure also has a role on the thermoelastic behavior of the dry sliding contact between the disc and the pads.

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Conflict of Interest

The authors have no conflict of interest to declare.