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Original Research Article

Structural and contact analysis of disc brake assembly during single stop braking event

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Abstract
An automobile disc brake system is used to perform three basic functions, i.e. to reduce speed of a vehicle, to maintain its speed when travelling downhill and to completely stop the vehicle. During these braking events, the disc brake may suffer of structural and wear issues. It is quite sometimes that the disc brake components fail structurally and/or having severe wear on the pad. Thus, this paper aims to examine stress concentration, structural deformation and contact pressure of brake disc and pads during single braking stop event by employing commercial finite element software, ANSYS. The paper also highlights the effects of using a fixed calliper, different friction coefficients and different speeds of the disc on the stress concentration, structural deformation and contact pressure of brake disc and pads, respectively. Results from the investigation could provide a better explanation of the variation in contact pressure distribution and in turn squeal generation. Thus, this study provides effective reference for design and engineering application of brake disc and brake pad.

Key words: Disc brake, Von Mises stress, Structural deformation, Contact pressure, Finite element

Nomenclature

M	vehicle mass, kg
v_0	initial velocity , m/s
t_{stop}	time to stop , s
R_{rotor}	effective rotor radius, m
R_{tire}	tire radius , m
μ	friction coefficient disc/pad
F_{disc}	rotor force, N
A_c	surface of the braking pad ,m
P	surface pressure for a single pad , MPa

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1. Introduction

Passenger car disc brakes are safety-critical components whose performance depends strongly on the contact conditions at the pad-to-rotor interface. When the driver steps on the brake pedal, hydraulic fluid is pushed against the piston, which in turn forces the brake pads into contact with the rotor. The frictional forces at the sliding interfaces between the pads and the rotor retard the rotational movement of the rotor and the axle on which it is mounted [1]. The kinetic energy of the vehicle is transformed into heat that is mainly absorbed by the rotor and the brake pad.

The frictional heat generated on the interface of the disc and the pads can cause high temperature. Particularly, the temperature may exceed the critical value for a given material, which leads to undesirable effects, such as brake fade, local scoring, thermo elastic instability, premature wear, brake fluid vaporization, bearing failure, thermal cracks, and thermally excited vibration as referred to [2,3]. Gao and Lin [2] stated that there was considerable evidence to show that the contact temperature is an integral factor reflecting the specific power friction influence of combined effect of load, speed, friction coefficient, and the thermo physical and durability properties of the materials of a frictional couple. Lee and Yeo [3] reported that uneven distribution of temperature at the surfaces of the disc and friction pads brings about thermal distortion, which is known as coning and found to be the main cause of judder and disc thickness variation (DTV). AbuBakar et al [4] in their recent work found that temperature could also affect vibration level in a disc brake assembly. Valvano and Lee [5] simulated thermal analysis on a disc brake with a combination of computer-based thermal model and finite element-based techniques to provide a reliable method to calculate the temperature rise, thermal stress and distortion under a given brake schedule. Wolejsza et al [6] performed analysis on the thermo-mechanical behavior of airplane carbon composite brakes using MSC/Marc finite element software which allows accurate simulation of the transient heat transfer phenomenon coupled to disc deformations caused by frictional sliding contact.

There are three types of mechanical stresses subjected by the disc brake. The first one is the traction force created by the centrifugal effect due to the rotational of the disc brake when the wheel is rotating and no braking force is applied to the disc. During braking operation, there are another two additional forces experienced by the disc brake. Firstly, compression force is created as the result of the force exerted by the brake pad pressing perpendicular onto the surface of the disc to slow it down. Secondly, the braking action due to the rubbing of the brake pad against the surface of the disc brake is translated into frictional or traction force on the disc surface which acts in the opposite direction of the disc rotation.

A disc brake of floating caliper design typically consists of pads, calliper, carrier, rotor (disc), piston, and guide pins. One of the major requirements of the calliper is to press the pads against the rotor and should ideally achieve as uniform interface pressure as possible. A uniform pressure between the pads and rotor leads to uniform pad wear and brake temperature, and more even friction coefficients as cited in [7]. Unevenness of the pressure distribution could cause uneven wear and shorter life of pads. It has also speculated that they may promote disc brake squeal. The interface pressure distributions have been investigated by a number of people. Tirovic and Day [8] studied the influence of component geometry, material properties and contact characteristics on the interface pressure distribution. They used a simple and nonvalidated, three-dimensional model of the disc brake. Tamari et al. [9] presented a method of predicting disc brake pad contact pressure for certain operating condition by means of experimental and numerical method. They developed a quite detailed model and validated the model by fitting the numerical deformations of the disc brake components with experimental results. Hohmann et al. [10] also presented a method of contact analysis for the drum and disc brakes of simple threedimensional models using ADINA software package. They showed a sticking and shifting contact area in their results. Like [8], validation of their model was not made. Ripin [11] developed a simple, validated threedimensional finite element model of the pad, and applied rather simple piston and finger force onto the back plate interface in his analysis. He studied the contact pressure distribution at the disc/pad interface, where gap elements were used to represent contact effect.

In this paper, structural analysis is performed on a simple finite element (FE) model of a real disc brake assembly to obtain the contact pressure distributions on the friction pads and the Von Mises Stress in disc interface by utilizing the ANSYS 11.0 FE software [12]. Sensitivity study on rotation of the disc, load pattern and coefficient of friction is also performed.

2. Finite element model and simulation

In this work, a three dimensional CAD and FE model consists of a ventilated disc and two pads with single slot in the middle as illustrated in Fig.1 and Fig. 2, respectively. The selected material of the disc is Gray cast iron FG 15 with high Carbon content and the brake pad has an isotropic elastic behavior whose mechanical characteristics of the two parts are presented in Table 1. The materials of the disc and the pads are homogeneous and their properties are invariable with the temperature.

A commercial FE software, namely ANSYS 11 (3D) is fully utilized to simulate structural deformation and contact pressure distributions of the disc brake during single braking stop application.

Table 1. Mechanical properties of the disc and pad

Properties	**Disc**	**Pad**
Young modulus E	138	1
Poisson's ratio ν	0.3	0.25
Density ρ (kg/m^3)	7250	1400
Coefficient of	0.2	0.2

A commercial FE software, namely ANSYS 11 (3D) is fully utilized to simulate structural deformation and contact pressure distributions of the disc brake during single braking stop application. Boundary conditions are imposed on the models (disc-pad) as shown in Fig. 3a for applied pressure on one side of the pad and Fig.3b for applied pressure on both sides of the pad. The disc is rigidly constrained at the bolt holes in all directions except in its rotational direction. Meanwhile, the pad is fixed at the abutment in all degrees of freedom except in the normal direction to allow the pads move up and down and in contact with the disc surface. In this study, it is assumed that 60% of the braking forces are supported by the front brakes (two rotors) as cited in [13]. By using vehicle data as given in Table 2 and Eqs (1)-(3), braking force on the disc, rotational speed and brake pressure on the pad can be calculated, respectively.

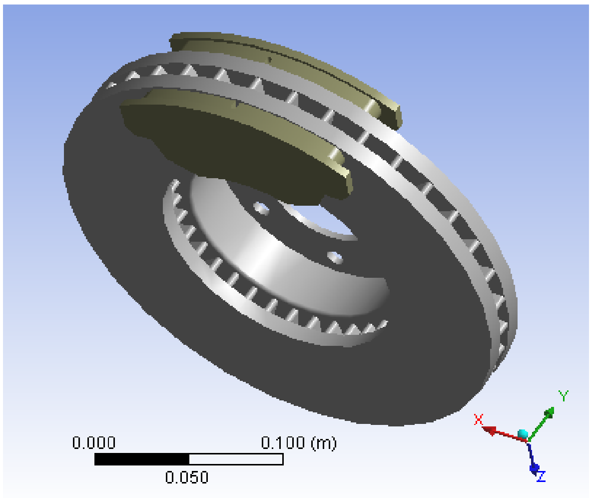


Fig. 1. CAD model of the disc and pads

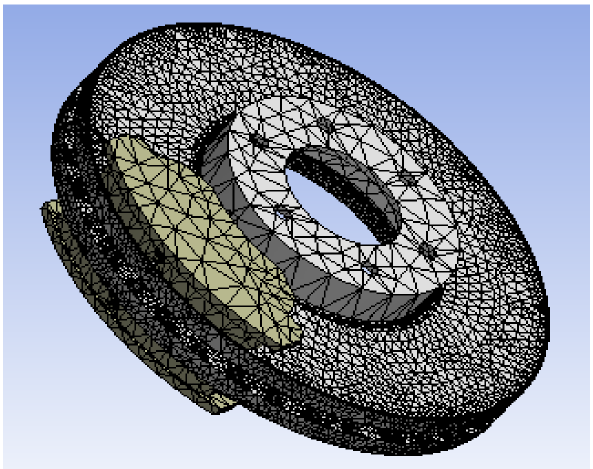
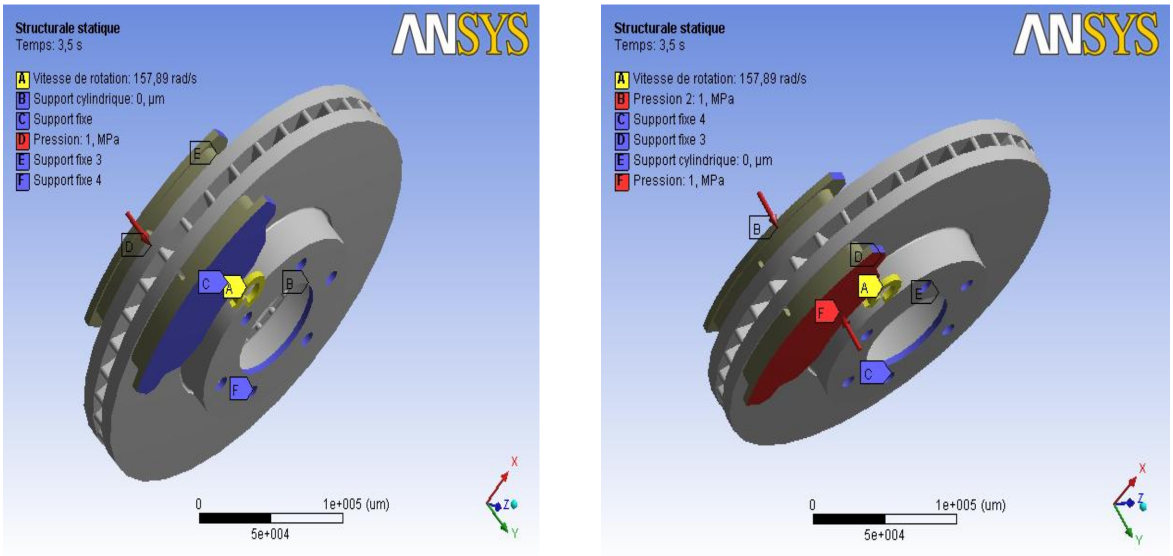


Fig. 2. FE model of the disc and pads



(a) One side (b) Two sides

Fig. 3. Boundary conditions and loading imposed on the disc-pads

Table 2. Vehicle data

Item	Value
Mass of the vehicle ,M [kg]	1385
Initial velocity – v_0 [m/s]	60
Time to stop t_{stop} s	45
Effective Rotor radius - R_{rotor}[mm]	101
Radius of the wheel - R_{tire}[mm]	380
The coefficient of friction disc/pads	0.2
Surface of the pad A_c [mm^2]	5246

$$F_{disc} = \frac{(30\%)\frac{1}{2}Mv_0^2}{2.\frac{R_{rotor}}{R_{tire}}\left(v_0.t_{stop} - \frac{1}{2}\left\{\frac{v_0}{t_{stop}}\right\}t_{stop}^2\right)} = 1047.36[N] \quad (1)$$

The rotational speed of the disc is calculated as follows:

$$\omega = \frac{v_0}{R_{tire}} = 157.89\ rad/s \quad (2)$$

The external pressure between the disc and the pads is calculated by the force applied to the disc; for a flat track, the hydraulic pressure is as referred to [14]:

$$P = \frac{F_{disc}}{A_c.\mu} = 1[MPa] \quad (3)$$

Where A_c is the surface of the pad in contact with the disc and μ the coefficient of friction.

3. Results and discussion

3.1. Von Mises Stress distribution

Fig.4. shows distributions of the equivalent Von Mises stress over braking period and it is shown that the highest stress occurs at the bolt holes at the time t = 0.25s. This is due to the disc having experience in torsion and shear modes. This high stress concentration can cause a rupture to the bolt holes.

3.2. Contact pressure distribution

Fig.5 illustrates contact pressure distributions of the inner pad at different braking times. It shows that the contact pressure increases gradually and reaches its maximum value of Pmax = 1.8 MPa at the end of braking period. It is believed that the rise in pressure on the contact surface can also cause a rise in the temperature of the disc and wear of the pads. At the leading side and inner radius of the pad, contact pressure is seen to be higher compared to the other regions.This is due to this area is mostly in contact with the disc surface. Fig 6 shows the evolution of contact pressures along angular positions of the pad. The maximum value of the contact pressure is located at the leading edge and at the level of the lower edge of the pad. Contact pressure distributions of the outer pad are depicted in Fig. 7. It shows that the maximum contact pressure is predicted in the middle of the pad with the value of 1.3 MPa. This is much lower than that obtained on the inner pad.

3.3. Effect of a fixed caliper

For a comparative study, the effect of a fixed caliper (disc with double pressure) is also simulated where it maintains the same boundary conditions used in the case of a single-piston caliper. Fig.8.shows the levels of equivalent Von Mises stresses in a section of a disc brake at the end of braking period. Unlike the case of the disc with a single-piston caliper, it is noted that the highest stress appears at the outer side of the fins with the value of 8.3 MPa. This is lower than the stress predicted at the bolt holes with the value of 31.4 MPa for single-piston caliper. It is also found that the stress is well distributed to the disc interface compared with stress predicted in Fig. 5.

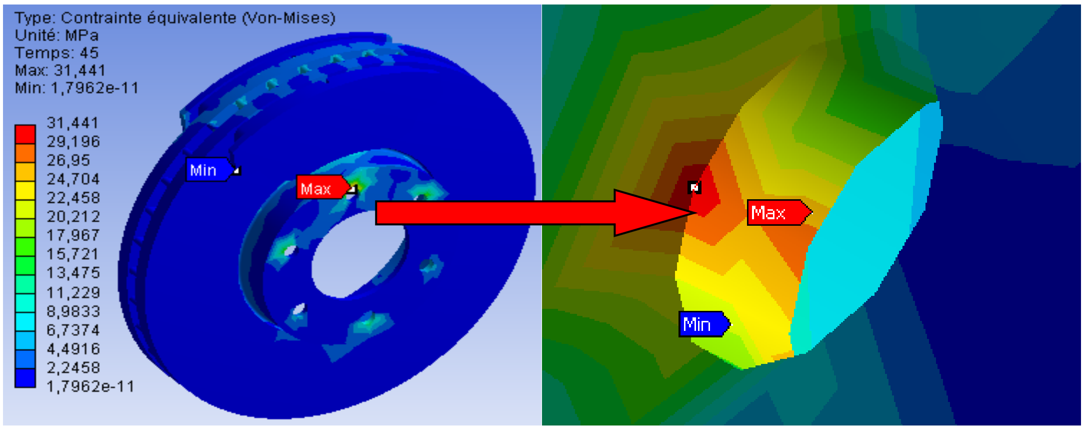
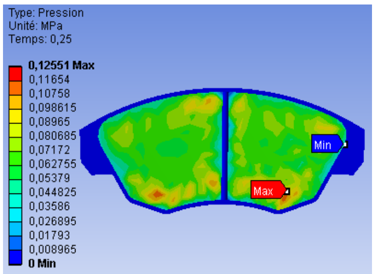
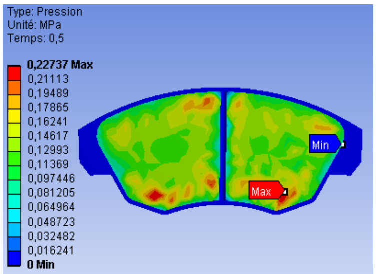


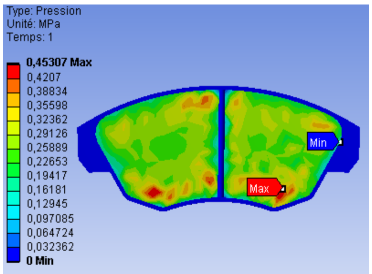
Fig. 4. Stress concentration at the bolt holes



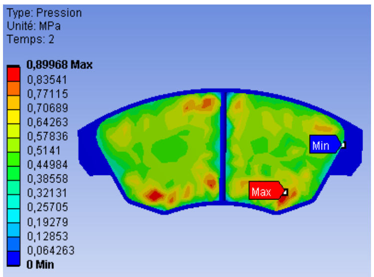
-a- *: at time t= 0,25 s*



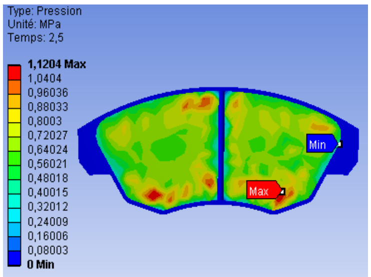
-b- *: at time t= 0,5 s.*



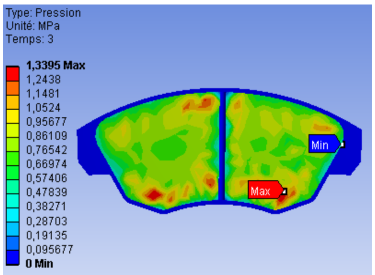
-c- *: at time t= 1 s.*



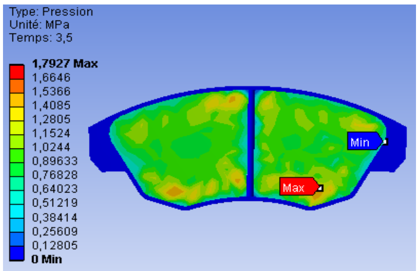
-d- *: at time t=2 s.*



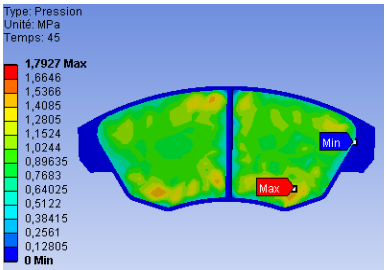
-e- *: at time t= 2,5 s.*



-f- *: at time t= 3 s.*



-g- *: at time t= 3,5 s.*



-h- *: at time t= 45 s.*

Fig. 5. Contact pressure distribution on the inner pad

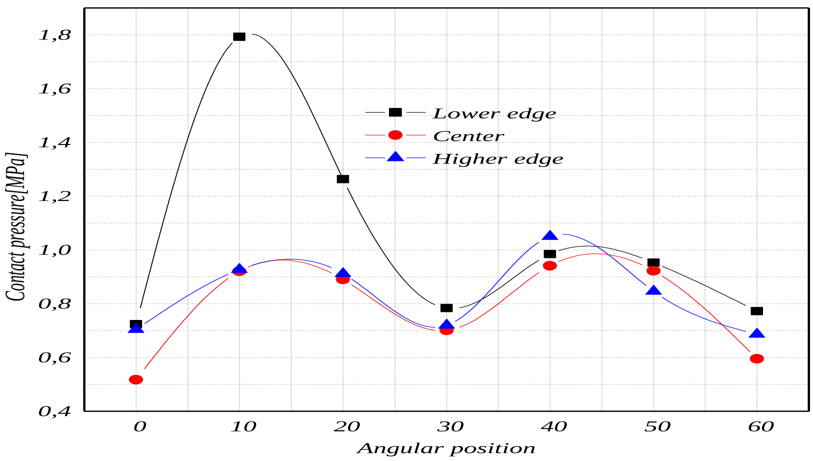
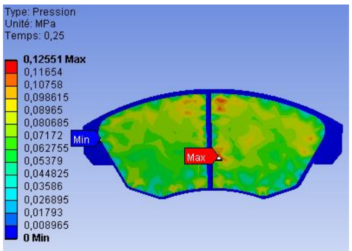
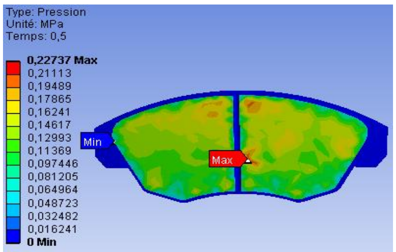


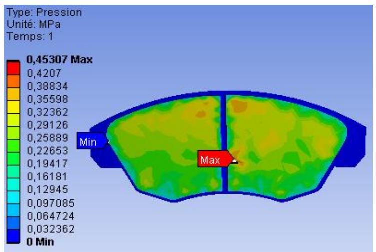
Fig. 6. Variation of contact pressures according to the angular position in the inner pad



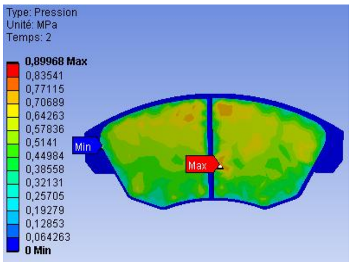
***-a-** : at time t= 0,25 s.*



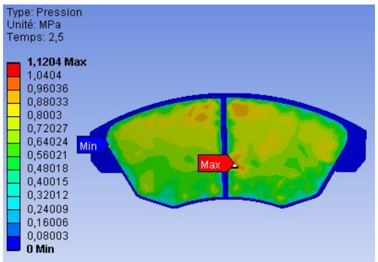
***-b-** : at time t= 0,5 s.*



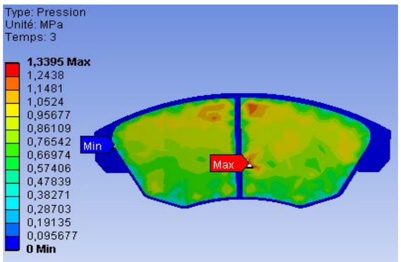
***-c-** : at time t= 1 s.*



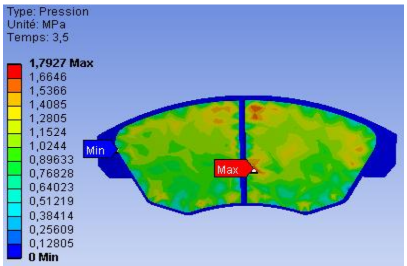
***-d-** : at time t=2 s.*



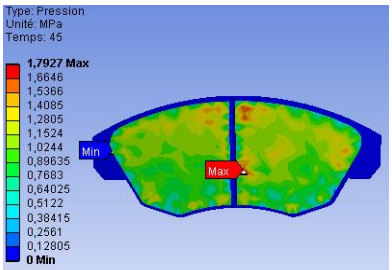
***-e-** : at time t= 2,5 s.*



***-f-** : at time t= 3 s.*



***-g-** : at time t= 3,5 s.*



***-h-** : at time t= 45 s.*

Fig. 7. Contact pressure distribution on the outer pad

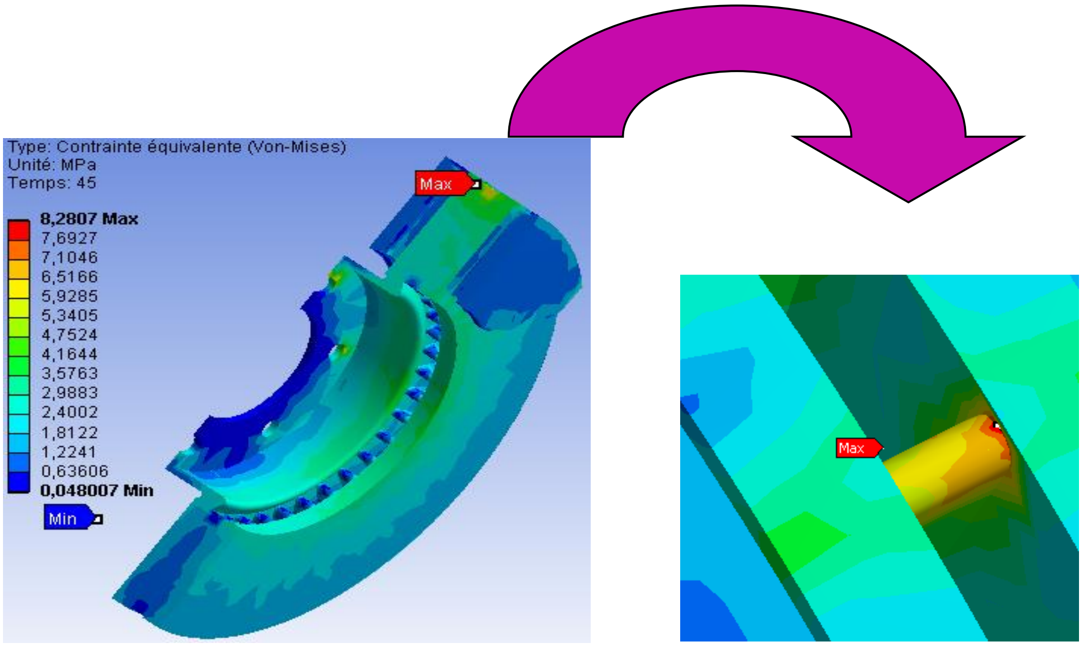


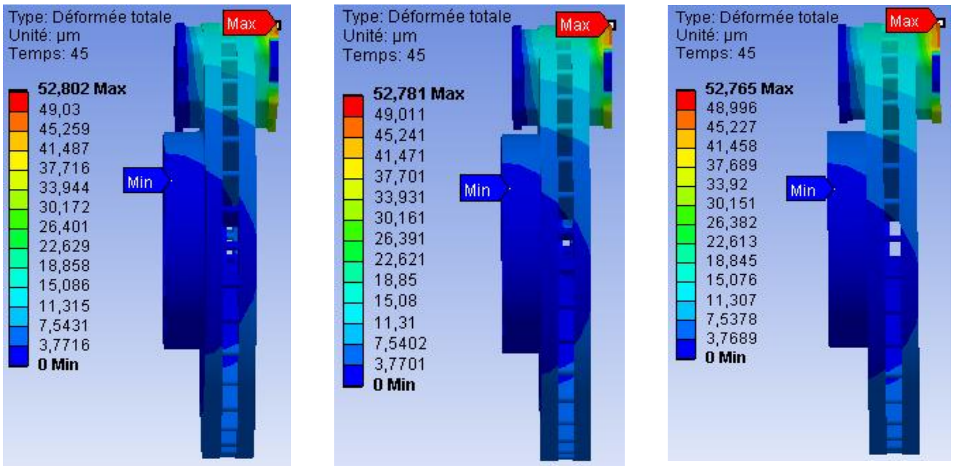
Fig. 8. Von Mises stresses

3.4. Effect of friction coefficient

It is interesting to see deformation behaviour of the disc and the pads with respect to variation of friction coefficient from 0.25 to 0.35. Fig.9. shows the different configurations of the total deformation of the model in the final stage of braking. It is clearly seen that the total deformation is slightly decreased with the increase of friction coefficient. Indeed, the high mechanical advantage of hydraulic and mechanical disc brakes allows a small lever input force at the handlebar to be converted into a large clamp force at the wheel. This large clamp force pinches the rotor with friction material pads and generates brake power. The higher the coefficient of friction for the pad, the more brake power will be generated coefficient of friction can vary depending on the type of material used for the brake rotor. If the value of the coefficient of friction is increased, the disc is slowed down by friction forces which are opposed to its movement, and the maximum deformation that it undergoes is less significant.

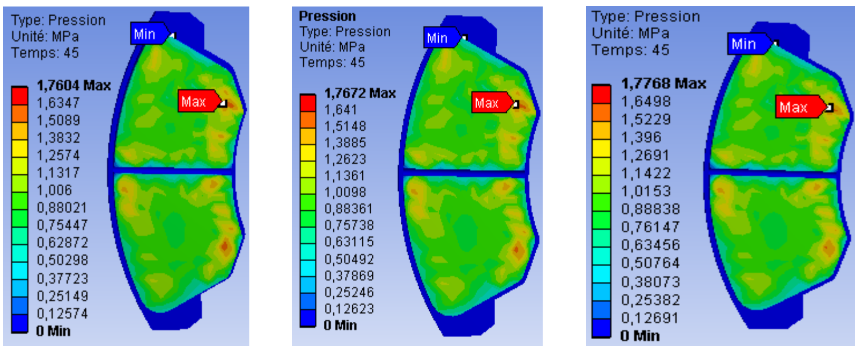
3.5. Effect of disc speed

Fig.10.shows prediction of contact pressure distributions at three different speeds of the disc. It is found that contact pressure distribution is almost identical in all three cases and its value increases with the increase of the angular velocity of the disc. This was also confirmed by Abu Bakar et al. [15]. It is believed that this increase can create the wear of the pads as they can leave deposits on the disc, giving rise to what is called "the third body." It is noted that the maximum contact pressure is produced on the pad at the leading edge.



-a- : μ =0,25 ***-b- :*** μ =0,30 ***-c- :*** μ =0,35

Fig. 9. Total deformation at the end of braking period



-a- : ω *=60 rad/s* ***-b- :*** ω *=90 rad/s* ***-c- :*** ω *=120 rad/s*

Fig. 10. Contact pressure distributions

4. Conclusion

This paper presents structural and contact analysis of a reduced brake model without considering thermal effects. The analysis is performed using commercial FE software package, ANSYS where the FE model only consists of a disc and two pads. From the single stop braking simulation it is found that:

- the bolt holes and outer side of the fins could first damage due to high stress concentration for single and double piston case, respectively
- contact pressure is predicted higher at the leading side compared to the trailing side and its value slightly increases with the increase of disc rotation speeds
- there is no significant change in disc-pad deformation with respect to the variation of friction coefficient

It is interesting to see how these prediction results match to the experimental data. This is the subject of the authors having the experimental means and works of future extension.

Regarding the outlook, there are two recommendations for the work related to disc brake that can be done to further understand the effects of thermo-mechanical contact between the disc and pads, the recommendations are as follows:

1) Tribological and vibratory study of the contact disc – pads;
2) Study of dry contact sliding under the macroscopic aspect (macroscopic state of the surfaces of the disc and pads).

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