Study Of The Thermo-Mechanical Behaviour Of Dry Contacts In The Brake Discs With A Grey Cast Iron Composition (FG260) «Application Of Software Ansys V13.0»

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Abstract

Vehicle braking system is considered as one of the most fundamental safety-critical systems in modern vehicles as its main purpose is to stop or decelerate the vehicle. The frictional heat generated during braking application can cause numerous negative effects on the brake assembly such as brake fade, premature wear, thermal cracks and disc thickness variation. The ventilated pad-disc brake assembly is built by a 3D model with a thermo-mechanical coupling boundary condition and multi-body model technique. The numerical simulation for the coupled transient thermal field and static stress field is carried out by sequentially thermal-structural coupled method based on ANSYS V13.0 to evaluate the stress fields and of deformations which are established in the disc had with the pressure of the pads. For this study various models of ventilated disc have been prepared and there coupled thermo-mechanical analysis is carried out.

Keywords: -Brake Discs, Pads, Heat flux, Heat transfer coefficient, Thermo-mechanical coupling, Von Mises Stress.

1. Introduction

The braking process is in fact the matter of energy balance. The aim of braking system is to transform mechanical energy of moving vehicle into the some other form, which results by decreasing of vehicle speed. The kinetic energy is transformed into the thermal energy, by using the dry friction effects and, after that, dissipated into the surroundings [1] In 2011, Ali Belhocine and Mostefa Bouchetara [1] did the study on Thermo-mechanical behaviour of dry contacts in disc brake rotor with a grey cast iron composition The modeling of transient temperature in the disc is actually used to identify the factor of geometric design of the disc to install the ventilation system in vehicles. The thermal-structural analysis is then used coupling to determine the deformation established and the Von Mises stresses in the disc. In 2012, Mesut Duzgun. [2] did a study on the Investigation of thermo-structural behaviours of different ventilation Applications on brake discs. In this study, the thermal behaviours of ventilated brake discs using three different configurations were investigated at continuous brake conditions in terms of heat generation and thermal stresses with finite element analysis. The results were compared with a solid disc. Heat generation on solid brake discs reduced to a maximum of 24% with ventilation applications. In 2012, M. Pevec et al [3] have studied prediction of the cooling factors of a vehicle brake Disc and its influence on the results of a thermal numerical simulation. In this study the common method that is used for predicting the temperatures in the brake disc during braking is numerical simulation analysis. With the help of Computational Fluid Dynamics, the flow through a vehicle ventilated brake disc of known geometry was determined, and the wall heat transfer coefficients for all vehicle speeds and brake disc. In 2012, Sung Pil Jung et al [4] have studied Thermal Characteristic Analysis and Shape Optimization of a Ventilated Disc. In this study, an analysis technique that can estimate the temperature rise and thermal deformation of the ventilated disc considering vehicle information, braking condition and properties of the disc and pad is developed. In 2010, Pyung Hwang and Xuan Wu [5] have studied Investigation of temperature and thermal stress in ventilated disc brake based on 3D thermo-mechanical coupling model. In this study, object of the present study is to investigate the temperature and thermal stress in the ventilated disc-pad brake during single brake. The brake disc is decelerated at the initial speed with constant acceleration, until the disc comes to a stop. The ventilated pad-disc brake assembly is built by a 3D model with a thermo-mechanical coupling boundary condition and multi-body model technique.

2. Heat flux entering the disc

In a braking system, the mechanical energy is transformed into a calorific energy. This energy is characterized by a total heating of the disc and pads during the braking phase. The energy dissipated in the form of heat can generate rises in temperature. Generally, the thermal conductivity of material of the brake pads is smaller than of the disc. We consider that the heat quantity produced will be completely absorbed by the brake disc. The heat flux evacuated of this surface is equal to the power friction. The initial heat flux q_0 [w/mm^2] entering the disc and stopping time t is calculated by the following formula.

$$q_0 = \frac{KE/4}{A \times t} \qquad (1)$$

$$t = \frac{V}{\mu \times g} \qquad (2)$$

Where KE - is kinetic energy developed at front axle in [J]. So KE/4 is energy developed at front brake disc on its one surface. A- Disc surface swept by a brake pad [mm^2], t - is a stopping time of vehicle in [s], V – Initial velocity of vehicle [m/s], μ - coefficient of friction between tire and road, and g - Acceleration of gravity (9.81) [m/s^2].

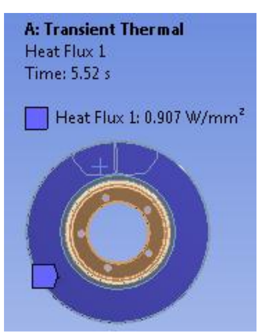


Fig.1. Application of Heat flux

The loading corresponds to the heat flux on the disc surface. The dimensions and the parameters used in the thermal calculation are recapitulated in Table No 1.

Table 1. Geometrical Dimensions and application parameters of automotive braking

Inner disc diameter, mm	152
Outer disc diameter, mm	271
Disc thickness (TH) ,mm	20
Vehicle mass m, kg	2200
Initial speed V, m/s	32.5
Deceleration *a* , m/s^2	5.88
One side Surface of disc swept by the pad A, mm^2	33596

The disc material is cast iron (FG260), with good thermo-physical characteristics and the brake pad has an isotropic elastic behaviour whose thermo-mechanical characteristics adopted in this simulation in the of the two parts are recapitulated in Table 2.

Table 2.Thermoelastic properties used in simulation

Material Properties	**Pad**	**Disc**
Thermal conductivity, *k* (w/m.°C)	5	57
Density, ρ (kg/m^3)	1400	7200
Specific heat, *c* (J/Kg. °C)	1000	460
Poisson.s ratio	0.25	0.28
Thermal expansion, (10^{-6}/ °C)	10	10.85
Elastic modulus,, *E* (GPa)	1	51
Coefficient of friction, μ	0.4	0.4
Operation Conditions		
Heat Flux, q (w/mm^2)	0.9070	
Braking Force (N)	25510	

3. Models prepared of ventilation disc

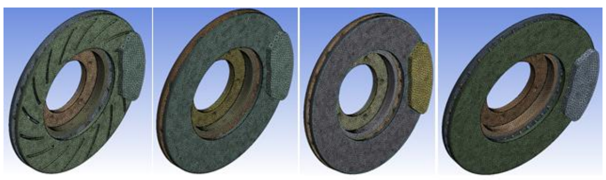
(a) (b) (c) (d)

Fig.2. Cut- sections of various models of ventilation disc

In this study, FEA was used to investigate the thermal-mechanical behaviour of four different ventilated brake designs: (a) 16 curved fins with 16 curved-slotted discs [Mass = 4.97 kg& Surface area = 240461.4248 mm^2] (b) 10 curved fin discs. [Mass = 5.23 kg & Surface area = 210962.092 mm^2] (c) 20 curved fin discs [Mass = 5.33 kg & Surface area = 229105.01 mm^2] (d) 36 redial fins discs [Mass = 5.41 kg & Surface area = 232259.446 mm^2]. Ventilated brake discs were designed according to the propeller-shaped methodology for the slot locations. For models of brake disc the plate thickness is kept constant i.e 7 and 6 mm and thickness of fin is 5 mm. And only the fin profile and length have changed. For model (a) 16 curved fin with 69.8 mm length and curve radius 120 mm and for cross slots 59.67 length, 3 mm width and 120 mm curve radius. (b) For 10 curved fins length is 105.72 mm and curve radius 120 mm. (c) for 20curved fins 69.8 mm length and curve radius 120 mm. (d) for 36 radial fins length is 53.50 mm.

4. Meshing of the disc

For FEA, three-dimensional (3D) constructions of brake discs, brake pads, and their assembly designs were modelled with 1/1 scale in a Pro-e software program, and then imported into Ansys V 13.0 workbench software program for the interactive thermo-mechanical analyses. Brake discs and brake pads were modelled by tetrahedron mesh types.



(a) (b) (c) (d)

Fig.3. Meshing of the disc

Models	**Node**	**Elements**
(a) 16 curved fins with 16 curved-slotted discs	80551	261036
(b) 10 curved fin discs.	78756	265351
(c) 20 curved fin discs	80235	271847
(d) 36 redial fins discs	80163	274158

5. Initials and boundary conditions

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

a) Total time of simulation = 5.52 [s]
b) Increment of initial time = 0.368 [s]
c) Initial Temperature of the disc = 28 [°C]
d) Materials = Cast iron FG 260
e) Convection = One introduces the values of coefficient of transfer of heat (h) obtained for each surface in the shape of a curve [1]

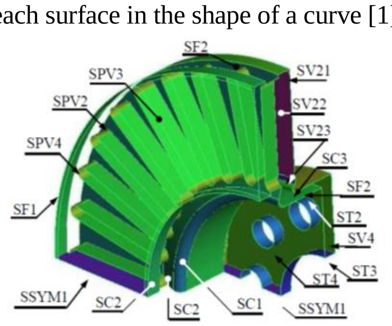


Fig.4. Various surfaces for a ventilated disc

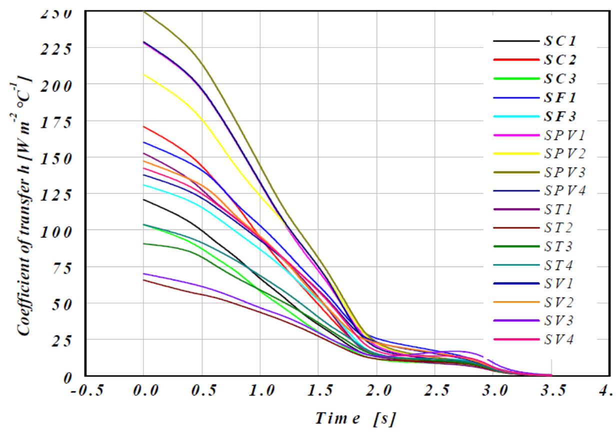
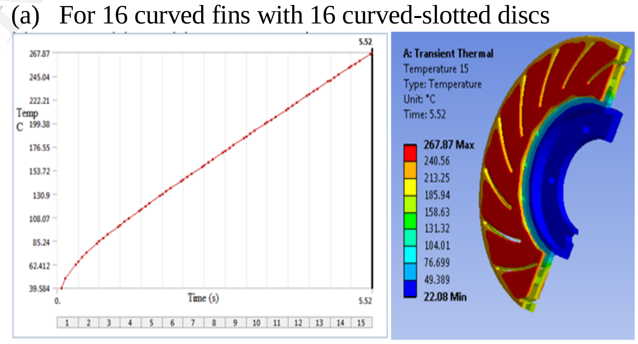


Fig.5. Variation of heat transfer coefficient (h) of various surfaces for a ventilated disc in transient case

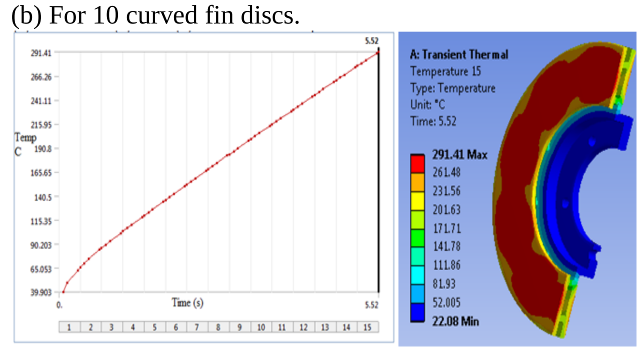
6. Results and discussions

Fig.6 shows the variation in the temperature according to time during simulation. From the first step, the variation in the temperature shows a great growth which is due to the speed of the physical course of phenomenon during braking, namely friction, plastic micro distortion of contact surfaces

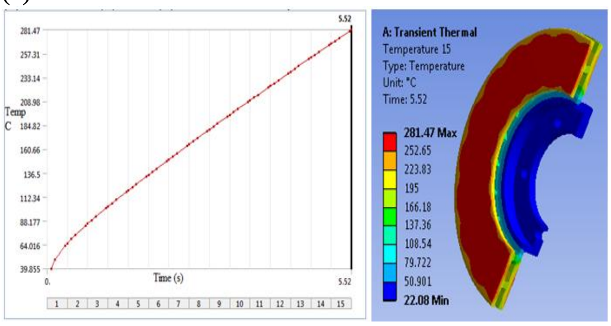
(a) For 16 curved fins with 16 curved-slotted discs



(b) For 10 curved fin discs.



(c) For 20 curved fin discs



(d) For 36 redial fins discs

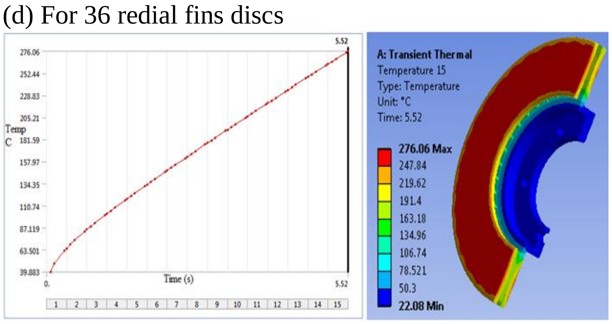


Fig.6. Temperature distribution of a various ventilation disc

From the above result we can see that there is minimum temperature developed in the 16 curved fin with 16 slots on its surface so as the surface area of the disc get increased there is reduction in maximum temperature developed. Also in case of curved fins as compared to the redial fins resistance to rotation get reduced and cooling get increased as air velocity get increased.

7. Coupled Thermo-Mechanical Analysis

7.1. FE model and boundary conditions

A commercial front disc brake system consists of a rotor that rotates about the axis of a wheel, a calliper–piston assembly where the piston slides inside the calliper, which is mounted to the vehicle suspension system, and a pair of brake pads. When hydraulic pressure is applied, the piston is pushed forward to press the inner pad against the disc and simultaneously the outer pad is pressed by the calliper against the disc. Numerical simulations using the ANSYS finite element software package were performed in this study. Various boundary conditions in embedded configurations imposed on the model (disc-pad), taking into account its environment direct, are respectively the simple case as shown in fig.8.The initial temperature of the disc and the pads is 28 °C and the surface convection condition is applied at all surfaces of the disc with the values of the coefficient of exchange from graph The heat flux into the brake disc during braking can be calculated by the formula described in the first part. The FE mesh is generated using tetrahedral element for the disc and pads.

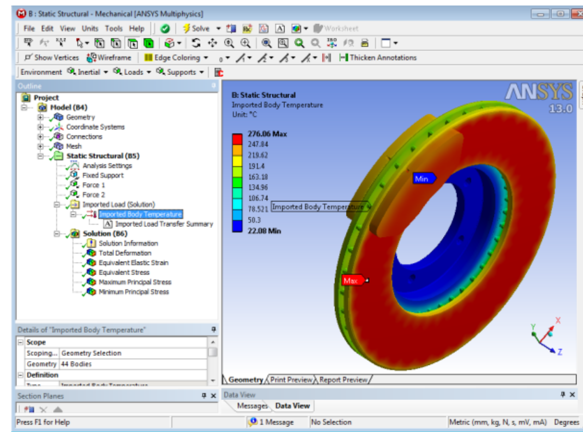


Fig.7. Thermal load imported in structural analysis.

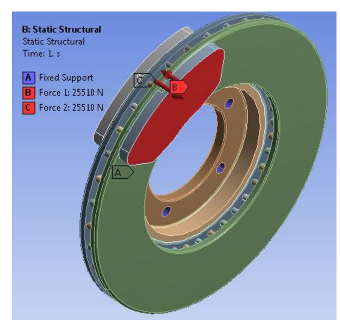
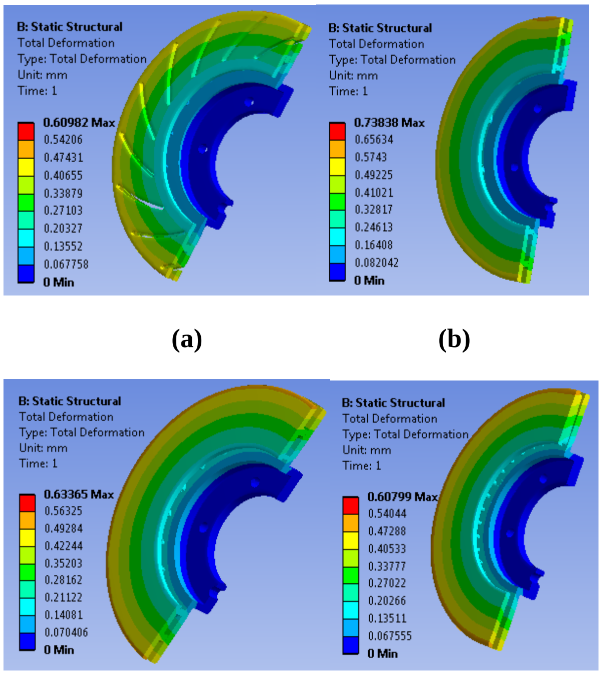


Fig.8. Boundary conditions and loading imposed on the disc-pads

After completion of transient thermal analysis the thermal load is imported in the static structural analysis as shown in above fig 7. Then the mechanical boundary conditions and loading are imported on the disc and pads as shown in fig 8.

7.2. Thermal deformation

Fig.9. gives the distribution of the total deformation in the whole disc for simulation. For this figure, the scale of values of the deformation varies from 0 mm to maximum value. One observes a strong distribution which increases with time on the friction tracks and the crown external and the cooling fin of the disc. Indeed, during a braking, the maximum temperature depends almost entirely on the storage capacity of heat of disc (on particular tracks of friction) this deformation will generate a dissymmetry of the disc following the rise of temperature.



(a) (b)

(c) (d)

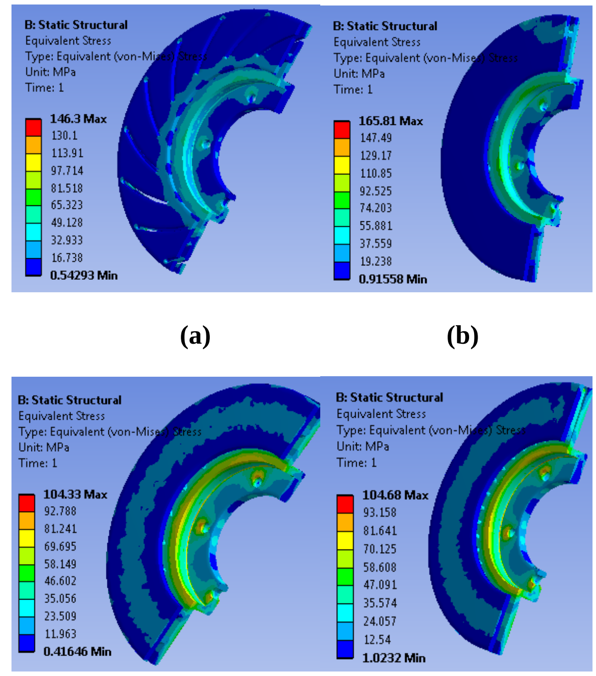
Fig.9. Total deformation distribution for various models of ventilation disc

The minimum deformation is in the model (a) and (d) i.e for disc with 16 curved fins with 16 curved-slots and 36 redial fins discs. The values are 0.60982 mm and 0.60799 mm respectively.

7.3. Von Mises stress distribution

Fig.10. presents the distribution of equivalent Von Mises stress, the scale of values varies from minimum to maximum MPa. The maximum value recorded during this simulation of the thermo-mechanical coupling is very significant that obtained with the assistance in the mechanical analysis dryness under the same conditions. One observes a strong constraint on the level of the bowl of the disc. Indeed, the disc is fixed to the hub of the wheel by screws preventing its movement. In the present of the rotation of the disc and the requests of torsion stress and sheers generated at the level of the bowl which being able to create the stress concentrations. The repetition of these requests will involve risks of rupture on the level of the bowl of the disc.

The minimum Von Mises stress distribution is in the model (c) and (d) i.e for disc with 20 curved fin discs and 36 redial fins discs. The values are 104.33 Mpa and 104.68 Mpa respectively.



(a) (b)

(c) (d)

Fig. 10. Von Mises stress distribution for various models of ventilation disc

8. Conclusion

In this publication, we presented the analysis of thermo-mechanical behaviour of the dry contact between the brake disc and pads during the braking process; the modelling is based on the ANSYS 13.0. We have shown that the ventilation system plays an important role in cooling discs and provides a good high temperature resistance.

The analysis results showed that, temperature field and stress field in the process of braking phase were fully coupled. The temperature, Von Mises stress and the total deformations of the disc increases as the thermal stresses are additional to mechanical stress which causes the crack propagation and fracture of the bowl and wear of the disc and pads. Regarding the calculation results, we can say that they are satisfactory commonly found in the literature investigations. It would be interesting to solve the problem in thermo-mechanical disc brakes with an experimental study to validate the numerical results.

So from above results it is seen that the profile of fins in ventilation disc plays important role for cooling of the disc. There are advantages of properly designed ventilation disc as it reduce the unsprung mass of the vehicle, reduction in the brake fade, period between two successive braking is get reduced and more important is if instance of radial fins we use the curved fins resistance to rotation get reduced and cooling of disc get increased as air velocity get increased.

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