Theoretical and Experimental Researches of Brake Discs' Thermal Stress

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Abstract: - The paper presents the results of the thermal stress researches made for two types of motor vehicles brake discs. The researches were made by processing the results obtained by a finite element analyse (FEA) and by the experimental tests of the thermal stress. The study was made for an intensive braking from 100 to 0km/h, a frequently met regime in nowadays traffic, making possible the comparison of two types of brake discs. Thus, it was possible to highlight the influence of the construction characteristics and material properties, of the studied brake discs, over the thermal stress.

Key-Words: brake disc, thermal stress, finite element analyse, FEA, thermography, braking.

1 Introduction

The continuous development of motor vehicles, meaning speed and acceleration performances, involves a rise of the average speed. This thing implicates a raised attention on traffic safety so the braking system must be further optimised. By doing this it will be easy to avoid some accidents even at low speeds, induced by the unpredictable appearance of some obstacles.

At the entrance and inside the cities the brakes are used more frequently. Thus, for a medium size city, the motor vehicle uses brakes about 30-40% from the total time spent in traffic.

For a braking, usually it is considered that, all the kinetic energy is transformed into thermal energy [1]. At an intensive braking the absorbed power by the disc brakes is 4-5 times higher than the engine's power. The high quantity of the heat resulted during braking, contributes to the aggravation of the brake's material qualities and speeds up the wear of brake discs and pads.

At a high temperature value, their efficacy is reduced due to fading phenomenon. Other effects are excessive wear, heat crazing, cracking, and grooved discs and changes in the material structure [2], [3] (hard spots, blue colouring).

All those effects will cause noise and vibration during braking, lowering value of friction coefficient and forward reducing brakes' efficacy, and further on, traffic safety. Nowadays the disc brake is in continuous optimisation [4] to erase comfort and raise traffic safety.

Thermal stress is a result of exterior constrains during temperature variation ΔT inside the brake disc, having no possibility to expand or constrict. It can be expressed as [5]

$$\sigma = -\frac{E}{1-\nu} \cdot \alpha \cdot \Delta T \,, \qquad (1)$$

where α – the linear thermal expansion coefficient of the brake disc material;

$\Delta T = T - T_a$, T – local disc temperature, T_a – ambient temperature, E – Young modulus, ν – Poisson coefficient.

During braking the heat is generated by the friction between disc and pads and is dissipated by convection and radiation (fig.1).

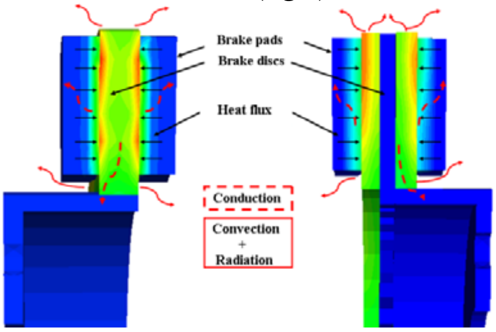

Fig.1 The phenomenological principle of thermal energy dissipation – radiation, convection and conduction

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The surfaces of disc which remains in contact with pads will have higher values of temperature, which can cause, in time, thermal stress inside the brake disc. The stresses are amplified in some cases by the pressure between pads and disc. While pads press the disc during braking, the heat generated will make both of them to expand.

If Poisson coefficient ν is considered negligible result

$$\sigma_1 = E_1(\alpha_2 - \alpha_1)\Delta T,$$
$$\sigma_2 = E_2(\alpha_1 - \alpha_2)\Delta T. \quad (2)$$

Total accumulated heat flux Q is obtained by dividing the brake power by swept area of the disc brake $Q(t)=kma(V_1-at)/S_1$; S_1 – swept area of the disc brake by brake pads; Q(t) – time varying heat flux [6].

The stress σ from the S_1 area can be written as

$$\sigma = \frac{2\pi \cdot Q}{\phi_0 \cdot \mu \cdot p \cdot r \cdot \omega}, \quad (3)$$

where $p = p_{\max} \cdot \frac{r_2}{r}$; p_{max} – maximum pressure distributed on pad; p – radial pressure on the disc brake; μ – friction coefficient, which varies with temperature for different materials [7].

A part of the heat flux Q is dissipated by radiation [7]

$$Q_{rad} = \sigma_0 \cdot \varepsilon \cdot A(T^4 - T_a^4), \quad (4)$$

where A =area of the disc radiation heat, and σ_0 = Boltzamn constant;

The time variation of stress from equation (3) can be written in the form

$$\sigma(t) = \frac{2\pi \cdot Q(t)}{\phi_0 \cdot \mu \cdot p \cdot r \cdot \omega}. \quad (5)$$

From two values of temperature T_1 and T_2 of brake disc, obtained for two values of radial speed ω_1 and ω_2, the thermal stress ratio can be written from equation (5) as follows:

$$\frac{\sigma_1(T_1)}{\sigma_2(T_2)} = \frac{(T_1^4 - T_a^4) \cdot \varepsilon(T_1) \cdot \omega_2}{(T_2^4 - T_a^4) \cdot \varepsilon(T_2) \cdot \omega_1}. \quad (6)$$

2 The Work Methodology

For temperature distribution, which influences thermal stress of brake disc, was made a case study for an intensive braking from 100 km/h to 0 km/h. It was made a finite element analyses and some experiments outside the laboratory, for two types of brake discs.

2.1 Finite Element Analyses

The finite element analyses were made for a complex geometry of discs and pads, which were the same as the real one from Dacia Logan. The aim of those analyses was to reproduce the exact complex shapes and sizes of discs and pads, thing rarely met in specialty literature [8].

Brake pads have a friction material thickness of 12 mm and a contact width, with the brake disc friction surface, of 41 mm.

Friction surfaces of disc have an outer radius of 234 mm and inner radius of 152 mm. For the simple disc the thickness is 11 mm. For the vented one the thickness of inner wall differs to the outer one, first is 7 mm and the second 8 mm. The vanes between the walls have a thickness of 6 mm, and are radial positioned. Based on those dimensions, the geometrical elements (discs and pads) were meshed in elements and nodes (fig.2). After meshing the assembly of disc and pads was made by specifying the position constrains. The whole assembly was finally transformed to an oprah-mesh element.

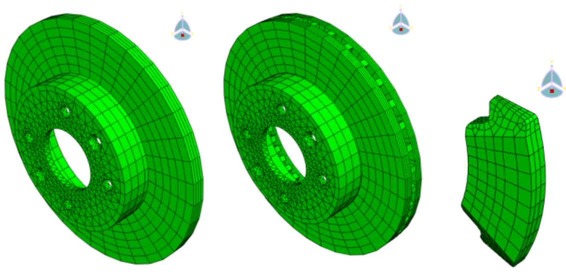

Fig.2 Oprah mesh element of disc-pads modelled in ABAQUS

Table 1 presents the number and type of elements and nodes from oprah-mesh elements.

Table 1 Meshing characteristics of oprah-mesh elements

	Simple disc	Vented disc
Number of nodes	3502	5179
Number of elements	3929	4468
Number of hexahedral elements C3D8RT	1680	2513
Number of tetrahedral elements C3D4	2249	1955

Brake pads are made of a mixture of elements glued by a binding component and the discs were made of cast iron. Thermal and mechanical properties are presented in Table 2, fig.3 and fig.4. Some parameters vary with temperature (Young's modulus, friction coefficient, expansion coefficient).

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Table 2 Material characteristic of the disc and the pads [10]

	Disc brake	Brake pads
Property	**Value**	**Value**
Density [kg/m^3]	7800	1150
Young Modulus [MPa]	209E3	Fig.3
Poisson Coefficient [-]	0.3	0.25
Expand Coefficient [K^{-1}]	1.1E-5	10^{-5}@20^{o}C; 30^{-5}@200^{o}C
Conductivity [w/mK]	48	0,9
Specific heat [J/kgK]	452	1200

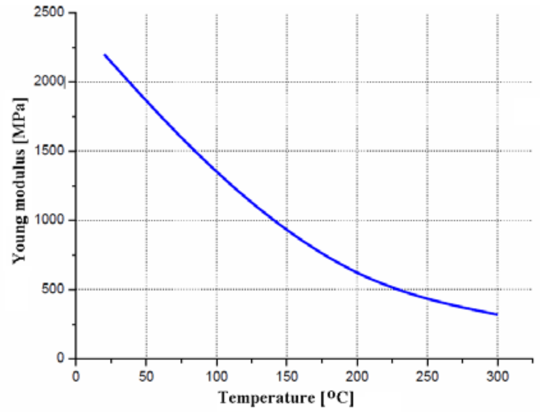


Fig.3 Variation of the Young modulus with temperature for brake pads [10]

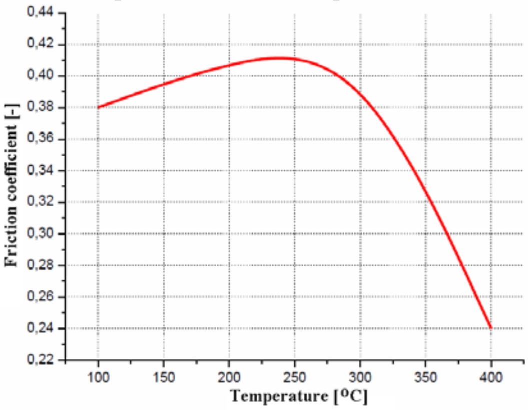


Fig.4 Variation of the friction coefficient with temperature for brake pads [10]

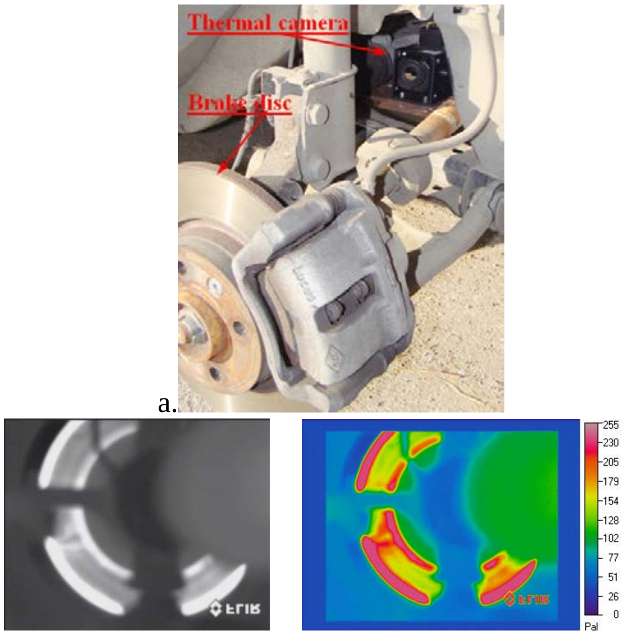
After declaring those properties the friction surfaces were declared. The simulation was divided in two steps. First was for the pressure of pads over the disc and thus the contact between them. This step propagates to the second one in which the angular speed of brake disc starts to decrease to zero. The angular speed of 89 rad/s is equivalent to 100 km/h of motor vehicle speed. Total time of this step was 4 seconds as the time of experimental braking tests.

Initially the pad has no degree of freedom. Over the exterior sides of the pad, a pressure of 3.6 MPa was applied. The pressure was computed for the total time of braking, and the dimensions of brake components [9]. In the centre of the disc was declared a reference point to the disc, for which was declared the angular speed of brake disc. The second step is declared as a “smooth step”, as to reduce the centrifugal forces effect over the model, at the beginning and the end of this step.

With this input data the analyses started and were run in ABAQUS/Explicit.

2.2 Experimental Tests Outside the Laboratory

Experimental tests were carried out, outside the laboratory, on two Dacia Logan motor vehicles to validate the finite element model. They were equipped with thermal cameras to record temperature variation and distribution over brake disc – simple and vented brake disc (fig.5).



b. c.

Fig.5 Position of thermal camera and the images taken with it
(a) thermal camera, (b) image taken in gray scale and (c) image transformed from gray scale in pseudocolor (c)

The used thermal camera took gray scale images (fig.5) of heat radiated from discs. These images were processed with specialized software to make possible the estimation of temperature repartition and forward to analyse thermal stress of brake disc.

To process the data, it was necessary to know the total time of braking, initial temperatures of discs

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and environment and the weather conditions (humidity, wind speed).

The tests were made at short time period between them on the same testing field making the state of the road having no influence over the results.

3 Results and Interpretation

Comparing finite element and experimental results (fig.6 and fig.7) it was possible to observe the non-uniform repartition of temperature caused by thermo-elastic behaviour of disc and the pads. The temperature rises at the upper part of the pads due to high slip speed between disc and the pads. This fact makes the temperature to rise in that area. This way the material will distend, while the pressure remains constant over the pads back, the contact will rise at the inner radius part so the temperature will rise too. For long braking the temperature shape will reverse [11]. This process is the effect of thermo-elastic instability (TEI) phenomenon which causes the pressure and temperature variation [12].

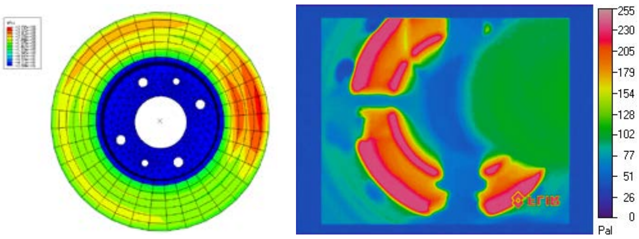

Fig.6 Comparison between theoretical images (obtained by FEA) and thermal images (results of tests) at the end of intensive braking for a simple brake disc

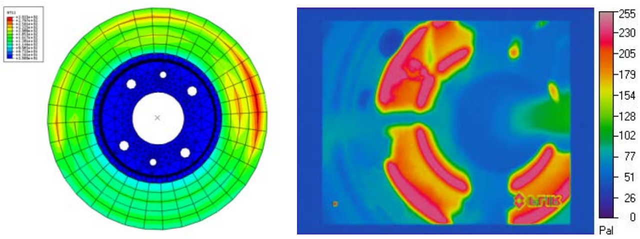

Fig.7 Comparison between theoretical images (obtained by FEA) and thermal images (results of tests) at the end of intensive braking for a vented brake disc

This temperature variation causes material structure changes, generating heat spots that can be amplified and thus result noises and vibrations [12].

Considering input conditions and the temperature distribution over the brake discs, which influence the thermal stress, the results of the finite element analyse and the tests are in good agreement [13], and make possible to observe the heat transfer inside the brake disc.

At the vented disc the heat will be released in the first phase from the friction surface. Due to conduction in the second phase the heat will be released inside the disc between vanes. The total surface of heat release is more than 60% bigger than the simple disc. For a high value of conductivity the heat will be released in a short period of time and in a high quantity.

Due to the difference between the thickness of the left and the right part of the vented brake disc (fig.8 and fig.9), the heat flux will reach faster, through the left part (which has a smaller thickness), at the interior of the brake disc. For the simple brake disc the heat will be dissipated only at the outside part, and the heat will be storage inside of it causing a higher thermal stress.

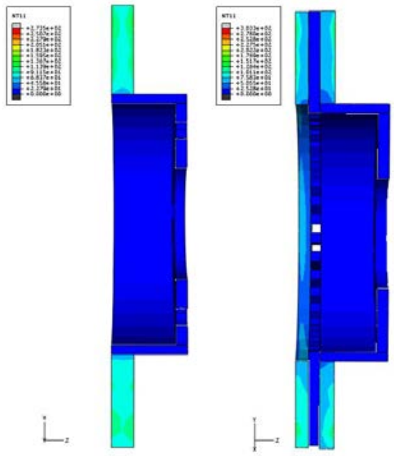

Fig. 8 Temperature distribution inside the brake discs during braking (at 50km/k)

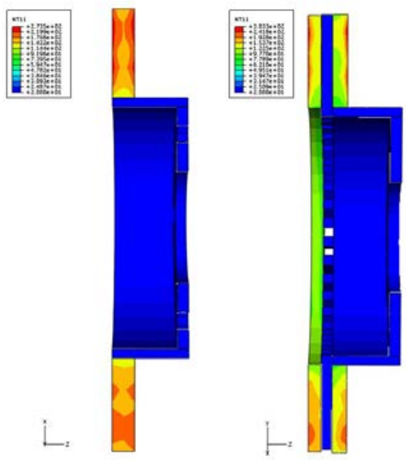

Fig.9 Temperature distribution inside the brake discs at the end of the intensive braking

At the end of the thermal stress simulation it was possible to analyse the heat flux transfer (conduction and radiation) as a function of the material thickness (fig.10). Thus during the life cycle of a vented brake disc, taking account the wear of the disc, the conduction of its material has a high influence over the thermal stress.

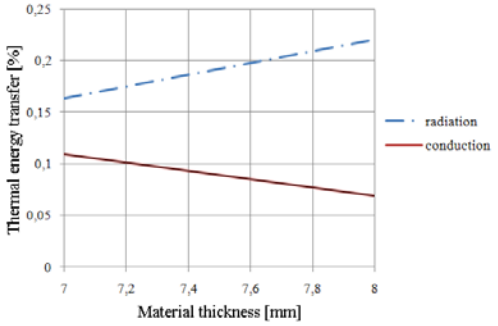

Fig.10 The influence of brake disc material thickness over the thermal energy transfer

Processing the thermal images obtained during experiments it was possible to draw the temperature variation of the two types brake discs (fig.11, fig.12).

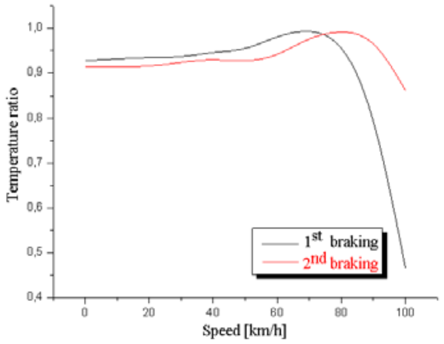

Fig.11 The temperature variation during intensive braking for a simple brake disc

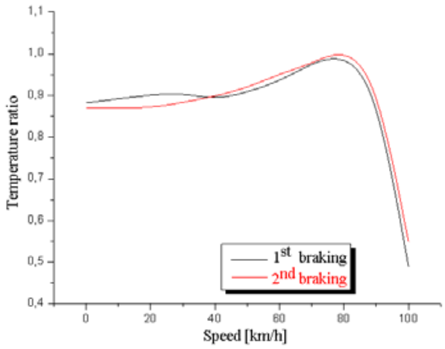

Fig.12 The temperature variation during intensive braking for a vented brake disc

There were made two sets of two intensive braking from 100 km/h, one set for one type of brake disc, to evaluate temperature variation during tests.

For the vented brake disc, a big quantity of the heat was dissipated before the second braking. Thus the temperature at the beginning of the second braking has almost the same value as the one at the beginning of the first braking.

Comparing temperature variation for both discs at the end of the tests, 33% of the heat, of the vented brake disc, was dissipated. The heat of the simple brake disc was dissipated in a proportion of 6%. Comparing the final temperatures of both discs, they differ in proportion of 11%, the simple disc temperature being the higher one. This explains the fact that the simple brake disc is exposed to high thermal stress during its life time.

4 Conclusions

For a half rotation of the simple brake disc (at the opposite part of the pads), the heat is dissipated, by radiation, about 26% over the friction surfaces. In case of vented disc, which has a dissipating area about 60% bigger than the simple one the heat will also be dissipated at the interior of the disc. Thus the disc is less thermally stressed.

Analysing the vented model, thickness plays a good part in heat transfer. Combined with interior surface state, the heat will be dissipated faster at a smaller thickness of side walls.

Optimising actual finite element analyse it's possible to make further studies to reduce thermal stresses of the disc brakes and further for the entire system.

References

[1] P.D., Milenković, J., Jovanović, A.S.,Janković, M.D., Milovanović, N.D., Vitošević, M.V., Dordević, M.M., Raičević, The Influence Of Brake Pads Thermal Conductivity On Passanger Car Brake System Efficiency, THERMAL SCIENCE: Vol. 14 (2010), [web site: http://www.doiserbia.nb.rs/img/doi/0354-9836/2010%20OnLine-First/0354-98361000 016M.pdf].

[2] Ali Belhocine, Mostefa Bouchetara, Simulation of fully coupled thermomechanical analysis of disc brake rotor, *WSEAS TRANSACTIONS on APPLIED and THEORETICAL MECHANICS*, Issue 3, Volume 7, July 2012, E-ISSN: 2224-3429.

[3] *** Brake Noise, Vibration and Harshness: Technology Driving Customer Satisfaction, AKEBONO, [web site: http://www.akebono

brakes.com/company/media_center/white_papers/akebono_NVH.pdf].

[4] Y.Kubota, K.Okubo, T.Fujii, Reduction of squeal on laminated brake disc fastened with distributed contact pressure, *Recent Researches in Engineering Mechanics, Urban & Naval Transportation and Tourism*, pp. 199-204, Cambridge, UK February 25-27, 2012, ISBN: 978-1-61804-071-8, Published by WSEAS Press.

[5] R. Limpert, *Brake Design and Safety*, SAE International,1999, ISBN - 1560919159, 978156091915.

[6] F.Talanti,F., and S. Jalalifar, Investigation of Heat Transfer Phenomena in a Ventilated Disk Brake Rotor with Straight Radial Rounded Vanes. *Journal of Applied Sciences* 2008, 8 (20): 3583-3592.

[7] Zhongzhe Chi, Thermal Performance Analysis and Geometrical Optimization of Automotive Brake Rotors, *A Thesis Submitted in Partial Fulfillment of the Requirements for the Degree of Master of Applied Science In Mechanical Engineering*, University of Ontario Institute of Technology, 2008.

[8] L.,Zhang, D., Meng, Zhuoping Yu, Theoretical Modeling and FEM Analysis of the Thermo-mechanical Dynamics of Ventilated Disc Brakes, *SAE 2010 World Congress & Exhibition*, April 2010, Detroit, MI, USA, Session: Brake Technology (Part 2 of 2), 2010-01-0075;

[9] *P., F., Gotowicki, V., Nigrelli, G., V., Mariotti*, Numerical And Experimental Analysis Of A Pegs-Wing Ventilated Disk Brake Rotor, With Pads And Cylinders, *10th EAEC European Automotive Congress* – Paper EAEC05YU-AS04, 2005 Boegrad.

[10] *** Abaqus/CAE © Dassault Systemes – User Guide, 2007;

[11] Kwangjin Lee, Barber, J., R., An Experimental Investigation of Frictionally - Excited Thermo-elastic Instability in Automotive Disk Brakes Under a Drag Brake Application, 1994 [web site: http://www-personal.umich.edu/~jbarber/JT1994.pdf].

[12] J.,-H., Choi, I., Lee, Transient Thermoelastic Analysis Of Disk Brakes In Frictional Contact, *Journal of Thermal Stresses*, pp.223–244, 2003, DOI:10.1080/01495730390179002.

[13] Ştefan Voloacă, Gheorghe Frăţilă, Concerns Regarding Temperature Distribution Obtained By Experiments And Finite Element Analyses For Two Types Of Brake Discs, U.P.B. Scientific Bulletin, Series D, Vol. 74, pp.33-42, Iss. 3, 2012, ISSN 1454-2358.