Proceedings of the 17th International Conference on Manufacturing Systems – ICMaS
ISSN 1842-3183
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CONSIDERATIONS ON THE MODELLING AND ANALYSIS OF BRAKE PAD

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Abstract: *The main part of a brake system, pads and disc, are generally considered as a source of the brake noise. Manufacturing parameters and composition of component materials can affect the final brake lining produced. In this paper the results of modelling and analysis of brake pad is discussed. A correlation between noise index and compressibility of brake pad is established. A static analysis is performed. Some results are presented that relate the displacement of the top and bottom parts of the brake pad and vibration modes of the lining assembly.*

Key-Words**:** *brake system, pad displacement, brake squeal.*

1. INTRODUCTION

In order to decrease noise and vibration of all vehicle systems and to eliminate this source of discomfort many experiments have been performed. Noise generated by the brake system is a common fact for all vehicle types. A classification of these noise levels is based on the frequency range of each noise.

Break squeal is generally defined as an unpleasant, self-induced, high-frequency (2000 – 10000) Hz, Chargin s.a [1]. Brake groan typically refers to noise levels below 500 Hz. The intermediate 500 Hz – 1000 Hz range can fall into either category depending on the opinion of the listener.

Much of the published work has included definition of brake squeal mechanism, definition of models to describe the dynamics of the brake system and the development of general methods to reduce brake noise.

It is agreed that brake noise is the result of vibration generated at the interface of the rotor and friction material. It has been suggested that the reason for brake noise is either due to vibration excitation of an unstable mode, or due to vibrations caused by large magnitude driving forces.

Liles [2] shows that the friction force instabilities generate the brake squeal, either from normal force vibration or from the negative slope of the speed and coefficient of friction plot.

Bergman, s.a [3] considers that the brake squeal not be generated if the coefficient of friction is kept below some critical level. For the tested pad/disc combination the level find by the authors was found to be 0.4. A small increase in the coefficient of friction in the close vicinity of the critical level causes a dramatic increase in squeal generation.

Nishiwaki [4] derived the equations of motion for a disc brake system using Lagrangian dynamics. Nishiwaki expanded the theory to suggest that all brake vibrations are generated by the same type of modal instability but at different modes depending on frictional input forces and modal displacements. The instability is generated due to a frictional force fluctuation as a result of normal force variations. The normal forces vary as a result of the interactions with the vibrating rotor and brake pads. The friction forces are used along with kinetic and potential energy expressions to develop the equations of motion using Lagrange's technique.

Yang and Gibson [5] present a comprehensive review of the research on brake noise and vibration. Are presented the researches, opinions and suggestions of the many authors on: noise and brake classification, research on brake squeal, methods of research, reduction of brake noise and vibration, brake materials and the future considerations

Hulten, s.a [6] presents the measured mode shape of a squealing drum brake. Also is presented the operational deflection shape while squealing under an operating condition.

2. FRICTION MATERIALS AND ELASTIC CONSTANT

A friction material composition is made up of as many as 10 – 20 raw materials, each with a unique volume percentage and function as strength and friction (steel, copper), friction (iron, copper, brass), strength and processing (mineral, cellulose), binder (phenol/formaldehyde), friction level (carbides), friction stability (graphite, rubber)).

The physical properties of this material influence the overall properties of the brake lining. Processing parameters and composition of friction materials affect brake noise.

In order to characterize the friction material, the next parameters can be measured: hardness, porosity, frequency and damping, and elastic constant that can be measured with ultrasonic waves.

In order to measure the first natural frequency and the rate of vibration amplitude decay that is proportional to damping for a given structure, an adequate test was used.

Young's module, shear module and Poisson's ratio are determined by measuring the transit time of shear and longitudinal modes using ultrasonic technique.

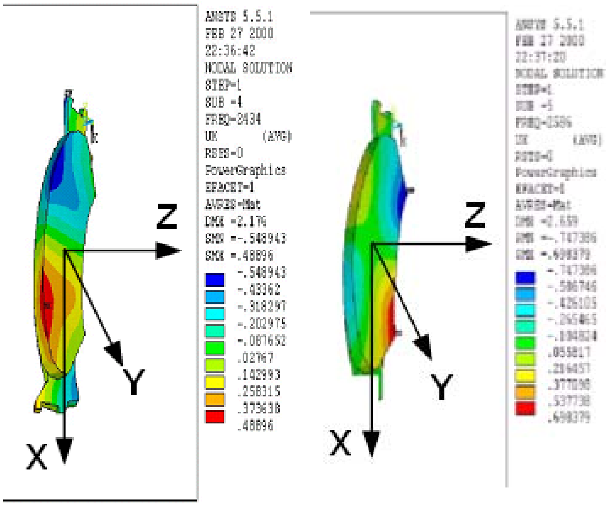

Fig. 1. Coordinates definition for sample lining.

The relation between the ultrasonic velocity and the engineering constants for anisotropic materials is somewhat more complex. In principle, it is possible to use ultrasonic to measure all nine independent elastic constant in orthotropic materials. Experience with friction materials suggests that they can be treated as transversely isotropic materials. As such, there are five independent elastic constants. In order to determine these constants, minimums of five independent measurements are needed. The typical composition of brake pads is referred at [7]:

a. structural materials: metals, carbons, glass and/or Kevlar fibres, added to obtain the mechanical stability;

b. matrix: in generally, phenolic resins are used, and sometime, different rubber type are used;

c. filler: added to decrease the cost and/or to improve the manufacturability; some fillers may affect the friction characteristics of the material;

d. frictional additives are added to control the coefficient of friction.

3. THEORETICAL MODEL

In order to find the equations of motion, Nishiwaki [4] develops a theoretical model of disc brake squeal, where the lining material it is assumed to be a spring without any mass installed in the direction of surface pressure.

Starting with relations for surface pressure variations that occur by relative displacements between the brake pads and the disc, kinetic energies, the following equation can be obtained:

$$M\ddot{X} + KX = \left(\frac{E_1}{h}\right)AX + \mu\left(\frac{E_1}{h}\right)BX\,, \qquad (1)$$

where:
E_1 represents Young modulus for lining
h represents lining thickness
Surface pressure variations, P_{op} and P_{ip} are:

$$R_{op} = \left(\frac{E_1}{h}\right)\left(w_{op} - w_d\right)\cdot r\cdot dr\cdot d\Phi\,,$$

$$R_{ip} = \left(\frac{E_1}{h}\right)\left(w_d - w_{ip}\right)\cdot r\cdot dr\cdot d\Phi\,, \qquad (2)$$

where:

R_{op} and R_{ip} surface pressure variations for outer pad and inner pad;

w_d, w_{op}, w_{ip} disc, outer pad and inner pad width.

Surface pressure variations, produce variations in frictional force, respectively:

$$F_{op} = \mu R_{op}$$
$$F_{ip} = \mu R_{ip} \qquad (3)$$

4. STATIC ANALYSIS OF THE BRAKE PAD

In order to perform the static analysis a brake system assembly is considered (Fig. 2).

Lining assembly, schematically presented, is considered as shown in Fig. 2. The pressure in the brake circuit was assumed to be in the range (1 – 9) N/mm^2. The model used for the analysis is represented in Fig. 3, and the generated mesh is shown in Fig. 4.

In this case the data used for analysis are: pressure in the calliper chamber, respectively the range (1-9) N/mm^2 is equivalent to the range (0.76-6.08) N/mm^2 on the backside of the shoe plate.

Area of active calculated for the calliper piston is 2450 mm^2, and area of the active surface of shoe plate (under piston) is 2210 mm^2. In this analysis 6 N/mm^2 value for the pressure in the calliper chamber is used.

Figure 5 shows the displacement fields in the brake pad and Fig. 6 shows the variation of E_z with displacement.

Figure 7 presents the generated mesh for the pad, and Figs. 8 – 11 the pad displacements for the studied case.

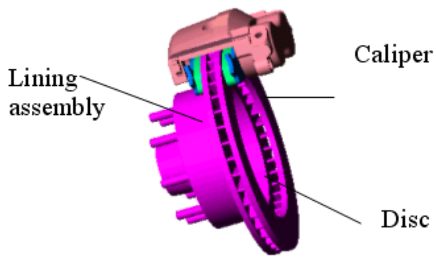

Fig. 2. Brake system assembly.

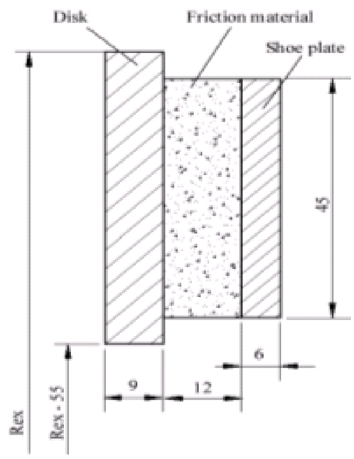

Fig. 3. Schematic representation for brake pad assembly.

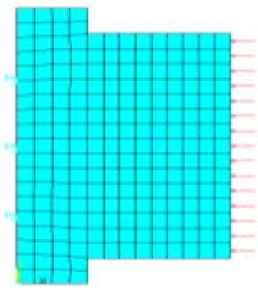

Fig 4. Generated mesh model for the calculation.

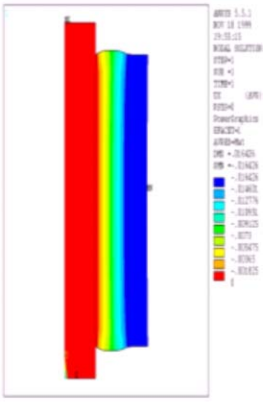

Fig. 5. Displacement fields in the brake pad.

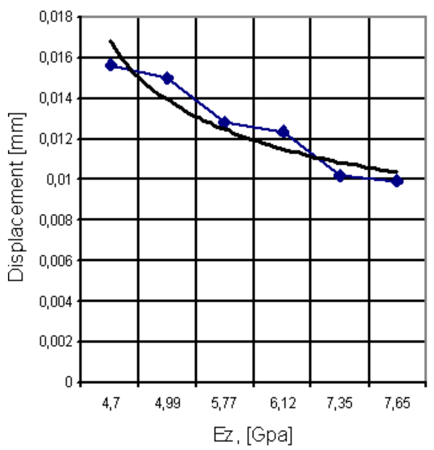

Fig. 6. Variation of E_z with displacement.

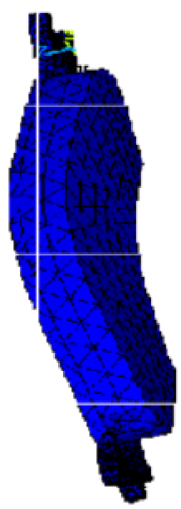

Fig. 7. Generated mesh for pad of this analysis.

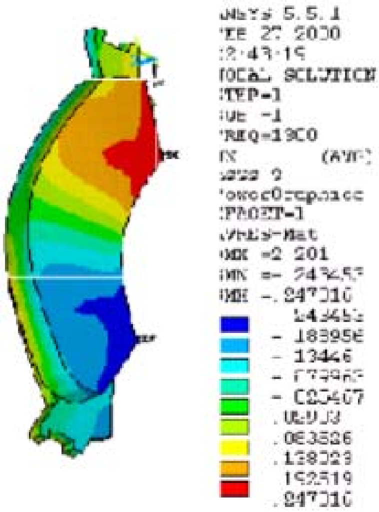

Fig. 8. Pad displacements I.

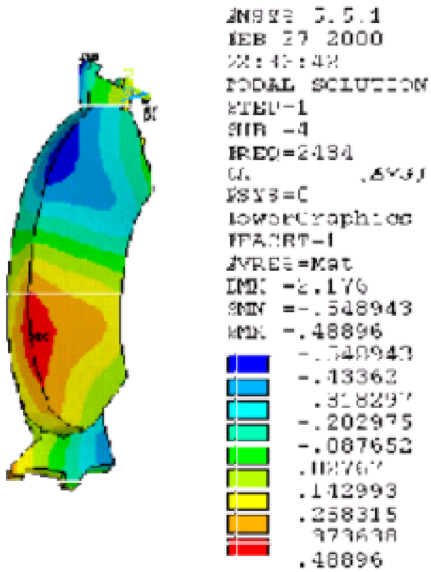

Fig. 9. Pad displacements II.

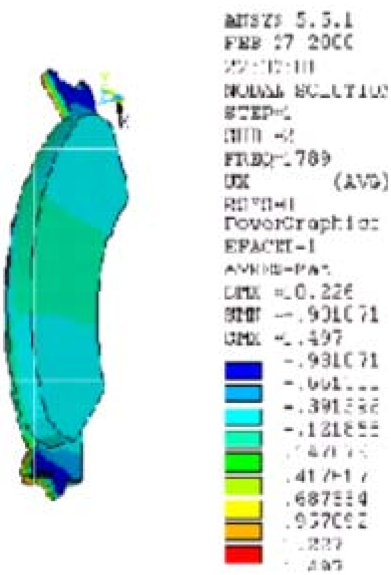

Fig. 10. Pad displacements III.

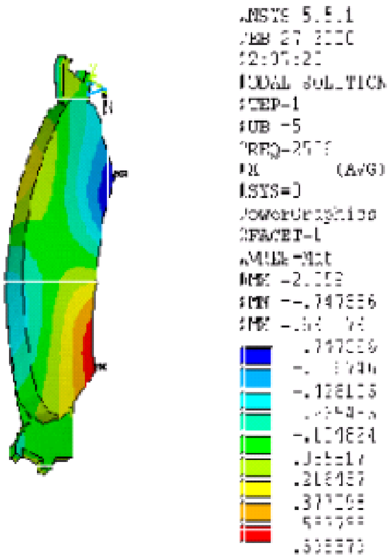

Fig. 11. Pad displacements IV.

Table 1

Mode Shape and Frequencies for the considered pad

***** INDEX OF DATA SETS ON RESULTS FILE

SET	TIME/FREQ	LOAD STEP	SUBSTEP	CUMULATIVE
1	2360.6	1	1	1
2	2410.9	1	2	2
3	3005.3	1	3	3
4	3386.3	1	4	4
5	3768.6	1	5	5
6	4812.7	1	6	6
7	4884.2	1	7	7
8	5095.5	1	8	8
9	5367.2	1	9	9
10	5734.2	1	10	10

The power regression $y = 0.0168x^{-0,274}$ illustrates that ~ 85% of variation in the displacement is accounted for by variability of Young's Modulus in the z direction.

In order to analyse the pad displacements, a common Finite Elements Analysis was developed. Figs. 8 and 9 present three mode shapes of vibration, as result of this analysis.

The frequencies values for ten vibration modes obtained in this analysis are presented in Table 1.

5. CONCLUSIONS

From the results of the analysis reported in this paper the following conclusions are given.

On the brake pad, for a passenger vehicle, under pressure, some displacements occur at the top and bottom part of the pad. The power regression curve shown in Fig. 5 indicates that 87 % of variation in displacement is accounted by the variability in the Young's Modulus of the lining material in the Z direction, E_z.

In order to analyse the pad displacement a static FEA analysis was developed. Mode shape and frequencies for the considered case are presented.

6. REFERENCES

[1] Chargin, M.L., Dunne, L.W., Herting, D.N. (1997). *Nonlinear dynamics of brake squeal*, Finite Elements in Analysis and Design 28, pp. 69-82.

[2] Liles, G. (1993). *Analysis of Disc Brake Squeal using Finite Element Method*, SAE, pp. 891150.

[3] Bergman, F., Eriksson, M., Jacobson, S. (1999). *Influence of disc topography on generation of brake squeal*, Wear 225-229, pp. 621-628.

[4] M. Nishiwaki (1993). *Generalized Theory of Brake Noise*, Proceedings of the Institute of Mechanical Engineers, Vol. 207, pp. 195-202.

[5] Yang, S., Gibson, R.F. *Brake vibration and noise: review, comments, and proposals.* Int. Journal of Materials and Product Technology, Vol. 12, Nos. 4-6, pp. 496-513.

[6] Hulten, J., Flint J., Nellemose, T., *Mode shape of a squealing drum brake*, P. 972028, SAE, pp. 1231-1245.

[7] Eriksson, M., Bergman, F., Jacobson, S. (1999). *Surface characterisation of brake pads after running under silent and squealing conditions*, Wear 232, pp. 163-167.

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