Effect of Disc Brake Squeal with Respect to Thickness Variation: Experimental Modal Analysis

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ABSTRACT

Disc brake or rotor squeal is an ongoing problem that occurs in the automotive industry. An undesirable disc brake noise problem can arise after a period of time of usage. The purpose of this paper is to investigate the structural dynamic behaviour of disc brakes with different wear thickness by using Experimental Modal Analysis. The wear thickness of disc brake rotors are 0.5 mm, 1.0 mm and 1.5 mm from the original thickness of 15.8 mm and 3.2234 kg weight. The modal parameters such as natural frequency, damping ratio, and mode shape are obtained in a free-free condition by using an impact hammer test. For original thickness of disc brake rotor, the first eight natural frequencies are 1256.4 Hz, 2486.9 Hz, 2654.9 Hz, 3092.1 Hz, 3348.7 Hz, 3407.0 Hz, 4130.0 Hz, and 5709.6 Hz. The results show that the natural frequency decreases when the thickness reduction increased at the same mode. It can be concluded that the wear effect of the disc brake rotor is one of the factors which may lead to the brake squeal problem due to the reduction of the natural frequency of the disc brake rotor.

Keywords: *Experimental Modal Analysis, Disc Brake, Disc Thickness Variation*

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Introduction

The brake system is one of the vehicle components that sometimes creates undesired vibration and unpleasant noise . The thickness of the disc brake rotor is reduced when it comes into contact with the brake pad due the effect of wear and tear and this is an important parameter for safety. The thickness of the disc brake rotor should not be lower than the minimum disc thickness predefined by the manufacturer because this can increase the probability of the deformation of the disc brake rotor. Besides that, a worse situation such as disc brake damage can occur at high intensity or during a long duration of braking [1, 2].

The goal of this study was to investigate the effect of wear on the structural dynamics behaviour of the brake rotor. Therefore, experimental modal analysis is considered in this paper and helps to avoid the difficulties that are inherent in FEM [3, 4]. So, it is important to investigate the structural dynamic behaviour of disc brakes caused by disc thickness variation at the beginning of the engineering design. The effect of disc brake squeal is the most sensitive and annoying type of brake noise to the frequency region of 1 kHz to 4 kHz, therefore the low frequency squeal is considered in this paper.

In practice, the accuracy of the disc brake rotor in FEM models is often limited by uncertainties about the actual geometry and material properties. Hence, experimental modal analysis is applied to correlate the measured vibration behaviour of disc brake components with that predicted by FEA from the previous studied by Daud [5]. The thickness variation for the disc brake showed the effect of the coupling mechanism between rotor and pad. This wear has a strong effect on the system instability and noise generation due to the change in coupling mechanisms. The wear effect also can change the geometry and stiffness properties of the brake pads and rotor, leading to brake squeal generation when the system is in the end of its lifetime [6].

In this paper, the problem of disc brake squeal may be addressed by reducing the thickness of the disc brake rotor at the surface which experiences the friction contact with the brake pad. Then, the effect of wear on disc rotor was investigated using experimental modal analysis. Based on the experiment, the natural frequency, damping ratio and mode shapes of the disc rotor which can lead to brake squeal can be determined and is useful for predicting and eliminating brake squeal.

Brief Review of Theory

The second order equation of motion for a brake system with a single degree of freedom system consists of the brake pad is defined using the following variables: m is mass, k is the spring stiffness, c is the damping, F is the friction force and μ is the coefficient of friction, with the equation:

$$m\ddot{x} + c\dot{x} + kx = \mu F \tag{1}$$

The schematic diagram the brake mechanism is shown in Figure 1. Ghazaly [7] used a stick-slip mechanism to represent the schematic diagram for the equation of motion of the brake system, since the disc brake rotates and the brake pad rests on a rotating disc brake.

Using Fourier transform on equation 1, we get:

$$(-\omega^2 m + j\omega c + k)x(\omega) = \mu F(\omega) \tag{2}$$

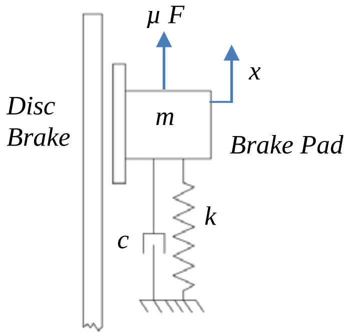


Figure 1: Schematic diagram of the disc brake and brake pad system

Where, $H = (-\omega^2 m + j\omega c + k)^{-1}$ and the frequency response function is,

$$x(\omega) = H(\omega)\mu F(\omega) \tag{3}$$

The natural frequency is:

$$\omega = \sqrt{\frac{k}{m}} \tag{4}$$

$$f = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{5}$$

Where ω is the angular natural frequency, f is natural frequency, k is stiffness and m is mass. The natural frequency is a function of stiffness and mass. Natural frequency is directly proportional to the stiffness and inversely proportional to the mass [8].

Methodology

Material Preparation and Machining Process

The brake rotor is cut by the CNC Lathe machine, Mazak 200MY Super Quick Turn with Mazatrol PC-Fusion CNC 640T as shown in Figure 2 to reduce the surface thickness of the brake rotor and to create the effect of wear after the impact hammer testing on the original thickness of the brake rotor has been done. The wear thickness of the disc brake rotor in this paper are 0.0 mm, 0.5 mm, 1.0 mm, and 1.5 mm. An impact hammer test is also conducted for all the wear thickness of the disc brake rotor.

A new disc brake type for B-segment or small car is used with the thickness and weight of 15.8 mm and 3.33 kg respectively. The thickness of the disc brake rotor is shown in Figure 3. In order to create 0.5 mm wear thickness on the disc brake rotor, 0.25 mm depth was cut on both the sides. This was because the brake pad was in contact with the both sides of the disc brake rotor during braking process. The procedures above were repeated again to create 1.0 mm and 1.5 mm wear thickness.

Figure 2: Disc brake fixed on chuck (A)

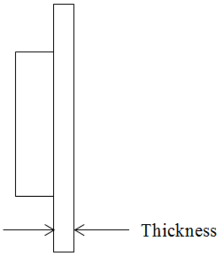


Figure 3: Side view of disc brake

Table 1 shows the physical properties of the disc brake rotor with wear thickness of 0.0 mm, 0.5 mm, 1.0 mm, and 1.5 mm. The mass of the disc brake rotor are measured and recorded after each machining process. The volume of the disc brake rotor is calculated by using $V = m/\rho$. Where V is volume, m is mass, ρ is density. The disc brake rotor is made up of grey cast iron. Grey cast iron has a density of 7200 kg/m^3.

Table 1: Properties of disc brake rotor

Thickness Reduction from original (mm)	**Mass (kg)**	**Volume (mm^3)**	**Final Thickness (mm)**
Original	3.33	4.625×10^{-4}	15.8
0.50	3.23	4.486×10^{-4}	15.2 – 15.3
1.00	3.05	4.236×10^{-4}	14.4 – 14.8
1.50	2.97	4.125×10^{-4}	14.0 – 14.3

The finished surface of the disc brake presented in Table 1 indicated that uneven surface variation of the disc brake could have happened during the disc brake rotation, it therefore considered to be induced by the hammering excitation vibration.

Experimental Modal Analysis

Experimental modal analysis is defined as the process of identifying the dynamic behavior of a structure of brake system in terms of its modal parameters such as frequency, damping ratio and mode shapes. The disc brake rotor is tested by using the impact hammer test in free-free condition. The impact hammer test is used to determine the modal parameters of the disc brake rotor using the impact hammer, accelerometer and LMS Dynamic Signal Analyzer. The impact hammer (Brüel & Kjær Type 8206-002), single axis accelerometer (Brüel & Kjær Accelerometer Type 4514-001), LMS Mobile Scadas Type SCM201V, and LMS Test.Lab Rev. 13 were used in this test. The model generation of the disc brake is created in the LMS Test.Lab Rev 13 according to the geometry of the actual disc brake. Then, the brake rotor is divided into 8 identical regions, with each region being 45° as shown in Figure 4.

Figure 4: The brake rotor setup with free-free boundary condition

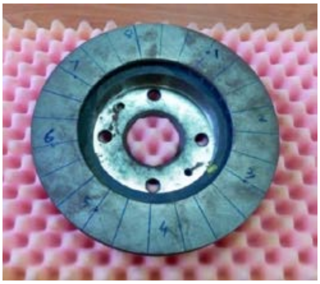


Figure 5: Point distribution of disc brake rotor.

For experimental modal analysis, the most popular boundary condition is free-free boundary condition. This is because the free-free boundary condition is the easiest to handle in an experiment and besides that, it has better descriptions on structural dynamic behavior of a single component without assembly. The experimental work done where the disc brake rotor is hung on the hanging support to create a free-free boundary condition as shown in Figure 4. The impact hammer and accelerometer are connected to the LMS as shown in Figure 6.

The accelerometer is fixed on the surface of the brake rotor and the impact hammer is used to excite different predefined points on the brake rotor. The method of the impact hammer test of this paper is roving hammer. The location of the accelerometer is fixed at point 1, and the impact hammer gives impact to the disc brake rotor from point 1 to point 8. The frequency response function is obtained from the LMS Test.Lab and resolved to extract the natural frequencies and mode shapes. The data obtained from the LMS data acquisition system is analysed using modal analysis.

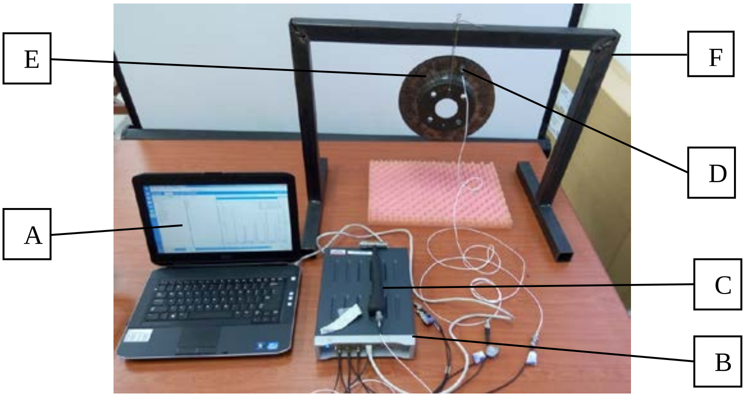
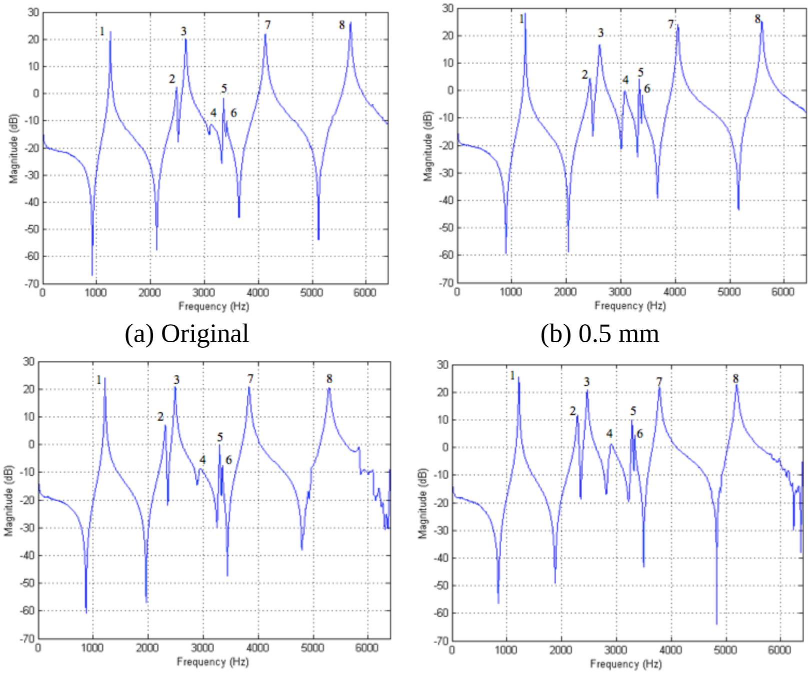


Figure 6: The experimental setup for impact hammer test. A) LMS Test.Lab; B) LMS Scadas Data Acquisition System; C) Brüel & Kjær Impact Hammer Type 8206-002; D) Brüel & Kjær Accelerometer Type 4514-001; E) Disc brake rotor; F) Hanging support.

Results and Discussion

In experimental modal analysis, the modal parameter estimation with stabilization diagram was obtained by LMS Lab.Test to find the stabilization of the modes. The frequency response function (FRF) for the disc brake rotor with original thickness, thickness reduction by 0.5 mm, thickness reduction by 1.0 mm and thickness reduction by 1.5 mm are shown in Figure 8. Peaks of the FRF show the modes of the disc brake rotor, there are in total 8 modes within the bandwidth of 6400 Hz for every thickness reduction. These FRFs also shows that most of the modal peaks reach 25 dB to 30 dB. With the aid of LMS Test.Lab Rev 13, natural frequency, damping and the mode shape of each mode are obtained. It can be seen that the value of the natural frequency decreases when the thickness reduction increases except mode 1. It indicates that after a period of usage of the disc brake rotor, the natural frequency of the disc brake rotor is decreased as explained in equations (5).



(a) Original

(b) 0.5 mm

(c) 1.0 mm

(d) 1.5 mm

Figure 8: FRFs for thickness reduction

It therefore can be concluded that a thickness reduction in disc brake rotor causes a relatively greater increase in stiffness compared to mass. This is because the stiffness is directly proportional to the thickness of the disc brake in the order of 3 while mass is directly proportional to the thickness of the disc brake in the order of 1. A reduction of thickness can cause a more significant reduction in stiffness. Based on the analysis, thickness reduction cause a more significant reduction in stiffness compared to mass.

Table 3 shows the experimental natural frequency of the disc brake rotor for different thickness reduction from mode 1 to mode 8. Dunlap et al. explained that the low-frequency squeal is typically occurs between 1 kHz and 5 kHz yet below the first circumferential mode of the rotor. The coupling of two or more modes of brake components producing optimum conditions for the occurrence of brake squeal. High-frequency brake squeal is defined as a noise which is produced by friction induced excitation causing by coupled resonances of the rotor itself as well as other brake components. It is typically classified as squeal noise occurring for frequency ranges between 5 kHz and 16 kHz [9]. The human ear is most sensitive to the frequency region of 1 kHz to 4 kHz, therefore the low frequency squeal is considered in this paper. Triches et al. stated that the low frequency squel is the most annoying type of brake noise [6]. In conjunction with Table 2, Figure 9 shows the graph of natural frequency against different thickness reduction.

Table 3: Natural frequency for different thickness reduction

Mode No.	**Natural Frequency (Hz)**			
	0.0 mm	**0.5 mm**	**1.0 mm**	**1.5 mm**
1	1256.4	1250.1	1216.5	1220.6
2	2486.9	2442.7	2311.6	2290.6
3	2654.9	2616.2	2490.4	2466.6
4	3092.1	3061.7	2910.6	2881.3
5	3348.7	3332.2	3295.9	3283.3
6	3407.0	3355.0	3347.3	3333.0
7	4130.0	4048.3	3834.1	3783.6
8	5709.6	5588.7	5290.4	5194.2

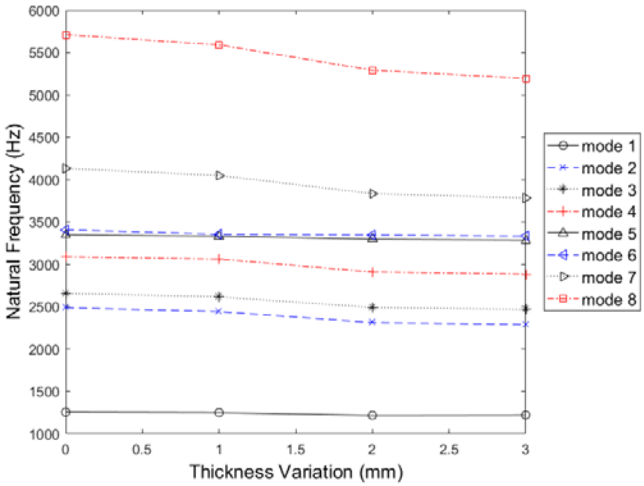


Figure 9: Natural frequency for different thickness reduction

Figure 10 shows the experimental damping ratio of the disc brake rotor for different thickness reduction. The damping ratio obtained from the modal analysis indicates that the disc brake structure exhibit at relatively low damping. The low damping can be also observed from the narrow peak of the FRF graphs. As the damping ratio is very small, therefore the damped natural frequency of the disc brake is approximately equal to the natural frequency of the undamped disc brake [10]. The energy was lost due to the fluid friction even if the disc brake is hung in a free-free condition. As long as the damping ratio is very small, the damped natural frequency of the disc brake is approximately equal to the natural frequency of the undamped disc brake.

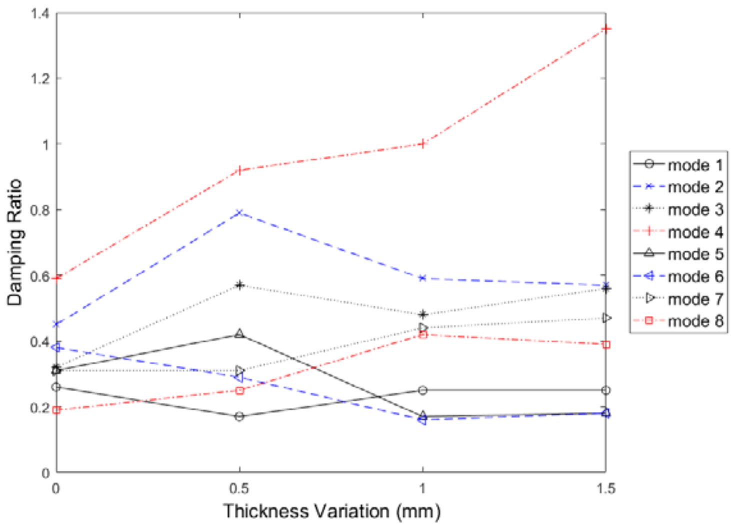
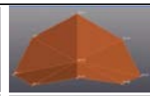
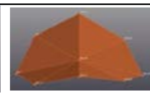
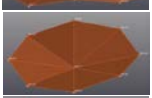
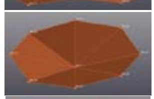
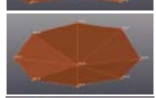
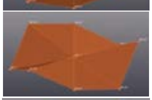
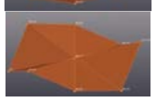
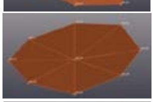
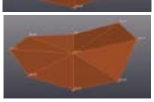
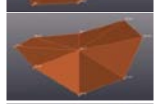
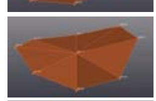
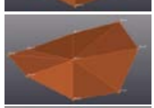
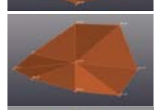
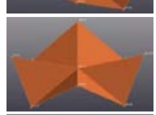
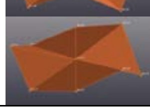
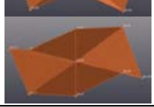
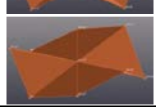
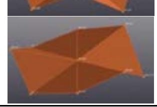


Figure 10: Damping ratio for different thickness reduction

Table 3 shows all 8 mode shapes from the measured data of the disc brake rotor with different thickness reduction of original (0.0 mm), 0.5 mm, 1.0 mm, and 1.5 mm. All the modes except for mode 4 are bending modes while mode 4 is a torsion mode. At the same mode, the mode shape is similar after thickness reduction of the disc brake rotor. A. Shahril [5] had examined the natural frequencies and mode shapes of the disc brake rotor with different wear thickness by using ANSYS. The experimental results obtained from this paper are used to compare with the FEM results obtained of the disc brake rotor with different thickness reduction by Shahril et al. [5]. By comparing the FEM results obtained Shahril et al. with the experimental results obtained from this paper, it can be seen that both of the results show that the natural frequency of the disc brake rotor decreases when the thickness reduction of the disc brake rotor increases.

The modes of the FEM results which have a closer value of natural frequency to the modes of experimental results are chosen to compare the experimental results with FEM results. The mode 1 and mode 6 from the FEM results are chosen to compare with the mode 1 and mode 4 in the experimental results respectively, a maximum difference of 10% between experimental and numerical results throughout to be acceptable. Figure 11 shows the comparison between the selected modes from FEM result and EMA results.

Table 3: Mode shapes for different thickness reduction using EMA

	Mode Shapes				
Mode No.	**(Original)**	**0.5 mm**	**1.0 mm**	**1.5 mm**	**Shapes**
1					1st bending
2					2nd bending
3					1st twisting
4					3rd bending
5					4th bending
6					5th bending
7					2nd twisting
8					6th bending

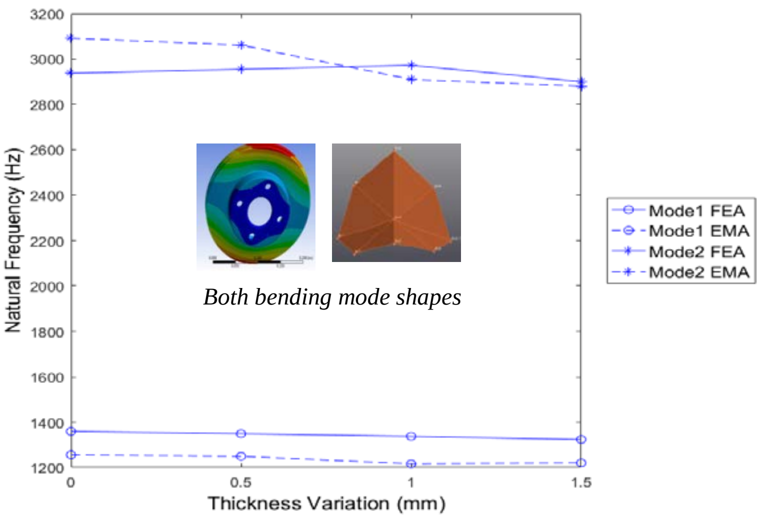


Figure 11: Comparison of FEA [5] and EMA for the natural frequency and thickness reduction

Conclusion

The experimental study of structural dynamic behavior of disc brake rotor with different wear thickness was carried out for disc brake rotor in free-free condition using impact hammer test. The modal parameters such as natural frequency, damping ratio and mode shape were obtained by using the impact hammer test. In this paper, the wear thicknesses of disc brake rotor are 0.0 mm, 0.5 mm, 1.0 mm, and 1.5 mm from the original thickness of 15.8 mm. For original thickness of disc brake rotor, the first eight measured natural frequencies were 1256.4 Hz, 2486.9 Hz, 2654.9 Hz, 3092.1 Hz, 3348.7 Hz, 3407.0 Hz, 4130.0 Hz, and 5709.6 Hz. The results show that the natural frequency decreases when the thickness reduction increases at the same mode. This was because the thickness reduction causes a reduction in the stiffness of the disc brake rotor which was more significant compared to the reduction in mass of the disc brake rotor. At the same mode, the mode shape was similar after thickness reduction of the disc brake rotor. It can therefore be concluded that the wear effect of the disc brake rotor was one of the factors which may leads to the brake squeal problem due to the decrement of the natural frequency of the disc brake rotor.

Acknowledgements

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