EXPERIMENTAL MODAL ANALYSIS OF BRAKE SQUEAL NOISE

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ABSTRACT

Disc brake squeal noise continues to be the major problem concern for the automotive industry. In this paper, the brake squeal phenomenon of brake rotor disc had been investigated using experimental modal analysis (EMA). The modal analysis excitation technique known as impact hammer test has been carried out to obtain modal parameter of brake disc structure namely the natural frequencies of the brake disc. Through the use of accelerometers, it is possible to measure up to four natural frequency squeal modes associated to the disc brake rotor. This technique can be used to verify the natural frequency obtained from finite element modelling to validate the computational modelling since the measurements were taken on the true structure of disc brake.

Keywords: Brake squeal, experimental modal analysis, impact hammer test.

1.0 INTRODUCTION

Disc brake squeal noise continues to be the major problem concern for automotive industry despite efforts to reduce its occurrence during the past decades. Despite of all effort, still many researches failed to provide reliable meaning of preventing the brake squeal phenomenon.

From theoretical perspective, the disc brake squeals noise can be classified as a friction induced type of vibration. The characteristic and understanding of this problem is very complicated due to the fact that the system is a transient phenomenon. The brake disc rotor which acting like a speaker is a moving component and the assembly brake component combine many component parts with complex interface.

Modal analysis is one of the methods used to understand the brake squeal phenomenon. Experimental modal analysis excitation technique known as impact hammer test can determine the natural frequencies of the brake disc related to brake squeal. This investigation is very important as brake squeal often involves modal coupling between various modes associated to natural frequency.

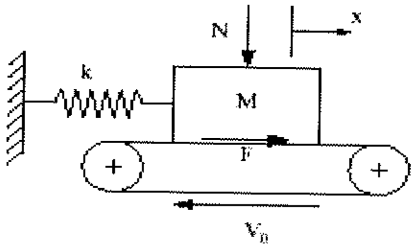
2.0 DEFINITION OF BRAKE SQEUAL

There are three general categories of noise associated with the automotive brake disc. These categories are classified according to the frequency range in which they occur.

i. Low Frequency Noise (100 – 1000 Hz)
ii. Low Frequency Squeal (1000 – 5000 Hz)
iii. High Frequency Squeal (5000 Hz >)

2.1 Low Frequency Noise

Low frequency disc brake noise typically occurs in the frequency range of 100 and 1000 Hz. The noise reside in this category are known as grunt, groan grind and moan. There are caused by friction material excitation at the brake rotor and lining interface where the energy is transmitted as vibrational response towards the brake corner and couples with other chassis component. To explain low frequency noises, the best-known model is referred to as non-linear stick-slip vibration as shown in figure 1. This simplified model shows a fragment of pad material rubbing against the disc is connected to the calliper by means of a viscous-elastic system, comprising a spring and a shock-absorber. This model assumes that the friction coefficient is unstable and varies in a linear manner with the rotational speed of the disc relative to the pad. The possibility of vibration emissions is in the range of 200 to 400 Hz.



"Fig. 1" A Simple Elastic Rubbing Surface Model

2.2 Low Frequency Squeal

For the frequency bandwidth range above 1 kHz to 5 kHz, the noise generated is classified as low frequency squeal. The mode of failure can be associated with frictional excitation couple with a phenomenon known as modal 'locking'. Modal locking of two or more mode of various structures producing optimum conditions for brake squeal as rotor typically vibrates with 2 to 4 nodal diameters.

2.3 High Frequency Squeal

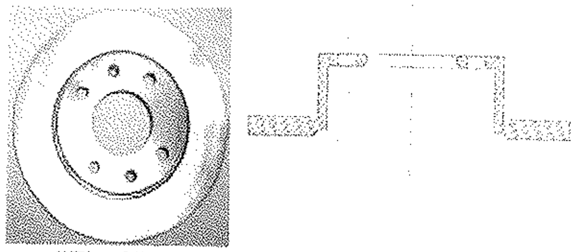
High frequency squeal is classified as squeal occurring above 5 kHz. The noise is produced by friction induced excitation imparted on coupled resonance of the rotor itself and also by other brake components. In high frequency squeal, Lang and Smales indicate that the disc nodes are much closer together and pad bending vibration becomes very important [6].

There are several factors that influence both high and low frequency squeal, which is the rubbing surface thickness, hat structure or vane pattern structure and materials used for production.

3.0 DESCRIPTION OF DISC BRAKE

The type of brake disc used in this project is in the form of top hat structure as shown in figure 2. It consists of two simple components, an annular disc and a cylinder. The design of connecting cylinder provides geometric offset for the mounting of brake disc to the vehicle. There are two flat annular discs.

The first type of annular disc attached around the inside cylinder providing mounting surface between brake disc and vehicle axle. The second annular disc attached around the outside of cylinder to provide braking surface. The mass of brake disc measured is equivalent to 3.15 kg.



"Fig. 2" Top Hat Structure Disc Brake

4.0 EXPERIMENTAL MODAL ANALYSIS (EMA)

The experimental approach to modelling the dynamic behaviours of structures through impact hammer test modal testing consists of these four steps:

a) Setting up the modal test
b) Taking the measurements
c) Analysing the measured test data
d) Documented results and compare with modelling data

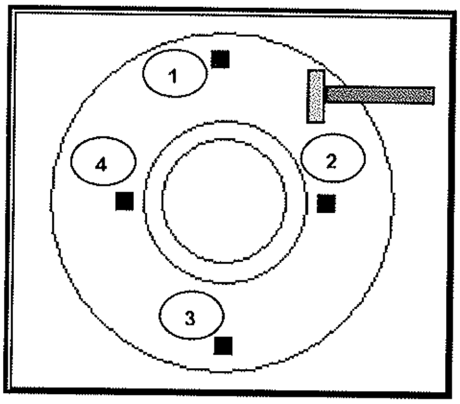
4.1 Experimental Device: Impact Hammer Test

A hammer is a device that produces an excitation force pulse to the test structure. It consists of hammer tip, force transducer, balancing mass and handle. The hard tip is selected as the solid metal structure like disc brake as all modes have higher frequency above 200 Hz by providing sufficient energy for high frequencies. When the tips strike on the circular disc surface, the pulse distributes the energy to a wide range of spectrum. The hardness of the tips together with the structure of rotor disc to be tested is directly related to the frequency range input pulse force.

An accelerometer is used to measured acceleration of rotor disc and the output of the signal is in the form of voltage. The signal transformed by signal conditioner before an analyser processes it. There are two aspects in the acceleration measurements, frequency and amplitude. The type of accelerometer used in this experiment is piezoelectric accelerometer.

4.2 Experimental Procedure

Brake rotor disc was clamped at the four-bolt hole of rotor's top hat connecting cylinder to simulate realistic boundary condition at the four location of mounting bolts. An accelerometer is attached on the surface of the circular disc. Repeatedly, the hammer is striking on the same position for 4 times to extract 4 different modes and natural frequencies. By positioning the accelerometers at four other location points on the top of circular disc (figure 3), the hammer is used to strike again to obtain another 4 set of modes and natural frequencies.



"Fig. 3" Accelerometer located positions

4.3 RESULTS AND ANALYSIS

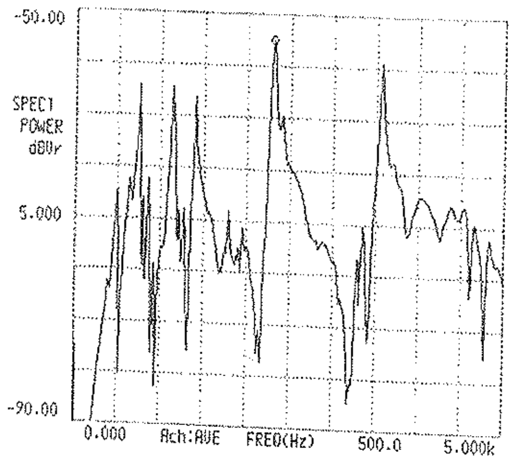
4.3.1 Frequency Response Function (FRF) Plots

Figure 4 on shows the computed set of test data of FRF from hammer excitation, which solved using FFT analyser. The response signal obtained from accelerometer with respect to the function of time is shown in figure 5.

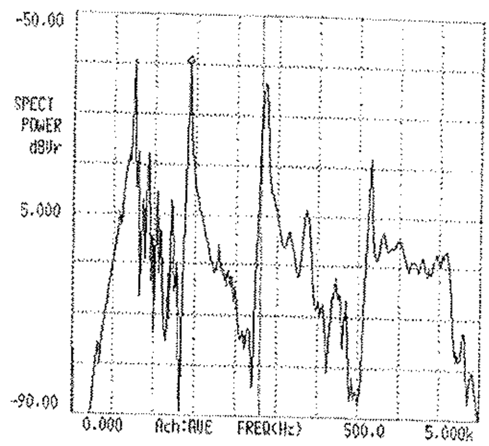
The graph in figure 4 shows the plot of spectrum power intensity of noise level in decibel unit over frequency. The spectrum is continuous and the band frequency range of measurement was selected to be 500 Hz with the frequency range from 0 to 5000 Hz. The plot of spectrum power intensity of noise level in decibel was decayed by 5 and 10 dB.

The FRF plots clearly indicate that at certain frequencies of disc brake, the excitation input force caused brake disc structure having narrow peaks and high value of spectrum power intensity of noise level. These peaks of the FRF plots are the natural frequencies of the brake disc. It can be seen that noise spectrum obtained from the experimental result shows peak intensity of disc brake range between 50 to 70 dB with natural frequency of 500 Hz to 4000 Hz. The first three modes of brake disc produced sharp resonance with high-level noise. Mode 1 show the highest level of noise, which was at 68 dB, and the lowest spectrum power intensity of noise level was at 50 dB for mode 4.

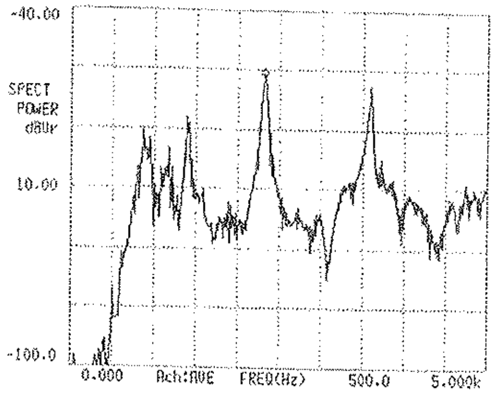
The first four modes of vibration and its natural frequencies are well separated which allowed the mode to be extracted from the peak resonance frequencies. The brake disc's modal damping, loss factor and mode shape therefore can be obtained by using peak picking method.



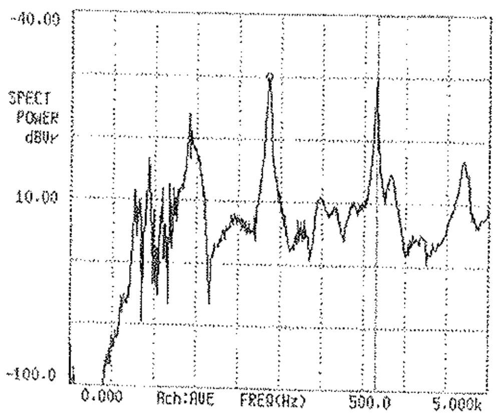
4(a)



4(b)

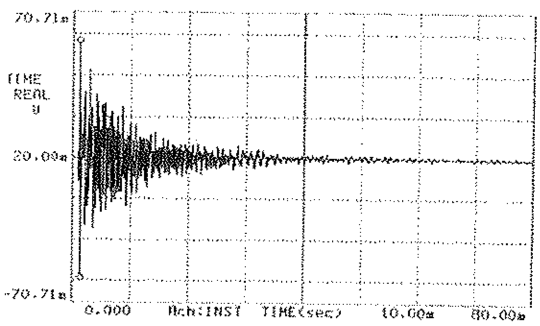


4(c)



4 (d)

***"Fig. 4"** Frequency response function FRF response signal when striking the disc brake at the same point for four different locations of accelerometer position*
(a) Position 1 (b) Position 2
(c) Position 3 (d) Position 4



***"Fig. 5"** Time Response Signal from Accelerometer*

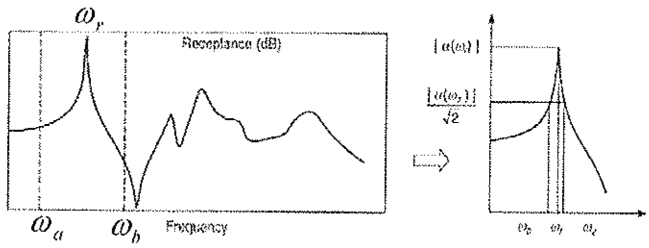
4.3.2 Assessment of FRF Data: Peck Picking Method

Peak picking method is a single degree of freedom method in view of the fact that each resonance of an experimentally determined transfers function. The approach is to compare the resonance region with an analytical transfer function of a damped single degree of freedom system. Steps for peak picking method are summarized as below:

a) Estimating Natural Frequency

The natural frequency of r^{th} mode is identified from the peak value of FRF

$$\left|\alpha_r(\omega_n)\right|_{\max} \text{ as } \omega_r = \omega_{peak}$$



***"Fig. 6"** Peak Picking Method*

b) Estimating Damping Ratio ξ and Loss Factor η

Because of ω_n is known directly from the peak location of the transfer function, the damping constant and loss factor can be computed firstly by determine the corresponding peak magnitude. Half power point at ω_a and ω_b are located from each side of the peak amplitude that is $\frac{\alpha_{\max}}{\sqrt{2}}$. Lost factor and damping ratio can be calculated as below.

Loss factor η: $\quad \eta = \dfrac{\omega_a - \omega_b}{\omega_r}$

Damping Ratio ξ: $\quad \xi = \dfrac{\omega_a - \omega_b}{2\omega_r}$

c) Q-factor

Q-factor measures the sharpness of resonant peak is defined by

$$Q\text{-}factor = \frac{1}{2\xi}$$

5.0 DISCUSSION

Table 1 shows overall extracted parameter obtained from the first four identified modes of vibration from FRF curve in figure 4(a)~4(d) identified by the four of vibration modes. FRF plots shows how modes can cause the disc brake structure to vibrate and produce noise. At small input force form the hammer can cause a very large response of natural frequencies. This is clearly indicated from the narrow peaks in FRF plots. Therefore when the disc is excited at one of the peak frequencies, the response of disc brake per unit force will be large. As results the brake disc modes will acts like amplifier.

The damping ratio obtain from modal data indicates that the structure of disc brake exhibit relatively low damping. As the damping ratio is very small (ξ < 1), therefore the damped natural frequency of disc brake is equivalent to the natural frequencies as $\omega_d = \sqrt{1-\zeta^2\omega_n}$. The term Q-factor that determines the sharpness of resonance frequencies was originated from the field of electrical tuning where the sharpness of the resonant peak is a desirable thing. The value increased along with the increasing value of natural frequencies in which the damping ratio becomes smaller and smaller.

Table 2 summarise the advantages and disadvantages of impact hammer test despite the facts that hammer test can be constructed weight ranging from few grams to several tons covering the frequency range between 0 to 5000 Hz. Listed below are the factor affected the experimental works:

i. Quality error
   - Structure support: Boundary condition of disc brake when it is not properly clamped at the centre hole of disc brake location.
   - Accelerometer: The sensitivity and its stability since the accelerometer dictates the signal to noise ratio and therefore large and stable sensitivity affect the measurement accuracy

ii. Quantity error
   - Signal processing error due to leakage, affect of window functions and discrete Fourier function.
   - Not all modes being excited due to excitation at a node

***Table 1** The first four mode of vibration obtained from impact hammer test*

Mode	Natural Freq. (Hz)	Half Power Point Frequency (Hz)		Loss factor η	Damping ratio ξ	Q factor
		ω_a	ω_b			
1	500	480	510	0.060	0.030	16.67
2	750	745	755	0.013	0.007	75.00
3	1150	1100	1180	0.070	0.035	14.38
4	1400	1390	1405	0.011	0.005	93.33

***Table 2** The first four mode of vibration obtained from impact hammer test*

	ADVANTAGE	DISADVANTAGE
Impact Hammer Test	▪ Very convenient to set up ▪ Provides wider frequency range ▪ Frequency range can easily be obtained ▪ No elaborate fixtures are required ▪ No variable of mass loading since mass loading can cause shift in modal frequencies from one measurement to one another ▪ In expensive for testing to set up	▪ There are limited energy in the frequency range ▪ High peak force might be required and test structure might damaged ▪ The force level only varies slightly between overload and underload level ▪ Not sufficient to excite at all modes ▪ It is not a steady state solution

6.0 CONCLUSION

From the experimental investigation, the vibration frequencies of a squealing disc brake's rotor are influenced by natural frequencies and modes of stationary rotor. Thus, brake squeal occurs in the vicinity of natural frequencies of the disc. More confidants can be placed from the results of finite element model if the measurements taken on the true structure of disc brake were performed using impact hammer test to validate the computational modelling.

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