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Original Article

Reducing brake noise, vibration, and harshness by damping layer on a brake pad

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Abstract

Brake pad is a product that plays a role in the automotive parts industry as a consumable, and directly affects traffic safety. The objectives of this study were to address the problems with brake noise, vibration, and harshness (NVH) from brake pads, by using a reinforcement material layer, and to raise the quality of brake pads used in the industrial sector. As a result, prototype brake pads with reinforcement by a damping layer were prepared and tested. The experiments revealed that brake pads with damping layer can reduce vibration and also brake noise at a given frequency by at least 10 percent when braking at speeds of 50, 100 and 130 km/h. Besides, this technique can be applied to the production of brake pads as demonstrated. However, the prototype brake pads are still in product development phase for testing in a variety of conditions, as regards both temperature factors and the usage period.

Keywords: Vibration and Harshness (NVH), brake noise, brake noise reduction, brake pads, damping layer

1. Introduction

Many countries currently have regulations with criteria and standards for auto parts. Therefore, industrial products, automobiles, and auto parts are also undergoing improvements. The automobile brake system comprises some vital auto parts relating to safety and consumer sensitivity. The confidence in brake system efficiency is the foundation of overall automotive efficiency. Research and development can lead to product improvements, particularly when data are generated for material tangible products. Further, brake pad improvement through the use of composites is something that all the stakeholders give a high priority.

The brake noise has been studied (Wallaschek, Hach, & Mody, 1999), and it can be classified by its characteristic frequency to four types as follows.

1) Judder: A phenomenon with low frequencies < 100 Hz, usually in vibration, e.g. vibration from speed reduction due to the brake or suspension system.
2) Groan and moan: A phenomenon in the bandwidth between 100 and 500 Hz. This phenomenon may contain more than one pure frequency and is independent from wheel speed, temperature, and pressure.
3) Howl: A phenomenon in the bandwidth from 500 Hz to 1 kHz, with similar properties to groan and moan.
4) Squeal: A vibration phenomenon between 1 kHz and 20 kHz that is classified into 2 ranges, i.e. low frequency squeal (1 kHz to 7 kHz) and high frequency squeal (7 kHz to 20 kHz). The vibration is caused by a violent resonance of the brake system.

The sources of squeal phenomenon can be identified. To clarify, the low frequency squeal is from out-of-plane resonance between the brake rotors and brake pad bending, whereas high frequency squeal is from in-plane resonance between the brake rotors and brake pads, as in Figure 2: Brake System Resonance (Triches, Gerges, & Jordan, 2004). In this case, it is usually found that brake rotors are stronger material than in the first case. As a result, vibrations against other equipment occurs, followed by resonance with a high-pitched noise.

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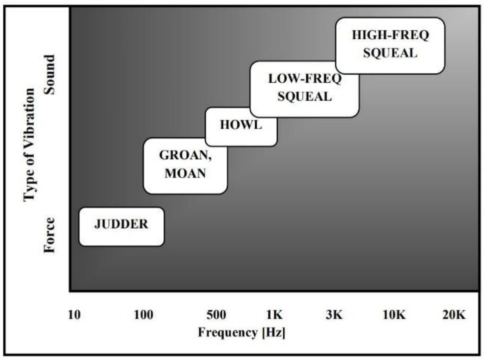

Figure 1. Phenomenon of brake noise and vibrations by frequency range (Wallaschek *et al*., 1999)

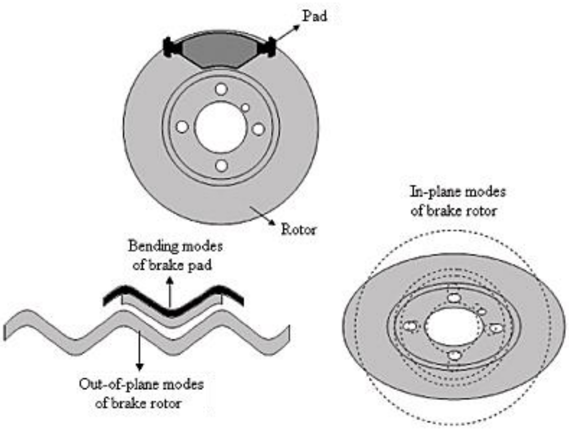

Figure 2. Resonance between the brake rotors and brake pads (Triches *et al*., 2004)

The notion of adding damping material to the brake system was initially introduced by Triches *et al*. (2004). The material could solve low frequency squeal, but came with a higher cost and longer production time. Therefore, properly sized damping materials adhering to the back of the brake pads were introduced by Zheng, Cai, and Tan (2004). This method was widely applied to solve brake noise during an initial period. However, the useful life was not long due to limitations related to temperature.

Later, there was an effort to apply analytical and numerical methods in related studies. A model for analyzing brake system vibrations that could induce vibrations automatically based on dry friction during movement was introduced by Shin, Brennan, Oh, and Harris (2002). Unfortunately, the model could simulate vibration behavior only in the out-of-plane mode. However, a study of vibration behavior in in-plane and out-of-plane modes based on coupled resonance equations was also reported by Ouyang, Mottershead, Cartmell, and Brookfield (1999). It reveals that the vibrations in these modes affect each other, i.e., interact.

Besides, a study with a more complex model was subsequently reported by Flint (2000). The model with a caliper, piston, brake pads, and a brake rotor was constructed and could calculate complex as well as ongoing vibrations. However, there were some limitations in terms of degree of freedom of movement for each part and deceleration of contacting parts, for which the values could not be changed during calculation. Furthermore, Liu *et al*. (2007) applied ABAQUS version 6.4 to study brake system resonance based on non-linear principles, in which variables were modified according to the conditions during the simulation. Simultaneously, Nack (1999) introduced a non-linear model with the use of MSC/NASTRAN. Equations for elasticity between brake pads and a brake rotor were set. Pressure and system velocity could be adjusted. Furthermore, machine learning-based methods were used to study detection and characterization of vibrations, to understand sensitivities and to predict brake squeal, by Stender *et al*. (2021). A recurrent neural network (RNN) was employed to learn the parametric patterns and it also identified the complicated patterns and temporal dependencies in the loading conditions.

Consequently, the study of tribological properties to find the link with vibrations was carried out by El-Butch (2000). It was found that the properties of friction materials affected pressure, speed, and sliding temperature, which could cause noise. Moreover, the effect of replacement of copper (Cu-free) in the brake-pad friction material on noise and vibration was demonstrated by Kalel, Darpe, and Bijwe (2021). The brake-pads made of stainless-steel particles and copper-based brake-pads do not significantly differ in brake-noise levels. Stainless-steel particles can be a suitable choice for Cu-free pads.

Simultaneously, the correlation between the size and mechanical properties of the surface contacts of the brake lining and the stick-slip characteristics were studied by Joo, Gweon, Park, Song, and Jang (2021). A Krauss tester and a scaled dynamometer were used to demonstrate the stick-slip characteristics. Avoiding the formation of large and high-strength contacts can be used to optimize brake lining. Then, the influences of carbon nanotubes on the properties of friction composite materials were studied by EL-kashif, Esmail, Elkady, Azzam, and Khattab (2020). The addition of carbon nanotubes into the friction materials could improve their mechanical properties such as hardness, strength, and modulus. A good combination of mechanical and tribological properties at 0.5**%** carbon nanotubes was suggested for reducing the noise and vibrations from the friction materials.
An optimization study to address brake squeals, aiming to minimize the strain energy of vibrating pads with constrained layer damping was undertaken again by Lakkam and Koetniyom (2012). The results can guide specification of the position of the constrained layer damping patch under pressure conditions. Moreover, the physical characteristics of brake discs or rotors (front– and back-vented brake discs) affecting the heat transfer were studied (Lakkam, Suwantaroj, Puangcharoenchai, Mongkonlerdmanee, & Koetniyom, 2013). This eventually led to the study of heat transfer on the front and back vented brake disc affecting vibrations (Lakkam, Puangcharoenchaib, & Suwantaroj, 2017).

2. Materials and Methods

2.1 Brake pad design

In the original design, the damping materials have been used by installing a backing plate as shown in Figure 3. There are four layers combined: 1) nitrile rubber, 2) carbon steel, 3) nitrile rubber, and 4) cold pressure adhesive. The 4 layers shim were designed to optimize anti-noise performance,

Figure 3. The original design for brake pads

add flexibility and also high temperature friction durability when braking, with a special adhesive. This principle is similar to the constrained layer damping technique (Lakkam & Koetniyom, 2012). However, the former design is costly and the damping materials can be lost and damaged in maintenance over a long term.

Brake pad design and production were implemented by Compact International (1994) Co., Ltd., a support company with production preparedness based on the idea of producing new pads. Original brake pads with only friction materials would be modified by adding a layer of friction materials mixed with a high-damping material. The approximately 2 millimeters thick damping layer based on nitrile rubber was inserted between friction material and backing steel plate, as shown in Figure 4, to absorb vibrations that cause noise. This design can eliminate the problem in terms of loss and damage due to maintenance over a long term. The materials and properties are shown in Table 1.

2.2 Test method

The brake noise reduction by alternative brake using added layers of friction materials with damping was studied. The operating conditions were simulated on a single dynamometer, as in Figure 5.

A wind tunnel was also installed to create air flow for the brake cooling system. The testing conditions were in compliance with JASO C 406 (2000) and were employed as a testing guideline. The conditions were referred from the table of JIS D4411 (2015) and adjusted to the test, as in Table 2.

A noise level meter was installed in compliance with SAE J2521 (2013), which states that a microphone must be installed in a plane not less than 10 cm from the source and not less than 50 cm above the ground, as in Figure 6. Noise levels were measured in weighting A (dBA) because these are

Figure 5. A single dynamometer, Faculty of Engineering, RMUTP

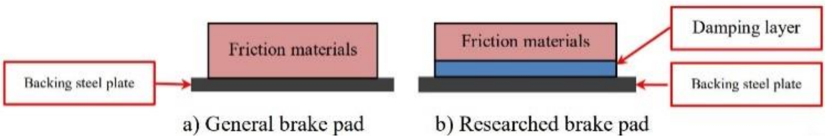

Figure 4. Brake pad design guideline

Table 1. Materials and properties

Model	Fiber (%)	Metal (%)	Lubricant (%)	Filler (%)	Damping layer	Damping ratio
A	13.5	8.0	17.5	61.0	N/A	0.0054
A1	13.5	8.0	17.5	61.0	Available	0.0189
A2	13.5	8.0	17.5	61.0	Available	0.0109

Table 2. Testing conditions (JIS D 4411, 2015)

Test items & sequence		Test conditions vehicle class				
		Initial speed (km/h)	Initial temperature (ºC)	Braking pressure (MPa)	Repetitions	Remarks
a	Initial measurement	-	-	-	-	Measurement of pad thickness
b	Burnish	65	80	3.5	200	
c	Effective test	50 100 130	no less 80 ±10	no less 1±0.02	6 or more at each test	
d	Final measurement and inspection	-	-	-	-	Inspect brake pad thickness

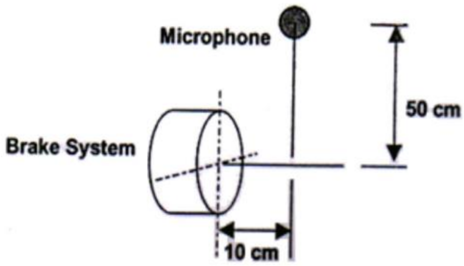

Figure 6. A noise level meter position (SAE J2521, 2013)

relative units responding to noises similarly to human hearing. The test was based on equations 1 and 2 as follows.

$$SPL \, or \, L_p = 20 \log\left(\frac{P}{P_0}\right) \tag{1}$$

where

SPL or L_p is sound level (dB)

P is sound pressure (Pa)

P_0 is referred sound pressure (20 μPa)

$$W_A = 10Log\left[\frac{1.562339 f^4}{\left(f^2+107.65265^2\right)\left(f^2+737.86223^2\right)}\right] + 10Log\left[\frac{2.242881\times10^{16} f^4}{\left(f^2+20.598997^2\right)\left(f^2+12194.22^2\right)^2}\right] \tag{2}$$

where

WA is weight A (dBA)

f is frequency (Hz)

As a result, sound and vibrations were recorded during the experiment in order to collect data for source analysis. This experiment focused mainly on out-of-plane resonance. The vibration acceleration in the out-of-plane direction (AZ) of the vibration sensor was detected, as in Figure 7, and the braking torque was measured as shown in Figure 8.

The air cooling and air flow in the wind tunnel were maintained at the level of 11 m/sec and were monitored. Brake pressure was controlled by mass release (Load) in order to press the mechanism parts of the brake master cylinder.

The time-domain signals of the measured sound pressure and vibration acceleration in the out-of-plane direction were obtained and calculated by Fast Fourier Transforms (FFTs) of these signals. This approach for sound and vibration research is suggested in Xiang *et al*. (2020).

3. Results and Discussion

The results focused mainly on the comparison between brake noise from general brake pads (No added damping layer) called “model A” and brake pads with added damping layer called “model A+”, which are brake pads with the same materials and ingredients of general brake pads. A layer of materials with high damping was added, which has been developed and produced by Compact International

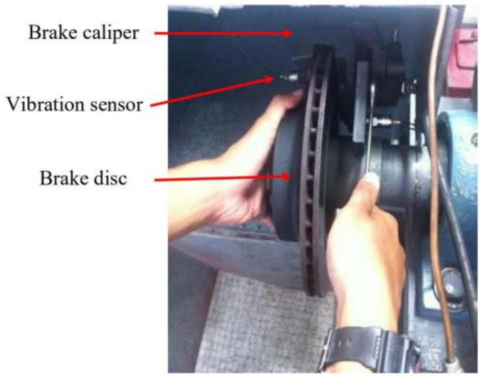

Figure 7. The installed position of the vibration sensor

Figure 8. Torque transducer installed in single dynamometer shaft

(1994) Co., Ltd. The analysis was based on maximum torque during braking as an indicator of the period of time for braking. For example, the result in case of regular brake pads when braking at 50 km/h is shown Figure 9.

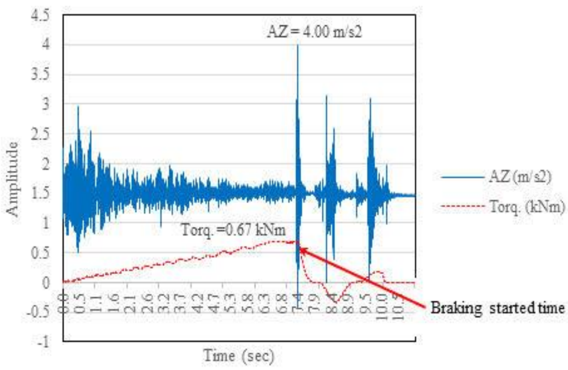

Figure 9. Torque and acceleration using general brake pads (Model A)

Figure 9 shows torque and acceleration during the braking. Torque gradually increases between 0 and 6 seconds of the testing time. Then, the simulated speed (50 km/h) and torque (0.67 kNm) were stable at 6.3 seconds of the testing time approximately. The braking started at the maximum torque during braking. In this moment, the maximum vibration is displayed at 4.00 m/s^2 and also sound from the experiment is recorded by a computer.

Similarly, Figure 10 illustrates torque and acceleration during braking. The simulated speed (50 km/h) and torque (0.43 kNm) are stable at 7.0 seconds of the testing time approximately similar to cases of Model A. However, the maximum vibration appears after braking approximately for a second by 1.88 m/s^2, which is reduced to one time. The experiment reveals that the constrained layer damping at backing plate is able to work under the braking pressure. This technique is widely used as a general design by manufacturers.

For brake pads with damping layer (Model A2), the maximum vibration is almost not different. Figure 11 illustrates torque and acceleration during braking. The simulated speed (50 km/h) and torque (0.65 kNm) are stable at 5.0 seconds of the testing time approximately. The maximum vibration appears after braking for approximately a second by 2.01 m/s^2 which is only different by 6% from the vibration result at 1.88 m/s^2 for the original design of brake pads (Model A2). The results reveal that the damping layer material is working under the braking pressure. It is not only consistent with theory of influences of strain energy on damping ratio for the brake pad system (Lakkam & Koetniyom, 2012), but each stress also acts through a different displacement. There is more possibility that the maximum flexibility causes more internal energy release in the brake pad (Mongkonlerdmanee *et al*., 2013).

Moreover, the experimental data were transformed from time domain into frequency domain in order to find the noise levels by frequency, emitted from the brake at certain times.

The comparison of Sound Pressure Level (SPL) between models A, A1 and A2 is gave 70, 70 and 61 dB A, respectively, without significant difference. The maximum SPL was slightly reduced. However, the maximum vibrations can be mitigated by both techniques (Model A1 and A2) as regards the maximum vibrations, as shown in Table 3.

Further, a comparison of brake noise is displayed in Table 4. According to all test results for vibration during braking and noise at the same frequencies between general brake pads (Model A) and brake pads with an added damping layer (Model A2), it was found that the use of a damping layer could reduce vibrations and noise, compared at the same frequency, by not less than 10% under the experimental conditions at speeds of 50, 100, and 130 km/h.

4. Conclusions

We have reported on overall results from experiments on vibrations and sound from braking. For experimental analysis, brake pads with a damping layer (Model A2) can reduce vibration in out-of-plane mode and sound (compared by frequency) at least by 10 percent from those of a general brake pad (Model A), and similar results were obtained for the original design of brake pads (Model A1) in terms of sound and vibration. The noise, vibration, and harshness (NVH) felt by consumers is improved to smoothen and soften the braking experience. Moreover, the brake pads that had inserted damping layer (Model A2) will not experience loss and damage due to maintenance over a long term. Simultaneously, the total production costs of the original design of brake pads (Model A1) and the brake pads with damping layer (Model A2) are only slightly different.

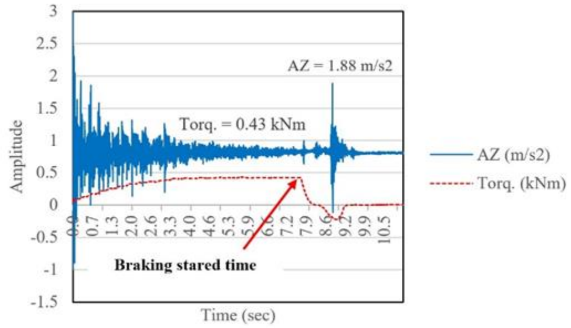

Figure 10. Torque and acceleration using original design of brake pads (Model A1)

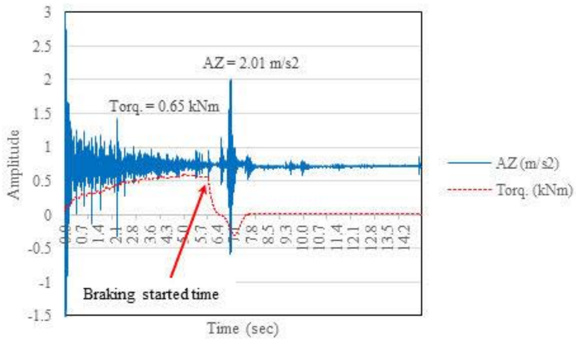

Figure 11. Torque and acceleration using added damping layer in brake pads (Model A2)

Table 3. Results on maximum vibration

Model of brake pads	A	A1	A2
Braking speed (km/h)	Max.vibration (m/s^2)		
50	4.00	1.88	2.01
100	1.64	1.26	1.29
130	1.97	1.42	1.48

Table 4. Results of Sound Pressure Level (SPL) measurement

Model of brake pads	A		A1		A+	
Braking speed (km/h)	Noise (dB A)	Frequency (Hz)	Noise (dB A)	Frequency (Hz)	Noise (dB A)	Frequency (Hz)
50	70	2,360	49	2,360	61	2,360
100	73	2,710	59	2,710	60	2,710
130	71	2,712	59	2,712	62	2,712

However, this is a preliminary study of the brake pad. This product also needs testing in real use over a long period. Perhaps there are more parameters affecting braking efficiency, such as heat transfer and friction properties, important over the lifetime. Moreover, the optimization of the damping layer (mass side or dimension) is also needed in terms of production, affecting the cost of brake pads as consumable products that play an important role in traffic safety.

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