M.H. Hussin et al. 2010. *Int. J. Vehicle Structures & Systems*, 2(2), 69-73
ISSN: 0975-3060 (Print), 0975-3540 (Online)
doi: 10.4273/ijvss.2.2.04

International Journal of Vehicle Structures & Systems
Available online at www.ijvss.maftree.org

Effects of Lining Thickness on Squeal in Drum Brake Assembly: Experimental Investigations

Muhammad Haidir Hussin[a], Abd Rahim Abu Bakar[a,b], Mohd Rahimi Jamaluddin[a], and Romain Szlapka[c]

[a]*Department of Automotive Engineering, Faculty of Mechanical Engineering, Universiti Teknologi Malaysia, 81310 UTM Skudai, Malaysia.*
[b]*Corresponding Author, Email: arahim@fkm.utm.my*

[c]*Institut Supérieur de l'Automobile et des Transports, 49, rue Mlle Bourgeois, F-58000 Nevers, France.*
Email: szlapkaromain@aol.com

ABSTRACT:
This paper presents the effects of brake lining thickness due to wear on drum brake squeal. Brake lining will be worn out and subsequently its thickness will be reduced after a few number of braking applications. Hence dynamic properties of the lining, such as its natural frequency, might be changed. In this work, two different sets of brake lining, i.e., new and worn lining are used to investigate its effect on squeal generation. First, modal testing is performed to determine natural frequencies of those brake linings at free-free boundary condition. Later, squeal tests are carried out using brake dynamometer and squeal frequency is measured up to 10 kHz. Several squeal results are plotted over brake operating conditions to observe the influence of different lining thickness. In addition to these, squeal mechanisms, i.e., modal coupling due to closeness of the natural frequency between drum brake components and negative damping due to negative friction-velocity slope that contribute to the squeal generation are also investigated and discussed.

KEYWORDS:
Drum brake, Squeal, Lining thickness, Modal testing, Squeal mechanism

CITATION:
M.H. Hussin, A.R. Abu Bakar, M.R. Jamaluddin, and R. Szlapka. 2010. Effects of lining thickness on squeal in drum brake assembly: Experimental investigations, *Int. J. Vehicle Structures & Systems*, 2(2), 69-73.

1. Introduction

Passenger cars have been one of the essential ground transportation for people to travel from one place to another. The braking system represents one of the most fundamental safety-critical components in modern passenger cars. Therefore, the braking system of a vehicle is undeniably important, especially in slowing down or stopping the rotation of the wheel by pressing brake linings against its rotating drum. Due to this braking operation, the brake system experiences a wear on the brake lining and very often generates an unwanted and yet annoying sound such as squeal. These two issues are typically very irritating to the drivers as they are not only required to replace the severe and quick worn out brake lining to a new one but also continuously facing with the high pitch squeal noise.

Brake squeal is one type of the brake noises apart from judder, creep-groan and many other noises [1]. It is commonly defined as a friction-induced vibration that occurs above 1 kHz and its sound pressure level exceeds 70 dB [2] or usually at least 20 dB above ambient noise level. Chen [3] in his recent review suggested two mechanisms, namely negative friction-velocity slope or negative damping [4-6] and modal coupling [7-9] that have strong influence on the squeal occurrence.

The effect of wear on disc brake squeal generation has been studied experimentally by a number of researchers in the past [10, 11]. However, there is very little investigation on the drum brake assembly and especially on the influence of lining thickness due to wear on squeal generation. Eriksson et al [10] studied the surface characterization of worn brake pads on squeal generation. They suggested that the squeal generation depends on the size of brake pad contact plateaus. Sherif [11] investigated the effect of surface topography of worn pad-Plexiglas disc assembly for squeal generation. She observed that a combination of glazed pad surface and run-in disc surface would generate a squeal whilst a smooth disc surface would not trigger squeal.

Fieldhouse et al [12-14] performed a series of investigations on the drum brake noise using holographic interferometry technique. They found that the drum and the backing plate strongly contribute to the high and low frequency squeals respectively. Felske et al [15] used the same technique of Fieldhouse and found that the squeal is potentially transmitted from the backing plate. Kung et al [16] performed experimental investigations on the low frequency squeal. They found that there is no squeal generated in the dynamometer. A low frequency squeal due to the out-of-plane motion of the backing plate is observed during the vehicle test.

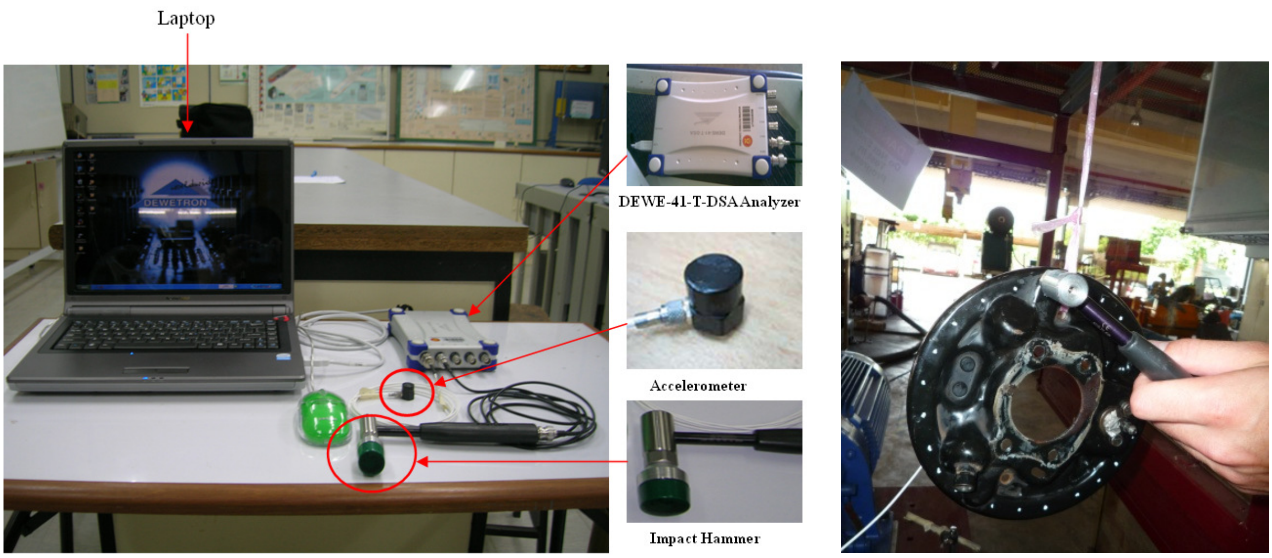
It is found that there is not much investigation on the effects of lining thickness due to wear on squeal generation has been made and published in the public domain. This paper attempts to investigate the effects of lining thickness (new and worn lining) on the squeal generation. First, modal testing is performed to determine the natural frequencies of these brake linings at free-free boundary conditions. Later, squeal tests are carried out using a brake dynamometer. The squeal frequency is measured up to 10 kHz. Several squeal results are plotted to observe the influence of different lining thickness for various brake operating conditions. In addition to these, investigations of squeal mechanisms such as the modal coupling and the negative damping that contribute to the squeal generation are discussed.

2. Modal Testing

For individual drum brake components such as a drum, a backing plate and two brake linings, roving impact hammer tests are carried out at free-free boundary conditions. As the dominant squeal frequency usually falls between 0.5 kHz to 10 kHz, the impact test is performed in the frequency range up to 10 kHz. The main aim is to obtain the natural frequency of the drum brake components and its associated mode shape. Fig. 1(a) shows the instrumentation for the modal testing. The impact hammer test that performs on the backing plate component of a drum brake assembly is shown in Fig. 1(b). A Kistler Type 9722A500 hammer and a Kistler Type 8636C50 uni-axial accelerometer are used and connected to a DEWE-41-T-DSA analyzer [17] in order to measure the signals. DEWESoft-6-DSA [17] software is used to obtain the natural frequency and its mode shape.

3. Squeal Tests

Brake dynamometer that is available in the Automotive Laboratory, Universiti Teknologi Malaysia is utilised to perform the squeal tests on the drum brake assembly. The brake dynamometer, as shown in Fig. 2, is driven by an 11 kW DC motor coupled with a 6:1 speed reducer that can provide a disc rotation up to 157 rad/s and a maximum brake torque of 400 Nm.



(a) Instrumentation used in the modal testing

(b) Impact hammer test on the backing plate

Fig. 1: Modal testing for the drum brake component

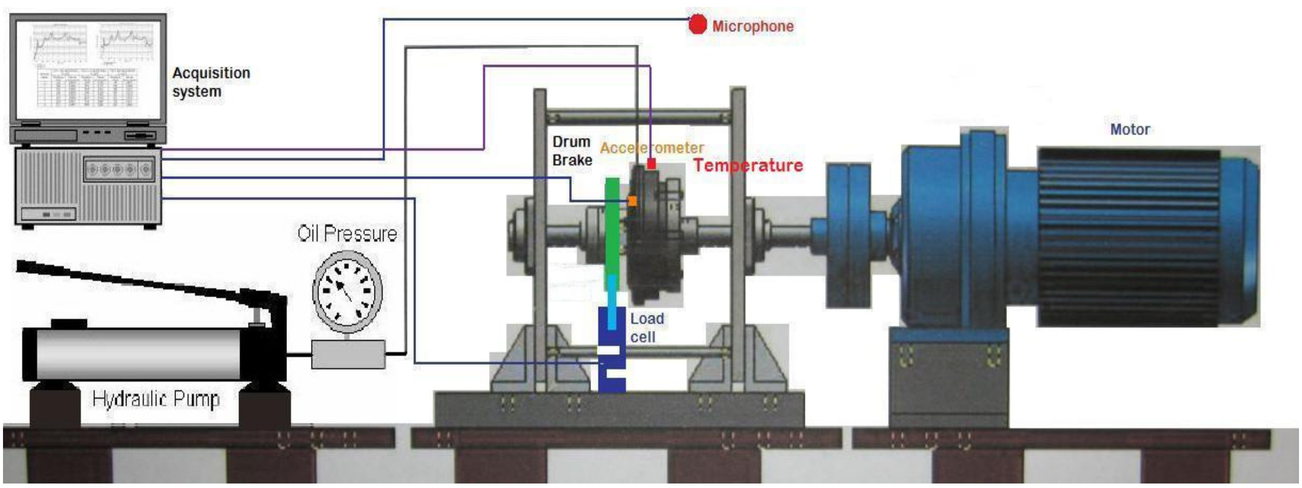


Fig. 2: Schematic diagram of the brake dynamometer

The dynamometer can accommodate any type of brake design by replacing the adapters and the support plates. A standard automotive master cylinder is used to provide the necessary hydraulic pressure to the brake system. A hydraulic brake-line pressure of 70 bars is sufficient to produce a maximum torque of the drum brake. A 3-phase Toshiba speed controller is used to adjust the speed of the drum or disc. A data acquisition system (DEWE-201) is used to monitor the drum temperature, braking torque, brake line pressure, sound pressure level, speed, and vibration response from the accelerometer located at the backing plate. The acceleration is then converted into frequency (<10 kHz) domain using Fast Fourier Transform (FFT) to obtain a power spectrum of the squeal frequencies. Microphone is also used to capture the sound pressure level in the event of squeal. This data can be compared with the accelerometer responses to obtain the squeal frequencies.

The brake squeal tests are undertaken at a constant brake-line pressure with varying dynamometer speeds and vice-versa. These tests are performed for more than 100 braking applications on each set of the brake lining. Thickness of the worn and new brake linings is 3.7 and 3.9 mm respectively. Prior to the squeal test, bedding-in process for the drum brake is performed for about one and a half hours and at an even speed of 6 rad/s with a brake-line pressure between 0.3 to 1.0 MPa. Both the speed and the brake line pressure are maintained until the temperature between the drum and shoes reaches 250° C. Having reached this temperature, the drum brake assembly becomes very hot and needs to be cooled down naturally at temperature of 50° C. After the bedding-in process, the brake assembly is more likely to generate a squeal. The squeal data is recorded only when the sound pressure level reaches 70 dB or higher [2].

4. Results and Discussions

4.1. Modal Analysis

Table 1 shows the natural frequencies (<10 kHz) of the brake lining (new and worn), the backing plate and the drum from the modal tests. The drum exhibits four natural frequencies; where as only three natural frequencies are observed for the braking plate and brake lining. The dynamic properties of the lining in terms of modal frequency have been slightly changed for the variations in the lining thickness. The first natural frequency of the new lining is 2% higher than that of the worn lining. However, the second and third natural frequencies of the worn lining are 9% and 3% higher those of the new lining. Some of the natural frequencies between the drum brake components are relatively closer especially the modes 2 and 3. Hence, it is expected that a squeal may be generated at these two modes.

Table 1: Natural frequencies of the brake components

Components / Modes	Natural frequency (kHz)			
	I	II	III	IV
3.7 mm lining (Worn)	2.26	4.41	8.28	-
3.9 mm lining (New)	2.31	4.02	8.08	-
Drum	1.5	4.93	8.06	9.53
Backing plate	2.07	4.07	8.4	-

4.2. Squeal Events

For the new and worn lining, squeal tests have been performed for more than 100 braking applications. A comparison of the measured response from the accelerometer and that of the micro-phone is shown in Fig. 3. The responses correlate well. Fig. 4 shows the sound pressure level for various squeal occurrences. It is found that the new lining has squeals approximately at 4.6 kHz, between 7 to 8.6 kHz, and between 9 to 9.5 kHz. For the worn lining, squeal occurs between 3.9 to 4.5 kHz and between 7 to 8.3 kHz.

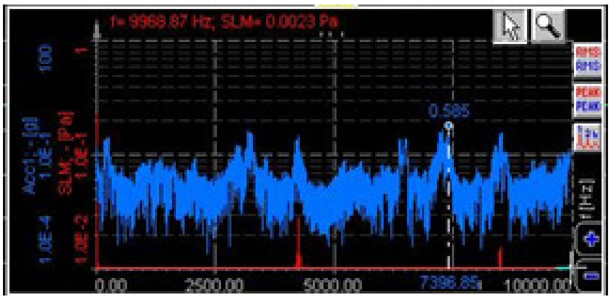


Fig. 3: Comparison of the responses from the accelerometer (in blue) and micro-phone (in red)

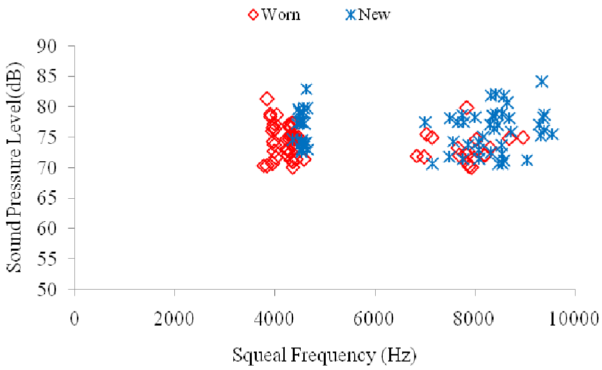


Fig. 4: Squeal occurrence recorded in the brake dynamometer

The distributions of the squeal occurrences are slightly different for the new and the worn lining. The new lining seems to generate additional squeals at higher frequency above 9 kHz. This might be due to the differences in the natural frequencies as given in Table 1. The worn lining tends to generate less squeal, i.e., 58 times compared to 65 times for the new lining as given in Table 2. For the new lining, there are 8 squeal events measured at a frequency of 9 to 9.5.

Table 2: Number of squeal occurrences

Frequency range (kHz)	Number of squeals	
	New	Worn
3.9 – 4.6	23	41
7 – 8.6	34	17
9 – 9.5	8	-

The first two squeal clusters, as shown in Table 2, are almost identical to the second and third modes. Thus, it can be stated that squeals at these modes are generated due to a modal coupling mechanism. This is agreed with previous works of Kung et al [16] and Ganguly et al [18]. The third squeal cluster frequency range of 9 to 9.5 kHz is close to the fourth mode of the drum. However, the question arises on why a squeal does not appear for the worn lining since the same drum is used for both the linings. Further investigations are required to find the reasons for having a squeal in this higher frequency range for the new lining.

A squeal can potentially be generated due to a negative friction-velocity slope that is commonly described as negative damping. Fig. 5 shows the response of the friction and drum speed for the squeal events. The new and worn linings have tendencies to produce a positive and negative damping respectively. Thus, it is understood that due to these characteristics, squeals are occurred for the worn lining. Theoretically, if a friction-induced vibration system has a positive damping then the system becomes stable and vice-versa. Hence, the new lining should not generate a squeal. However, tests from the dynamometer indicate differently, i.e., more squeals are generated for the new linings than those of the worn one. From this work, it can be concluded that a negative damping is not a sole mechanism to generate the squeals. This finding agrees with the previous work of Eriksson [2].

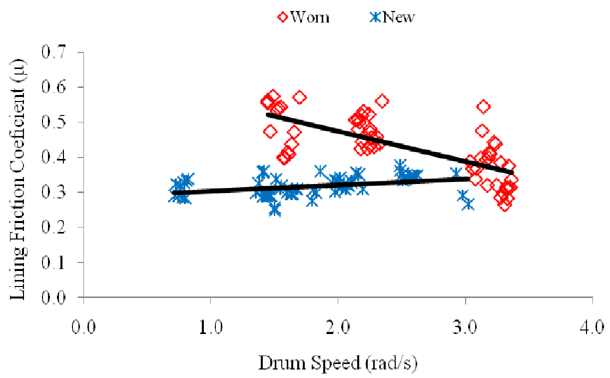


Fig. 5: Characteristics of friction against drum speed

The following Sections detail the squeal behavior against brake operating conditions such as drum temperature, hydraulic pressure, drum speed, friction coefficient, and humidity for the new and worn linings. Fig. 6 shows a comparison of the temperature following the braking application during squeal tests. The new lining has generated squeals at much lower temperature than the worn lining. The new lining starts to generate squeal at a temperature of 50^o C whilst the worn lining does only at temperature of 100^o C. A highest temperature of 250^o C is observed for the worn lining.

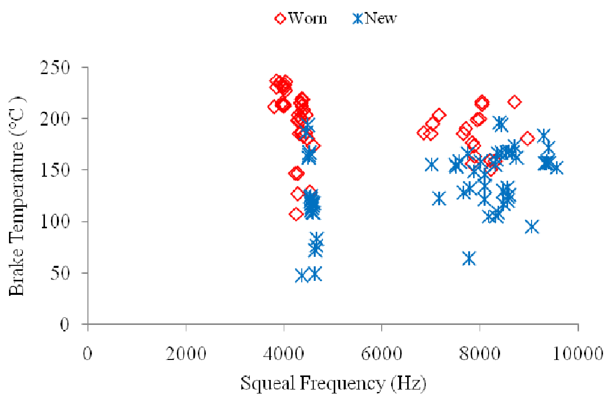


Fig. 6: Distribution of drum temperature against squeal frequency

Fig. 7 shows that both the worn and new linings have almost identical hydraulic pressure distribution at a squeal frequency of 4 kHz. The pressure is scattered for the frequency between 7 to 9.5 kHz. The new lining starts to generate squeal at 2 MPa whilst the worn lining does at 3.0 MPa. The new lining tends to produce a squeal at a lower speed of 0.5 rad/s than the worn lining (1.5 rad/s) as shown in Fig. 8. An highest speed of 3.5 rad/s is recorded for the squeals in the worn lining.

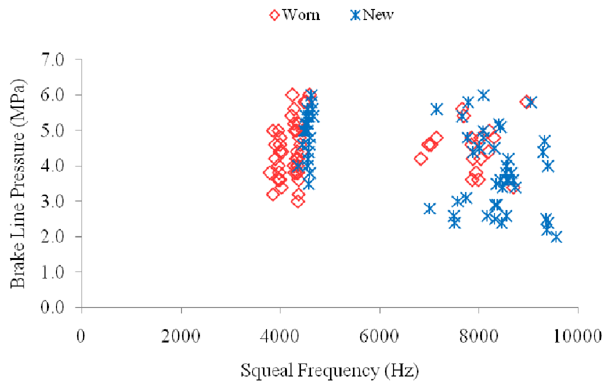


Fig. 7: Distribution of hydraulic pressure against squeal frequency

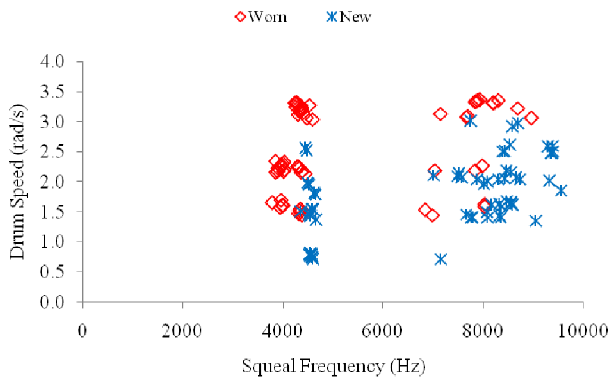


Fig. 8: Distribution of drum speed against squeal frequency

Distribution of the friction coefficient against the squeal frequency is shown in Fig. 9. It is noticed that the friction coefficient for the new lining is less than that of the worn lining. The friction coefficient of the new lining distributes closely at 0.3. For the worn lining, the friction coefficient is scattered between 0.28 and 0.6. However, it seems that most of the squeal events are generated almost at the same relative humidity between 60 to 80 % as shown in Fig. 10. It shows that humidity is significant in generating a squeal.

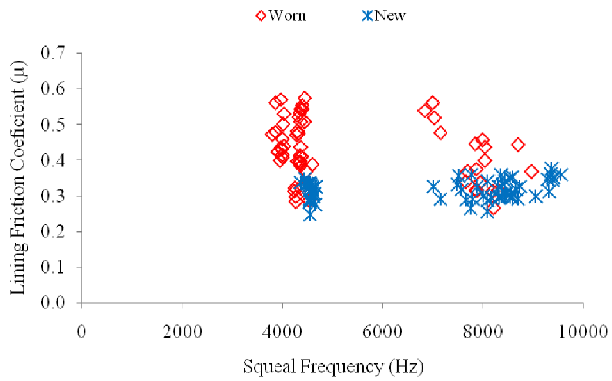


Fig. 9: Distribution of friction coefficient against squeal frequency

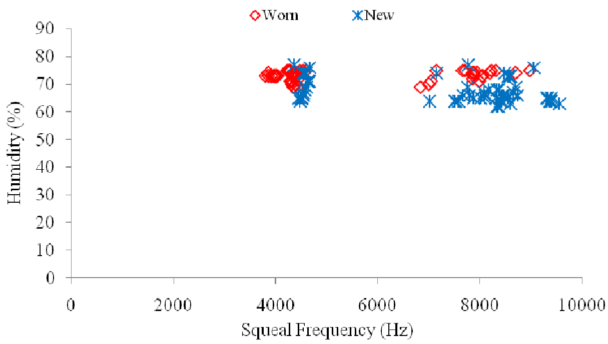


Fig. 10: Distribution of humidity against squeal frequency

5. Conclusions

This paper investigates the effects of brake linings on the squeal occurrence. Two sets of the brake lining are used in this work, i.e., a new lining with 3.9 mm thickness and a worn lining with 3.7 mm thickness. First, modal testing is conducted to obtain modal frequencies of drum brake components including the new and worn linings. Later, squeal tests are carried out with more than 100 braking applications for each lining. From the results of modal and squeal tests the following conclusions are made:

- There are slight differences in the natural frequency between the new and worn lining.
- The natural frequencies of the linings, drum and backing plate are almost identical for the second and third modes. This behaviour might result in a modal coupling effect that subsequently generates a squeal at these frequencies. Indeed, the squeal tests have shown that there are squeals in these two modes. Hence, it can be said that the squeal is potentially due to a modal coupling mechanism.
- The new lining exhibits a positive damping whilst the worn lining shows a negative damping. Theoretically, the new lining supposedly to show no squeal but the squeal test results differ this. This indicates that the negative damping mechanism is not the main factor in generating squeal, primarily in the current work.
- The distribution of temperature, pressure, friction coefficient, drum speed and humidity against the squeal frequency are slightly different for both the worn and new linings. These parameters for the worn lining are mostly higher than those of the new lining. It is also observed that both the linings generate a squeal only if the humidity falls between 60 to 80%.

It should also be noted that current findings are based on only two different sets of lining (thickness of 3.7 and 3.9 mm) and more sets of lining should be used to verify these findings. This is under authors' intention for future work.

ACKNOWLEDGEMENTS:

The authors would like to thank the Malaysian Ministry of Science, Technology and Innovation (MOSTI) and Universiti Teknologi Malaysia (UTM) for their continuous support in the research work. This research was fully supported by a research grant (No.: 03-01-06-SF0427).

REFERENCES:

[1] N.M. Kinkaid, O.M. O'Reilly, and P. Papadopolous. 2003. Review of automotive disc brake squeal, *J. Sound & Vibration*, 267, 105-166.

[2] M. Eriksson. 2000. *Friction and Contact Phenomena of Disc Brakes Related to Squeal*, Ph.D. Thesis, Acta Universitatis Upsaliensis Uppsala.

[3] F. Chen. 2009. Automotive disc brake squeal: An overview, *Int. J. Vehicle Design*, 51(1/2), 39-72.

[4] H.R. Mills. 1939. *Brake Squeal*, The Institution of Automobile Engineers, Report No. 9162B.

[5] R.A. Fosberry and Z. Holubecki. 1955. *An Investigation of the Cause and Nature of Brake Squeal*, Report No.1955/2, The Motor Industry Research Association.

[6] Y. Yuan. 1995. A study of the effects of negative friction-speed slope on brake squeal, *ASME Des. Eng. Tech. Conf.*, 84(1), 1153-1162.

[7] M.R. North. 1972. *Disc Brake Squeal – A Theoretical Model*, Tech. Report 1972/5, The Motor Industry Research Association.

[8] G.D. Liles. 1989. Analysis of disc brake squeal using finite element methods, *SAE Paper 891150.*

[9] A. Akay, J. Wickert, and Z. Xu. 2000. *Investigating Mode Lock-in and Friction Interface*, Final Research Report, Carnegie Mellon University.

[10] M. Eriksson, F. Bergman, and S. Jacobson. 1999. Surface characterization of brake pads after running under silent and squealing conditions, *Wear*, 232 (2), 163-167.

[11] H.A. Sherif. 2004. Investigation on effect of surface topography of disc/pad assembly on squeal generation, *Wear*, 257 (7/8), 687-695.

[12] J.D. Fieldhouse. 2000. A study of the interface pressure distribution between pad and rotor, the coefficient of friction and caliper mounting geometry with regards to brake noise, *Proc. Int. Conf. Automotive Braking*, ISBN: 1-86058-261-3, 3-18.

[13] J.D. Fieldhouse, N. Ashraf, C. Talbot, T. Pasquet, P. Franck, and G. Rejdych. 2006. Measurement of the dynamic center pressure of a brake pad during a braking operation, *SAE Paper 2006-01-3208.*

[14] J.D. Fieldhouse, N. Ashraf, and C. Talbot. 2008. The measurement and analysis of the pad/disc interface dynamic center of pressure and its influence on brake noise, *SAE Paper 2008-01-0826.*

[15] A. Felske, G. Hoppe, and Matthai. 1980. A study on drum brake noise of heavy duty vehicle. SAE Paper 800221.

[16] S.W. Kung, G. Stelzer, and A.K. Smith. 2004. A study of low frequency drum brake squeal, *SAE Paper 2004-01-2787.*

[17] http://www.dewetron.com/int/?no_cache=1 (visited in June 2010).

[18] S. Ganguly, H. Tong, G. Dudley, F. Connolly, and S. Hoff. 2007. Eliminating drum brake squeal by a damped iron drum assembly, *SAE Paper 2007-01-0592.*