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MODELING OF BRAKE SHOE IN DRUM BRAKE SQUEAL

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ABSTRACT

In general, drum brake squeal is excited by friction induced vibration at the drum-shoe interface. The instability of the vibration depends on various parameter especially friction coefficient. This paper shows the development of minimal model of a drum brake squeal under binary flutter instability which is caused by the velocity independent friction coefficient. The vibration of the brake shoes is considered as the primary unstable component which contributed to the drum brake system squeal. Experimental work is carried out to indentify the value of parameters of this model and also verified the critical value of unstable vibration for squeal. The binary flutter instability occurs when two initially separated and different natural frequencies of the system coupled at a common frequency to produce one unstable mode. The coupling of the modes in this model is between the first torsion mode and the second torsion mode of the leading brake shoe. The effect of the friction coefficient, damping coefficient and the location of centre of contact pressure are investigated.

Keywords: Friction, vibration, binary flutter, instability, squeal

1. INTRODUCTION

The occurrences of automotive brake noise has been studied since the 1920s (Lee, Yoo et al. 2001) and yet none was able to completely eliminate brake noises although large improvement has been made. Brake noise is annoying to vehicle passengers and greatly influence the customers comfort level in the vehicle. According to manufacturers of brake pad materials, up to 50% of their engineering budget was spent on noise, vibration and harshness issues (Abendroth and Wernitz 2000).

Drum brake squeal is defined as a self-excited friction induced vibration with frequency exceeding 1000Hz. Typically, drum brake squeal is excited at low speed and high friction condition when the brake shoes are pressed against the rotating drum (Hulten 1995). The occurrence of the brake squeal is sensitive to the change of friction coefficient, speed, location of centre of pressure, stiffness, damping coefficient and also the geometry of brake assembly (Ouyang, Nack et al. 2005).

Based on the previous studies on stability of brake squeal, there are four possible mechanisms to excite brake squeal. These are binary flutter instability of mode coupling (Hoffmann, Fischer et al. 2002; Teoh and Ripin 2010), negative friction-velocity characteristic that cause negative damping (Shin, Brennan et al. 2002), sprag-slip (Sinou, Thouverez et al. 2003) and follower of friction force (Popp, Rudolph et al. 2002). Some researchers studied the effect on brake squeal with more than one mechanism. (Kang, Krousgrill et al. 2009; Teoh and Ripin 2011).

The critical friction coefficient to excite binary flutter instability of drum brake squeal can also be predicted by using reduced-order characteristic equation to compute the complex eigenvalue of the system (Huang, Krousgrill et al. 2009). A drum brake shoe with non-uniform cross section is used to study the effective method to reduce drum brake squeal by partially changing the contact area and the shapes of the shoes (Lee, Yoo et al. 2001). The results showed that the occurrence of drum brake squeal is highly dependence on the dynamics characteristics of the drum brake system. The slightly modification of the cross-section area of the brake shoe able to quench the drum brake squeal.

Although a lot of publications of the analysis drum brake squeal can be found, none has verified the model of mode coupling mechanism experimentally. In this paper, experimentally verified model of the drum brake shoe is developed to study the characteristic of mode coupling mechanism. Besides, the numerical and experimental results are compared to provide better association with the model of the real drum brake system.

2. METHODOLOGY

2.1 Experimental modal analysis

Experimental modal analysis is carried out to determine the mode shapes and natural frequencies of the brake shoe. Twelve accelerometers (Dytran, 3224A2) are installed on each brake shoe. The brake shoe is knocked using impact hammer (Kistler, 9724A5000) to produced input force in order to construct frequency response function (FRF) using LMS Data acquisition system. Besides, the Operating Deflection Shape (ODS) analysis is carried out on brake shoe during squealing condition and the squeal noise frequency recorded using microphone.

According to the modal analysis results shown in Figure 1, the first two natural modes of the brake shoe are the first and second torsion mode of the brake shoe with frequency of 1231Hz and 2420Hz. These modes have the damping ratio of 0.52% and 0.67% respectively.

Mode 1 **Mode 2**

ω_{n1}=1231 Hz ω_{n2}=2420 Hz
ζ=0.52% ζ=0.67%

Figure 1 The first two modes of the brake shoe in free-free condition

Spectrum analysis is carried out to investigate the excited frequency and mode shape of the pair of brake shoes as shown in Figure 2. By comparing the squeal frequency in Figure 3(a) and the vibration frequency of the leading brake shoe in Figure 3(b), the squeal frequency is identical with the excited frequency of the leading brake shoe. Both have the highest amplitude at frequency of 1850Hz. Thus, the leading brake shoe is believed to have the greatest contribution to the occurrences of brake squeal. Based on the ODS result shown in Figure 4, the excited mode of leading brake shoe during squealing is a torsion mode with the frequency of 1850Hz. This mode has the frequency between first and second torsion mode as in Figure 1. Thus, we suggested that the occurrence of drum brake squeal is excited by mode coupling between the first and the second torsion mode of the leading shoe.

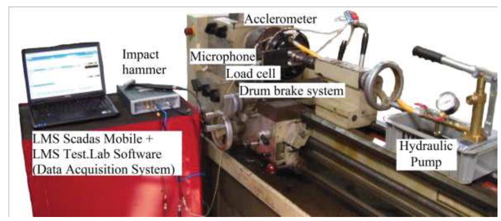
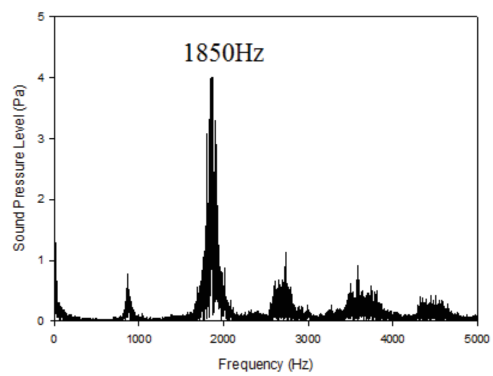
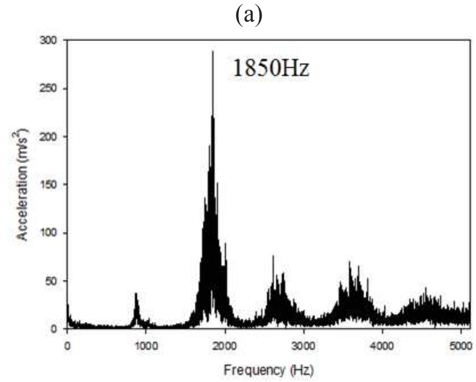


Figure 2 Experiment setup for spectrum analysis



(a)



(b)

Figure 3 Fast Fourier Transform (FFT) during squealing. (a) Sound Pressure Level of squeal (b) Vibration level of leading shoe.

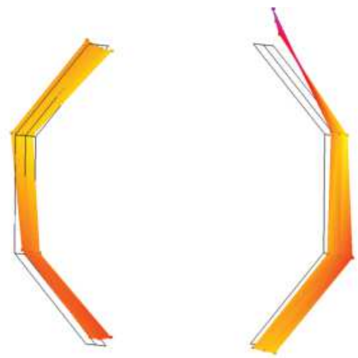


Figure 4 ODS of the trailing brake shoe during squealing at 1850Hz

2.2 Modeling of drum brake shoe

(a)

(b)

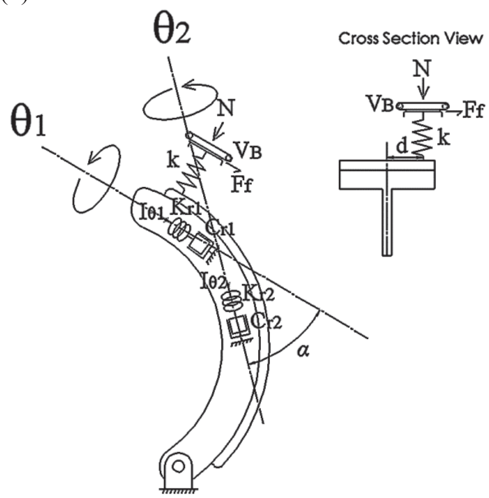


Figure 5 (a) The leading drum brake shoe; (b) Model of leading drum brake shoe

The two degree-of freedom model shown in Figure 5(b) is modeled based on the leading drum brake shoe as shown in Figure 5(a). This model consist of rotation about θ_1-axis and rotation about θ_2-axis. The rotation about each axis represent first and second torsion mode of the drum brake shoe. This model can be easily associated with the real drum brake system where the relationship between coupling mode can be shown and the effect of friction force is included in the system. The angle between two axis is denoted by α. The moment of inertia for each modes are represented as $I_{\theta 1}$ and $I_{\theta 2}$ due to the different moment of inertia. Meanwhile, the torsion stiffness and damping coefficient of each modes are represented as K_{r1}, K_{r2} and C_{r1} C_{r2}. The contact stiffness of the shoe on the rotating drum is denoted by k with sliding velocity of V_B. The contact force acted on brake shoe is denoted by N and the friction force F_f. The location of centre of contact pressure is denoted by d.

The equation of motion of the model is derived as in Eq. (1-2). Meanwhile, the friction force which is acting on the brake shoe is derived as in Eq. (3). Transient analysis is carried out to investigated the stability of the system with different parameters using Dormand-Prince method (Dormand and Prince 1980). The values of torsion spring stiffness, torsion damping coefficient, contact stiffness sliding velocity and contact force are extracted from experiment results. The moment of inertia for each axis is generated from Finite Element model of brake shoe. Meanwhile, the other parameters are subjected to change in order to investigate their effect on system stability.

$$I_1\ddot{\theta}_1 + C_{r1}(\dot{\theta}_1 - \dot{\theta}_2\cos\alpha) + K_{r1}(\theta_1 - \theta_2\cos\alpha) = Nd \quad (1)$$

$$I_2\ddot{\theta}_2 - C_{r1}(\dot{\theta}_1 - \dot{\theta}_2\cos\alpha) - K_{r1}(\theta_1 - \theta_2\cos\alpha) + C_{r2}\dot{\theta}_2 + K_{r2}\theta_2 = Nd\cos\alpha - F_f(d\sin\alpha) \quad (2)$$

Where friction force,

$$F_f = \mu \times N = \mu \times kd[\sin(\theta_1 - \theta_2\cos\alpha)] \quad (3)$$

3. RESULT AND DISCUSSION

3.1 Influence of friction coefficient to the occurrence of drum brake squeal

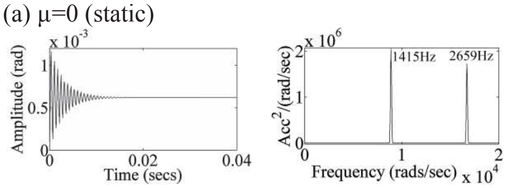
The friction coefficient is known to has the most influence on mode coupling mechanism. Therefore the transient analysis carried out for different value of friction coefficient, μ ranged from 0 to 0.5, while the location of centre of pressure, d remain unchanged at 0.010m. Since the brake squeal is a self-excited friction induced vibration, the growing oscillations on the graph rotation-time represent unstable or squealing condition. In the other hand, the damped oscillations represent stable or non-squealing condition.

Based on time history results as shown in Figure 6(a)-(c), the oscillations damped with time which represented the stable non-squealing condition. The damping rate of these oscillations decreases when the friction coefficient increases from 0 to 0.40. When the friction coefficient is larger than 0.45, the system vibration increases with time which represented the unstable or squealing condition as shown in Figure 6(d). The limit cycle of the unstable torsional vibration is 0.26rad when friction coefficient, μ is 0.45. This self-excited vibration able to overcome the system damping and maintain the limit cycle if the condition remain unchanged.

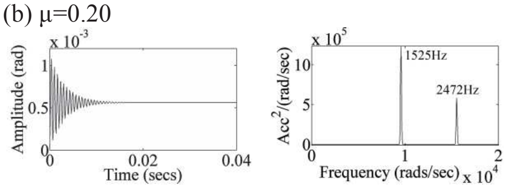
The graph of power spectral density in Figure 6(a) shows that the brake shoe have two modes during contact and static condition with frequency of 1415Hz and 2659Hz respectively. These modes represent the first and second torsional mode of the brake shoe. The power spectral density results in Figure 6(a)-(d) show that the two initially separated modes come closer to each other when friction coefficient increases until they coalesced to produce a single high magnitude unstable mode at μ=0.45 with frequency of 1767Hz. The unstable mode has a magnitude 1.0×10^5 time greater than the stable mode.

Time History **Power Spectral Density**

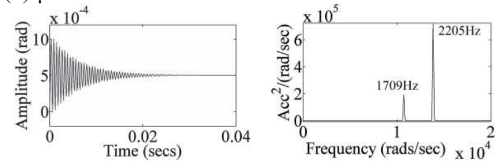
(a) μ=0 (static)



(b) μ=0.20



(c) μ=0.40



(d) μ=0.45

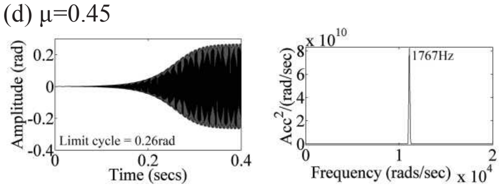


Figure 6 Time History and Power Spectral Density results for different value of friction coefficient, μ.

3.2 Influence of location of centre of pressure to the occurrence of drum brake squeal

In addition to the friction coefficient, the location of centre of pressure also influences the mode coupling instability of drum brake squeal. Since the brake shoe's lining of the typical passenger car is 0.03m in width (0.015m from CG), the stability analysis is carried out for location of centre of pressure $0 < d < 0.015$m from CG. The fiction coefficient, μ is set as 0.50 for this analysis.

The time result in Figure 7(a) shows that the vibration of the brake shoe remain stable (damped oscillation) for d=0.005m although the friction coefficient is increased above it critical value. The power spectral density showed the two modes with frequencies of 1478Hz and 2550Hz. These frequencies are almost identical as the modes 1415Hz and 2659Hz at static condition (Figure 6(a)). Although these modes come closer to each other when μ increases to 0.50, but their affect is still insufficient to couple them together since they are still far away from each other. This result indicated that the coupling of torsional modes is impossible if the location of centre of contact pressure is near to the centre of gravity (CG) of the shoe.

When the location of centre of pressure is 0.010m from CG as shown in Figure 7(b), the unstable vibration is excited at frequency of 1767Hz with limit cycle of 0.90rad. This frequency is identical to squeal frequency as in Figure 6(d) which indicated the unstable vibration is excited by mode coupling mechanism. However, when the value of d increase to 0.015m as in Figure 7(c), the unstable vibration detected with higher limit cycle (2.2rad) at frequency of 1760Hz and 2733Hz. The sudden increase of the limit cycle of the unstable vibration is caused by another mode which is excited at frequency of 2733Hz which is different from the coupled mode of 1760Hz. Due to the higher torque produced by the contact force at *d*=0.015m, the frictional contact produced another unstable mode which at the same time is superimposed on the coupled mode. Fortunately, the mode 2733Hz is very difficult to produce in real condition since the contact will have to be only at the edge of the lining of the brake shoe.

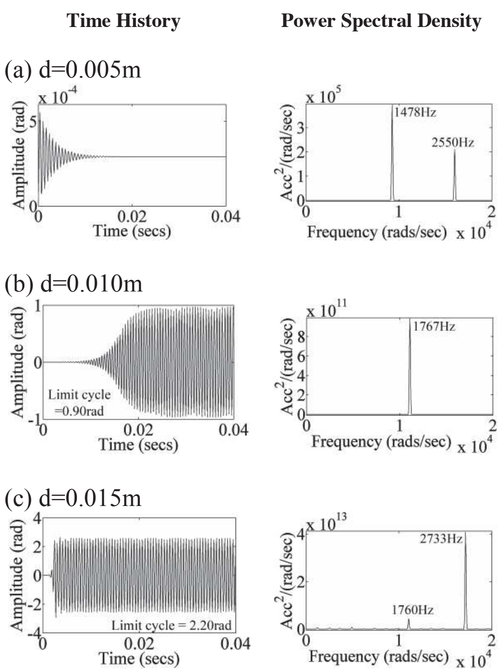


Figure 7 Time History and Power Spectral Density results for different location of centre of contact pressure at μ=0.50.

3.3 Comparison of experiment and simulation results.

Table 1 listed the frequencies of mode of the experiment and simulation. The first and second torsion mode of the brake shoe generated from the model in Figure 6(a) is 1415Hz and 2659Hz respectively. These modes are 14.9% and 9.9% different from the experimental modal analysis. The natural frequencies from the model are slightly higher than the experiment value, since the experimental modal analysis was carried out in free-free condition while the model was simulated in contact condition. Thus, the gap between them is contributed by the contact stiffness between the drum and the brake shoe. The squeal frequency recorded from both methods are different by only 4.5%. Thus, this model is proven to have the characteristics of actual mechanism of drum brake squeal.

Table 1 Comparison of experiment and simulation results

	Experiment	Simulation	Error (%)
1st torsion mode	1231 Hz	1415 Hz	14.9
2nd torsion mode	2420 Hz	2659 Hz	9.9
Squeal mode	1850 Hz	1767 Hz	4.5

4. CONCLUSION

a) The drum brake squeal is modelled after the instability of the brake shoe and excited by the mode coupling mechanism between first torsion mode (1231Hz) and the second torsion mode (2420 Hz) of the leading brake shoe with frequency of 1850Hz.
b) The mode coupling mechanism between first and second torsion mode is excited when the value of friction coefficient is higher than 0.45. This indicated that the drum brake squeal is detected only when the friction coefficient is higher than 0.45.
c) The location of centre of contact pressure influence the system stability if it is away from centre of gravity of the shoe.

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NOMENCLATURE

C_{r1}	Torsional damping coefficient respect to θ_1-axis, 0.24 kg.m^2.s^{-1}.rad^{-1}
C_{r2}	Torsional damping coefficient respect to θ_2-axis, 0.39 kg.m^2.s^{-1}.rad^{-1}
d	Location of centre of contact pressure from C.G, 0.005m to 0.015m
F_f	Friction force between rotating drum and leading brake shoe, N
I_1	Moment of inertia respect to θ_1-axis, 2.00×10^{-4} $kg.m^2$
I_2	Moment of inertia respect to θ_2-axis, 3.13×10^{-4} $kg.m^2$
k	Contact stiffness between the rotating drum and leading brake shoe, $2.00\times10^{8} N.m^{-1}$
K_{r1}	Torsional stiffness respect to θ_1-axis, 2.50×10^{4} $N.m.rad^{-1}$
K_{r2}	Torsional stiffness respect to θ_2-axis, 5.50×10^{4} $N.m.rad^{-1}$
N	Contact force acting on leading brake shoe, 500 N
V_b	Sliding velocity between leading brake shoe and rotating drum, 1.0 $m.s^{-1}$
α	Angle between θ_1-axis and θ_2-axis,45°
θ_1	Angular displacement respect to θ_1-axis, rad
θ_2	Angular displacement respect to θ_2-axis, rad
$\dot{\theta}_1$	Angular velocity respect to θ_1-axis, $rad.s^{-1}$
$\dot{\theta}_2$	Angular velocity respect to θ_2-axis, $rad.s^{-1}$
$\ddot{\theta}_1$	Angular acceleration respect to θ_1-axis, $rad.s^{-2}$
$\ddot{\theta}_2$	Angular acceleration respect to θ_2-axis, $rad.s^{-2}$
μ	Coefficient of friction, 0 to 0.50