NUMERICAL ANALYSIS OF THE EFFECTIVENESS OF BRAKE INSULATOR IN DECREASING THE BRAKE SQUEAL NOISE

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ABSTRACT

Brake functions when two different materials are in contact to reduce a motion. Due to surface irregularity, this contact at high revolution and contact force produces irritating noise called brake squeal. This paper presents the study of introducing brake insulator into the brake assembly in order to reduce the noise. Different configurations of insulators are used in the Finite Element Analysis (FEA). The effectiveness of the brake insulator is analyzed using different type of materials. The finite element model of the brake is developed based on actual drum brake dimensions. FEA is used for modal analysis to predict the modal frequencies and mode shapes. Various friction coefficients, wheel speeds and brake forces are considered in the analysis. The squeal is shown by positive real part of the baseline graph. The accompanied slip rate in the baseline model of the insulator increases the brake squeal noise significantly.

KEYWORDS: *Brake insulator; Brake squeal; Finite element analysis; Modal analysis; Brake insulator damping*

1.0 INTRODUCTION

Brake is one of the important components in transportation. In advancement of vehicle, the improvement of brake has focused on increasing brake power and reliability. However, purification of vehicle acoustic and comfort has greatly increased the benefaction of brake noise to this aesthetic and environmental concern. Brake noise is an irritant to the users. Most of them believed that brake noise is symptomatic of a defective brake and this problem will lead them to claim a warranty from the company that produced the vehicles although the brake functioning as well as it had been designed. Brake noise or generally called brake squealing has no precise definition. Thus, in brake part design and manufacture, noise generation and suppression have become conspicuous consideration. Despite, as noted by previous researchers (Abendroth & Wernitz, 2000), (Hamid et al., 2014), many makers of materials for brake pads spend up to 50 % of their engineering budgets on noise, vibration and harshness

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issue. There are few terminologies for brake noise such as squeal, groan, chatter, judder, moan, hum and squeak. However, the terminology that often be used is squeal. The phenomenon of squeal is probably the most common, disturbing to users and environment, and its cost the manufacturer in term of warranty. There are no precise definitions for brake squeal, but it is frequently agreed that squeal occurred at frequency above 1000 Hz (Lazim et al., 2014).

Brake squeal is a phenomenon of dynamic instability that occurs at one or more of the natural frequencies of the brake system. The research on this field had started since 1950’s where the busses was banned from using the road because producing a squealing or noisy sound from the brake (Crolla & Lang, 1991).

There are a few approaches used in predicting the probability of the squeal occurrence which are theoretical, experimental and Finite Element (FE) approaches. Besides, there are several methods also proposed to suppress or reduce the squeal occurrence which are; structural modifications, active control and adding damper (Kinkaid et al., 2003). From the three methods, adding damper is the most efficient method and it may be applied by changing the material with high damping material or by adding insulator to the pad or shoe, which depend on what type of brake that will be used.

Drum brake produces significant amount of noise compared to disc brake. Motorcycle brake which is using the drum brake type is used for this study. Brake is one of the most important things that need to be considered when producing a vehicle. The squealing sound that produced from the brake not only contributed to the noise pollution, but also make the users are not comfortably used the vehicle. They thought that the brake might be broken down and the vehicle are not safely be used which will lead them to claim a warranty from the company that produced the vehicle. Frictional contact is one of the important sources of squeal noise in braking systems. A merging of two real modes in an unstable complex one as the friction coefficient increases describing the general scenario for squeal occurrence. This phenomena is called modal coalescence, which the example can be seen in review of (Giannini, 2008). Many researchers have done the experimental works on laboratory or industrial brake setups. For example, Nakata et al. (2001) shows that squeal involved a mode of the pads coupled with a mode of the disc. Even if this phenomena is likely to be known, numerical modal analyses remain insufficiently predictive to improve the design of brake systems (Hervé et al., 2008).

Despite, in FEM models contrary to lumped models, important phenomena such as gyroscopic effects, circulatory forces and damping behavior are generally counted (Flint et al., 2010). All those phenomena will result in stabilize or destabilize the system because they affect the modal eigenvalue. Among the methods, increasing the system damping has been shown to be very effective to control squeal noise (Kappagantu, 2008). A common way to do that is by applying the shims on the pad back plates (Liu et al., 2007).

In the previous research, great efficiency inside a vehicle subjected to road tests (Festjens et al., 2012). Several brake systems components have a high damping behavior such as brake shims. Because of their glue and rubber layers those components have viscoelastic behavior (Triches et al., 2004). The use of damping material glued on the back-plate of the pads may reduce squeal propensity (Fritz et al., 2007). While the general idea among the researchers is that the damping increase the stability of the system, Junior et al., (2008) in his paper shows that, if damping is equally distributed on the two modes involved in the mode coupling phenomenon, the only effect is a shift of the curves towards the negative real parts. If damping is spread non-equally over the two modes, a shifting and a smoothing effect can be seen on coalescence curves. If the ratio in damping between the two modes is sufficient, an increase in damping tends to make the brake unstable for a lower value of the friction coefficient (Kappagantu & Denys, 2008).

According to Nakata et al., (2001) the insulator is a laminated plate attached to the pad back plate, and its application leads to an increase in the pad stiffness. However, the pad stiffness is very important to determine the stability characteristics of the brake system. Therefore, the effect of the additional damping provided by the insulator must be greater that the effect of the stiffness modification. This condition, however, is a variable that depends on the brake system analyzed.

Insulators usually are a multi-layer construction of steel, visco-elastic and rubber materials that are positioned between the backing plate and piston fingers of the braking system. As the insulator vibrates along with the backing plate, the different layers tend to bend and unbend resulting in the shearing and extension/contraction of the visco-elastic material. This causes vibration energy to dissipate in the form of heat (Flint et al., 2010).

There are a few squeal mechanisms to describe the squeal generation. The most common mechanisms are friction characteristic, sprag-slip model and modal friction-induced vibration (mode coupling). Friction-induced vibration is the most popular mechanism and widely used among the researchers to predict the squeal generation. This mechanism assumed that the coefficient of friction varied when the normal force of the friction is varied due to geometric coupling, then under some condition a pair of natural frequency may merge then squeal will be occurs.

As it’s mentioned before, the squeal is excited more at low than at high speeds and squeals occur only over limited ranges of pressures and are most common at temperatures below 100 oC. From the literature also, it is clear that there are three methods to investigate the squeal generation which are experimental methods, analytical methods or numerical methods. Experimental methods reflect real situation of the system but it takes time and somehow expensive. Analytical methods does not representing an actual system and is limited to one particular frequency of squeal. Numerical methods (finite element methods) provide a reasonable results and similar to the experimental results with short time.

The complex eigenvalue analysis noted, most common method to evaluate system stability. That attributed to, the complex eigenvalue provide fast and reliable results for the brake system as compared with the other methods (transient solution, Routh criterion). The information about the stability of the system facilitates solve the problem with a suitable solution. In this paper, in order to reduce the brake noise, several type of brake insulators are introduced in the assembly of drum brake with different thickness. Using FEA software ABAQUS, the effectiveness of these insulators in reducing the noise is numerically analyzed.

2.0 METHODOLOGY

The squeal that generate attributed to the system instability. In the previous section, several methods for analyzing the stability of the system have been introduced. However, in this section will focus more on the complex eigenvalue analysis method. Complex eigenvalue analysis can determine which system vibration modes are unstable. Addition of damping material in the brake system will be used to stabilize unstable mode (Triches et al., 2004). The FE software used to predict the squeal generations of a drum brake system by using complex eigenvalue analysis is ABAQUS. A 3D solid model is generated based on an actual rear drum brake assembly. Previous design and analysis using commercial software are practiced (Abdullah et al., 2014), (Abdullah et al., 2013a), (Abdullah et al., 2013b), (Abdullah et al., 2012). The drum brake assembly is taken from Yamaha LC. The brake assembly consists of a drum, brake shoes, back plate, beam, springs and braking lever. However in this project, only related parts are modelled for stability analysis, i.e., the drum and the shoes.

The drum is the most important part in the brake system. The real part has two bearings and bays in between, while it's simplified in the FE model and built as one part. Shoe in the real system consist of two parts the shoe body and the friction material and they attached together by adhesive. In the FE model it's also consisted of two parts attached together. There are two springs in the brake system they will not modelled, just their effect will add between the two shoes. The stiffness used is 10 kN/m for one side, and 20 kN/m for another side following the design in (Hamid et al., 2014), (Lazim et al., 2014). Table 1 shows the FE model parts with their element specifications and mechanical properties.

Table 1. FE model parts with their specifications

	Drum	**Shoe & lining**
Component		
Type of element	Tetrahedral C3D4	Tetrahedral C3D4
No. of elements	93499	44106
No. of nodes	14796	2208
Density (ρ) kg/m^3	3400	2700 , 2100
Young's Modulus (E) GPa	70	71 , 8
Poisson's Ratio (v)	0.3	0.3 , 0.3

The FE modal analysis can be done by run one step in ABAQUS with free-free boundary condition setting the interested frequency range (1 to 10 kHz). There are several factors determine the natural frequency and the mode shape of the FE model, changing these factors may help to determine the squeal occurrences in the FE model result. First is the geometry, since the geometry was built based on the real component it will be not that suitable to change the geometry as a way to reduce the error. Next is the material properties that have been assigned to the FE model developed. There are three parameters we are dealing with, which are the modulus of elasticity, density and Poisson's ratio. Tuning these three parameters is the method used in this project to validate the model, this method called model updating.

In order to determine the stability of the drum brake system complex eigenvalue available in ABAQUS software will be used. The positive real parts of the complex eigenvalues demonstrate the degree of instability of the drum brake system which means probability of squeal occurrence. The stability of the drum brake system will determine without insulator first to check the unstable frequencies, and then an insulator will attach to the shoe to evaluate the effectiveness of the insulator on the stability of the brake system. In order to observe the complex eigenvalue analysis using ABAQUS, four main steps are recommended (Festjens et al., 2012). They are given as follows:

i. Nonlinear static analysis for applying brake-line pressure.
ii. Nonlinear static analysis to impose rotational speed on the drum.
iii. Normal mode analysis to extract natural frequency of undamped system.
iv. Complex eigenvalue analysis that incorporates the effect of friction coupling.

As it is mentioned in the literature the complex eigenvalue problem can be given in the following form:

$$([M]s^2 + [C]s + [K])\{\emptyset\} = \{0\} \tag{1}$$

Where: *M*, *C* and *K* are mass, damping and stiffness matrices, respectively. For underdamped systems the eigenvalues always occur in complex conjugate pairs. For a particular mode the eigenvalue pair is:

$$s_{i1,2} = \sigma_i \pm j\omega_i \tag{2}$$

Where σ_i is the real part and ***w***i is the imaginary part for the ***i***th mode. The motion for each mode can be described in terms of the complex conjugate eigenvalue and eigenvector:

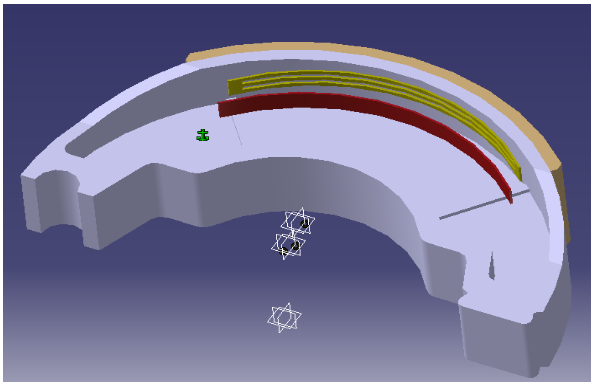
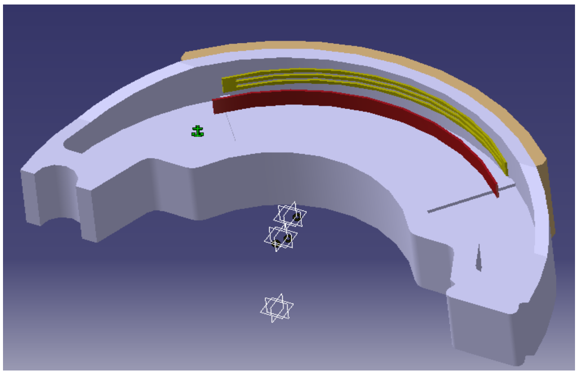
$$\{u_i\} = \{\emptyset_i\}e^{\sigma it}(e^{j\omega it} + e^{-j\omega it}) \tag{3}$$

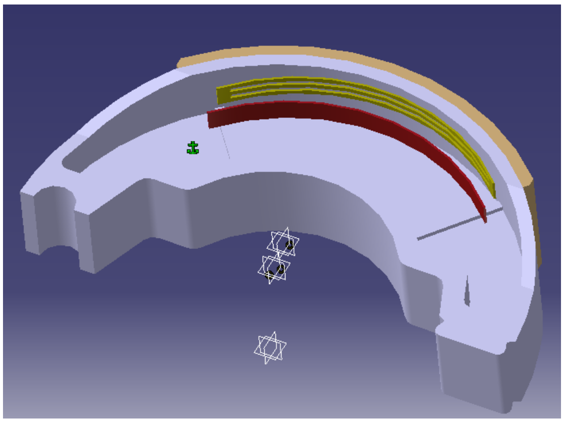
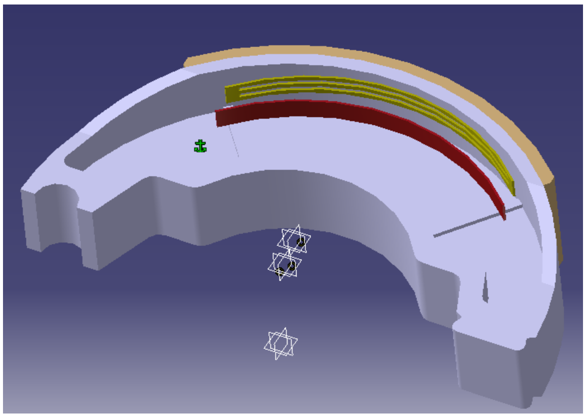
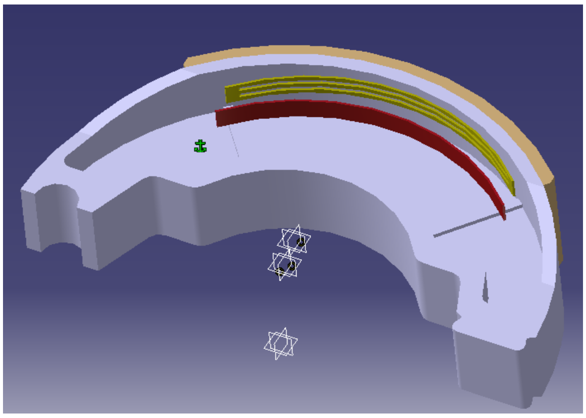
Thus, σ_i and *wi* are the damping coefficient and damped natural frequency describing damped sinusoidal motion. If the damping coefficient is negative, decaying oscillations typical of a stable system result. However a positive damping coefficient causes the amplitude of oscillations to increase with time. Therefore the system is not stable when the damping coefficient is positive. By examining the real part of the system eigenvalues the modes that are unstable and likely to produce squeal are revealed (Du et al., 2015b).

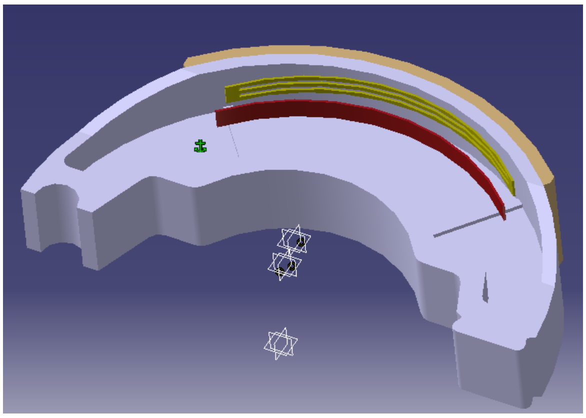
2.1 Insulator Configuration

In reducing the squeal occurrences, insulator will be used in stability analysis. The insulator which consists of different material layer will be attached to the inner side of the shoe. In this project, four pair of sample insulators with different material will be tested. Table 2 shows the proposed configurations of the insulators with their description, where the yellow color refers to the polymer material and the red color refers to the iron material.

Table 2. Different Insulators Configurations

Sample No.	**Description**	**Configuration**
Sample 1	1 Frame rubber (yellow), 0.5 mm thickness & 1 Steel plate (red), 0.5 mm thickness.	
Sample 2	1 Frame ABS (yellow), 0.5 mm thickness & 1 Steel plate (red), 0.5 mm thickness.	

Sample 3	1 Rubber plate (yellow), 0.5 mm thickness & 1 Aluminum plate (red), 0.5 mm thickness.	
Sample 4	1 Frame ABS (yellow), 0.5 mm thickness & 1 aluminum plate (red), 0.5 mm thickness	
Sample 5	1 Rubber plate (yellow), 0.5 mm thickness & 1 composite plate (red), 0.5 mm thickness	

Sample 6	1 Frame ABS (yellow), 0.5 mm thickness & 1 composite plate (red), 0.5 mm thickness	

Complex eigenvalue analysis can be performed to get the unstable frequencies of the system without insulator then test the system with the insulator. The stability analysis will perform with different coefficients of friction for the baseline. The positive real parts of the complex eigenvalues indicate the degree of instability of the drum brake system where it indicates the probability of squeal occurrence.

Brake is one of the most important things that need to be considered when producing a vehicle. The squealing sound that produced from the brake not only contributed to the noise pollution, but also make the users are not comfortably used the vehicle. They thought that the brake might be broken down and the vehicle are not safely be used which will lead them to claim a warranty from the company that produced the vehicle. Besides, the development to suppress and eliminate the squeal produced in the brake has been the target of many researchers for recent years. However, there is no fully solution for this problem and the squeal only can be suppressed but cannot be eliminated. Theoretical and Finite element (FE) approaches are the methods that will be used to determine the squeal noise.

2.2 Finite element (FE) model of a baseline drum brake unit.

A 3D (CAD) model of a drum brake system shall be developed that based on a real brake system. This CAD model will be used in the finite element (FE) software called ABAQUS to create a baseline brake FE model. The component and assembled FE model will be analyzed using modal analysis to predict its dynamic properties such as modal frequencies and modes shapes.

2.3 Stability analysis of the baseline FE model.

Stability analysis shall be performed on the FE model in order to predict squeal noise using complex eigenvalue analysis within 1 to 10 kHz frequency range. Various friction coefficients, wheel speeds and cable forces should be considered in the stability analysis. The positive real parts of the eigenvalue indicate degrees of squeal noise in the brake unit. Both squeal frequencies and modes of the brake shoes will provide useful information to establish FE model of the insulator and to determine appropriate amount of damping for the insulator to prevent squeal occurrences.

2.4 Finite element (FE) model of brake insulators

The insulator FE model shall be developed based on different design configuration such shapes, number of layers and thickness of layer. Damping properties of the insulator can be estimated using Rayleigh damping, structural damping and loss factor.

3.0 RESULTS AND DISCUSSIONS

The result displayed the squeal occurrences after a few parameters was set up such as coefficient of friction, μ, the sliding velocity between the drum and the brake shoe, v and the frequencies of interest (1 kHz to 10 kHz). The result will also lead in determining which coefficient of friction that influencing more squeal to occurred. From the result collected, the model which has the highest number of squeal occurred will be suppressed by using brake insulator.

3.1 Stability Analysis Result

Figure 1 shows the predicted result of the squeal occurrence of the FE model of the drum brake system. The result of the project will be compared between the baseline models which have a static coefficient of friction without slip rate and another one is baseline models with the presence of slip rate. The complex eigenvalue analysis performed by setting a set of boundary condition, where the frequency of interest is between 1 to 10 kHz, the rotational speed of drum is set to be 40 rad/s, the displacement of the cam for brake applying is set to be 4mm and the coefficient of friction, μ is varied from 0.3 to 0.5. Based from Figure 1, is clearly can be seen that baseline model without slip rate that contains coefficient of friction, μ=0.30 have no squeal generation. However, the squeal is detected when the coefficient of friction, μ is 0.4 and 0.5. There is one squeal detected for μ=0.4 at frequency 5677.7 Hz and μ=0.5 at frequency 5659.9 Hz. Previously in literature review, it is state that this phenomenon occurred is due to the magnitude of stiffness matrix with increasing friction coefficient. The addition of the friction coupling forces at the friction interface result in the stiffness matrix for the system containing unsymmetrical off-diagonal coupling term. From this stability point of view, this coupling is considered as the reason of brake squeal occurred (Hervé et al., 2015). Baseline model with slip rate produced more squeal occurrence compared to baseline model without slip rate. There are 7 unstable mode detected for μ=0.3 and μ=0.4, and 6 unstable mode detected for μ=0.5. Previously, it is state that this phenomenon occurred due to the change of the friction force direction. When the velocity of vibration changed the direction alternately, the friction force will also change the direction accordingly if the magnitude of the sliding velocity is less than the magnitude of the vibration velocity. In this situation, a repeating sequence of friction force in opposite direction is produced to maintain the vibration in undamped condition. Sliding velocity with friction coefficient decreased critically when the vibration velocity decrease. The system will be stable when the vibration velocity decrease with the increasing of friction coefficient since higher friction force provides higher vibration resistance to the system. Figure 2 shows relationship between frequency and the squeal generation occurred for baseline model which contains coefficient of friction, μ and slip rate.

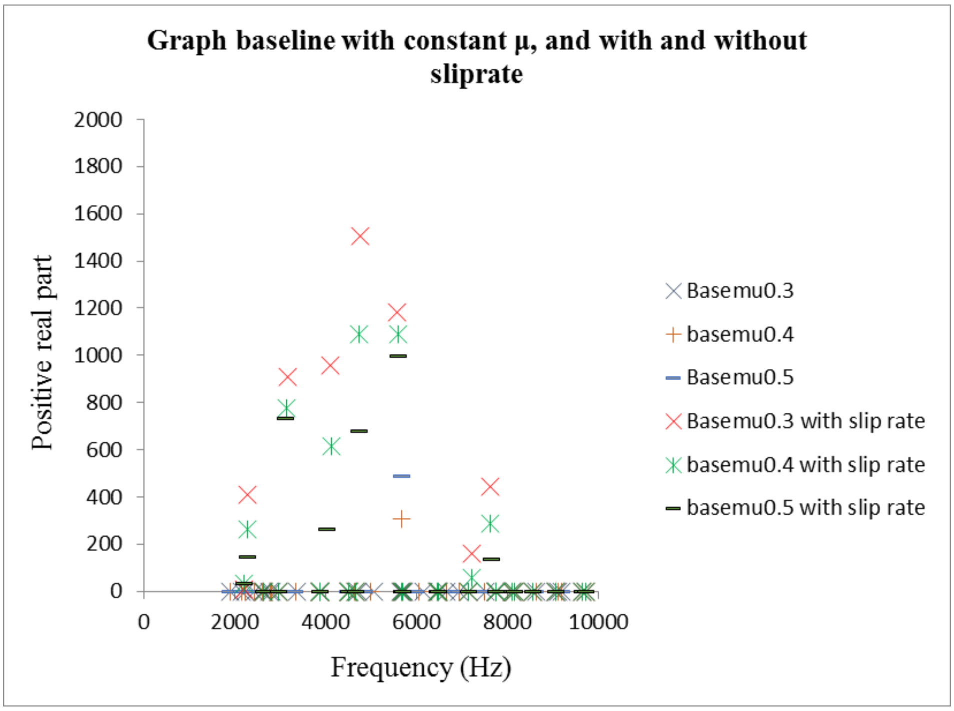


Figure 1. Comparison baseline models with and without slip rate.

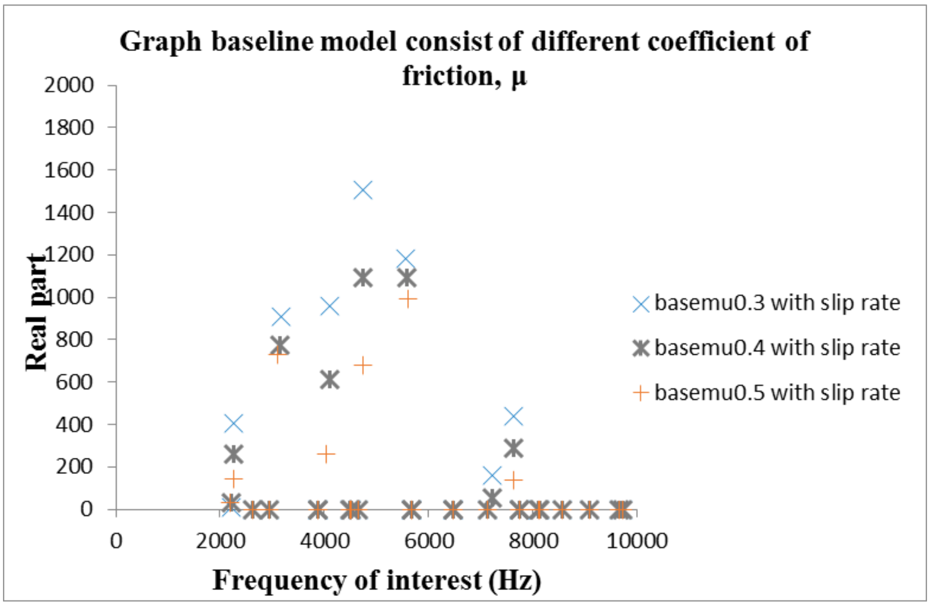


Figure 2. Relationship between frequency and brake squeal generation.

Based from the above figure, is clearly can be seen that baseline model which contains slip rate and friction coefficient of 0.3 and 0.4 produced more positive real part compared to the other model. Thus, baseline model which consists of slip rate and friction coefficient of 0.3 or 0.4 will be models that need to be suppress. In this case, the model chose is baseline model with friction coefficient of 0.4. The squealing generated based on the friction coefficient, μ and the existence of vibration velocity has been discussed. From the discussion, the baseline model for friction coefficient of 0.4 has been chose to be suppressed. However, another tested is conducted by using the same model to determine the brake squeal occurrence causes by the difference operating condition; which is different value of velocity. The friction coefficient, μ of 0.4 produced the most unstable mode. By using the same model, the velocity of the system is changed. This tested is conducted to observe the effectiveness of velocity in producing brake squeal. The rotational speed used are 30 rad/s, 40 rad/s and 100 rad/s. Figure 3 show the result of the test. The result shows that the baseline model with rotational speed of 30 rad/s, 40 rad/s, and 100 rad/s have same number of unstable mode. However, the baseline model with rotational speed of 30 rad/s recorded the highest brake squeal generated at frequency of 4741.8 Hz. Thus, the baseline model with μ=0.4, and rotational speed of 30 rad/s will be choose as the model that need to be suppressed.

3.2 Brake Insulator Result

The system can be stabilized by increasing the damping (Du et al., 2015a). There is a few common methods used to increase the damping of the system; first by choosing highly damped material for the drum brake system or the second, adding a damping material to the drum brake system. Four insulator configurations have been tested against squeal generation, the test conducted under friction coefficient of 0.4 and the speed of 30 rad/s. The purpose is to obtain the effectiveness of these insulators to suppress the squeal. However, in this case, the test will be focuses more on applying different types of material on brake insulator, but not the damping properties. The materials that have been tested are rubber, Acrylonitrile butadiene styrene (ABS), steel, aluminum and composite. Figure 4 show the stability analysis of drum brake system for insulator with different types of iron material and ABS. Based from Figure 4, it shows that one squeal occurred for all three pairs of material at the frequency range between 6 kHz to 7 kHz. Previously on the data shows on Figure 3, there are at least six squeal occurred at the frequency range between 2 kHz to 10 kHz. However, all three materials in Figure 4 succeeded in suppressing the squeal occurred and only one squeal is detected.

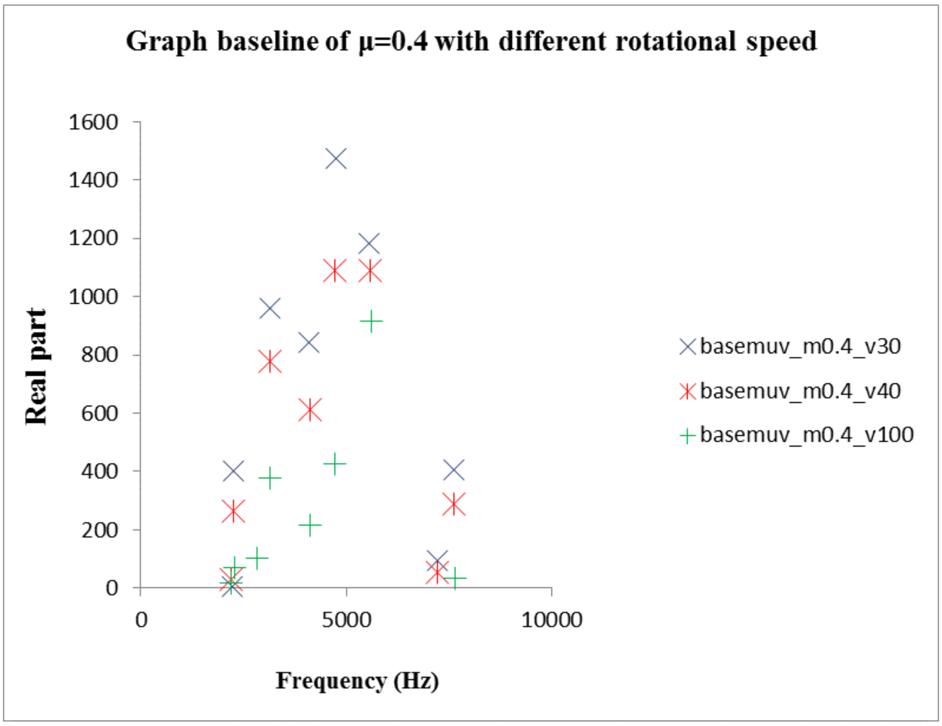


Figure 3. Relationship of frequency and brake squeal occurrence for different speed.

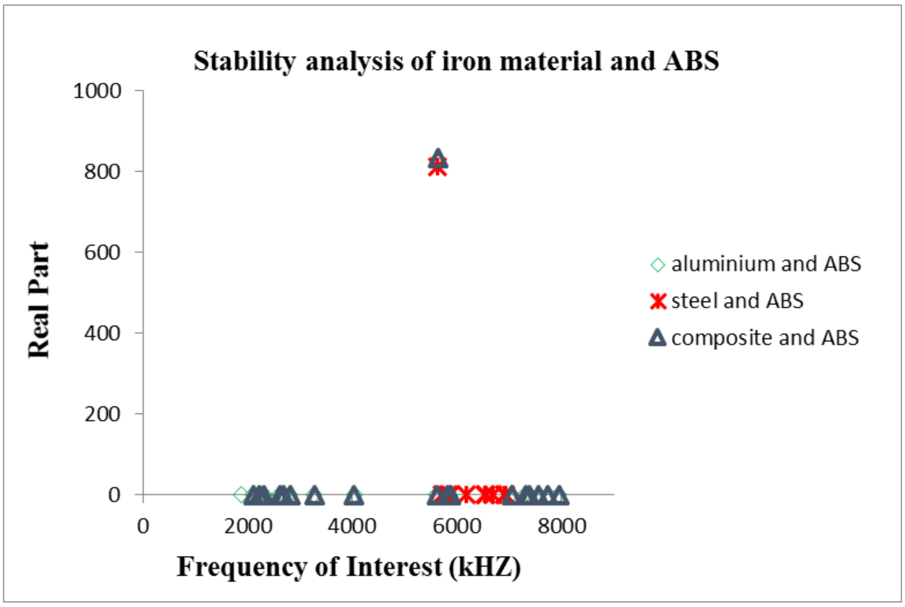


Figure 4. Stability analysis of drum brake system for insulator with different types of iron material and ABS.

Based from Figure 5, it shows that a few squeal occurred for all three pairs of material. Steel and rubber recorded the highest squeal occurred compared to the others pair. However, all three materials in Figure 5 succeeded in suppressing the squeal occurred and only one squeal is detected for pairs of rubber with composite and aluminum.

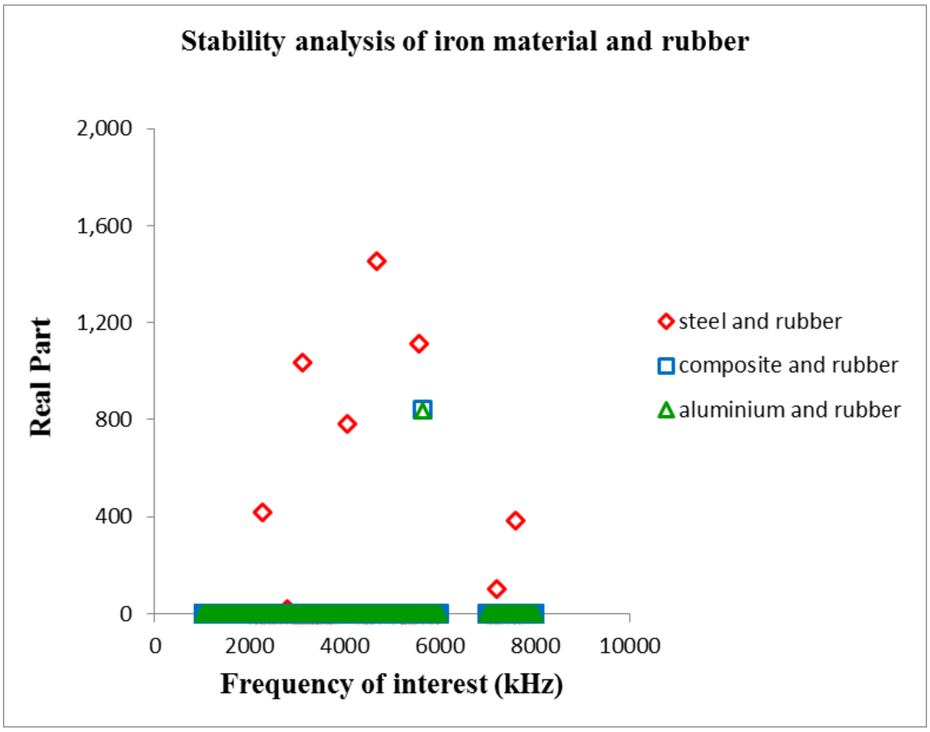


Figure 5. Stability analysis of drum brake system for insulator with different types of iron material and rubber.

Figure 4 and Figure 5, both show that the material used successfully reduce the squeal generation. However, compared to Figure 4 and Figure 5 displayed more precise result. This can be prove from the data collected from ABAQUS software where the pairs of iron material with rubber read more data compared to the pairs of iron material with ABS. Table 3 shows the example of squeal occurred for rotational speed of 30 rad/s and coefficient of friction of 0.4.

Table 3. Data collected for v=30 rad/s and μ=0.4.

Mode Number	Real Part	Frequency (Hz)
1	4.1933	2212.2
2	402.44	2266.4
3	959.27	3153.3
4	844.7	4102.8
5	1475.7	4741.8
6	1181.6	5568.5
7	94.606	7213.7
8	404.7	7622.3

The brake insulators that have been used should reduce the squeal generated near the frequencies values as shown in Table 4. Example, let's have a look at data collected for aluminum pair with ABS and rubber at frequencies range of 2 kHz to 3 kHz.

Table 4. Data collected for pair of aluminum and ABS

Mode number	Real Part	Frequency (Hz)
1	0	2109.9
2	0	2221
3	0	2318.7
4	0	2620.9
5	0	2696.1
6	0	2814

4.0 CONCLUSIONS

The complex eigenvalue available in ABAQUS software utilized to predict the stability of the drum brake system of the motorcycle, the stability analysis performed with different coefficients of friction to study the influencing of the friction coefficient. The worst condition was associated with μ= 0.4 and v= 30 rad/s, where there were eight unstable frequencies detected. Apply an insulator is the proposed method used to reduce the unstable frequencies. The insulator consists of multi layers of rubber and iron plate attached together, then they glued to the inner side of the shoes body. Six insulator with different pairs of iron material and polymer have been tested, and all of them could reduce the unstable frequencies. However, the best pair of insulator is the pairs that have the least value of the positive real part in the complex eigenvalue analysis. A pairs of composite body and aluminum with rubber recorded the least value of positive real part. Thus, a pairs of composite body and aluminum with rubber is the best material to suppress the squeal generated.

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