Jurnal Mekanikal
December 2009, No. 29, 67-83

SUPPRESSING DISC BRAKE SQUEAL THROUGH STRUCTURAL MODIFICATIONS

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ABSTRACT

Brakes squeal remains an elusive problem in the automotive industry. There have been many proposals proposed to tackle these problems but very few are effective to suppress and eliminate it. This paper proposes a solution to suppress disc brake using the finite element method. First a three dimensional finite element (FE) model of a real disc brake assembly is developed and validated. Then, complex eigenvalue analysis made available in commercial FE software package is performed to determine stability of the brake system where positive real parts of the complex eigenvalue indicate unstable system and in turn exhibit squeal generation in the brake assembly. Then, various disc modifications are proposed to reduce the brake squeal. A good modification should be able to reduce and eliminate squeal at various brake operating conditions.

Keywords: *Complex eigenvalue, disc brake, finite element, squeal, structural modification.,*

1.0 INTRODUCTION

Since vehicle comfort has become such an important factor to indicate the quality of a passenger car, eliminating or reducing the noise and vibration of a vehicle structure and system seems to provide a leading edge in the market to vehicle manufacturers. With progress made towards other aspects of vehicle design refinement against vehicle vibration and noise through improvement, refinement in brake vibration and noise is inevitable. This can be seen from the literature where the awareness on the brake vibration and noise issues started as early as 1930's using theoretical, numerical, and experimental approaches and sometimes combination of any of them. Kinkaid *et al.* [1] conducted a comprehensive review covering the study of disc brake from its mechanisms, experimental studies, the models developed and methods to eliminate squeal. In a recent review, Chen [2] provides guidelines to suppress and eliminate squeal occurrence. This includes

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optimization of the damping, minimizing the impulsive excitation and reducing the modal coupling. These three guidelines have been implemented by many researchers and thought to be essential for squeal reduction approaches.

Fosberry and Holubecki [3] observed that offsetting the pad towards the leading edge, supporting the side of the friction material rather than a backing plate and using a split disc which an annular ring riveted to the main disc could reduce squeal occurrence. Ishihara *et al.* [4] proposed a method for reducing low-frequency disc brake squeal. The fixed type, four opposed piston, disc brake was used in their experiments. The theoretical study was also performed to identify factors that generate squeal. They observed that friction coefficient and pressure fluctuation that coupled with relative displacement in the normal direction of friction surface were the great influence in squeal. Changing friction coefficient of material, the contact position between the pad and disc, the vibration characteristics of the disc and caliper, and the linear stiffness of pad material could reduce squeal. Additionally, changing rotor shape and material were confirmed experimentally reduced in squeal. Dunlap *et al.* [5] provided general solutions for every categories of disc brake noise. For low frequency noise they suggested a lining material modification where lining composition such as filler, abrasive, fibre and lubricant had strong influence in groan noise. In order to address low frequency squeal, they proposed to decouple the caliper and the disc modes where the disc material was changed from gray cast iron to dampen iron. While for high frequency squeal, increased in the brake rotor stiffness could reduce squeal propensity. They also observed that brake pad geometry and contact pressure had significant effect on brake squeal. Finally, the commented that brake noise insulators had capacity to provide significant reductions in squeal level. Nakajima and Okada [6] found that changing the bolt position towards end of the mounting bracket was effective to reduce squeal.

Dessouki *et al.* [7] discussed the prevention method of the disc brake squeal. They differentiated three classes of disc brake squeal as (1) caliper bracket induced squeal (2) pad induced squeal and (3) disc induced squeal. They proposed that the common countermeasure for caliper bracket induced squeal was to introduce mass loading to the caliper bracket or alternatively to stiffen the bracket. For pad-induced squeal, they proposed chamfers, shorter length pads and insulators. For disc-induced squeal, used slice cuts in the radial direction, increase cheek thickness and disc damping could prevent in-plane squeal. Nishiwaki *et al* [8] believed that brake squeal was strongly influenced by the natural frequency and mode of the disc. In order to prevent squeal to occur they modified the disc by removing a number of vanes or fins. They observed the disc with 2, 3, 4 or 6 vanes eliminated could prevent squeal occurrence. Matsuzaki and Izumihara [9] proposed a slit disc where they cut two radial slots into the rubbing surface of the disc to reduce squeal. Fieldhouse *et al.* [10] proposed disc asymmetry to reduce disc brake squeal. Sets of radial holes are drilled into the disc rim to generate disc asymmetry. It is thought that the disc asymmetry could decouple bending modes and hence reduce the propensity of squeal occurrence. Kung *et al.* [11] calculated modal participation factors that could examine the modal coupling mechanisms. The results showed that the disc contributes 23 percent of the total system motion. They found that changing the disc material would eliminate dynamic instability of the disc brake where it might decouple the modal interaction of the disc. Park *et al* [12] presented an approach to estimate and reduce disc brake squeal. First, they employed complex Modal Assurance Criterion (MAC) to identify most influence component towards squeal generation. They found that the disc and the caliper are most dominant components on the unstable modes as shown in Figure 1. In their work, only the caliper is modified, simulated and tested. They found that modifying the caliper shape could eliminate squeal at 7 kHz.

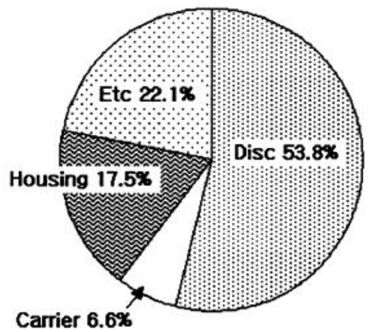


Figure 1: Component contribution factors for squeal frequency at 7 kHz [12]

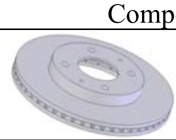
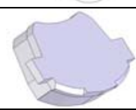
It is clearly shown from aforementioned studies that the disc has major contribution towards squeal generation. Thus, this paper attempts to eliminate brake squeal through disc modifications using finite element method. The main objectives of these modifications are to separate merging modes by modifying the mass and stiffness distributions, to reduce excitation and to alter modal characteristics [2]. Prior to that a 3-dimensional finite element model of a real disc brake system is developed and then validated in terms of natural frequency and its associated mode shape. In order to examine instability of the disc brake assembly, ranging from 1 kHz to 10 kHz, commercial finite element software ABAQUS is utilized. First, the baseline FE model is simulated to predict unstable frequencies. Several disc modifications are proposed and simulated. A good modification should eliminate positive real parts in the complex eigenvalue at various brake operating conditions.

2.0 MODELLING OF DISC BRAKE ASSEMBLY

A detailed three dimensional finite element (FE) model of a vented disc brake assembly has been studied in this research as shown in Figure 2. All disc brake system components, except a rubber seal (attached to the piston), two rubber washers (attached to the guide pins) and pad insulator, have been included in the model. The finite element model has up to 35,000 solid elements and has approximately 120,000 degrees of freedom. Details of the disc FE model are given in Table 1.

Figure 2: Finite element model of a disc brake assembly

Table 1: Descriptions of the disc brake components

Components		No. of Elements	No. of Nodes
	Disc	8177	7866
	Caliper	8138	5135
	Carrier	7204	3803
	Piston	383	678
	Back Plate Friction Material	2735	1804
	Guide Pin	811	665
	Bolt	112	186
	Clip	990	2090

2.1 Validation of the FE Model and Squeal Test

In order to obtain dynamic characteristics of baseline disc brake components or assembly, it is common practice to perform modal analysis. Squeal tests are conducted to verify prediction from the complex eigenvalue analysis.

2.1.1 Validation at Component Level

In order to validate the FE model of the disc brake components, an experimental modal analysis (EMA) is performed on the individual brake components. The method is used to provide an independent means of verifying the results obtained from the FE modal analysis in the free-free boundary condition. The impact test method is carried out to obtain natural frequencies and the associated mode shapes of the brake components. By adjusting the Young's modulus and the density, the numerical and experimental natural frequencies for free-free boundary condition of the disc brake components become very close and they are listed in Table 2. The material properties of the disc brake components are given in Table 3. Tirovic and Day [13] highlighted that an accurate representation of the component model forms one of the validation stages for good squeal correspondence between experiments and predictions.

Upon completion of the modeling, all the disc brake components must be brought together to form an assembly model. Contact interaction between disc brake components is represented by node-to-surface contact elements except for the disc/pads interface where surface-to-surface contact elements are employed. This selection is due to the fact that contact pressure distributions at the disc-pad interface are more significant than the other component contact interfaces.

Table 2: Comparison of natural frequencies at free-free boundary condition

Components	Mode	EMA (kHz)	FE (kHz)
Caliper	1	2.50	2.54
	2	4.90	4.89
	3	5.60	5.54
Carrier	1	1.49	1.53
	2	2.37	2.31
	3	5.1	5.03
Guide Pin and Bolt	1	3.35	3.16
	2	6.84	7.23
Disc	1	1.22	1.25
	2	2.86	2.89
	3	4.62	4.57
Brake Pad	1	3.40	3.56
	2	6.90	7.10
Piston	2	2.90	2.89

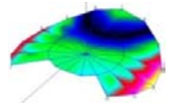
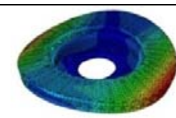
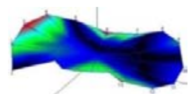
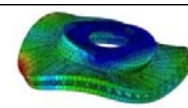
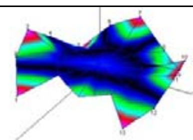
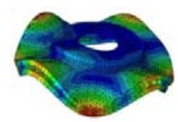
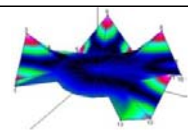
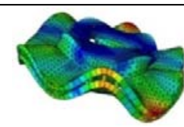
Table 3: Material properties of disc brake components

Components	Density (kgm^{-3})	Modulus Young (GPa)	Poisson's ratio
Disc	5579	110	0.3
Caliper	6162	111.29	0.3
Carrier	8000	170	0.3
Piston	8500	135.6	0.3
Guide pin	7656	93	0.3
Bolt	6219	95.7	0.3
Clip	7806	190	0.3
Friction material	2519	8.7	0.3
Back pad	8333	210	0.3

2.1.2 Validation at Assembly Level

The second validation stage is to capture dynamic characteristics of the assembled model. The previous separated disc brake components are assembled together to form the assembly model. A combination of node-to-surface and surface-to-surface contact elements is used to represent contact interaction between disc brake components. In the experimental modal analysis, a brake-line pressure of 0.4 MPa is imposed to the stationary disc brake assembly. A similar condition is also applied to the FE brake assembly model. In this validation, measurements are taken on the disc as it has a more regular shape than the other components. It is found that the predicted results are close to the measured data as shown in Table 4.

Table 4: Measured and predicted mode shape for the disc assembly

Mode	EMA	FEA
Bending	 1.29 kHz	 1.313 kHz
3 Nodal Diameter	 3.26 kHz	 3.298 kHz
4 Nodal Diameter	 4.68 kHz	 4.76 kHz
6 Nodal Diameter	 8.51 kHz	 8.408 kHz

2.1.3 Squeal Tests

Disc brake squeal tests are carried out using a brake dynamometer. The brake dynamometer is driven by a 11kW dc motor providing up to 157 rad/s disc rotation and 400 Nm of torque that has recently being developed at the Automotive Laboratory, Universiti Teknologi Malaysia. There are approximately fifty (50) squeal tests have been conducted at various operating conditions such as different brake-line pressure and disc speeds. The experimental study is carried out using a new and unworn brake pad. It is found that squeal frequencies are dominant at 4.4 kHz, 6.7 kHz and 8.9 kHz as shown in Figure 3.

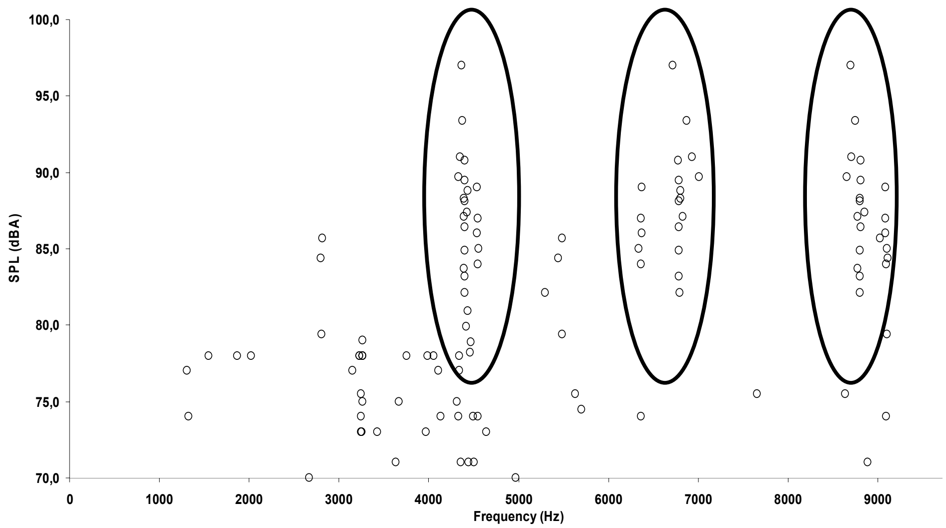
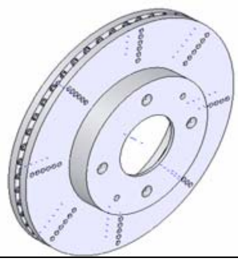
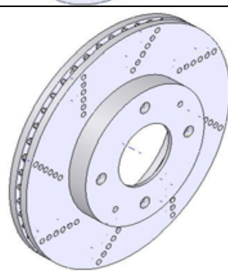
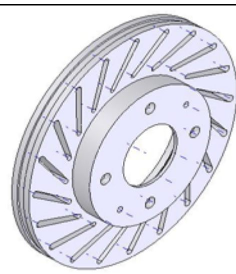
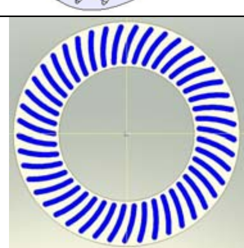


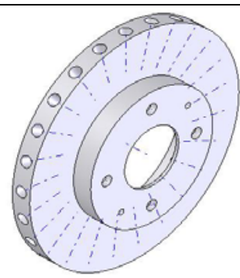
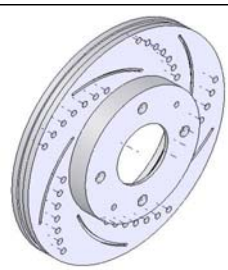
Figure 3: Sound pressure level versus squeal frequency

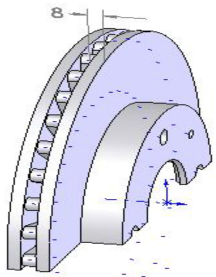
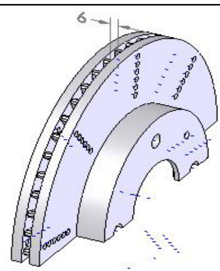
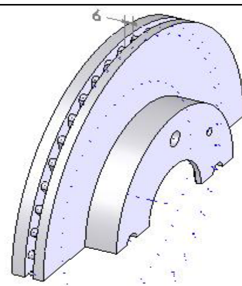
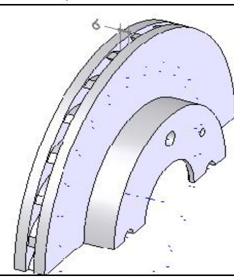
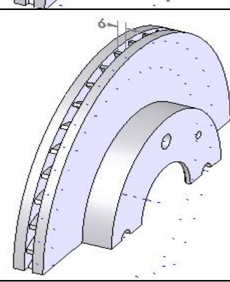
2.2 Disc Modifications

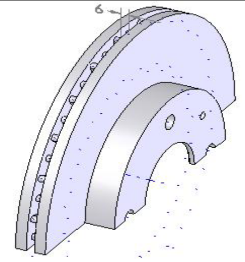
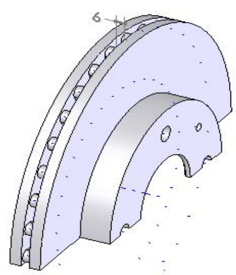
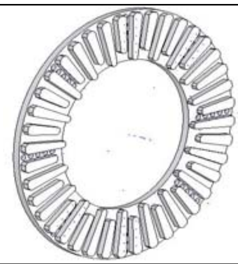
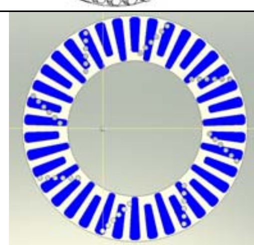
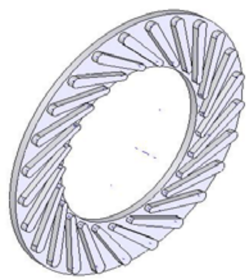
In this section, various disc modifications are proposed in order to obtain improved squeal performance of the disc brake assembly. This is achieved when either the positive real parts of complex eigenvalue of the baseline model are reduced or disappeared. Structural modifications can be done on the brake pad lining, carrier and caliper. There are 19 disc modifications being proposed including modifications on shape and angle of the vanes, drilling holes on the disc surface, changing cheek thickness and using a solid disc. Details of the modifications are listed in Table 5.

Table 5: Disc structural modifications

No	Modifications	Description
Baseline		Original disc brake with chamfered vane. The vanes are arranged 1 short and 1 long side by side.
DM01		Original disc brake has been added with 6 holes. There are 8 sets of 6 holes being arranged equally on disc.
DM02		Original disc brake has been added with 7 holes of 45^o. There are 8 sets of 7 holes being arranged equally on disc. The cheek thickness has been reduced from 8 mm to 6 mm
DM03		Slotted disc. The slot angle is 20^o and being arranged clockwise.
DM04		Disc with curve vanes. 45 vanes have been arranged clockwise with 7^o angles.

DM05		Vanes with blade-type have been arranged anticlockwise. Inlet of air flow is small but the outlet of air flow between the vanes is large. 25 vanes have been arranged.
DM06		Vanes with blade-type have been arranged anticlockwise. Inlet of air flow is small but the outlet of air flow between the vanes is large. 40 vanes have been arranged.
DM07		25 cone vanes being made to the disc. Inlet of air flow is small but the outlet of air flow is big in diameter.
DM08		Uneven vanes position. The vanes have been position up and down to each other and filleted.
DM09		Solid disc. There are 7 curving holes and dented curve with depth of 3 mm.

DM10		Size of vanes increases to 11 mm with 7^{o} angles. Cheek thickness remains the same.
DM11		The cheek thickness of DM01 is reduced to 6 mm.
DM12		The cheek thickness of DM04 is reduced to 6 mm.
DM13		The cheek thickness of DM05 is reduced to 6 mm.
DM14		The cheek thickness of DM06 is reduced to 6 mm.

DM15		The cheek thickness of DM08 is reduced to 6 mm.
DM16		The cheek thickness of DM10 is reduced to 6 mm.
DM17		Vanes size of the original disc has been increased to 11 mm in diameter. 6 holes have been drilled to the disc. Cheek thickness remains the same.
DM18		Vanes size of the original disc has been increased to 11 mm in diameter. 6 holes have been drilled at angle of 45^{o} to the disc. Cheek thickness remains the same.
DM19		Vanes with 20^{o} angles in anticlockwise direction. Size of vanes has been increased to 10 mm.

3.0 STABILITY ANALYSIS

The complex eigenvalue analysis available in ABAQUS is utilized to determine instability of the disc brake assembly. The positive real parts of the complex eigenvalue indicate the degree of instability of the disc brake assembly and are thought to indicate the likelihood of squeal occurrence. The essence of this method lies in the asymmetric stiffness matrix that is derived from the contact stiffness and the friction coefficient at the disc/pad interface [14].

Typically, the complex eigenvalue analysis is simulated in the implicit version while the explicit version is used for the transient analysis. In order to perform the complex eigenvalue analysis using ABAQUS, four main steps are required [15]. They are given as follows:

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

3.1 Results and Discussions

In this work, preliminary operating conditions of numerical complex eigenvalue analysis for the disc brake assembly are set to brake line pressure of 3 MPa, rotating speed of 9 rad/s and friction coefficient of 0.35. For the baseline model, it is found that there are nine unstable (squeal) frequencies predicted in the complex eigenvalue analysis as compared to three in the squeal tests. Even though complex eigenvalue analysis generates more unstable frequencies, nevertheless three of them are quite close with the squeal frequencies as listed in Table 6. It is seen that the disc is dominated by out-of-plane modes for all squeal frequencies as shown in Figure 4. Since out-of-plane mode typically has two identical frequencies at different phase which leads to mode coupling and it is essential to break the symmetrical feature of the disc. Thus, by modifying the disc structure as listed in Table 5, it can separate merging modes, reduce excitation and alter modal characteristics of the disc.

Table 6: Predicted results for the disc brake assembly

Speed (rad/s)	Friction Coefficient	Pressure (Mpa)	Exp		FE	
			Frequency (Hz)	Amplitude (ms^{-2})	Frequency (Hz)	Real parts
4.63	0.49	0.4	4336	2.60	4467	1.2952
			6730	0.85	6376	92.63
			8708	0.89	8778	504.86

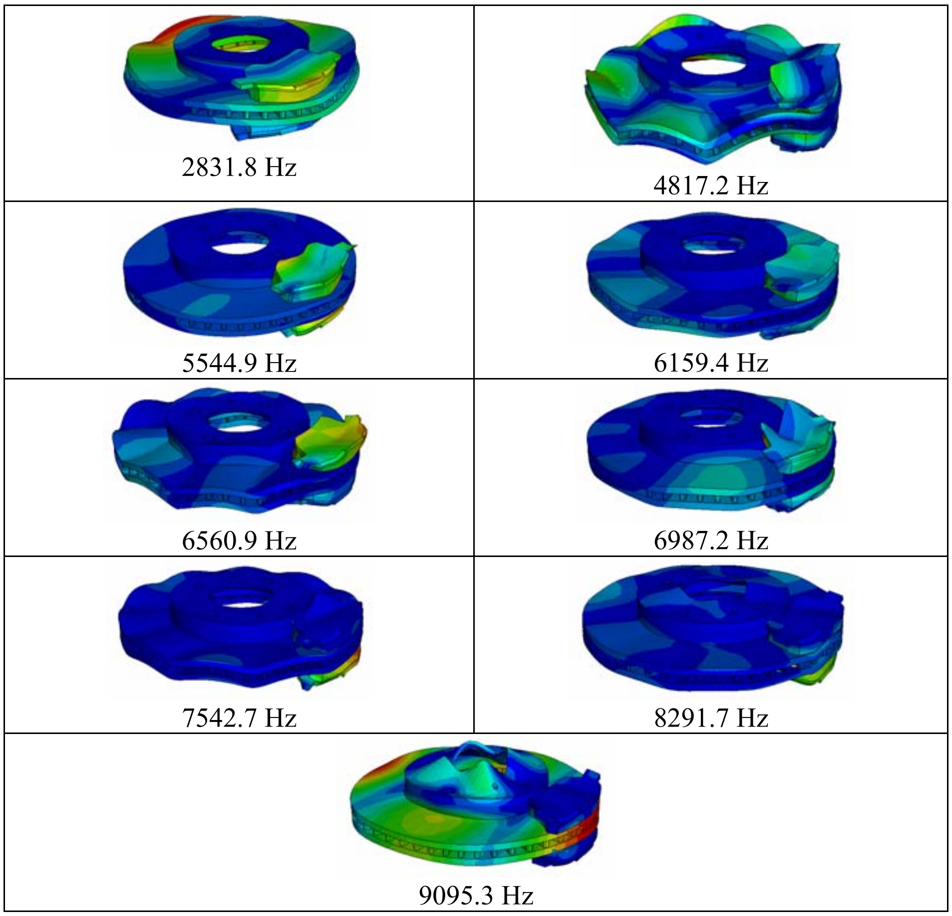


Figure 4: Unstable (squeal) mode shapes for the baseline model

The first modification (DM01) is made by drilling six holes with equally distributed at eight locations on the disc surface as shown in Table 5. It is found that such modification does help to reduce squeal at low frequency, i.e. below 5 kHz while high squeal frequencies remain as indicated in Figure 5. The second modification (DM02) is done by reducing cheek thickness to 6mm together with 45 degree of seven holes drilled equally at eight locations. This modification seems to show a non-promising result where they are eight squeal frequencies predicted in the complex eigenvalue analysis. More squeal frequencies are predicted for modification DM03 as shown in Figure 5. This modification is based on the slotted disc as illustrated in Table 5. Similarly, modifications DM04, DM07, DM08, DM09, DM12, DM18 and DM19 are not capable of reducing squeal as predicted in the baseline model. This indicates that those modifications are not a good solution to prevent squeal. Details of those modifications are described in Table 5. As for modification DM05, there are only three squeal frequencies predicted compared to nine for the baseline model. However, these three frequencies cover both low and high frequency range. It is also similar to modification BDM14. There are five disc modifications found to be reducing low squeal frequency, which similar to DM01. The modifications are DM10, DM11, DM13, DM16 and DM17. It is found from Figure 5 that only modification DM15 can totally eliminate those nine squeal frequencies as predicted in the baseline model. This suggests that reducing cheek thickness to 6mm and using uneven vanes position can be a good solution to eliminate squeal. In order to check its effectiveness against squeal, this modification is simulated again at different brake operating conditions as described in Table 7.

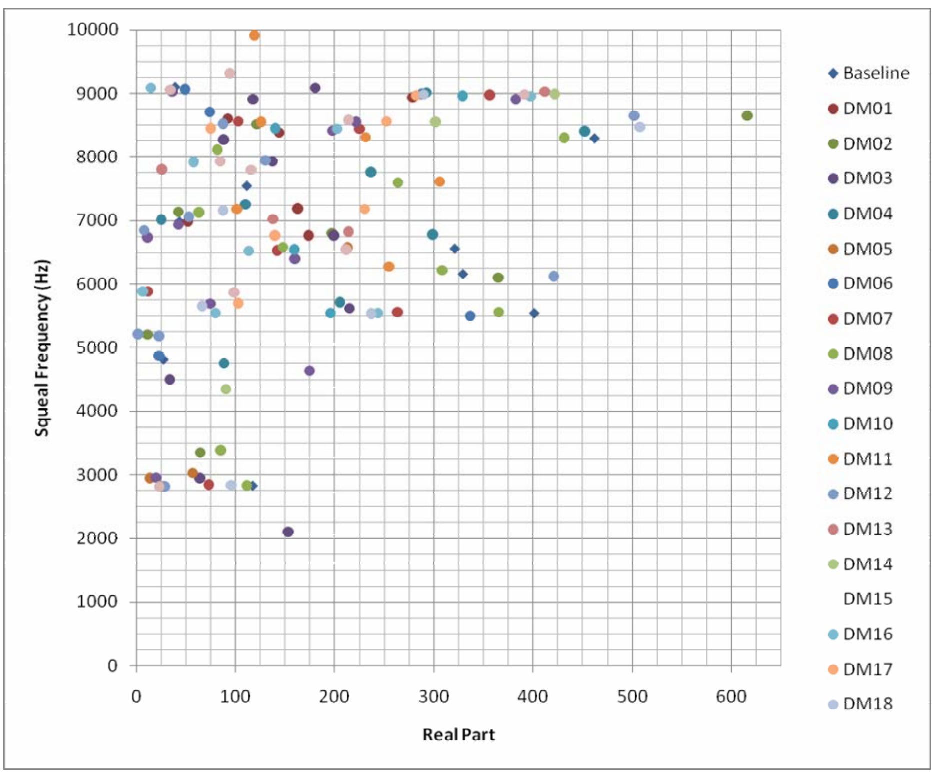


Figure 5: Predicted results for different disc modifications

Table 7: Predicted squeal frequencies at different brake operating conditions

Conditions	Pressure (MPa)	Speed (rad/s)	Friction Coefficient	Predicted Squeal Frequency (Hz)
BC1	6	9	0.4	None
BC2	10	9	0.5	None
BC3	30	9	0.5	None
BC4	40	9	0.7	None

Modification DM15 is first simulated at pressure of 6 MPa, rotating speed of 9 rad/s and friction coefficient of 0.5. From complex eigenvalue analysis, there are no squeal frequencies predicted at these particular operating conditions. It is interesting to see that for different operating conditions, i.e., BC2, BC3 and BC4 there are also no squeal predicted. This indicates that modification DM15 is a good solution to eliminate squeal generation.

4.0 CONCLUSION

This paper presents suppression approach of brake squeal through structural modifications using the finite element method. Various disc modifications have been proposed and simulated using complex eigenvalue analysis. Initially, there are nine unstable frequencies (squeal) appeared in the baseline model and they are dominated by out-of-plane modes. Having proposed 19 disc modifications including modifying cheek thickness, vanes arrangement, size and shape, drilled holes and changing to a solid disc, it is found that only DM15 can totally eliminate positive real parts which indicate no squeal in the disc brake assembly. This modification is also capable of preventing squeal occurrence for different brake operating conditions and thus indicates its effectiveness against squeal generation.

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). The first author would like to thank his colleagues, Director of Ungku Omar Polytechnic and the Chief Director of Polytechnic Education Department.

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