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On the Study of Disc Cheek Thickness as a Design Parameter for Reducing of Brake Squeal Noise

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Abstract

There is much research about mechanism generation of brake squeal noise. Recently many researchers have focused on study on mode coupling mechanism where out-of-plane and in-of-plane disc vibration modes together are coupled in some braking conditions to generate brake squeal noise. This paper is a study on 3 configurations brake disc where a cheek thickness is considered as a main design parameter for reducing the mode coupling effect. The Design A is designed with the same cheek thickness of out-board and in-board sides of the disc friction surfaces. The Design B and Design C discs are designed with the different cheek thickness of the disc friction surfaces, these designs are considered for reducing the mode coupling effect where the contact friction force draws the modes to coalesce. Stability robustness due to braking condition and manufacturing uncertainties is considered. Disc stiffness change shows more stability robustness of the Design A and Design B. Stability robustness due to increase of contact friction coefficient are found on Design A and Design B. The brake squeals analysis utilized Complex Eigenvalue Analysis (CEA) from the finite element package is performed as the simulation results. Vehicle tests of the 3 discs are performed for obtaining the field test results to compare and discuss with simulation results on this design parameter.

Keywords*:* Modal Coupling, Brake Squeal Noise, Complex Eigenvalue Analysis.

1. Introduction

Brake squeal noise is continuing on study by many researchers especially from industrial [1-3]. Due to complexity of the brake squeal mechanisms and many brake system components involved, there are several techniques used for reducing brake squeal noise such as changing composition and microstructure of gray cast iron [4] to change the natural frequency of the brake rotor without changing the rotor geometry [1]. The study on brake squeal also is divided onto low frequency squeal and high frequency squeal where the latter is concerned most in interaction between brake pad and disc [5]. However recently, some researchers have studied on mode coupling of the brake disc and mode locking between pad and disc mechanism [1] of the brake disc. The mode coupling mechanism is occurred between friction force direction of the in-plane mode and the sound radiated direction of the out-plane mode. The symmetrical shape of the brake disc can also result in the doublet modes [6] and these modes can cause the instability of the brake disc during friction-induced force.

Since the availability of the finite element package of brake squeal analysis, there are many researches on studying for design parameters and braking conditions Liu et. al [7] utilized the complex eigenvalue method to analysis of brake disc squeal where the system parameters, hydraulic pressure, rotational velocity of the disc, frictional coefficient of the contact area between the disc and the brake pad, investigated. The avoidance of the brake squeal can be performed on either the disc or the pad design. Fritz et al [8] also performed the complex eigenvalue analysis for understanding the relation between damping and mode coupling patterns of the disc brake system. Ouyang et al [9] have conducted the transient analysis to assess the brake system more than the stability issue but also the influence of the temperature distribution and other nonlinearities. The techniques can pinpoint into more complicated parameters involved in the real situation.

In this research work, there is the problem of high frequency brake squeal occurred on the original design of the brake system. The brake disc is redesigned to enhance more stability to the brake squeal event. The complex eigenvalue method is used mainly for studying stability robustness. The test vehicle according to SAE J2625 [10] is performed for relating the test results with the simulation results.

2. Problem statements

In this study, the original design of the brake disc has been available for commercial uses for some period of time. However, there are some addressed issues on brake squeal problem. In Fig. 1, during vehicle brake test, it showed the time-frequency spectrum, so-called spectrogram, measured from accelerometer at the caliper unit. The dominant frequencies are around 7,500-7,800 Hz and 15,000 – 15,600 Hz from many customer vehicles, where the accelerometer is placed on the brake caliper in the perpendicularly direction. The acceleration amplitude is shown in decibel unit with the maximum amplitude of red color corresponding to 53 m/s^2. These frequencies are occurred as harmonic frequencies. The brake squeal conditions, hydraulic pressure of 5 MPa and contact friction coefficient of 0.5, have been confirmed from the bearing-type dynamometer testing laboratory facilities. This result useful to clarify which condition occur brake squeal and how much intensity of that condition.

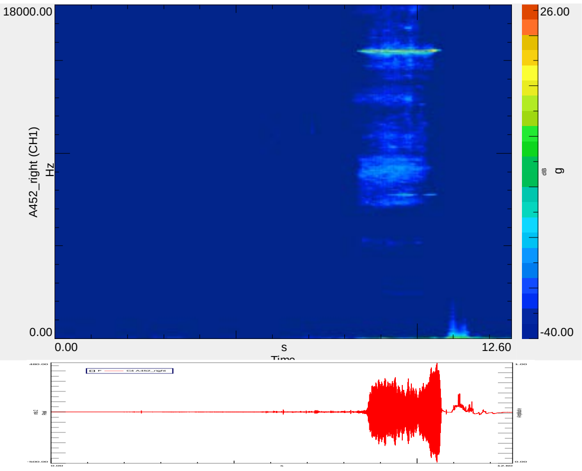


Fig. 1 Spectrogram of brake caliper acceleration during brake deceleration of the claimed brake system

3. Simulation with the complex eigenvalue method

3.1 Preparation of the finite element models

The investigation of this high frequency squeal has been studied by uses of the complex eigenvalue method with such a condition obtained from the dynamometer test see Table 1 for the conditions. Angular velocity is the disc rotational velocity applied in steady state condition. Table 2 provides the material properties of the brake disc and pad assembly model from laboratory testing. Due to availability of brake disc and pad prototypes, the modal testing data from these components are used to update the finite element model constructed from SimXpert software, such that the models can predicted very close natural frequencies to the prototypes, such deviation between test results and model prediction within 3 percent. The proportional damping value of this study is 0.005, this value get from laboratory testing and properly meshing should there are two layers for the thickness of the brake disc and pad, and the mode shapes of vibration can be predicted up to the 20,000 Hz, maximum range of high frequency brake squeal. Fig. 2 shows the finite element model of brake disc and pad system for the complex eigenvalue method. The picture show properly meshing for brake squeal analysis.

Fig. 2 Finite element model of brake disc and pad assembly for complex eigenvalue analysis.

Since the introduction of its commercial software package, the complex eigenvalue method [11] has been used widely as a simulation tool for brake squeal analysis. The method can account for applied brake pressure on the pad, contact friction coefficient between brake disc and pad, and disc angular velocity. The output results is come with the complex eigenvalue where the real part can be negative value, stable mode, or positive value, unstable mode. The positive real part can be represented as the negative damping ratio of the unstable mode.

Table 1. Braking conditions for complex eigenvalue analysis

Angular Velocity (Rad/s)	Mu	Pressure (MPa)
6.28	0.5	5

Table 2. Material Properties of Brake disc and pad

Part	E (GPa)	Density (kg/m^3)	Poisson's ratio
Disc	114	7300	0.3
Pad lining mat.	9.5	3000	0.3
Pad Back plate	200	7300	0.3

3.2 Simulation results and analysis of the original disc design

From Fig. 3, there are many unstable modes resulted from the complex eigenvalue method. The positive real parts or modes located on the right plane are unstable. However, there are three modes with higher negative damping ratio, see Fig. 4.

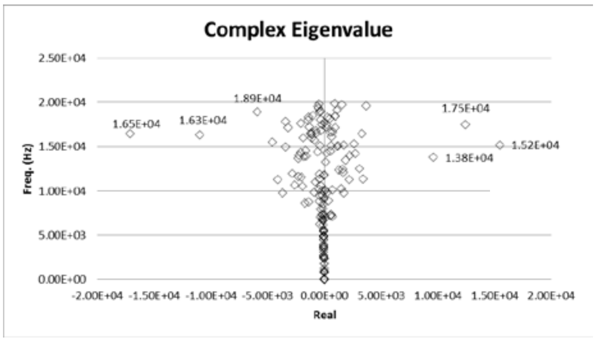


Fig. 3 Complex eigenvalue of original disc design with braking conditions from Table 2.

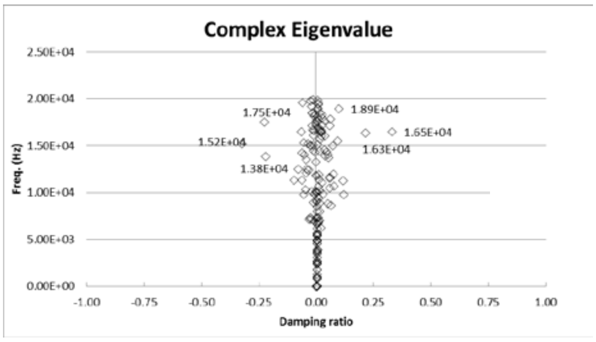
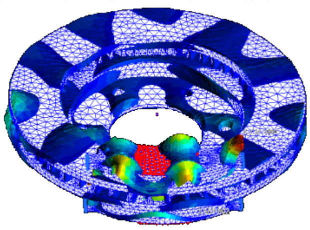
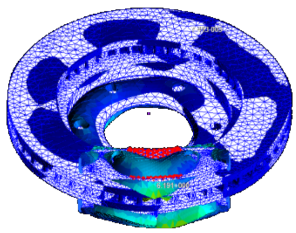


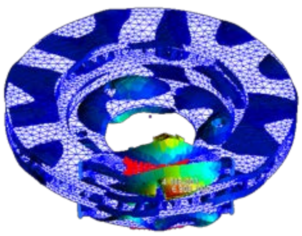
Fig.4 Negative damping ratio of the complex eigenvalue



13,800 Hz (unstable)



15,200 Hz (unstable)



17,500 Hz (unstable)

Fig. 5 Dominant unstable mode shape at 13,800 Hz, 15,200 Hz, and 17,500 Hz of the original disc

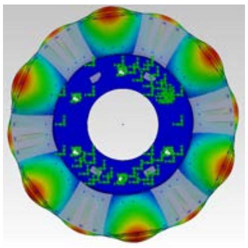
The maximum negative damping ratio around -0.3, is at the vibration mode shown in Fig. 5 at frequency of 15,200 Hz. The unstable mode shapes of the three unstable mode are similar behaviour such that the disc vibrations are in in-of-plane direction and the out-of-plane motion is occurred at the brake pad. These unstable modes froms simulation are close agreement to the spectrogram of test results shown in Fig. 1.

If considers at the disc and pad normal mode, in Fig. 6, with fixed boundary condition, the unstable mode at 15.2 kHz has the distinct shape as the in-plane radial mode with 6 nodal diameter lines and 1 nodal circumferential line (IPR6+circumferential nodal line) of the normal mode at 14,316 kHz. The unstable shape of the brake pad is corresponding to the shape of pad normal mode at 15,692 kHz. The friction excitation in this range of frequency probably come from the 3rd circumferential mode at nearby frequency of 15.517 kHz. The out-plane mode with 6 nodal diameter lines can be excited by harmonically couple with mode in-plane radial with 6 nodal diameter lines (6ND). The brake squeal noise can result from modulation of these harmonic frequencies [6] where occurred from the test result shown in Fig.1

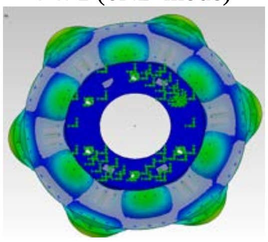
3.3. Validation with vehicle test results

The further vehicle test adapted from SAE J2625, “Automotive Vehicle Brake Squeal Test Recommend Practice”, has been adapted to repeat the test conditions found from actual field testing such that the other designed discs can be examine later to theirs stability robustness with this test. The original brake disc and pad system has been installed on the vehicle. The Link Engineering data acquisition unit and transducers, on vehicle system are used to monitor for vehicle speed, deceleration rate, pad temparature, and the pedal hydraulic pressure. The accelerometers are attached to the brake caliper on both sides of the front brake system.

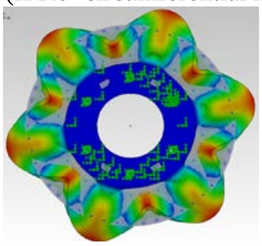
The test matrix is comprised of brake pad temperature ranged from 50 °C to 250 °C, initial vehicle speed before braking at two values of 20 km/hr and 40 km/hr, and deceleration of 0.1 g’s, 0.2 g’s and 0.3 g’s.



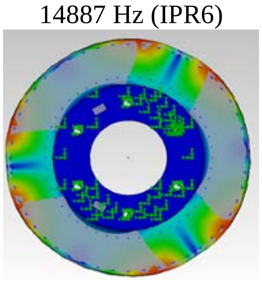
7471 (6ND mode)



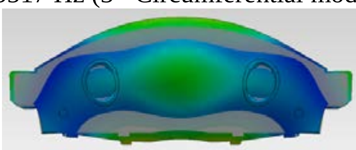
14316 Hz (IPR6+circumferential nodal line)



14887 Hz (IPR6)



15517 Hz (3rd Circumferential mode)



15692 Hz (2 longitudinal nodal lines)

Fig. 6 Significant normal mode shapes of brake disc and pad involved to brake squeal

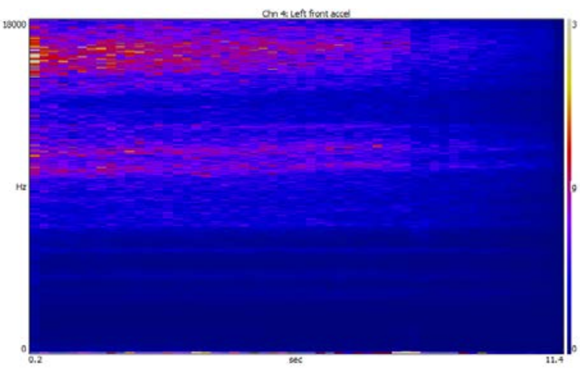


Fig. 7 Acceleration spectrogram measured from brake caliper on test vehicle according to SAE J2625 with squeal noise

The Fig. 7 shows the spectrogram of measured acceleration with initiating of squeal noise. The brake conditions of Fig. 7 are at high temperature of 250 °C, 40 km/hr, and 0.3 g's, which it is the worst squeal noise. The maximum scale of vibration level from the accelerometer is of 30 m/s^2 with yellow color (level varied from dark to bright color). There are the distinct noise frequencies around 9,900 Hz, 11,000 Hz, and also around 15,000 – 17,000 Hz. The first two squeal frequencies are corresponding the normal modes from the finite element model as the in-plane radial with 4 and 5 nodal diameter lines, or IPR4 at 8,943 Hz and IPR5 at 11,966 Hz, respectively. These modes are double the frequencies and of the same nodal diameter line of the out-plane modes, 4ND at 4,281 Hz and 5ND at 5,870 Hz. These harmonic frequency modes are coupled each others and the response can become modulate these pairs of modes together [6].

Mode locking where the similar deformed shapes among the in-plane and out-plane modes of the brake disc and the bending modes of the pad, results in self-excite each other and generates sqeual mechanism. During interaction between brake disc and pad, the symmetry geometry of these components, provides many close modes. The modes are drawed to coalesce and turn into one stable and one unstable modes with negative damping [6]. Mode spliting technique is to avoid the symmetry shape of the brake disc [12]. This will lead to further disc design to solve for the squeal problem as mentioned in the next section.

In summary, the complex eigenvalue method can be used to examine the stability of the brake system in steady state condition. There is the transient simulation [9] that can show how the self-excite mode occurs. However, due to availability of test facilities as describe above, this research work will use only the complex eigenvalue method as the tool for designing the brake disc. The brake squeal occurred during vehicle test with braking conditions at high temperature where the contact friction coefficient is tend to increase [6] and the natural frequencies of the disc can be lowered up to 3 percent [6] depended on its materials and geometry. Moreover, the finite element simulation can be performed for stablity robustness of the brake system due to these system parameter changes.

3.4 Stability robustness of disc brake squeal due to disc temperature and contact friction coefficient changes

As mentioned above, the brake disc in operation will subject to varying brake parameters. In practice, stability robustness of disc brake squeal are concerned to many parameters as recommended [6]. For example, in-plane and out-plane disc modes should be well seperated to avoid mode coupling. Contact friction coefficient should be tested for brake squeal in range of 0.2 to 0.65. The varying in natural frequencies of both brake disc and pad should be taken into account by using some field data from vehicle manufacturer.

For brake disc design, the complex eigenvalue method can account for varying in stiffness or natural frequencies of the disc and contact friction coefficient. The typical design of this disc model which made from cast iron, has a dynamic modulus of elasticity about 120 GPa. The deviations of the modulus are of -5%, +5% and +10% of the typical value to be considered. Varying of the friction coefficient values of 0.3, 0.4, and 0.5, are studied. Fig. 8 and 9 show the results from above two varying parameters, repectively.

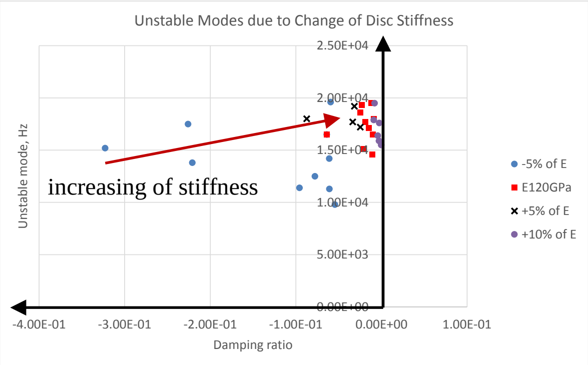


Fig. 8 Negative damping ratio varying with change of disc stiffness

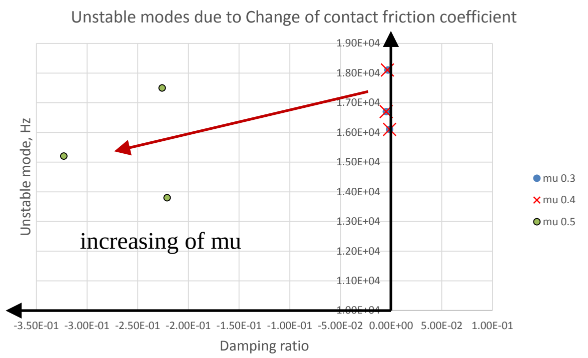


Fig. 9 Negative damping ratio varying with change of contact friction coefficient

Fig. 8 shows the natural frequencies of the brake disc and pad assembly with the damping ratio. More the negative damping, more the unstable or squeal noise. Increasing of the disc modulus of elasticity, the system will be more stable. In the practice, varying of disc temperature will results in lower natural frequencies. As well agreed the results from simulation with vehicle testing, the severity brake conditions are at high temperature where the disc natural frequencies can be lowered and the contact friction coefficient can be higher. Then the friction coefficient can draw the modes to coalesce and results in unstable modes [6]. Lin [12] had pointed out that the symmetry of disc geometry and also the pad, should be avoided such that the modulation of the modes can be reduced.

4. Further disc design to reduce brake squeal noise

There are two other disc designs named as Design A and Design B. Fig. 10 shows dimensions of the original disc design. Design A is aimed to reduce weight of the disc and also to counterfeit with the squeal noise by different thickness of outboard and inboard disc thickness. The fin number also is designed to change the nodal diameter line of the bending out-plane modes. The other parameters are designed to cope with the performance of the brake system. Design B is prepared from another designer where the first purpose is to solve for brake judder problem. Also due to unequal disc thickness, the Design B is taken into the complex eigenvalue analysis for examining its performance on noise and vibration. Table 3 summarizes all important configurations of the new disc designs.

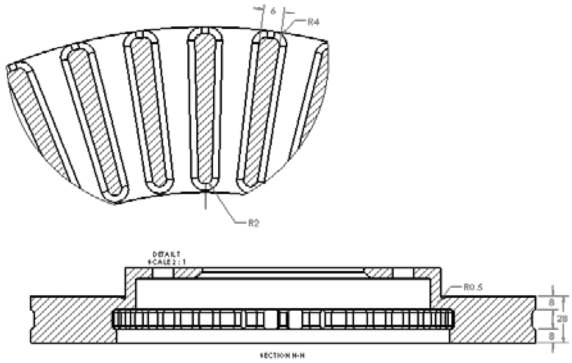


Fig. 10 Design parameters of brake disc

Table 3: Design parameters for brake squeal improvement

Design Parameters	Original disc	Design A	Design B
Outboard thickness (mm)	8	6	9.3
Inboard thickness (mm)	8	7	11.3
Fin thickness (mm)	12	15	8
Fin number (pcs)	48	42	48
Weight (kg)	7.09	6.25	8.24

Fig. 11 shows the damping ratio of all brake design such that the Design A and B are much more stable even there are also the unstable modes from the prediction but the modes close to the marginally of frequency axis.

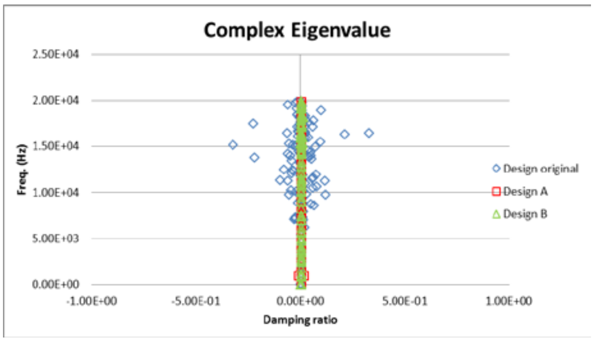


Fig.11 Comparison of unstable modes among original design, Design A, and Design B.

The varying of disc modulus of elasticity, is studied with the Design A and Design B. There is no unstable mode for conditions of -5% and the typical modulus but there are some unstable modes near the marginally stable axis (see Fig. 12 and 13 for Design A and B respectively). In practice, these small unstable can be dampened with introducing of other friction damping from the system assembly.

The Fig. 15 show result of spectrogram measured from brake caliper on test vehicle according to SAE J2625 with squeal noise (Design B). Found that Design B can reduce brake squeal problem. The result testing not found any spectrogram of brake squeal when have testing in the same condition with Design A.

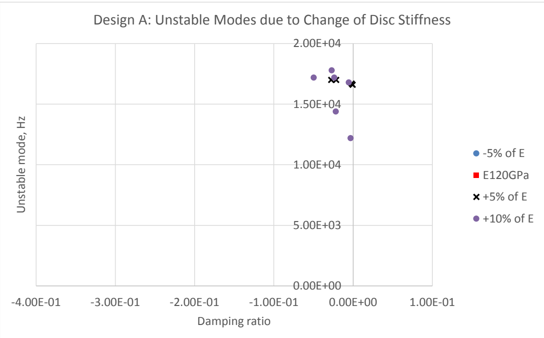


Fig. 12 Negative damping ratio varying with change of disc stiffness: Design A

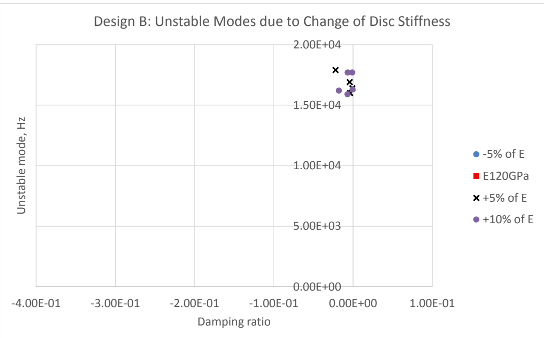


Fig. 13 Negative damping ratio varying with change of disc stiffness: Design B

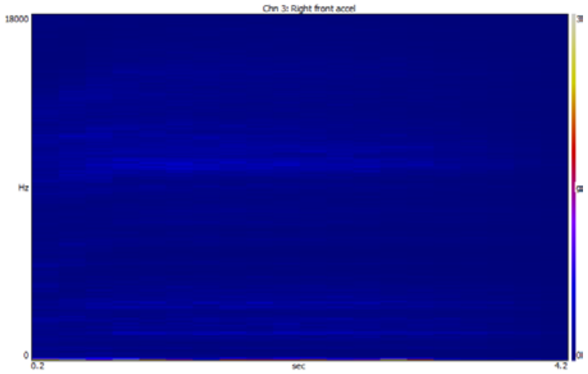


Fig. 14 Acceleration spectrogram measured from brake caliper on test vehicle according to SAE J2625 with squeal noise (Design A)

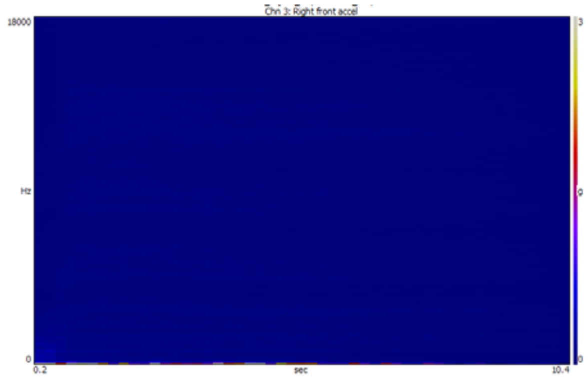


Fig. 15 Acceleration spectrogram measured from brake caliper on test vehicle according to SAE J2625 with squeal noise (Design B)

The vehicle test equipped with new disc Design A and B, are performed according to SAE J2625. The spectrogram at the same worst case of brake conditions, is shown for both discs. The Design B (Somboon Advance Technology designer) tends to be more robust to squeal noise than the Design A where there is a little sign of unstable mode at 9900 Hz, but less significant.

5. Conclusions

This research study is to investigate of the commercial brake disc problem. High squeal noise is occurred from several claimed customers, then this is probably from design aspects. The complex eigenvalue analysis utilized the finite element model of the brake disc and pad assembly, is used mainly to investigate of the brake squeal noise and to study the stability of the brake system. The original design of disc have found that both simulation results and vehicle test results are well agreed with behavior of the squeal mechanism. The coupling of in-plane and out-plane modes could lead to brake squeal noise if other conditions are involved, mode locking due to the same deformation shape of the brake disc and pad, modulation of the modes occurred as harmonic frequencies of two modes. Also, the symmetry of in-bound and out-bound friction surface geometry probably causes the mode coalesce resulted in unstable modes. Suggested as the design criteria in many literature reviews especially in reference [6], this information has been adapted to perform the complex eigenvalue analysis and design the vehicle test matrix. The proposed disc designs have been verified from simulation and vehicle test for suppression of the unstable modes.

6. Acknowledgement

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