Research Article

Stability Improvement of Brake Disc to Mode Coupling at High Frequency Squeal

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Received: 4 February 2020; Revised: 13 July 2020; Accepted: 26 August 2020; Published online: 9 November 2020

Abstract

In this research study, the high-frequency squeal noise of a brake disc was found to occurred at a frequency of about 15 kHz. The potential root cause has been studied where mode frequency coupling and shape locking mechanism of brake disc and brake pads components are the main investigated topic. From the vehicle field test and the Dynamometer test, the braking condition, friction coefficient and braking pressure, have been confirmed to be used in numerical experiments. The updated finite element model (FEM) with the modal testing data of the existing brake components are formulated for the Complex Eigenvalue Analysis (CEA). In this study, the modification is based on in-board and out-board cheek thickness of the brake disc. Two of nine modifications of the brake disc cheek thickness are proposed with the method of separation the brake disc out-of-plane and in-plane modes and the method of avoiding shape locking between the brake disc and the brake pads modes. The constructed prototypes are verified with the vehicle field test and well agreed with the CEA.

Keywords: High frequency brake squeal, Complex eigenvalue analysis, Mode coupling, Modal testing and analysis, Finite element analysis, Brake disc

1 Introduction

Brake squeal noise prediction using the Finite Element Analysis (FEA), is widely used by many researchers [1]. The Complex Eigenvalue Analysis (CEA) is mostly referred as the technique used in the commercial FEA software such as ABAQUS and NASTRAN SimXpert 2011. Nevertheless, the technique has required the braking conditions and the agreement of the Finite Element Model (FEM) and the existing brake component dynamics to provide adequate information to investigate the root cause of the squeal mechanisms [2].

Squeal mechanisms are categorized into mode coupling, mode lock-in, stick-slip instability, negative friction velocity slope, and sprang-slip [3]. In industrial, the Dynamometer test is performed with the test matrix [4] for identifying the noise concerns at the early stage of the brake prototype. Even passed the test, the prototype is not always guaranteed for noise during operation. In addition, the vehicle field test [5] has contributed to the real situation of the brake uses. The noise data is collected from many mileages and is processed to examine the noise occurrence.

The squeal noise is divided into low-frequency squeal (LFS), around 1 kHz to 4 kHz, and high-frequency squeal (HFS), around 4 kHz to 20 kHz [6]. This is to identify the brake component, which involved in the squeal mechanisms. The HFS has most contributed from brake disc and pad dynamics where the LFS incorporated with the caliper. Many researchers have addressed the mode coupling mechanism as the root cause of the HFS. Due to complexity of the squeal mechanism, many engineering tools and schemes are required to implement for the root cause investigation. Park *et al*. [7] have introduced the approach for investigating the mode coupling of the LFS brake system. The study has implemented four kinds of techniques, the Dynamometer test with 3-D scanning laser for operating deflection shapes (ODS), the brake component modal testing, the vehicle testing, and the simulation with CEA. The modification of the brake disc has implemented on the swan neck thickness and fin thickness. Yokoyama *et al*. [8] have used a 3-D laser scanning vibrometer to investigate the HFS at 11.9 kHz. The 2nd in-plane mode is the dominant mode during the noise event. The FEM of brake disc has been studied to confirm the mode. The modification has been focused on the fin shape. Joo *et al*. [9] have studied on the three FEM model configurations of the brake system consisted of brake disc, brake pads, caliper, hub and knuckle. They have pointed out that the CEA may provide only the tendency of the squeal occurrence but not account for prediction of squeal noise in some complexity circumstances. Chen *et al*. [10] have studied on mode coupling of the brake disc in-plane mode and out-of-plane mode around 6 kHz. Modifications of brake pads with chamfer are used to reduce the possibility to excite the brake disc in-plane mode. Cha *et al*. [11] have demonstrated the separation of the brake disc in-plane and out-of-plane modes by changing the vane fillet as the design parameters. The normal mode FEA has been performed for parameter sensitivity to the mode separation which affected most to the in-plane modes.

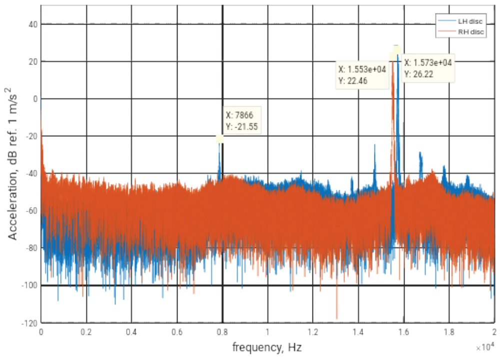
From the literature review, this study is to utilize the tools, the commercial FEM software SimXpert 2011 with CEA Brake Squeal module, the modal testing, the Dynamometer test, and the vehicle field test to verify the HFS problem from the carmaker request. The 16-inch brake disc and brake pad are modeled for the numerical simulation. The original design is studied for the root cause of the squeal mechanism. Found on this study that the original design problem is the coupling between brake disc out-of-plane and in-plane modes at high frequency. As reviewed, the countermeasure is diversified into many geometry modifications of the brake disc, depended on the specific brake design and constraints. In this study, the variation of the brake disc cheek thickness, both in-board and out-board, has been studied to overcome the HFS where the intensity of vibration modes considered as the constraints. The sensitivity of stable region for the squeal noise, is represented to the designer as the example for the design guideline. The modified prototypes are then performed with the vehicle field test to confirm the countermeasure.

2 Statement of Problems and Methodology

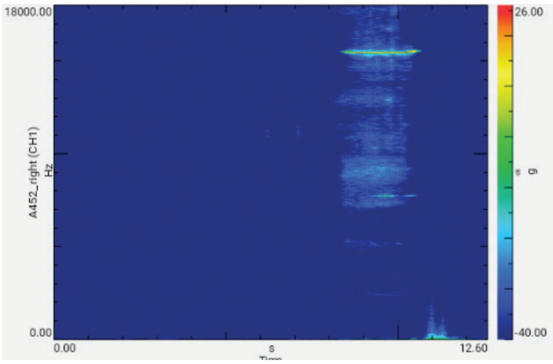
Most contributed brake parts to the HFS noise are from interaction of brake disc and pads [12]. The carmaker has requested for investigating on the customer vehicle about the squeal noise. The noise mechanism is needed to identify and to remedy based on brake disc modification. The HFS noise has been reproduced with the bearing-type Dynamometer. The similar brake condition during the vehicle test is applied on the test. The test data is to be compared with those from the numerical technique called CEA such that the noise-generating mechanisms can be identified and confirmed. Then the brake disc will be redesigned to avoid the mode coupling/locking or to increase the brake system stability to the HFS occurrence. The followings are the engineering tools used in the study and the preliminary results from investigation.

2.1 *On-Vehicle testing*

The vehicle with the claimed brake system is instrumented with accelerometers, Bruel & Kjaer type 4397 on the calipers for both left and right front corners, where the brake disc and brake pads vibrations are transferred to the measurement point. The 1/2 inch microphone, Bruel & Kjaer type 4189, is installed at the left of the driver ear. The vehicle under test is the 3200 cc engine capacity equipped with the 16 inch ventilated brake disc with the one-piston floating caliper. The on-vehicle test schemes for noise reproduction is performed with several driving conditions ranged from initial braking vehicle speed from 30 km/h to 80 km/h and braking deceleration between 0.3g's to 0.5g's. The vibration signal is captured by setting the frequency range of interest of 25 kHz. Figure 1 showed the spectrum and the spectrogram of the detected brake squeal event. The high acceleration and squeal noise occurred at intervals 8.5–10 s with the vibration level more than 20 g's, probably the squeal mechanism [13]. The squeal is occurred at the low vehicle speed (25–30 km/h) and low deceleration (0.3 g's). In the



(a)



(b)

Figure 1: (a) Spectrum plot of the caliper vibration during brake event from both left (red) and right (blue) hand side of the front brake system and (b) spectrogram for the right brake system.

spectrum and spectrogram plots, the dominant peaks are at 15.73 kHz and 7.86 kHz, respectively, where the first frequency is double of the latter. The same squeal condition during the vehicle test is confirmed with the laboratory test with, the bearing type Dynamometer tester followed to the SAE J2521, test standard.

2.2 *Dynamometer testing*

The claimed brake disc, brake pads, and caliper system are used in the bearing-type Dynamometer testing. The further parameters of interest are consisted of braking pressure and friction coefficient where can be obtained from the controller unit of the tester. The matrix of testing and method of accelerometer and microphone installation, are followed to SAE J2521.

The test results have indicated the occurrence of high-frequency squeal noise at 15,350 Hz under braking pressure of 3 MPa, simulated vehicle speed around 30 km/h, and friction coefficient around 0.3. These braking conditions are later used as the nominal reference conditions for the numerical experiments with the CEA module. The braking temperature of 100°C is not included in the simulation due to the simulation software limitation.

2.3 *Modal testing and analysis*

Frequently, modal testing and analysis is widely used to obtain the dynamics characteristics of the brake components as a single part or the assembly module in the vehicle. As shown by many researchers, it is difficult to relate only the component mode of vibrations to the brake squeal under operating conditions. Most of them have used the 3-D scanning laser vibrometer to obtain such the operating characteristics. In this study, the design requirement is to improve the stability of the brake disc to squeal occurrence such that the numerical experiments need to be set up for comparing with the original design of the brake disc. The modal testing and analysis are performed to extract the modal data from the existing prototype such that the FEM can be updated with the testing data.

The modal data comprised of natural frequency, mode shape, and damping ratio, are extracted from the testing data. The brake disc and brake pads are supported by the soft sponge to simulate such a free-free boundary condition and minimize the additional damping to the components, see Figure 2. The impact hammer with hard tip Bruel & Kjaer type 8204 is used for excitation. The three accelerometers Bruel & Kjaer type 4397 are installed on the brake disc for measuring the Frequency Response Functions (FRFs), divided into out-of-plane modes (normal direction to brake disc friction surface), in-plane radial modes (radial direction of the brake disc) and in-plane circumferential modes. These sensors are mounted on the brake disc part by adhesive glue as shown in Figure 2. To describe the brake disc vibration mode shapes within the frequency range 4–16 kHz, the brake disc dimension is divided into 168 points with the 14 nodal diameter (14ND) lines (28 portions along the disc perimeter).The brake pads, an accelerometer sensor is installed in normal direction to the back-plate surface as shown in Figure 2. The 90 measurement points are defined on the back-plate surface of the brake pads with the same portion

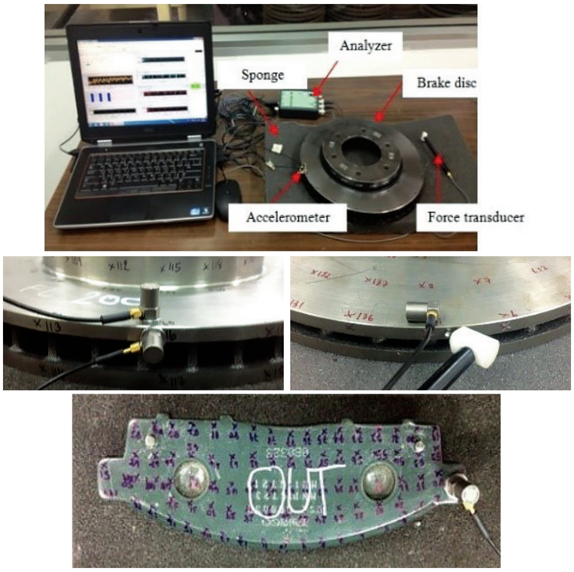


Figure 2: The arrangement of modal testing and the excitation and measurement points of the brake disc and brake pads.

along to the brake disc. The Bruel & Kjaer analyzer type Proton+ is used for collecting the FRFs which are processed with the ME-Scope modal analysis software to examine the mode shapes. The results will be discussed on the next section.

2.4 *Complex Eigenvalue Analysis (CEA) method for stability analysis*

Nowadays, CEA has become a common and widely used tool for brake designers to predict the stability of brake system. Compared to the normal modal analysis where the FEM is constructed as the symmetry system matrices, the CEA has included the friction dynamics where the friction forces and nonlinear contact stiffness during due to brake pressure stepping [14], can be included in the simulation model, resulting in the asymmetry system matrices.

To revise the concept of analysis, Hoffmann *et al*. [15] have presented a simplified single mass model with two degree of freedom, in-plane (x) and out-of-plane (y) directions (see Figure 3). The k_1 and k_2 are linear spring constants for supporting the single mass, m and k_3 is the normal contact stiffness between the mass and the sliding surface or representing as the brake pad extensional stiffness. A coulomb friction

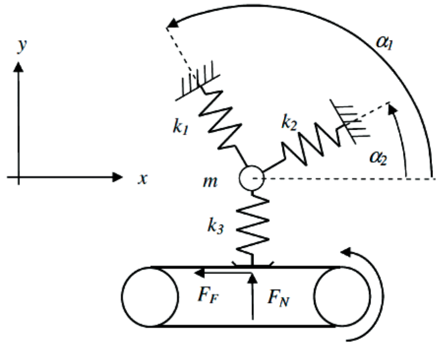


Figure 3: Two degree of freedom analyzed by Hoffman, *et al*. [15].

force, F_F, is applied with constant friction coefficient, μ. Then the system equation of motion is formed as Equation (1).

$$\begin{bmatrix} m & 0 \\ 0 & m \end{bmatrix}\begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix} + \begin{bmatrix} k_{11} & k_{12}-\mu k_3 \\ k_{21} & k_{22} \end{bmatrix}\begin{Bmatrix} x \\ y \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix}$$

$$\text{or } \boldsymbol{M}\ddot{\boldsymbol{q}} + \boldsymbol{K}\boldsymbol{q} = \boldsymbol{0} \quad (1)$$

Where $\boldsymbol{M}$ is the mass matrix, $\boldsymbol{K}$ is an asymmetric stiffness matrix as described above and $\boldsymbol{q}$ is a displacement vector. This results in asymmetry stiffness matrix due to the introduction of friction, where the eigenvalues of the system equation can result in merging of two real modes to two complex conjugate modes for some specific value of μ.

The negative damping coefficients of the complex eigenvalues is used for indicating instability of the brake system vibration, to compare with the brake squeal occurrence in the Dynamometer matrix test. The system equation [16] of the brake disc and brake pads FEM including damping matrix $\boldsymbol{C}$, can be written as

$$(\lambda^2 \boldsymbol{M} + \lambda \boldsymbol{C} + \boldsymbol{K})\boldsymbol{\varphi} = 0 \quad (2)$$

Where $\boldsymbol{C}$ is a damping matrix that consisted of material damping contribution and $\boldsymbol{\varphi}$ is eigenvector or mode shapes at a specific value λ, a complex eigenvalue that consist of real and imaginary parts. For damped structure, the eigenvalue pair of mode i,

$$\lambda_{i_{1,2}} = \alpha_i \pm j\omega_{d,i} \quad (3)$$

Where α_i is the real part, $\omega_{d,i}$ is the imaginary part or damped natural frequency, and j the imaginary notation. The α_i is associated with the damping ratio, ζ in term of $-\zeta\omega_{ni}$ where ω_{ni} is the undamped natural frequency at mode i. The negative value of the real part or positive damping ratio will ensure that the free response of the system will decay or stable. In contrast, the positive value of the real part or negative damping ratio (also damping coefficient) results in system response instability such that the vibration amplitude will radiate the surrounding air as the squeal noise.

The system Equation (2) then can be considered as two methods of analysis for the brake disc and pad assembly. Firstly the normal mode analysis is to understand the excitation mode from the disc in-plane motion and the response mode from the disc out-of-plane motion. Secondly, the complex mode analysis, where the system Equation (3) incorporated with asymmetry stiffness matrix due to contact friction between brake disc and pad, is to understand the stability of the system.

Complex Eigenvalue Analysis or Brake Squeal Module of the commercial Finite Element software is commonly performed by 4 steps [17] as follows: nonlinear static analysis for applying brake pressure, nonlinear static analysis to impose rotational speed on the brake disc, normal mode analysis to extract the system characteristics and complex eigenvalue analysis that incorporates the effect of friction coupling. The contact friction between the brake disc and pad assembly depends to the applied contact pressure and sliding velocity. Equation (4) provides the contribution of surface friction to the damping and stiffness matrices of the system equation as follows:

$$d\tau_i = \left(\mu + \frac{\partial \mu}{\partial p} p\right) n_i dp + \frac{\partial \mu}{\partial \bar{\dot{\gamma}}} p n_i n_j d\dot{\gamma}_j + \frac{\mu p}{\bar{\dot{\gamma}}}(\delta_{ij} - n_i n_j) d\dot{\gamma}_j \tag{4}$$

where

τ_i	tangential stress
p	contact pressure
$\dot{\gamma}$	relative sliding velocity
$\mu = f(p, \dot{\gamma})$	friction coefficient
$n_i = \frac{\dot{\gamma}_i}{\dot{\gamma}}$	normalized slip direction $i = 1, 2$

Radial and circumferential directions

$\bar{\dot{\gamma}} = \sqrt{\dot{\gamma}_1 + \dot{\gamma}_2}$ equivalent sliding velocity.

There are three terms in the right-hand side of the Equation (4) contributed to the asymmetry stiffness matrix, the damping matrix due to dependent of friction coefficient in direction parallel to the contact surface, and the damping matrix in direction perpendicular to the contact surface, respectively.

3 Analysis of the Mode Coupling Mechanism

In this section, the study of brake disc in-plane and out-of-plane mode coupling is investigated to assure which is the coupling mode reflected to the claimed brake system. Then the stability improvement of the brake system based on brake disc design is introduced.

3.1 *Preparation of the finite element model and simulation parameters*

In this study, the brake squeal module within the commercial software named SimXpert 2011, is used for the CEA. The brake disc and brake pads are discretized or meshed with tetrahedral elements where the brake disc and brake pads thickness are divided at least 2 layers. The model has 65,275 elements and 92,000 degrees of freedom (DOF). The FEM of the brake disc and the brake pads, are validated and updated for its dynamic material properties with modal testing data of the existing brake disc-pad prototype. The FRF measurements are identified the natural frequencies and their mode shapes using the ME-Scope software. The tuned FEM is referred to the material tuning process for its mass properties and dynamic Young's modulus. The identified vibration modes are comprised of the out-of-plane mode defined in term of the number of the nodal diameter lines (ND), the in-plane modes in radial (RI) and in circumferential (CI) directions. The natural frequency prediction with the tuned FEM are within 3 percentage deviated from the test data for the modes ranged from 4–16 kHz of the brake disc out-of-plane and in-plane modes and the pad out-of-plane modes.

3.2 *Simulation results and mode coupling analysis*

In study of the several claimed brake systems, the braking conditions obtained from the Dynamometer test matrix are about 0.3 to 0.5 friction coefficients, 10 to 30 km/h, and 3 to 5 MPa. For numerical experiment study, the braking conditions have been extended as

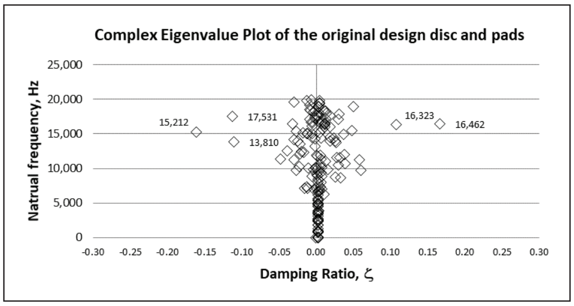
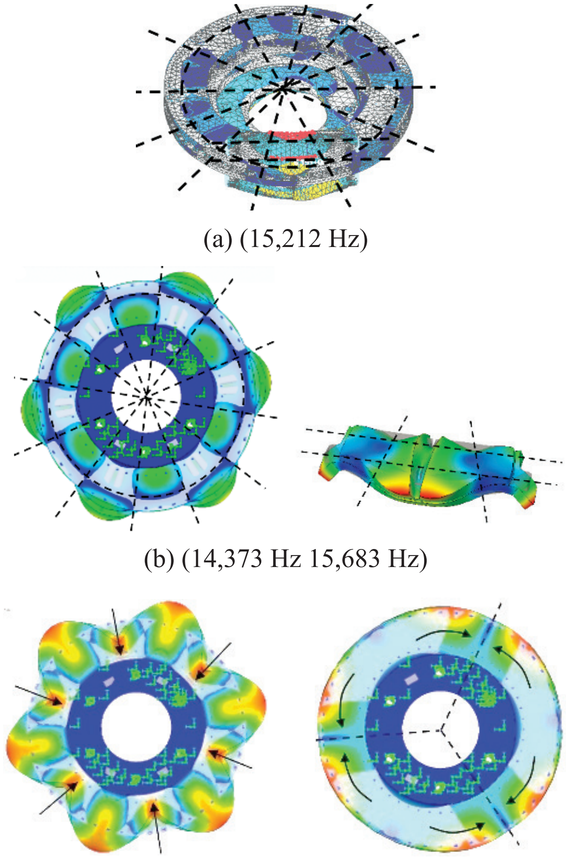


Figure 4: Complex eigenvalue plot of the original design disc and pads with 0.5 friction coefficient and 5 MPa braking pressure.

a set of parameters in the CEA, the contact friction coefficient between 0.1 to 0.6 with 0.1 increment and the braking pressure between 1 to 5 MPa with 1 MPa increment. In all cases, the brake disc sliding velocity is assigned to rotate constantly as 6.28 rad/sec, angular velocity about its centerline axis or equivalent to 10 km/h. To provide marginally stability for all modes, the FEM is incorporated with structural damping of 0.005.

Among results of the CEA study, the original design brake disc-pad FEM has indicated clearly the unstable behavior or the solution with the negative damping coefficient at the conditions of 0.5 contact friction coefficient and 5 MPa braking pressure. The condition is along with the extracted condition from the Dynamometer test.

Figure 4 showed the complex eigenvalue plot such that the distinct negative damping ratio occurred at the mode frequencies of 15,212 Hz, 17,531 Hz, and 13,810 Hz, respectively. The most unstable mode at 15,212 Hz is close to the field test data (Figure 1) where its corresponding mode shape is shown in Figure 5(a). The assembly shape is shown as mode-locking between the brake disc with fixed support at the hat surface and the brake pads with fixed at the edges except longitudinal direction such that there are the circumferential nodal lines occurred in both components. This mode is corresponding to the out-of-plane normal modes of the brake disc and brake pads as shown in Figure 5(b). The brake disc mode with fixed support at the hat surface, is shown the mode shape with 6 nodal diameter lines (6ND) and 1 nodal circumferential line (1NC (or (6ND+1NC mode). The brake pads mode with free support, is shown as the mode with 2 nodal diameter lines (transverse lines)



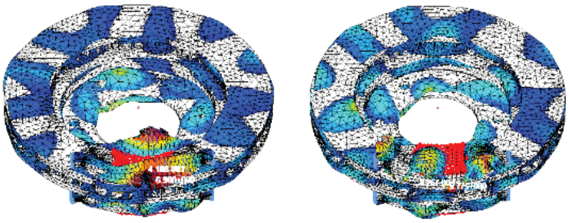
(a) (15,212 Hz)

(b) (14,373 Hz 15,683 Hz)

(c) (15,057 Hz 15,741 Hz)

Figure 5: (a) Complex Eigenvalue Analysis: unstable mode shape at frequency of 15,212 Hz. (b) Normal Mode Analysis: Out-of-plane modes (Left) Disc mode: Mode 6ND+1NC at the frequency of 14,373 Hz (Right) Pad mode: at the frequency of 15,683 Hz. (c) Normal Mode Analysis: (In-plane modes) (Left) the 6th radial in-plane mode, 6RI (15,057 Hz) and (Right) The 3rd circumferential in-plane mode, 3CI (15,741Hz).

and 2 nodal circumferential lines (longitudinal lines). Those modes are the out-of-plane motion where the air around the disc can be radiated as squeal noise. However, there must have the excitation mode or the in-plane modes coupling to the out-of-plane mode near the unstable frequency. Figure 5(c) showed the in-plane normal modes related to the nodal lines and coupling in frequency with the out-of-plane modes. The 6RI mode is the in-plane motion mode with 6 portions of altered radial motion. The 3CI mode is the in-plane motion mode with 3 portions of altered circumferential



(a) (17,531 Hz) (b) (13,810 Hz)

Figure 6: Complex Mode Analysis: subsequent unstable modes at the frequency of (a) 17,531 Hz and (b) 13,810 Hz.

motion. From the field test result (Figure 1), there is also the squeal noise frequency around 7–8 kHz. These noises seem to be the subharmonic frequencies of the 15–16 kHz modes. In normal mode analysis, the mode is the 6ND out-of-plane.

Compared to Figure 5(a), the subsequent unstable modes for the lower negative damping mode at 17,531 Hz and 13,810 Hz as shown in Figure 6. For the CEA mode at 17,531 Hz, the brake pads are in twisting motion where the unsymmetrical brake pads motion is suppressed with the contact area pressure. For the CEA mode at 13,810 Hz, also the unsymmetrical brake pads motion is in bending. From above unstable CEA modes, the brake disc mode shapes are not distinct to the ND normal mode. From this evidence, this study will redesign the brake disc to avoid the mode-locking pattern, the mode shape of the brake disc and the brake pads not the similar shape at the coupling frequency of both component, and to separate the brake disc out-of-plane and in-plane modes such the coupling mechanism to the excitation minimized.

4 Improvement of the Disc-pads Stability to Squeal Mechanism

The original design is come with the same disc thickness of 8 mm on both cheeks and friction annular surface area. Initially, the nine set of in-board and out-board cheek thickness pair are selected and studied for their brake performance before squeal study. The change in their normal mode frequencies are investigated as a primarily study. The CEA is then performed to evaluate their stability. Finally, the two set of modification will be manufactured as the prototypes to verify with the vehicle field test followed to SAE J2526 test standard.

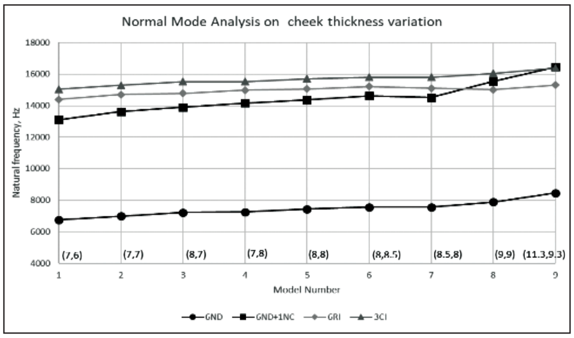
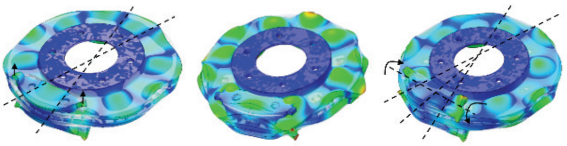


Figure 7: Normal mode analysis of brake disc: cheek thickness variation.

4.1 *Modification of disc cheek thickness to frequency and shape decoupling*

The purpose of the brake disc modification was prevention of mode coupling/locking brake squeal noise at frequency around 15 kHz. From the mode coupling/locking described in the previous section, the brake disc cheek thickness will be redesigned to decouple the out-of-plane modes from the in-plane modes and also the brake pads mode. The variation of brake disc cheek thickness, a pair of in-board and out-board sides, is considered. The bending of cheek in out-of-plane motion, is contributed most from the cheek flexural rigidity. For the nine set of the modified thickness, the model number 1 to 9 are a pair of (7,6), (7,7), (8,7), (7,8), (8,8), (8,8.5), (8.5,8), (9,9), (11.3,9.3), dimension in mm, for in-board and out-board, respectively. The original design is of the cheek thickness, model number 5 or (8,8). The model number 9 or (11.3,9.3) is from the existing brake disc prototype available from another design purpose. It may not proper for weight and cost reduction strategy in design. The minimum cheek thickness is limited to the strength design. From Figure 7, the normal modes of the brake disc involved to the out-of-plane motion (dark lines) are most affected the increase of thickness, which the natural frequencies are increased. This change also effects to the mode coupling in frequency between the in-plane (6RI mode) excitation and the out-of-plane (6ND + 1NC) response.

To study the normal mode-locking shape pattern, the assembly brake disc-pads model with fixed support at the hub, is studied as shown in Figure 8 where the 6ND + 1NC mode shape is examined. It found that



(a) (b) (c)

Figure 8: Assembly normal mode analysis of modified brake discs (a) Out-of-phase Pad motion (model number 2 to 7). (b) No pad motion in the upper part (model number 8). (c) Pad motion in 2nd bending + torsion (model number 9).

the original design (model number 5) has the two pad mode shapes, out-of-phase each other Figure 8(a) and the same for model number 2 to 7. However, there is no the 6ND + 1NC mode for the model number 1. The brake pad motion at the 6ND+1NC mode has been altered for the Model Number 8 and 9 as shown in Figure 8(b) and (c), respectively. The 6ND+1NC modes of the model number 8 and 9, are higher frequency than the 6RI mode compared to the model number 1 to 7 (Figure 7). Noted that only the model number 1 and 2 have the additional mode in the middle between the 6ND+1NC mode and the 6RI mode (the middle mode not shown in the trend plot of Figure 7).

In conclusion of the assembly normal mode analysis, the model number 1 and 2 are provided the separation of the brake disc out-of-plane and in-plane modes. The model number 8 and 9 are also provided the unlocking pattern between the pad and the brake disc at the squeal frequency.

The CEA is performed for all model number to study about their stability. It found that the stable models are the model number 1 or (7,6) and model number 9 or (11.3,9.3).

Figure 9 shows the complex eigenvalue plots for all model number. The CEA mode shapes at the 6ND+1NC mode of both model number 1 and 9 are shown in Figure 10. There is no brake pads motion for the 6ND + 1NC mode shape of model number 1 or no coupling excitation between brake disc in-plane and out-of-plane modes. The model number 9 is no brake

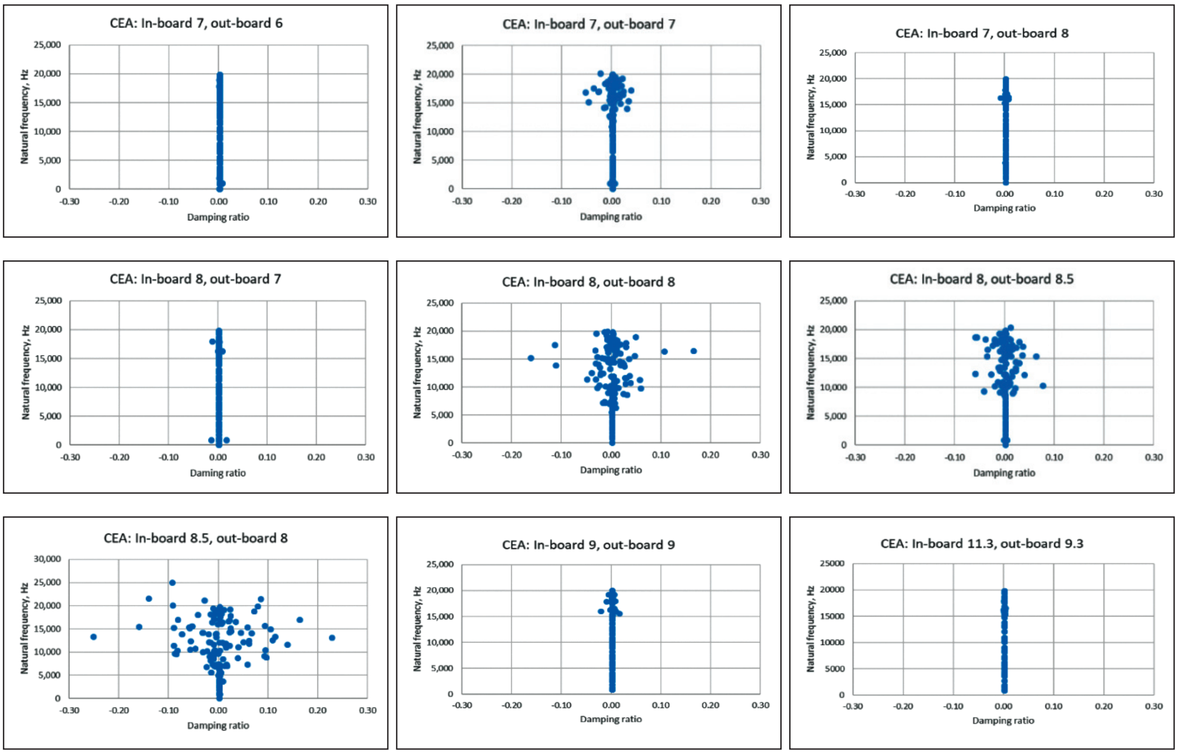
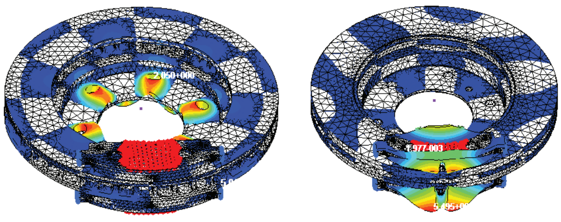


Figure 9: Complex Eigenvalue Analysis: a complex eigenvalue plot for 9 modified models.



(a) (b)

Figure 10: Complex Eigenvalue Analysis (CEA) of 6ND+1NC mode for (a) model number 1: 14,984 Hz and (b) model number 9: 15,939 Hz.

disc circumferential nodal line (1NC) where the brake pads are, or the mode shape pattern is not locking between brake pads and brake disc.

4.2 *Verification of the modified prototype with the vehicle field test*

The model number 1 is constructed as the prototype used for verification and compared with the existing prototypes, the model number 5 or the original design, and the model number 9. All brake disc prototypes are the new disc which are simulated experimentally as the used disc with the Dynamometer test matrix according to SAE J2521. Then, the vehicle field test matrix followed to SAE J2625, is performed and the spectrogram of brake caliper vibration as a generation of noise shown in Figure 11. The horizontal axis is the time scale from 0.2 to 4 s, the vertical axis is the frequency axis ranged from 0 to 18,000 Hz, and the color bar is ranged from 0 g's to 3 g's acceleration vibration level. These results are obtained from one of thirty test conditions of the test matrix, with pad temperature of 150°C, vehicle speed of 40 km/h, and vehicle deceleration of 0.3 g's. It found that the modified model number 1 and 9 are more stable to squeal noise generation. This is agreed with the CEA results.

Furthermore, Figure 12 provides the negative damping coefficients plot for extended pairs of the cheek thickness. The 49 pairs including the 9 pairs mentioned before, are displayed for the sensitivity analysis to squeal noise generation. The maximum negative damping coefficient or unstable region occurs at a pair (8.5,8). With this mapping, the brake system designer can use as the guideline for selecting the pair of cheek thickness that more robustness (low value of the coefficient) to the squeal noise generation.

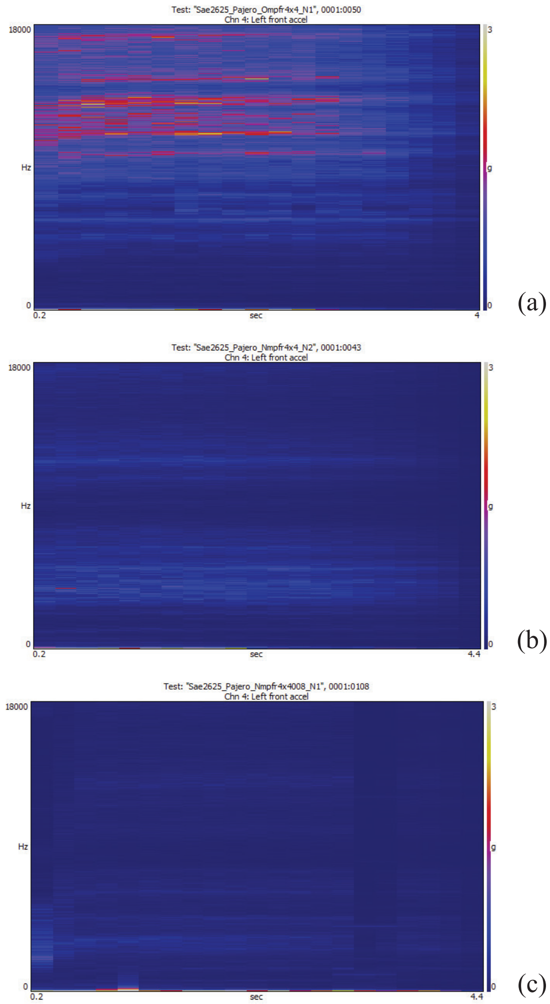


Figure 11: Spectrogram during field test: (a) Original design, with cheek thickness (8,8) mm (b) model number 1, with cheek thickness (7,6) mm (c) model number 9, with cheek thickness (11.3,9.3) mm.

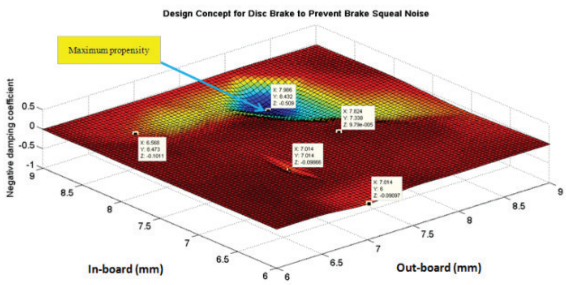


Figure 12: Sensitivity analysis of high frequency brake squeal due to 49 pairs of cheek thickness variation.

5 Conclusions

This study is to redesign the brake disc of specific car model to overcome the high frequency brake squeal noise. The normal mode analysis and CEA are performed to understand the squeal mechanisms, where the FEM of the brake disc and brake pads are well formulated and updated with the modal testing results of the existing components. This study provides the depth analysis of the way to overcome the coupling between the brake disc out-of-plane and in-plane modes where the latter is the excitation and the former is the response as appeared in the Model Number 1. The brake disc modification is to change only the cheek thickness. The nine sets of symmetry and asymmetry pairs of the in-board and out-board cheek thickness are studied. The method is presented to improve the system stability by avoiding the mode shape locking between the brake disc and the brake pads as appeared in the model number 9. The prototypes of the original brake disc and two of the modified brake discs are used to verify with the vehicle field test. The test matrix followed to SAE J2526 is performed and found that the modified brake discs are more stable to the generation of squeal mechanism, and well agreed with CEA results.

Acknowledgement

Authors would like to thank the National Metal and Materials Technology Center (MTEC), Thailand and Somboon Malleable Iron Industrial Co., Ltd. for co-funding to this research study.

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