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Understanding the Conceptual Framework of Pad Structural Modification for Disc Brake Squeal Noise Suppression Based on System's Dynamic Response

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ABSTRACT – *Brake squeal noise solutions have been an essential subject in the automotive industry. It is among the most prevalent issues with car braking systems. For decades, the braking force that relies on the frictional force provided by the contact between the friction pad and rotor is still being used, thus inheriting the same fundamental issue of friction-induced brake squeal noise. Many proposed solutions and methods have been available from many research papers and articles. This paper attempts to summarize the most relevant literature on existing studies and theories to help understand the conceptual framework of the brake pad structural modification methods to reduce brake squeal noise. The review covers the underlying principles of the mode-coupling theory, methodology, and the technique's effectiveness on vibrational characteristics and behavior analysis. A visual representation of a conceptual framework is finally presented at the end of this paper. The system's dynamics response study is found adequate to explain the relationship of structural pad modification towards the brake squeal noise propensity. However, more detailed reviews are required from the study of the system's vibrational energy, another major factor for accurate prediction to thoroughly understand the fugitive phenomena of the disc brake squeal noise.*

KEYWORDS: Brake squeal, noise suppression, conceptual framework, review, dynamic response, mode-coupling, structural modification

1. INTRODUCTION

The importance of silent braking has become apparent by looking at the trend of the European Union's (EU) new vehicle requirement, which regulates the noise level limit to be reduced continuously year by year since 1978, from 82 dB (A) down to 72 dB (A) in 2016, and latest to 68 dB (A) by 2026 (VCA, 2022). Furthermore, with the popularity of electric vehicles (EVs), cabin quietness has become increasingly necessary because as it radiates much less noise than Internal Combustion Engine (ICE) cars, other external noises have become more easily audible. Therefore, it would dramatically change the quality association and acceptance level of brake noise perceived by drivers, passengers, and people in the surroundings.

Thompson (2011) illustrates a wide range of brake Noise, Vibration, and Harshness (NVH) occurrences over a broad spectrum of frequencies, as depicted in Figure 1. Several terminologies are found famously in many works of literature, such as squeal, groan, chatter, judder, moan, hum, wire-brush, and squeak. Squeal noise is the most prevalent as the driver can hear it easily within the audible range between 1,000 and 18,000 Hz as a pure single-tone, high-pitched annoying, and irritant noise (Day, 2014). Its amplitude can be measured between 60dB to 120dB within a meter distance (Parra et al., 2010).

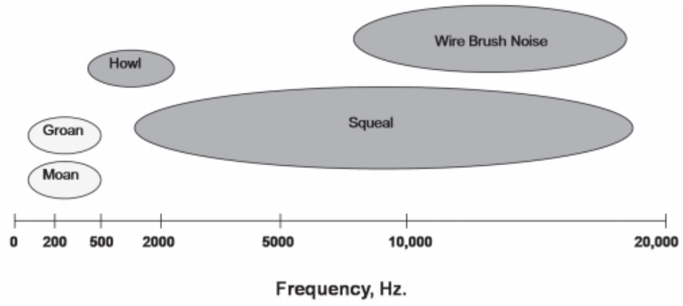


FIGURE 1: Category of brake noise based on frequency range (Thompson, 2011)

For many years, the quick fix solution suggested in the customers' domain for squeal noise is simply changing the brake pads based on a common understanding that the friction interface significantly contributes to the general brake noise generation. Unfortunately, this countermeasure might temporarily solve the problem but could jeopardize the level of performance and introduce another noise under different conditions.

Many have accepted that frictional-induced dynamic instabilities cause the most brake noise from the brake components (Chen, 2007). Watany and Saad (1999) found a good correlation between the vibration resonance frequency measured for the disc brake components (pad and caliper) and the measured squeal noise related to the sound pressure level. In the open literature, numerous methods or strategies have been proposed to reduce or eliminate brake squeals. Researchers employed various methodologies to understand the squeal noise phenomena experimentally or theoretically using analytical or numerical methods.

There are two approaches to studies regarding the brake squeal mechanism: the excitation source and the system response (Day, 2014). The study of the excitation source focuses on the origin of vibrating signal input, while the latter focuses on the system response that amplifies the vibration. Both studies direct different approaches to suppress the noise. One is tackling the root cause, and another is tackling the effect. Many researchers have focused on the system response in recent years, which is easier to tackle than the fugitive nature of the source, which is more complicated because of the constant changing of the surface topography of the brake pad lining, as highlighted by Bakar et al. (2010). It is also because of inherent non-linearities, uncertainties in the material properties, contact and boundary conditions, and system complexity.

Various suggestions were proposed, including well-known techniques such as structural modifications of the brake disc, caliper, and brake pad, insertion/removal of pad shims (constrained layer damping material), clips, grease, and a new formulation of friction materials development. Chen (2009) provides guidelines for suppressing and eliminating the prevalence of squealing. Researchers adopt three fundamental approaches to reduce squeal: damping optimization, impulsive excitation minimization, and mode coupling reduction. However, Jeong et al. (2016) presented results that installing the damping kits may unduly increase the softness in the brake system, leading to underperforming braking perceived quality and concerns about the shim's effectiveness in the long term without proper consideration in design. This is why structural modification gained more popularity among all other methodologies.

Researchers have studied the effectiveness of modifying the brake pad surface profile toward squeal propensity (Wang et al., 2016). Different kinds of structural modification in trials were investigated, such as grooves, chamfers, or combinations. While structural change might affect the tribology and vibration behavior of the system, it is a challenge to reduce or eliminate brake squeal noise while not compromising the vehicle's braking performance. Thus, it led to other studies, such as those conducted by Dąbrowski et al. (2018) and many others, to investigate the impact of the brake pad surface treatment and finishing on frictional performance, tribology, and vibration.

This paper aims to summarize the most relevant literature on existing studies and theories to help understand the conceptual framework of the brake pad structural modification methods to reduce brake squeal noise. Various techniques are reviewed in the following sections, including experimental modal analysis, finite element analysis, structural modification, and experimental diagnostic and validation.

2. MODE COUPLING INSTABILITY MECHANISM

Most researchers agreed it is because of the friction via the rotating motion between the pad and disc, inducing the dynamic instability in the system that contributes to the self-excited and self-sustained significant forcing function in the brake system. Stick-slip, sprag-slip, mode coupling instability, and the negative slope of the relationship between frictional force and relative sliding speed are a few of the fundamental theories on squeal mechanisms that are extensively discussed and reviewed by Kinkaid et al. (2003), Ouyang et al. (2005), and Chen (2009). They highlighted that the theories should be combined or unified to understand the squeal phenomena better. In the earliest concept, squeal was believed to be due to the negative slope relationship of dynamic friction coefficient (μ) variation against sliding velocity (v) at the frictional interface. Du and Wang (2017) believed that the negative slope friction effect is primarily associated with in-plane vibrations and plays a role in developing additional in-plane unstable modes. The 'stick-slip' mechanism is a periodic excitation that results from the higher static μ than the dynamic μ. It was later proved that a constant friction coefficient could also generate squeal. While the mode coupling hypothesis has become more general and unanimous among researchers, the topic of mode coupling will be more focused on for further discussion. It is based on the theory of the critical basic assumption that the contact modeling of the variable normal force is a constant or no necessary change of coefficient of friction. Thus, it presents the primary concept of understanding the system response related to mode coupling instability and the mechanism based on these fundamental conditions.

Mode coupling, or several names used in literature such as binary flutter, non-conservative displacement dependent forces, or mode lock-in, is a phenomenon due to coupled two adjacent vibration modes in parts experiencing friction. It can occur by the coupling between sub-components, which are the rotor's and brake pad's mode vibration (Geometrically Induced Instability). It can also occur by the coupling of the individual component's different modes, such as the rotor's in-plane and out-of-plane modes' alignment (Binary Flutter Instability) (Chen et al., 2003). Jarvis & Mills (1963) identified the contribution of geometric coupling, where the normal vibrational modes of the contacted pad and disc, coupled mathematically by a continuity equation over the entire surface, led to non-linear oscillations. The natural frequencies of two vibrational modes approach each other until they form the stable and unstable modes. The unstable mode oscillates in a limit cycle and triggers a brake squeal, while the stable mode decays due to damping. This instability can create a significant vibration response when under suitable conditions because each component coalesces adjacent vibration modes, including the caliper and other connected components.

The study by Triches et al. (2004) confirmed the geometrical disc-pad mode coupling. The brake components and geometry modal analysis conducted found that a system mode was created by the coupling of the brake pad's 3rd bending mode and the disc's 6th bending mode. The pad's 3rd bending mode appears at 6,650 Hz, while the 6th bending mode of the disc appears at 7,320 Hz. The wavelength approximated is 100mm for the pad and 112mm for the disc. Therefore, those two bending modes create mode frequency coupling and shape locking mechanism, as shown in Figure 2 below.

As mentioned previously, the mode coupling mechanism can also occur on the symmetrical component, such as the symmetrical rotor, which holds two modes of the same order with a very similar frequency (Fieldhouse et al., 2004). The alignment and connection between the rotor's in-plane and out-of-plane mode vibration are the leading and intrinsic sources of disc brake squeals at high frequencies, according to Chen et al. (2002). However, this mechanism cannot cultivate the squeal on its own. To couple them together and generate an arousal force, there must be a precise amount of pressure, friction, and warmth. Therefore, Wagner et al. (2014) investigated how the braking disc's asymmetry (slots, uneven vane spacing, and weights) could separate the double natural frequencies.

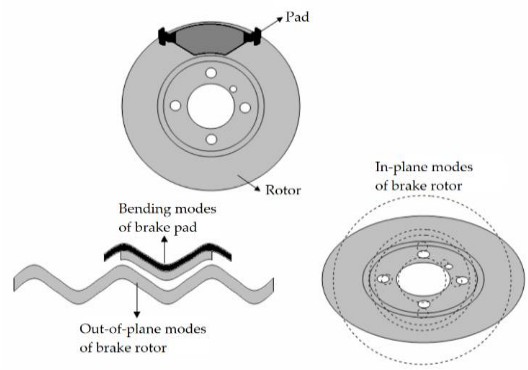


FIGURE 2: Mode coupling between the brake pad and disc (Triches et al., 2004)

3. ANALYTICAL MODEL OF MODE COUPLING DYNAMIC EQUATION

The theoretical modeling of a physical system results in the equation of motion that describes how it moves with time. It leads to the depiction of differential equations consisting of inertial force, damping force (energy dissipation), and elastic (restoring) force by a simple mass-spring-damper single degree of freedom (SDOF) system equation and schematic diagram Figure 3 below:

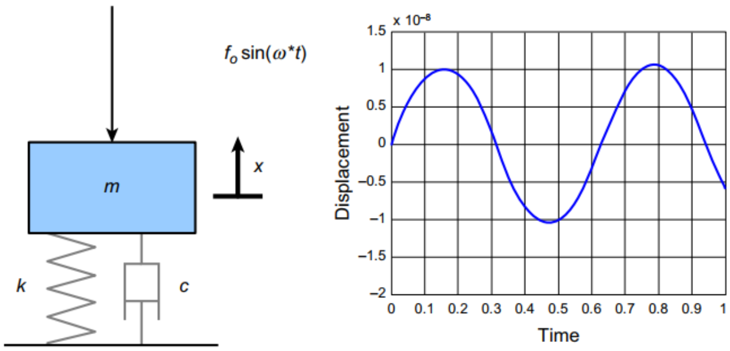


FIGURE 3: Mode SDOF damped forced vibration system and response (Day, 2014)

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = f(t) \tag{1}$$

where,

m = mass
c = damping coefficient
k = stiffness coefficient
$x(t)$ = mass displacement
$\dot{x}(t)$ = mass velocity
$\ddot{x}(t)$ = mass acceleration
$f(t)$ = applied force

Various researchers have introduced several analytical models to describe the dynamic characteristics and study the instability of a vehicle disc brake system concerning modal coupling mechanisms. It presented valuable insights into understanding the parameters and factors impacting the vibration behavior of the system resulting from the analysis of various models. The advancement started from a relatively simple model of single mass with single-degree-of-freedom (SDOF) models. Then, it expanded towards more complex multiple mass, lumped parameter multiple degrees of freedom (MDOF) models.

One of the most reviewed modes coupling minimal models is by Hoffmann et al. (2002). They proposed 2-DOFs to simulate the mode coupling instability's underlying physical mechanism of in-plane and out-of-plane modes from an intuitive perspective by the simplified notation of x-axis and y-axis displacement, respectively, as seen in Figure 4 below.

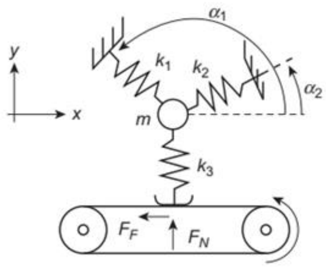


FIGURE 4: Mode coupling model by Hoffman et al. (Hoffmann et al., 2002)

A mass block depicted as a particle m is pressed by a conveyor belt moving at a constant speed with a constant normal force F_N. Two coupled linear springs with stiffnesses k_1 and k_2, and angles α_1 and α_2 each supports the block. With a Coulomb-type friction force F_F and a constant coefficient µ, another linear spring k_3 works tangentially and represents the normal contact stiffness between the mass and the sliding surface. As a result, the system's equation of motion is given as follows:

$$\begin{bmatrix} m & 0 \\ 0 & m \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix} + \begin{bmatrix} k_{11} & k_{12} \\ k_{21} & k_{22} \end{bmatrix} \begin{Bmatrix} x \\ y \end{Bmatrix} = \begin{Bmatrix} F_F \\ F_N \end{Bmatrix} \tag{2}$$

with,

$$k_{11} = k_1 cos^2\alpha_1 + k_2 cos^2\alpha_2,$$
$$k_{12} = k_{21} = k_1 sin\alpha_1 cos\alpha_1 + k_2 sin\alpha_2 cos\alpha_2,$$
$$k_{22} = k_1 sin^2\alpha_1 + k_2 sin^2\alpha_2 + k_3,$$

The frictional force can be approximated or linearized as $F_F = \mu k_3 y$, which leads to a homogeneous system with an unstable stiffness matrix equation system. The steady slip state small perturbations are negligible and can be ignored.

$$\begin{bmatrix} m & 0 \\ 0 & m \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix} + \begin{bmatrix} k_{11} & k_{12} - \mu k_3 \\ k_{21} & k_{22} \end{bmatrix} \begin{Bmatrix} x \\ y \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix} \tag{3}$$

According to the theory, the friction force converts energy from the portion of the system that is sliding in friction into vibrational energy in the other parts of the system. As a result, the mass moves both vertically and horizontally in unison. When a structure's dynamic and contact properties create an asymmetric stiffness matrix, unstable mode coupling creates squeal noises as some adjacent eigenvalues and eigenvectors merge. This unstable mode coupling occurs with a specific range of contact stiffness and coefficient of friction values. According to Rusli and Okuma (2007), a more significant coefficient of friction enhances the likelihood that mode coupling would be induced in a structure. Hoffmann et al. (2002) claim that the friction force functions as a cross-coupling force that brings motion parallel to the contact surface and motion normal to it together and that the system's corresponding structural cross-coupling forces must be balanced for instability to begin to occur. Thus, a non-linear modeling approach could include this as a potential representation of a "source" model.

Von Wagner et al. (2007) reviewed previous minimal models while introducing a complementary new model. They commented that in many articles, the initiation of disc brake squeal is described as the instability of the trivial solution caused by non-conservative frictional forces even with a constant coefficient of friction. Therefore, a minimal model of disc brake squeal must contain at least two degrees of freedom. However, many mathematical models proposed based on the mode coupling principle cannot be directly used to analyze the generation of friction-induced vibration between two bodies. Furthermore, associating model parameters with a specific physical meaning for reality friction pairs is challenging. Therefore, Lyu et al. (2017) developed a specific three-degree-of-freedom dynamic model of the beam-on-disc system to simulate the vibration behavior.

4. EXPERIMENTAL MODAL ANALYSIS METHOD

Experimental modal analysis is a technique used to determine the vibrational properties of an actual structure or system where measurements are taken and analyzed to evaluate the vibrational modes of the system. Each vibration mode consists of three modal parameters or properties which together describe the system's dynamic characteristics: (a) modal frequency – resonant frequency of oscillation for the mode, usually expressed in Hertz (Hz); (b) modal damping – a measure of the dissipation of vibration energy or the rate at which the vibration decays, usually expressed as a percentage on the critical damping (the optimum damping level which sufficient to prevent free vibration); and (c) mode shape – deformation or displacement pattern of the structure for the mode, described by complex-valued displacements.

Experimental modal analysis can be done through an impact hammer test where modal parameters can be determined experimentally from frequency response function (FRF) plots. The frequency response function is a ratio of the system input (driving force) to the system output (acceleration at one or more points) of a linear system over a frequency range, as shown in Figure 5 below.

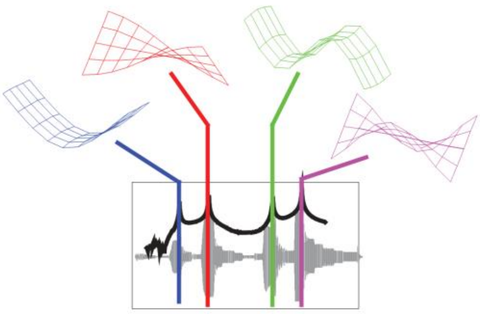


FIGURE 5: Frequency response function (Avitabile, 2017)

Mathematically, FRF can be represented by the following equation:

$$\mathrm{F_j}(\omega) \rightarrow \mathrm{H_{ij}}(\omega) \rightarrow \mathrm{X_i}(\omega) \tag{4}$$

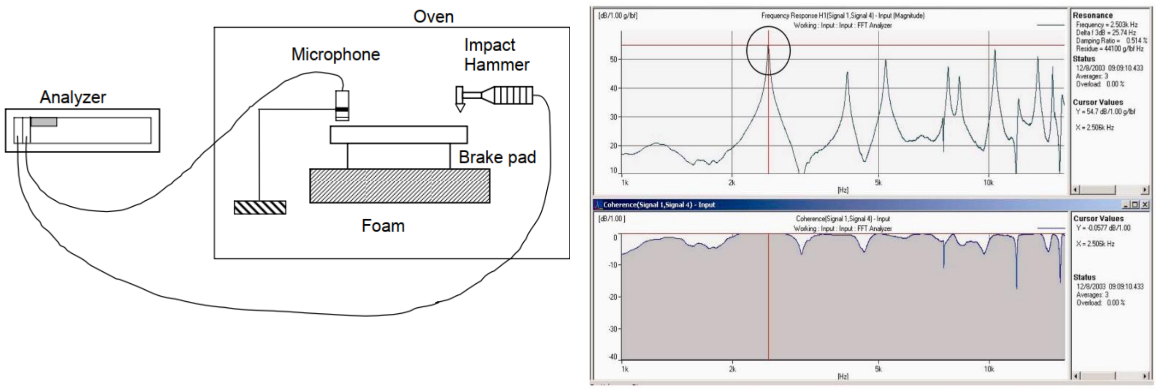
$$\mathrm{H_{ij}}(\omega) = \frac{\mathrm{X_i}(\omega)}{\mathrm{F_j}(\omega)} \tag{5}$$

where,

$\mathrm{H_{ij}}(\omega)$: is the frequency response function,
$\mathrm{F_j}(\omega)$: is the excitation (input) in the frequency domain,
$\mathrm{X_i}(\omega)$: is the response (output) in the frequency domain.

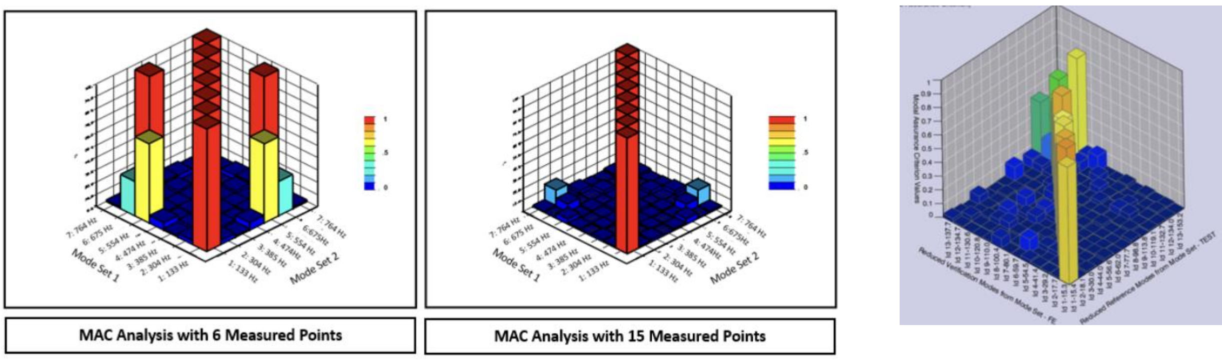
The FRF is represented as a function of frequency and is complex in form, having a real and imaginary component. The FRF given in the above equation is compliance since the numerator is measured in displacement units, and the dominator is a unit of force. SAE International provided guidelines for conducting Brake Pad & Disc experimental modal analysis named J2598 Automotive Disc Brake Pad Natural Frequency and Damping test (SAE, 2020) and J2933 Verification of Brake Rotor Modal Frequencies (SAE, 2011), respectively. The brake pad modal test procedure applies to vibration modes between 500Hz and 16kHz. The first three natural frequencies, $\mathrm{f_n}$ (n=1,2,3), and the accompanying loss factors, η are the parameters that are being measured. Figure 6 below is an example of a disc brake pad's FRF and Coherence.

The appropriate measurement location is determined using the Modal Assurance Criterion (MAC) procedure to ensure all mode shapes are correctly captured. This technique defines the degree of consistency and similarity between mode shape vectors. MAC is used to describe how similar the mode shapes are for a given mode pair using a scale of 0 to 1 (e.g., 0% to 100%). Everything on diagonally should be 100%, and all off-diagonals should be near zero. It is used in two stages. The first is ensuring that the accelerometer location is sufficient to uniquely identify all modes from FE modal analysis. Secondly is to evaluate how well FE simulation and Test models correlate. Figure 7 below shows MAC results after adding additional points and comparing the FE and Test Modes.



(a) (b)

FIGURE 6: (a) Brake pad's impact hammer setup; (b) Pad's FRF (top) and coherence (Bottom) (SAE, 2020)



(a) (b)

FIGURE 7: (a) MAC with a different number of measuring points; (b) Comparison between FEA vs. experimental results (Siemens DISW, 2019)

The pad often exhibits fewer vibration modes within the same frequency range than the disc. Additionally, the damping loss factor values associated with the pad vibration modes are higher than those associated with the rotor because the pad's friction material provides far more damping than the cast steel used in the disc (Triches et al., 2004). Consequently, disc modes tend to be the primary determinant of squeal frequency. During measurement, the brake pad to be tested must be isolated and supported on a foam pad as specified in SAE J2598 to prevent errors in damping measurements. In addition, it is essential to set the pre-trigger delay to capture the entire input force pulse. The complete response must be recorded on the response channel where windowing cannot be used. Some researchers have traditionally used an exponential decay window on the response channel. Although this can be useful in some studies, this measurement has shown that such windowing leads to errors (Thompson, 2011).

Since the dynamic magnification factor is inversely proportional to the damping level, the damping value is essential to the size of the quantified response at resonance. Calculating the modal frequency response throughout the half-power bandwidth or using the 3dB approach, as illustrated in Figure 8, is one way to confirm that the assumed modal damping ratio is accurate. It is implemented as a damped single-degree-of-freedom system under direct-force excitation. For lightly damped structures ($\zeta < 0.1$), the critical damping ratio ζ, the half-power bandwidth ($f_2 - f_1$), and the resonance frequency f_n, roughly follow this relationship:

$$\zeta = \frac{f_2 - f_1}{2f_n} < 0.1 \tag{6}$$

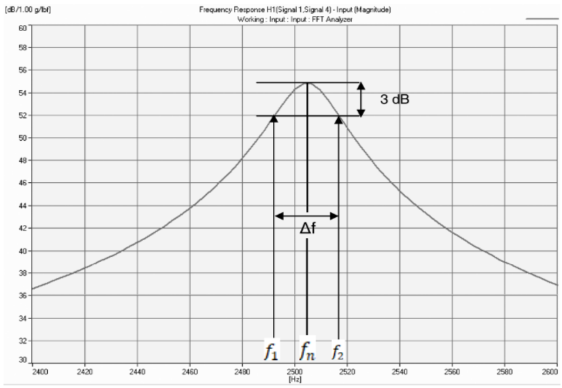


FIGURE 8: Half-power bandwidth damping (SAE, 2020)

These approximations are less precise when multiple modes are contributing to the response. They are nevertheless helpful in confirming that the damping values incorporated into the FE model align with the system's real damping. In addition to the commonly used accelerometer & hammer method, alternative methods are also available, such as the laser vibrometer using the hammer method or the non-contact shaker method. Chen et al. (2005) compared accuracy and highlighted each technique's main related factor.

5. FINITE ELEMENT APPROACHES

The finite element (FE) method has become preferred for investigating brake squeal. It has significantly increased in popularity because of the poor experimental methods for anticipating squeal early in the design process. Furthermore, realistic modeling of the various braking system components has an advantage over analytical techniques. It accurately represents complex geometries, boundaries, and loading conditions (Trichês Júnior et al., 2008). With the FEA tool, numerical modal analysis can be carried out; thus, the modal parameters of all brake components can be determined using the normal mode analysis. In addition, as different vibration modes caused the mode coupling between brake components rather than individual parts, the real advantage of FE modeling is the ability to model an entire braking system to predict its stability and, thus, its tendency to squeal (Papinniemi, 2007).

As FEA has become a standard analytical tool across many areas of engineering and with the advance in modern computing capacity, it has made it possible to analyze structures with a great deal of accuracy and detail beyond the hand calculations in the past. Essentially the FEA procedure converts a continuous physical structure into a discrete model featuring a finite number of degrees of freedom (DOF). In mathematical terms, the structure is represented as a linear system of n equations, where n is the number of DOF. These equations are treated as a matrix equation of degree n, and powerful tools from matrix algebra are used for the solution procedures. The general process of generating a validated FEA model and its underlying steps is shown in the flow chart in Figure 9 below.

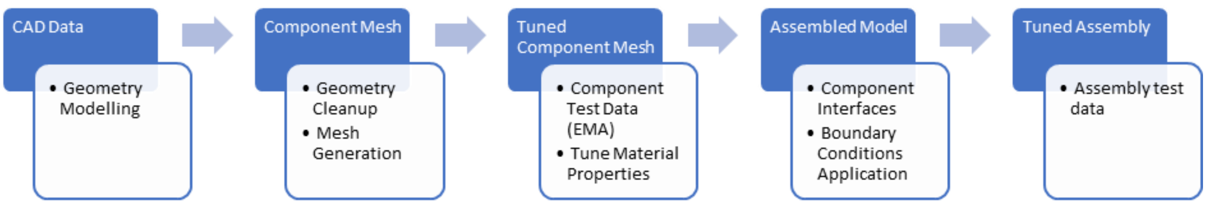


FIGURE 9: The general process of FEA model generation

Various FE models were constructed beginning from the simplified model of a disc and one or two pads, extending to the complete disc braking system, which includes a disc, a pair of pads, a caliper, a piston, a carrier, and two bolts and guide pins respectively (Lina et al., 2011). In recent years, with the computational capability of multi-body dynamics (MBD) simulation, the modeling has become more complex, which includes the entire disc brake corner with suspension links, steering knuckle, and the disc brake assembly (Nouby et al., 2011) to provide more realistic boundary conditions. Figure 10 below depicts the FE modeling progression through the years.

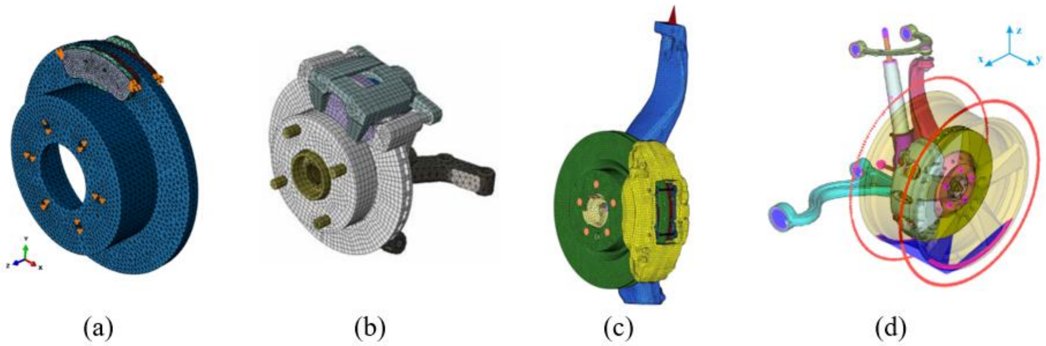


FIGURE 10: Progression of FE Model: (a) Simplified Model (Williams, 2015); (b) Corner Module (Nouby, Abdo, et al., 2011); (c) Suspension Strut (SAE, 2020); (d) Complete Corner Assembly (Huemer-Kals et al., 2017)

5.1 Modeling Correlation, Tuning, and Updating

The predicted results of the finite element model of a composite structure with different types of connections are highly dependent on the mesh size of the finite element model. The complexity has forced engineers to seek the most efficient techniques for selecting the appropriate mesh size to obtain accurately predicted results from normal modes analysis (Omar et al., 2018). Apart from reducing the computational time, it is significant for the model mesh to be developed to obtain a more uniform distribution of the contact stiffness. Oberst et al. (2013) conducted an extensive numerical study of mesh convergence, choice of element types, element sizes, and boundary, which found that the number and the type of expected unstable vibration modes converge as mesh density increases.

Material parameter tuning is also essential for accurately representing the actual component. Based on the Oberst and Lai (2015) pad-on-disc model study, they pointed out that the different material definitions, whether isotropic or anisotropic, can result in considerable variation in the radial, tangential, and rotational pad mode's eigenfrequency. Chantalakhana et al. (2019) updated their disc and pad models with the modal test results, treating the pad lining material as orthotropic and other materials as isotropic to analyze mode coupling effectively. SAE International provides a standard guideline named SAE J3013 Friction Material Elastic Constant Determination through FRF Measurements and Optimization (SAE, 2019). It describes the steps and the variables to be controlled to estimate the friction material's anisotropic elastic constants using pad assembly FRF data and procedure for optimization. Then the elastic constants acquired will be used as inputs for the brake NVH simulation to establish a correlation of the vibration of the brake pad assembly between the simulation and measurements.

Williams (2015) considered the more advanced FE model updating technique, which is the most crucial step for creating a FE model of a brake system. Ewin (2000) described the model updating procedure, which involves using test data collected from the actual structure to improve, rectify, or update an original theoretical model built to understand the dynamics of the structure. Two methods are established to update the finite element model: direct and iterative. Direct methods update the finite-element model accordingly without considering changes in physical parameters. Models generated by direct methods frequently depict the quantified parameters without considering the structure under study. The mass and stiffness matrices generated as a result have no physical significance and cannot be connected to physical changes in the finite parts of the original model. As a result, the matrices are typically fully populated and not sparse, and the connectivity of the nodes is not guaranteed. Contrarily, physical parameters are adjusted with iterative methods until the finite element model accurately reproduces the quantitative data. This characteristic of iterative approaches results in finite element models that guarantee node connection and have mass and stiffness matrices with physical significance (Marwala, 2010).

The most common application of experimental modal testing and FE Normal Mode analysis is to compare predictions regarding a structure's dynamic behavior with those observed in practice. Two different comparison methods are available to validate a theoretical model (i.e., FE Model) over EMA. First, there are comparisons based on response and modal properties. Although the response properties of a tested structure can be generated directly in EMA, some finite element software packages make generating Frequency Response Function (FRF) plots more difficult. Furthermore, modal property comparisons are perhaps the most common and convenient in current practice, where natural frequencies and mode shapes correlate (graphically or numerically) between prediction and EMA results. Ewin (2000) clearly outlines this process and the practical steps. At the component level, the first of these steps is to perform a direct and objective comparison of specific dynamic properties measured versus predicted via modal testing. For example, each disc brake component's free-free boundary conditions are used to gather data on its modal properties, including its natural frequencies, mode shapes, and FRFs. Secondly, Finite Element software is used for numerical modal analysis of all components. A correlation is then made to quantify the differences (or similarities) between the two data sets. The sources of any discrepancies between the two models are also identified and located. The final step is to perform a tuning procedure or 'model updating' by adjusting or modifying one result set or another to bring them closer together and reduce the error. This final step mentioned can involve, for example, changing the material properties, boundary conditions, spring or mass stiffness, and damping values.

All of the brake components are assembled on the assembly level. All connections between parts are defined using a combination of node-to-surface and surface-to-surface elements. A specific brake line pressure is applied to the stationary brake rotor to represent the actual application in the assembly. Again, experimental modal testing is utilized with the corner set of the braking system mounted on a dynamometer. Adjustments are made to the spring stiffness values that link the components in which the predicted modes of the assembled system were tuned until the data was matched to the EMA results.

5.2 Prediction Using Normal Modes Analysis

Normal Modes Analysis studies a system dynamic characteristic (modal analysis) via the FEA method based on free-free vibration eigenvalue & eigenvector problem solutions. The dynamic character is defined independently where each of the modes of a system has a specific frequency, with particular damping and characteristic deformation. When given an excitation at its natural frequency, the structure is subjected to a deformation related to the mode shape of each mode associated with the geometry of the components, along with their associated distribution of mass and stiffness. In the free-free modal analysis, each brake component's modal parameters, such as mode shape, are identified, as shown in Figure 11.

A specific study on modal coupling and its effect on brake squeal using normal mode analysis was conducted by F. Chen et al. (2002). They explained that a brake system's modal coupling could include the combination of the rotor's in-plane and out-of-plane modes or a combination of components, such as between pad and caliper modes. The tangential mode (also called circumferential mode) and radial mode, or combined, are examples of rotor in-plane modes as the rotor vibrates along its surface. On the other hand, out-of-plane modes are oscillation along the normal direction of its surface, producing nodal diametric mode, nodal circle mode, or combined. Their results demonstrate a good agreement between the rotor in-plane and out-of-plane mode coupling frequencies and the vehicle squeal frequencies. However, the rotor and pad modes predominate for a high-frequency squeal, such as one above 3 kHz.

A recent study by Chantalakhana (2019) successfully verified the correlation between the normal mode analysis of updated material properties and the dynamometer test result to verify the mode coupling mechanism. Using a similar methodology, Yeamdee et al. (2019) also investigated the effect of the slot and chamfer shape of the brake pad on mode coupling. Although it can vary slightly with temperature, friction coefficient, and brake pressure, the coupled squeal frequency usually is relatively close to the aligned frequency. Therefore, it suggests that up-front design actions can limit squeal propensity using modal alignment frequencies as potential squeal frequency indicators. Furthermore, this suggested that this modal alignment frequencies determination technique is helpful before performing the time-consuming process and more accurate prediction of Complex Eigenvalue Analysis (CEA).

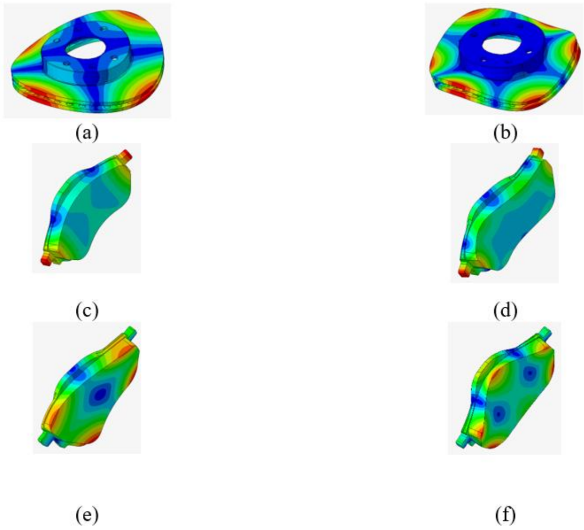


FIGURE 11: Normal Mode Analysis of disc and pad

5.3 Instability Analysis using Complex Eigenvalue and Dynamic Transient Method

In principle, brake squeal prediction involves evaluating the stability of the brake system. Two common methods for assessing the stability of a large-scale finite element model can be determined by observing the evolution in the time domain via Dynamic Transient Analysis (DTA) or by determining the location of poles in the Complex Eigenvalue Analysis (CEA). The CEA uses Laplace Transform mathematical technique to transform the time-domain differential equation into the equivalent frequency-domain algebraic equation. The transformation shows the complex modes contained in the imaginary part, which represents the cyclic frequency, and the real part which represents the damping of the mode. The brake squeal research community has widely used CEA to study friction-induced squeal in automotive disk brake assemblies. The analysis employs a non-linear static pre-stressed normal modes analysis simulation sequence and then comes with a complex eigenvalue extraction algorithm to identify the dynamic instabilities (Ballinger, 2016). If we represent an unstable system response in the time domain, we can see that the diverging amplitude of the oscillation continues to grow indefinitely. A braking system in self-excited instability is non-linear and exhibits limited cycle behavior. An initial disturbance around the ground state grows exponentially before the dissipative effect compensates for friction work and is converted into vibration energy (Meehan & Leslie, 2021).

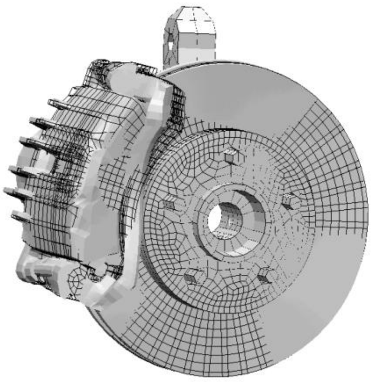
The complex eigenvalue method can calculate the dynamic instability of a braking system caused by two coupling modes. The degree of instability is indicated by the positive real parts of the complex eigenvalues, which predict the tendency to squeal. Once the potential occurrence of squeal is identified, design modifications can be tested numerically to evaluate their effectiveness in reducing squeal. The analysis can be demonstrated using simple mass/spring/damper lumped parameter models, presenting the single degree of freedom (SDOF) system dynamic response caused by the $f = f_o \sin(\omega t)$, sinusoidal input, as shown previously in Figure 3. The system transfer function can be used to calculate the frequency response.

$$\frac{X}{f_o} = \frac{1}{-\omega^2 m + i\omega c + k} \tag{7}$$

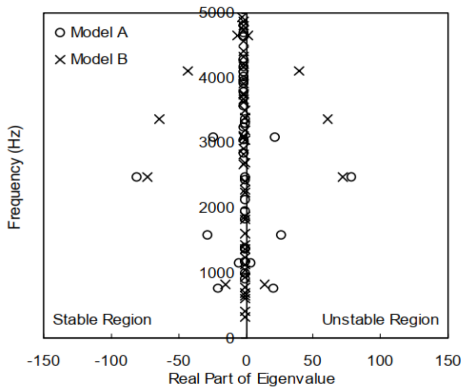
From the characteristic equation below, the roots or eigenvalues can be computed to determine system stability, with a negative real part indicating stability.

$$-\omega^2 m + i\omega c + k=0 \tag{8}$$

The inclusion of an asymmetric friction matrix derived from the inclusion of tangential friction in the contact pressure analysis is the basis of the CEA (Nouby, Sujatha, et al., 2011). The process predicts the complex eigenvalues, with the real part indicating the degree of instability and the imaginary part of the corresponding frequency. The capability to identify the unstable modes and corresponding frequencies in short simulation times increased the usage of the method as the most common predictive tool to investigate squeal problems. Figure 12 below is an example studied by Kung et al. (2000), who carried out the CEA on two assembly models. The imaginary component of the eigenvalues on the vertical axis represents the natural frequency. The real component of the eigenvalues, which measure damping or instability, is represented on the horizontal axis. Systems have modes with negative damping when the real component of the eigenvalues is positive. Therefore, the unstable area is in the right half plane, while the stable area is in the left.



(a)



(b)

FIGURE 12: Example of CEA (Kung, Dunlap, et al., 2000) (a) 2.5 kHz complex mode form. Shaded means deformed; wireframe means undeformed; (b) Complex eigenvalues of two different models for comparison.

The ability of CEA to predict brake instabilities that could result in brake noise is well established. However, sometimes it misses the unstable frequencies and is prone to overstate instabilities that do not exist in the real braking system. The over-prediction of unstable modes results from insufficient damping in the model compared with the real braking system (Esgandari et al., 2013). The CEA analysis mainly uses a linear model, despite the strong non-linearities associated with the contact problem (Lü & Yu, 2016). Damping characteristics must be tuned into the model to guarantee that the FEA model of the brake unit accurately represents the system's actual behavior. A more advanced approach recently used, such as Modal Amplitude Stability Analysis (GMASA), complements the CEA method’s weakness. The CEA method cannot estimate vibration amplitude and suffers from under or over-prediction of modes involved in system response in the non-linear dynamic response and its quasi-periodic oscillations estimation (Denimal et al., 2021).

On the other hand, the DTA has several advantages. It offers more insight into the transient non-linear character and can predict the amplitude of a limit-cycle motion. It also allows time-varying properties to be considered. These advantages make it possible to find a solution in the time domain. Nouby et al. (2011) determined the system’s instability through divergent vibration time response and compared it to CEA results. The output time domain data are transformed into frequency domain data by the MATLAB software using the Fast Fourier Transform (FFT) technique. Concerning the drawbacks, design iterations are challenging because one run requires significantly more time and memory capacity than the CEA and does not reveal information about unstable modes.

It is essential to ensure a more accurate prediction of squeal propensity by introducing the operating, physical, and environmental conditions that affect the friction dynamics of the disc-pad contact surface. The methodology considers both the thermal effects of heat generation and the interface pressure distribution due to variation of topography at specific times during the braking period. It requires an integrated dynamic study of the fully coupled non-linear contact and the thermo-mechanical analysis, which the present review does not cover due to its complexity.

5.4 Modal Participation Factor of the Pad

The modal participation factor measures how strongly a given mode contributes to the structure's response when subjected to force/displacement excitation in a specific direction. It is useful in defining the systems mode shapes to identify the most contributed components to be selected and improved. Finite element modal analysis uses the mode participation factor measurement to determine the number of modes to be extracted and to find the most critical natural frequencies or modes from the millions of natural frequencies and modes from finite element models with an infinite degree of freedom. It measures the amount of mass moving in each direction for each mode. A high value in a direction indicates that the mode will be excited by forces or excitations in that direction.

Kung et al. (2000) pointed out that the vibration behavior of the complete brake system cannot be directly derived from the dynamic characteristics of the individual components. Instead, an unstable mode can originate from two closely coupled component modes that depend strongly on the nature of friction coupling. When this occurs, both modes typically have closely spaced frequencies and mode shapes that are geometrically compatible. As introduced earlier in section 4, Modal Assurance Criteria (MAC) can be used to calculate the component participation factors based on the complex algorithm related to the unstable modes. It can provide information on modal participation factors to the mode coupling during brake squeal (Allemang, 2003). For a vibration system that consists of n components, one of the modes is considered a combination of all component eigenvectors.

The pad contribution factor is typically in a range of 10%-30% based on a complex MAC algorithm calculated by Kung et al. (2000). According to previously reported modal participation analysis, as shown in Figure 13, the influence of pad design modification on the unstable modes may be less than 20%. However, there is still value in examining the pad dynamic problem as it is more economical, feasible, and straightforward than all the other components. Furthermore, despite the challenges, many works of literature have shown that we can improve the squeal noise problem through a friction pad design modification.

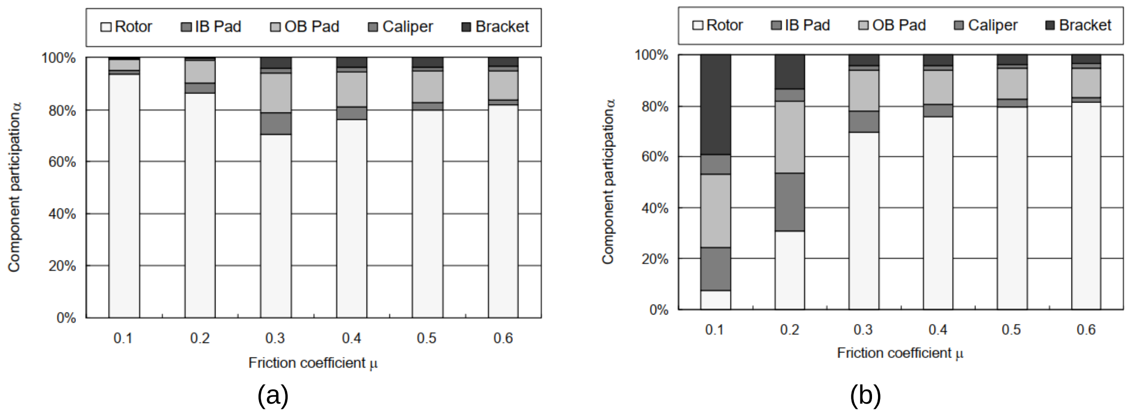


FIGURE 13: Modal participation factors at various friction coefficients. (Kung, Saligrama, et al., 2000) (a) Component participation factors for 1st Mode; (b) Component participation factors for 2nd Mode

6. STRUCTURAL MODIFICATION

Finding an effective design modification is a trial-and-error task if the modal coupling mechanism is not understood. The example of structural design modification using chamfer and slot (also called groove) on the brake pad and the modification on the disc rotor surface and ventilation chamber is shown in Figure 14.

(a)

(b)

FIGURE 14: Example of the modification; (a) Brake pad (Performance Review Institute, 2022); (b) Disc rotor (Brembo, 2015)

6.1 Geometrical Influence & Impact

Liu et al. (2008) use the FEA and dynamometer test to investigate the effect of slot and chamfer. The shape modifications introduced are a slot at the center, 21mm parallel spacing chamfers at the leading and trailing sides, and V-chamfers at the top and bottom centers. The result showed that the instability of most unstable modes is reduced when those modifications are applied. Furthermore, when those modifications are applied, the instability of the most unstable mode is noticeably reduced. Bergman et al. (2000) experimented with drilled, slender, and short pad configurations to see how they affected the occurrence of squeals. They found that the short brake pad consistently had a lower squealing probability than both other types of brake pads.

An approach of using chamfered and slotted pads was also presented by Lina et al. (2011) to tackle brake squeal. They develop a three-dimensional FE model of the existing disc brake system to predict squeal and conduct the simulation using the complex eigenvalue and transient analysis. The comparison against the brake dynamometer data was found to be correlated. They suggested three modifications to the pad and repeated the simulation and testing for comparisons. A pad with chamfers and diagonal slots is shown to be able to suppress squeal in both prediction and squeal tests.

By altering the design parameters for the vane fillet, Cha et al. (2018) have shown how to separate the brake disc's in-plane and out-of-plane modes. In addition, they performed the normal mode FE analysis for parameter sensitivity to the mode separation, which most affected the in-plane modes. Wang (2016) also conducted a similar study with various orientation degrees of grooves cut on the pad-on-disc system and later concluded a similar outcome with other researchers.

Li et al. (2020) analyzed the influence of the characteristic parameters of the brake pad on the structural modal. The modal analysis was used to calculate the brake pad's mode frequency and the mode shape with different thicknesses, the depth and width of the groove, and the chamfer. According to the simulation results, the brake pad's thickness, the brake pad's groove depth, and the brake pad chamfer's size influence the brake pad's modal properties.

6.2 Tribological Effect

Based on the literature reviewed, there is strong evidence that the chamfers' design changes the pad and disk contact, and slots change the flexural properties of the pad assembly. Despite the successfulness of the structural modification in reducing the squeal propensity, it was also found that, at the same time, modifications can influence the tribological behavior of the friction material. Based on studies by Wang et al. (2016) on several different groove cut patterns of a disc surface, they conclude that there is a significant consequence on the wear behavior of the noise generation by modifying the contact area between the disc and the friction material. Park (2018) studied the optimization method of pad abutment or point of contact using the design of the experiment by utilizing regression equations via varied values of factors within the iteration of the program.

7. EXPERIMENTAL DIAGNOSTIC & VALIDATION

Over the past 20 years, the accuracy of predictive design analysis methods has improved significantly with the increased sophisticated computer-based modeling and simulation capabilities, which allow various aspects of brake design optimization and verification to be performed in virtual tests. However, experimental testing is the only way to prove a design or obtain vital information for an accurate design prediction, so actual testing still plays an important role. Experimental procedures are always required to determine not only the subjective quality (sound quality) of the noise but also its occurrence (location, environmental, vehicle conditions, frequencies, periodicity) and intensity (sound pressure level, loudness).

The braking NVH measurement can be performed in the laboratory or on the road, where both methods have advantages and disadvantages. Road testing is beneficial to ensure the final quality of the actual vehicle in a real-world environment. However, it is often expensive, time-consuming, and indecisive due to the small number of vehicle samples. Usually, the results are obtained very late at the end of the development process, where required changes would significantly impact on the overall development time. On the other hand, for laboratory testing, the preferred operating conditions for the dynamometer test are tough to meet and require more sophisticated equipment. However, laboratory testing can provide quick and easy access to diagnostic and troubleshooting, allowing more data measurement flexibility than road tests (Glisovic & Demic, 2012).

The leading equipment of any NVH measurement system is the transducers, the data acquisition, and the data analysis systems, as shown in Figure 15 by Thompson (2011). These transducers transform sound or vibrations into electrical signals that can be recorded and processed to interpret physical phenomena. The data acquisition and analysis systems convert the time-based raw signals from the transducers into calibrated time or frequency domain data for storage and further analysis of the squeal noise level and its occurrence (Thompson, 2011).

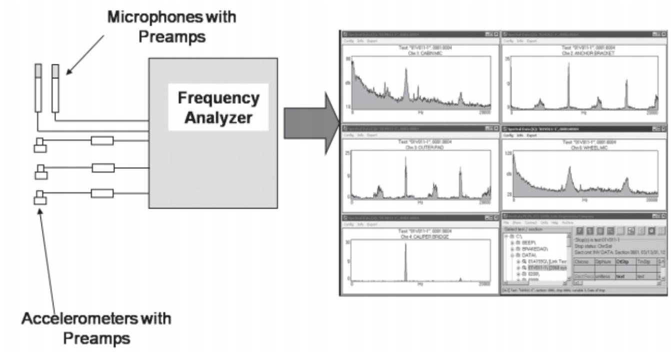
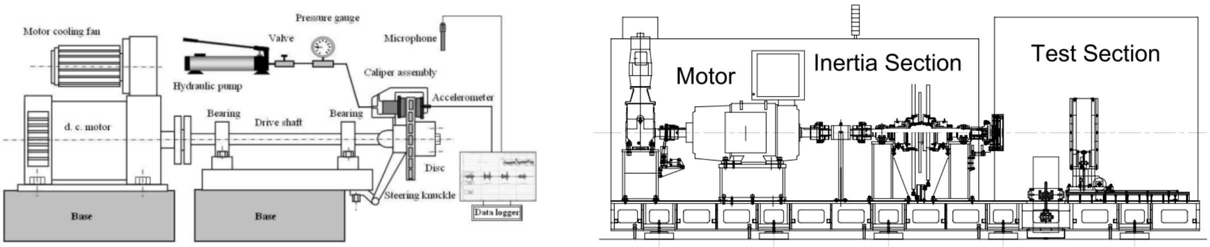


FIGURE 15: Key NVH Equipment by Thompson (Thompson, 2011)

7.1 Brake Dynamometer Squeal Test

The internationally recognized laboratory brake noise measurement standards are the SAE J2521 Disc and Drum Brake Dynamometer Squeal Noise Test Procedure (SAE, 2013). Based on the original proposal by Bosch (Blaschke & Rumold, 1999), it has provided a standard guideline that has enabled consistent and meaningful assessments that have been critical to avoiding brake squeal problems. This NVH matrix test uses a dynamometer to test the noise on a shaft or a chassis/roller. The shaft-type dynamometer is the most widely used because it is the most compact and efficient way to conduct various cases in the study.

In the past, drag-type dynamometers were primarily designed for drag tests to reproduce brake noise. However, the influence of the car’s inertia should be presented to carry out a realistic simulation of stopping a vehicle. Furthermore, inertial mass often results in smoother operation at low speeds when noise problems are common. It can be done in two ways. The conventional method involves mounting the discs on a rotating shaft whose rotational inertia matches the linear inertia of the car. In a more contemporary method, the electric motor simulates the vehicle's inertia. Modern brake dynamometers can offer both techniques (Glisovic & Demic, 2012) to give the user the most flexibility possible while emulating the car brake system. Figure 16 shows one typical example of a shaft-type dynamometer schematic diagram comparing with and without an inertia wheel.



(a) (b) With Inertia Wheel

FIGURE 16: Typical Example of Shaft-Type Dynamometer; (a) Without inertia (Wadageri et al., 2016); (b) With an inertia wheel (Thompson et al., 2002)

The test criteria described in SAE J2521 are widely used to guide the brake squeal simulation or correlation against FEA. Esgandari and Olatunbosun (2016) use the dynamometer test to confirm and correlate the damping effect and flexibility of the rubber and adhesive material of brake shims or insulators based on the Rayleigh damping method and tuning to eliminate the majority of over-predicted instabilities by the CAE method. Lazim et al. (2016) studied the embedding mechanisms of particles of various sizes in the case of a brake pad/disc system to correlate their impact on friction and dynamic behavior. The investigation focuses on characterizing the worn surface of friction material squealing with external silica sand particles (SSP). Arvin et al. (2017) studied the effectiveness of constrained layer dampers with different configurations at a hydraulic pressure range of 5 bar to 30 bar and temperature range of 50°C to 200°C on a brake test dynamometer. These studies use the same SAE J2521 to ensure consistency and comparison with other researchers or future works. Poletto et al. (2017) employed six brake noise indexes to assess the squeal produced by four pad materials subjected to the SAE J2521 test procedure. The semi-metallic pad was found to be the noisiest friction material, followed by low-metallic, non-asbestos organic (NAO), and, finally, low-metallic.

7.2 Non-contact Measurement

Besides dynamometer tests, non-contact displacement measurement techniques and laser metrology can also be performed in laboratory conditions. For example, Doubled-Pulse laser Holographic Interferometry (DPHI), Electronic Speckle Pattern Interferometry (ESPI), and Laser Doppler Vibrometer measurement are practical tools for determining the natural frequencies, vibration mode shapes, and forced response to identify the cause of the brake noise and evaluate engineering solutions but is not discussed in detail for this article. These methods offer significant advantages over traditional transducers, such as accelerometers, partly because of their non-intrusive and non-contact operating nature. In addition, it allows the visualization of a vibration pattern and further complements the combination usage of acoustic holography and high-speed cameras. However, the experiments are primarily expensive and time-consuming.

7.3 On-Road Vehicle Noise Evaluation

One of the essential parts of the vehicle braking system test is the actual confirmation of the repeatability and the control ability of braking under all conditions of operation and use. The legislature also requires car manufacturers to demonstrate that the vehicle meets the legal requirement for type approval or self-certification. Unfortunately, many aspects of brake performance and operations cannot be accurately predicted on a bench or virtually meet all the requirements mentioned. In most cases, the material data, properties, and detailed information about each related part under the actual operating conditions are not known precisely. Figure 17 shows the standard chassis dynamometer designed for brake NVH testing.

A wide range of on-road tests run for brake noise evaluation to investigate and screen the potential of the brake noise. Sometimes it includes severe or specialized operating conditions designed to elicit brake noise that can last from a few hours to a few months to complete. Therefore, one of the most important goals of the on-road measurement is to measure the critical operating conditions when a noise event occurs and evaluate the annoyance and potential customer reaction. SAE J2625 is one of the internationally recognized procedures of vehicles' brake squeal noise test. It provides a squeal evaluation method. It focuses on a low-speed test with a range of temperatures and pressures as part of the evaluation with the instrumented assessment of the propensity of the squeal generated, as shown in Figure 18 below.

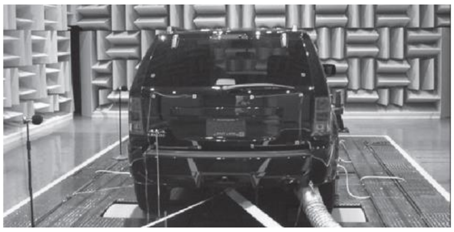


FIGURE 17: Noise evaluation in anechoic chamber (Thompson, 2011)

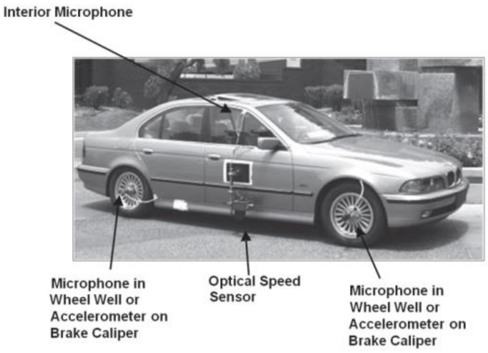
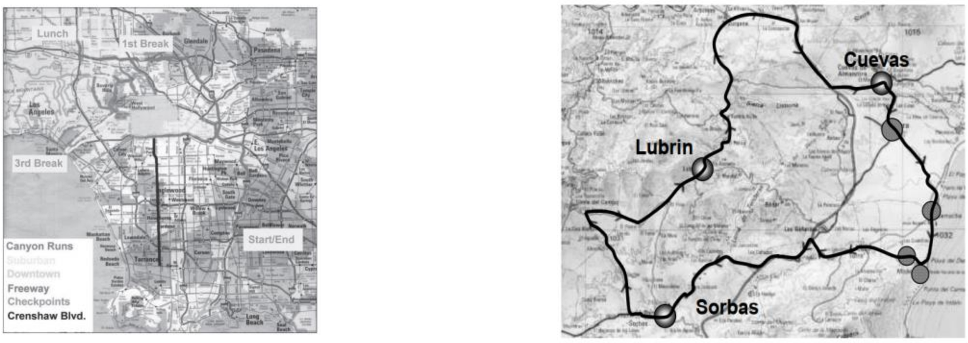


FIGURE 18: Vehicle noise evaluation setup (Thompson, 2011)

Public road testing is performed to evaluate brake NVH and wear where brake issues are commonly found. One widely used screening procedure is the city traffic test of Los Angeles City Traffic (LACT) procedure shown in Figure 19 (a). The approach is to run a vehicle for an extended period in urban traffic with routes typically selected to provide operating conditions that induce squeals. There are several such test routes in Los Angeles, Detroit, Phoenix, and other urban centers in the United States. Oberst and Lai (2012) performed their final validation of their solution effectiveness by the 25-day LACT actual vehicle test. In Europe, the predominant route is Mojacar in Spain, as shown in Figure 19 (b). There are many other city traffic routes worldwide. Manufacturers have their screening procedures explicitly developed to cater to their specific market requirements or customers' daily usage representation at a specified location or geographical area, such as the Shanghai City Traffic Test. For example, Shanghai GM uses the Huangshan test to validate brake NVH and wear performance for the Chinese market signoff (Q. Wang et al., 2014).

Figure 20 shows an example of an instrumentation system developed by Applus+ IDIADA in Spain for the Mojacar Brake Noise Evaluation test route, allowing automatic analysis of the occurrence, frequency, and sound pressure level (SPL) of the brake noise. Noise can be triggered with different parameters. The brake line pressure, temperature, and speed conditions are automatically recorded and analyzed when the noise occurs (Del Sol & Mateu, 2013). Furthermore, it can also carry out the correlation of data with subjective evaluation from expert drivers. However, few researchers have conducted comparison studies between different test cycles, such as comparing Mexico City traffic against the LACT route (Menendez, 2012).

At least five NVH measurements must provide valuable data for a comprehensive on-road test. It is a measurement at each corner by an accelerometer. There must be at least one interior microphone. In addition to these NVH measurements, it is crucial to quantify the operating conditions at which the noise occurs. To this end, measuring vehicle speed, brake temperature, line pressure, and some indication of brake applications is necessary. Many studies have attempted to simulate the road-testing profile into the dynamometer. For example, Grochowicz et al. (2007) simulate road testing by considering different driving resistances and real road profiles in a brake noise dyno test and have established a global test procedure. Structural modification on the brake pad might have changed its operational durability, thus requiring actual on-road testing for confirmation. Suppliers or manufacturers must comply with parts, assembly, and product consistency; reliability is also critical.



(a) (b)

FIGURE 19: Example of public road evaluation test route; (a) LACT, USA (Thompson, 2011); (b) Mojacar, Spain (Mody et al., 2002)

(a) (b)

FIGURE 20: dbBrake Noise Evaluation Acquisition System by Applus+ IDIADA (Del Sol & Mateu, 2013). (a) dbBrake system; (b) Rubbing thermocouple on the disc surface

8. SQUEAL SUPPRESSION EFFECTIVENESS

Even though the structural modification's primary motivation is to prevent unwanted modes from occurring or shifting to some noisy frequencies, the inevitable fact is that the shift of some existing noise frequencies might create a new noisy one when a modification is made. However, it can still be a powerful, effective tactic combined with optimization or inverse methods (Ouyang et al., 2005). Furthermore, many experimental studies manage to provide an agreement against the actual squeal measurement. A successful FE model for pad design optimization requires finding a suitable method to model the friction coupling between rotor and pad, identifying the critical pad design parameters and beneficial design changes, and finally using this information to update the prototype FE model (Dai et al., 2002). Because so many additional influencing elements must be considered for an effective prediction, the finite element analysis prediction approach is still challenging. For example, it requires consideration, such as geometrical imperfection (Bonnay et al., 2015) that affects the contact interface pressure distribution (Bakar et al., 2003) and the thermo-mechanical contact analysis due to its "rotating heat source" effects (Hassan et al., 2008).

Wang et al. (2014) conducted experimental studies on the effect of a grooved disc on tribological properties, apart from vibration and noise properties. The result shows that grooves on the disc surface can significantly affect tribological behavior regardless of the contact configurations. Dabrowski et al.'s experiment (2018) on a brake dynamometer summarizes that the modification of the surface of the brake pads, other than reducing squeals, also affects their friction characteristics. Both chambers and slots have had a positive effect on their friction characteristics. The modification reduced the maximum and average coefficient of friction, but the minimum coefficient was increased. The disparity between the maximum and minimum values is also reduced. It was also found that the coefficient of friction has become more stable and settled in low and high temperatures under high-speed deceleration braking conditions. Thus, the difference in the friction coefficient for various braking application conditions is negligible for modified pads compared to unmodified ones. The characteristics found above make the brake feel more predictable and dependable for the driver throughout using the brake pad life. These two example studies suggest that further investigation is required to summarize each structural design's effect on the brake pad's tribological properties.

9. CONCEPTUAL FRAMEWORK SUMMARY

All the literature presented earlier can be summarized into a visual representation of a conceptual framework, as shown in Figure 21 below. Each relevant piece of information is noted with asterisk marks for easy reference for the reader to follow.

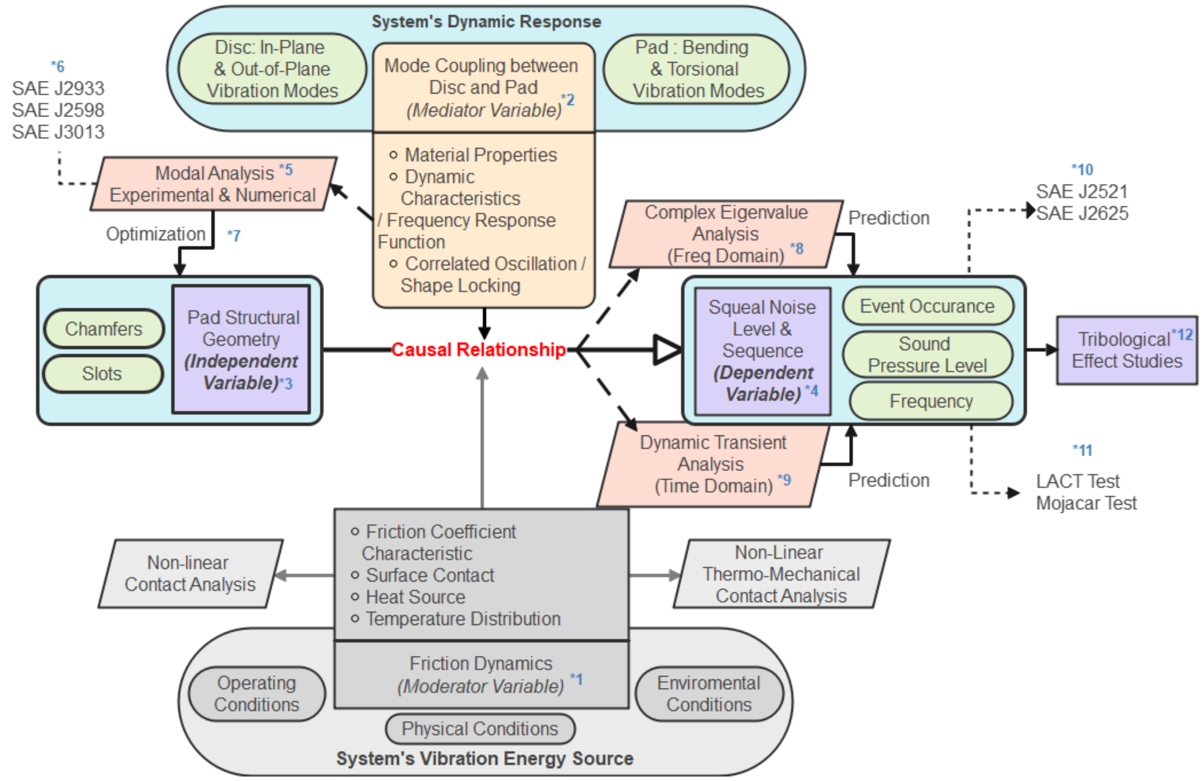


FIGURE 21: Conceptual framework of pad structural modification for squeal noise suppression

The two main aspects contributing to the brake squeal generation are the friction dynamics (moderator variable)*1 which provides the energy source for the system to vibrate, and the mode coupling between the disc and pad (mediator variable)*2 which contribute to the system's dynamic response in the form of vibration. In addition, the pad's structural geometry parameters alteration, such as chamfers and slots, are the independent variable*3 factor that creates a causal relationship towards the dependent variable*4 of intensity and frequency of squeal noise production.

In this review, the summary of all the works of literature focuses on the system's dynamic response, which focuses on the vibration mode coupling of correlated oscillation structure between the disc's out-of-plane and pad's bending mode. Details parametric studies were conducted on the material properties, dynamic characteristics via the frequency response function, and the correlated mode shape. The prediction and optimization of the structural geometry modification can be achieved from the Modal Analysis studies*5 on each component's free-free vibration and the assembly between disc and pad via experimental impact test and numerical studies using the FEA software.

SAE provides the recommended practice*6 on the disc and pad experimental modal analysis based on the J2933 and J2598, respectively, and the determination of the pad's elastic constant based on the J3013 document guideline. Several geometrical modifications are proposed based on variations of slot and chamfer design. Various optimization methodologies*[7], including parametric studies, are conducted to separate detected close modes between disc and pad to reduce squeal propensity.

Then, the squeal propensity can be further predicted and evaluated via the Complex Eigenvalue Analysis*[8] (CEA) in the frequency domain or Dynamic Transient Analysis*[9] (DTA) in the time domain. The predictions can be further compared against the experimental result from the SAE recommended practice guideline*[10] based on the J2521 Dynamometer Squeal Noise test and the J2625 Vehicle Brake Squeal Test. Finally, on-road vehicle noise evaluation is conducted to validate the real-world application's effectiveness on specified routes*[11] such as city driving of LACT, or a combination of urban and extra-urban driving, such as Mojacar Spain. In addition, a few researchers conduct additional research*[12] on brake pad surface finishing and treatment effects on frictional performance, tribology, and vibration.

The second aspect of the system's vibration energy source was not in the scope of this review. However, it is essential to emphasize that the solution on the fugitive nature of the source in a great effort to predict the squeal with better accuracy and to understand and simulate the actual event of the squeal by considering the parameters such as friction coefficient characteristics, surface contact, heat source, and temperature distribution. Even though it is more complicated because of the requirement for non-linear contact analysis and uncertainties in the operating, physical and environmental conditions, a similar review is recommended to understand the conceptual framework in this study area.

10. CONCLUSION

This paper summarizes the most relevant literature on existing studies and theories to help understand the conceptual framework of the brake pad structural modification methods to reduce brake squeal noise. It is found that more detailed reviews are required on the system's vibration energy source area of study. It is essential to ensure a more accurate prediction of squeal propensity by including the operating conditions, physical condition, and environmental condition that affects the friction dynamics of the disc-pad contact surface. In addition, it will involve a more advanced non-linear contact analysis and thermo-mechanical contact analysis. The tribological impact research area is also not covered in this review, attracting significant extensive studies in recent years. However, it is concluded that the system's dynamics response study is adequate to give a clear explanation and representation to understand the underlying principles related to the mode-coupling theory, methodology, and the effectiveness of the structural modification technique on the aspect of vibrational characteristics and behavior analysis for the disc brake squeal noise suppression. The paper has developed and presented a visual representation of a conceptual framework.

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