Model Updating of an Automotive Disc Brake System

Jack J. R. Williams

An increase in customer expectation of quality in automobiles has caused a surge of research into the refinement of noise, vibration and harshness concerns over the last 20 years. Brake squeal has been at the forefront of this research as it results in exorbitant warranty costs to automotive manufacturers. Finite element analysis provides an indispensable tool for investigating brake squeal. It is able to realistically represent the complex geometries and boundary conditions of a brake system as well as indicate the stability of design changes without the need for a prototype. Model updating is a vital step in creating a validated finite element model of a brake system. It is accomplished by utilising modal testing of brake components to validate the finite element model.

The aim of this project was to create a finite element model of a real brake system that is validated using model testing of brake components. This was accomplished by creating finite element models using the software ABAQUS, completing extensive modal testing on all brake components and updating the models using a Rayleigh damping model and specialised model updating software FEMtools. A new method of testing and updating brake pads was also developed. A model of the brake pads and the brake rotor was subjected to complex eigenvalue analysis using ABAQUS in an attempt to identify potential squeal modes which appear as unstable vibration modes. Brake squeal was successfully predicted at corresponding frequencies using the updated model.

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Nomenclature

\( \mu_s \)  static friction coefficient
\( \mu_k \)  kinetic friction coefficient
NVH  noise, vibration and harshness

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I. Introduction

With an increased focus on the refinement of noise, vibration and harshness (NVH) concerns in automobiles over the last few decades, brake noise has become a major subject of research for the automotive industry (Papinniemi 2007). Disc brake noise is one of the leading contributors to warranty costs of original equipment manufacturers’ (OEM) NVH departments. Customers often believe that disc brake noise is indicative of faulty or unsafe brakes, despite the system usually functioning as designed in all other aspects, and demand that it be fixed (Roches 2011). However, on rare occasions noise can be caused by faulty brakes making it a safety concern. In addition, brake squeal in particular can cause micro-cracking in the lining of the pads leading to premature wear and the self-excited vibration of brakes may cause failures of high performance, lightweight alloy rims (Oberst 2011). In any case, noise must be reduced or eliminated to increase safety and customer satisfaction.

In an attempt to define the sound emitted, various names have been given to disc brake noises. Such names include grind, judder, grunt, moan, groan, squeak, squeal and wire brush (Kinkaid, O’Reilly & Papadopoulos 2003). Brake squeal has received by far the most research due to its disturbing nature to both passengers and the environment as well as exorbitant warranty costs to the OEMs. It is generally agreed that brake squeal is a sustained, high-frequency vibration of brake system components during a braking action resulting in noise audible to vehicle occupants and passers-by (Kinkaid, O’Reilly & Papadopoulos 2003). Brake squeal is generated when the brake system enters into an unstable vibration mode that causes self-excited vibration of the system (Oberst & Lai 2011b). Despite the large amount of research invested into brake squeal, no single theory exists which is able to explain or predict its occurrence. Perhaps the biggest obstacle to brake squeal research is the fugitive nature of brake squeal as it can be transient and non-repeatable under the same apparent conditions (Oberst & Lai 2011a).

As outlined in Papinniemi et al. (2002) there are three methods of analysing disc brake squeal:

1. Analytical approach to study isolated mechanisms;
2. Numerical simulations to approximate the structural continuum; and
3. Experiments to validate theory and numerical results.

This project will be focused on numerical modelling and experimental validation for the purposes of investigating disc brake squeal.

Finite element modelling has recently gained widespread use within the automotive industry to model disc brake squeal. It provides an important tool for simulating brake systems and investigating brake squeal. Unfortunately there is currently no method available to predict squeal early on in the design phase. Modal testing provides a reference to the model, a means of accurately determining the modal parameters of a physical structure without making any assumptions about it (Wall 2003). The finite element model must be optimised to more closely match modal testing results in order to assure the validity of its representation of the structure. This process is called model updating and will be explored in detail throughout this project.

Finite element modelling, experimental modal analysis (also referred to as modal testing) and model updating were completed on a real brake system. The brake system is a prototype designed by Chassis Brakes International (CBI) for the Ford Ranger. Furthermore, an indication of the stability of the brake system was demonstrated through a simplified complex eigenvalue analysis.
II. Previous Research

A. Brake Squeal

From a stability point of view, brake squeal is caused by a friction induced self-excited vibration of the brake system (Chen 2009). Analysis of instabilities involves a combination of triggering sources and squeal tendencies of the system. In general it is a combination of these two variables that is needed to cause squeal. The major theories behind the mechanisms of brake squeal are explained in detail in Oberst (2011). For completeness, these are stick-slip, sprag-slip, negative gradient, hammering effect, mode coupling, damping, gyroscopic effects, transient bursts of lateral in-plane vibration, and surface waves. However, none of these theories can individually explain every aspect of brake squeal (Oberst 2011). According to Chen (2009), brake squeal mechanisms can be grouped into the following categories: 1) stick-slip, sprag-slip and creep slip, 2) modal lock-in or coupling and 3) moving loads. The first group is caused by an intermittent sticking phase in which a static friction coefficient $\mu_s$ applies and kinetic friction coefficient $\mu_k$ takes over. The pads undergo a dig in and release process caused by geometric instabilities or rough surfaces. The second group occurs when two modes are coalescing or merging to form a complex and unstable mode. A moving load scenario may occur due to the sliding motion of a traditional floating calliper design as this lateral loading can excite resonances.

B. Numerical Analysis

The advantage of finite element modelling over analytical methods is that the different parts of the brake system are modelled realistically. It allows for accurate representation of complex geometries, boundary and loading conditions (Junior, Gerges & Jordan 2011). It can be used to perform numerical modal analysis, hence finding the modal parameters of all the brake components. However, the coupling between brake components causes vibration modes different to those found in individual parts. The real benefit of FE modelling is in its ability to model an entire brake system in order to predict its stability and hence its squeal propensity (Papinnieni 2007).

The first major work on FE modelling of brake squeal was by Liles (1989). Liles built a large scale FE model of a brake system. He first modelled each component and importantly manually updated these individual models to match modal testing data. He then assembled the brake system, creating the less certain connections between components through engineering intuition and iterative testing. This method of updating the models of components to match experimental results has become the standard process for modelling brake squeal in the years since, and will be discussed in more detail below.

Since then, various studies have been undertaken using FE modelling of brake systems of varying degrees of complexity and validation. Studies by Junior et al. (2011), Oberst (2011), and Liu et al. (2007) used simplified finite element models of a disc brake assembly to minimise brake squeal with a disc and one or two pads. Whereas Papinniemi (2007), Dai & Lim (2006) and Abu-Bakar et al. (2005) utilised finite element models of the entire disc brake assembly consisting of a disc, piston, calliper, carrier, two pads, two bolts and two guide pins. Furthermore, investigations by Nouby, Sujatha & Srinivasan (2011) and Roches (2011) utilised a finite element model which includes the entire disc brake corner with a steering knuckle and wheel hub in addition to the disc brake assembly. All of these models were validated using the same manual updating process outlined by Liles (1989). In addition, many of these researchers also validated their model at the system level using modal testing of the assembled system as well as actual measured boundary conditions such as pad roughness and pressure contact analysis. The updating of the assembly is an important extra effort which has been previously overlooked by researchers such as Kung et al. (2000) and Liles (1989).

C. Modal Testing

Modal testing is accomplished by measuring the vibration response of a structure whilst vibrating it with a known excitation (Ewins 2000). Using modal testing, the forced or free dynamic response of a structure can be reduced to a discrete set of modes. The modal parameters (modal frequency, modal damping and mode shape) constitute a complete dynamic description of the structure within the frequency range of interest (Dossing 1988). Using a frequency domain model, a system descriptor $H(\omega)$ called the frequency response function (FRF) can be found

$$H(\omega) = \frac{X(\omega)}{F(\omega)}$$

Where $X(\omega)$ and $F(\omega)$ are the response and input spectra respectively. The FRF describes the dynamic properties of the system. FRFs are used in three forms, the compliance, mobility and accelerance. These
correspond to displacement per unit force \( \frac{X(\omega)}{F(\omega)} \), velocity per unit force \( \frac{\dot{X}(\omega)}{F(\omega)} \) and acceleration per unit force \( \frac{\ddot{X}(\omega)}{F(\omega)} \) respectively. Accelerance is the most commonly used as this is the output of accelerometers that are used frequently for modal testing (Døssing 1988).

D. Model Updating

Model updating is the single most important step for creating a FE model of a brake system. Without it, the model is only useful as a computational representation of the geometry of the brake system. Model updating as defined by Ewins (2000) is the process in which an initial theoretical model constructed for analysing the dynamics of the structure can be refined, corrected or updated using test data measured on the actual structure. Model updating can be achieved using direct matrix methods or indirect, physical property adjustment methods (Ewins 2000).

The first updating stage was first demonstrated by Liles whom updated his FE models of components manually in 1989. Model updating involves using modal testing of each disc brake component with free-free boundary conditions to obtain their modal parameters. Numerical modal analysis using FE software of all brake components follows. Next the models are subjected to a tuning procedure in which their material properties, boundary conditions, spring or mass stiffnesses and/or damping are adjusted to reduce the relative error in the predicted natural frequencies, mode shapes and/or FRFs. This has generally been achieved using a trial and error process, however a least squares or Bayesian method is superior for optimising the model in terms of accuracy and time taken. For the most part researchers are deliberately unclear on how and to what extent model updating was undertaken, by simply saying that their model was 'validated' with modal testing.

The second stage of model updating is at the assembly level. Individual brake components are assembled in the FE model. All connections between parts need to be defined using a combination of node-to-surface and surface-to-surface elements and the stationary brake rotor can be subject to a certain level of brake line pressure. Modal testing is again utilised at an assembly level, with the brake system mounted on a test rig. By adjusting the values of the spring stiffness linking the components, the predicted modes of the assembled system are tuned to match the modal testing results. This method was used successfully by Nouby, Abdo, Mathivanan & Srinivasan (2011), Abu-Bakar & Ouyang (2008), and Roches (2011). Lastly, the contact pressure distribution and friction between the pads and disc should be properly investigated experimentally to obtain a fully validated FE model of the brake system.

E. Stability Analysis

FE modelling is used to perform two types of stability analysis for brake squeal: complex eigenvalue analysis (CEA) and time domain analysis (TDA). Nack (1995), Nack (2000) explained the idea behind CEA - to determine the necessary condition for a system to become unstable and grow into a state of limit cycles. He went on to conclude that if a mode has a negative real part in its eigenvalue, it will not have a chance to grow into limit-cycle and thus cause sustained noise. Therefore any mode whose eigenvalue has a positive real part has the potential to become unstable. Time domain analysis follows the progression of displacements in the time domain to determine the unstable state. This process is used to determine the limit-cycle motions but is very time and computationally intensive. As such it is not an approach which is widely used in the automotive industry, but is gaining popularity (Nouby, Sujatha & Srinivasan 2011).

Complex eigenvalue analysis (CEA) is currently the preferred method for predicting the stability and squeal propensity of brake systems. It can give good indications of the effectiveness of design changes by providing an indication of the stability of the system. The governing equation for CEA is written as

\[
(\lambda^2 M + \lambda C + K)\psi = 0
\]  

(2)

Where \( \lambda \) is the eigenvalue, \( M \) is the mass matrix, \( K \) is the unsymmetrical stiffness matrix, \( C \) is the damping matrix and \( \psi \) is the eigenvector. For a particular mode, its eigenvalue is found as

\[
\lambda_{1,2} = \alpha_i \pm i\omega_i
\]  

(3)

Where \( \alpha_i \) is the real part, the damping coefficient and \( \omega_i \) is the imaginary part, the damped natural frequency (Nouby, Abdo, Mathivanan & Srinivasan 2011). According to a study completed by Nouby, Sujatha & Srinivasan (2011), there are four main steps to performing CEA using Simulia’s ABAQUS:

1. Nonlinear static analysis for applying brake-line pressure,
2. Nonlinear static analysis to impose rotational speed on the disc,
3. Normal mode analysis to extract natural frequency of an undamped system, and

4. Complex eigenvalue analysis that incorporates the effect of friction coupling.

Using this method, Nouby et al. successfully found the unstable modes of a brake system of the entire disc brake corner. In a similar manner, Junior et al. (2011) successfully predicted the squeal propensity of their brake assembly using CEA, also taking heat generation and dissipation into account. Abu-Bakar et al. (2005) used CEA to predict the stability of their brake system for different brake pressures. Hassan et al. (2009) also accounted for heat generation in their simplified pad on disc model. CEA was used to accurately predict the squeal propensity of their brake system. Esgandari et al. (2013) went one step further by utilising Rayleigh damping applied to the material of components to fine tune the unstable modes predicted by CEA. They then went on to improve their prediction by testing and modelling the damping shims of the brake pads. The damping shims are very difficult to model with accuracy and as such are almost always excluded from analyses.

III. Experimental Modal Analysis

Modal testing was completed on the prototype system from CBI. All components on the right hand rear side were tested including the brake rotor, mounting bracket, brake calliper, inner pad and outer pad. This section will outline the procedure of these experiments.

A. Experimental Set Ups

After much trial and error, the most successful process for conducting modal testing was found. The standard method used to conduct modal testing involved the following steps:

◦ A hole was drilled and tapped into the component.

◦ The component was suspended using elastic cord.

◦ An electrodynamic shaker was connected to the component by threading a steel rod (or stinger) into the hole in the component.

◦ The shaker was connected to a power amplifier and the amplifier to the data collection software.

◦ The component was subjected to chirp excitation.

◦ The response was measured by a scanning laser vibrometer at many points across the surface of the component.

B. Brake Rotor, Brake Calliper and Mounting Bracket

The set up for modal testing of the brake rotor can be seen in figure 1. The brake rotor was suspended horizontally and a hole was drilled and tapped into the friction surface. Measurements were taken by measuring the vibration response of the brake rotor using a Polytec Scanning Laser Vibrometer (PSV-400). Only a portion of the brake rotor needed to be measured due to the symmetry of the structure and mode shapes. A Bruel and Kjær (B&K) type 4809 electrodynamic shaker was used to excite the brake rotor. The distance between the shaker and the brake rotor was measured before and after the shaker was threaded onto the stinger to minimise the effect of the weight of the shaker mass loading the brake rotor. The shaker was connected to a B&K type 2706 power amplifier to amplify the excitation signal generated by the PSV software. The excitation signal after amplification was then fed back into the Polytec type OFV5000 controller for processing. Using the shaker, the brake rotor was subjected to broadband excitation with the frequency range of 50 Hz to 20 kHz using a periodic chirp excitation signal. 371 scan points were measured using the PSV across one third of the top hat section and friction surface of the brake rotor and 50 complex averages of each point were taken. A resolution of 6.25 Hz was used in the scans to ensure that none of the peaks of the frequency response function were cut off. Zoom FFT scans were completed at a few of the natural frequencies of the brake rotor to further ensure the resolution of the test was fine enough to accurately capture the heights of the peaks of the resonances.

Reliable and repeatable results were achieved using this method. The averaged FRF of the brake rotor can be seen in figure 2. 53 mode shapes between 50 Hz and 20 kHz were found during modal testing of the
brake rotor. Lessons learned during modal testing of the brake rotor meant subsequent components were more straightforward to test.

![Averaged Frequency Response Function - Brake Rotor](image)

**Figure 2:** Averaged frequency response function and coherence of the measurement from modal testing of the brake rotor

The ceramic piston and rubber seals could not be removed from the calliper housing without being sacrificed, potentially damaging the calliper in the process. So these components remained in the housing during modal testing. A hole was drilled and tapped into the back of the cylinder of the calliper so that the stinger could be threaded into the cast aluminium structure. The calliper was excited using periodic chirp excitation and was hung up using elastic cords. 800 scan points were measured using the PSV across the 'back' of the calliper housing. 25 experimental mode shapes were found up to 20 kHz. As with the calliper and rotor, the bracket was hung up using elastic cords, excited using a stinger threaded into the side and scanned using the PSV. 291 points were measured in a scan that found 30 mode shapes. Photographs of the modal testing of all components are included as appendix A. The averaged FRFs of the calliper and mounting bracket are shown as figure 3a and 3b.

**C. Brake Pads**

The brake pads proved to be the most difficult to perform modal testing upon. After completing scans of the pads, it was discovered that the model updating software was incapable of validating the finite element model of the pad sub-assembly. The software was unable to converge to a solution due to the orthotropic material properties of the pad lining and in particular the added complexity of separate materials of the backplate and pad lining. To counter this, it was decided to sacrifice a few of the available pads, to independently test two beams machined out of the backplate and pad lining. Through preliminary numerical modal analysis in ABAQUS, the importance of maximizing dimensions of the steel backplate beam became apparent. This is because the natural frequencies of a small steel beam were at very high frequencies (due to the small size and the high speed of sound in steel) that are more difficult to test. Thus it was decided to test an entire backplate with the damping shim and pad lining removed as well as a pad lining beam...
and pad lining plate. The School of Engineering and Information Technology (SEIT) main workshop was commissioned to ‘dismantle’ a few of the brake pads.

A hole was drilled and tapped into the backplate and it was threaded onto the stinger. The backplate was hung up using cotton thread by its ‘ears’. The electrodynamic shaker was utilised to excite the structure and the response was measure across 200 points using the close up unit on the PSV. The pad lining plate was tested in a similar fashion except bees wax was used to connect the plate to the stinger and 300 scan points were measured. The set ups for these experiments can be seen in appendix A. The averaged FRFs of the backplate and pad lining plate are shown as figure 3c and 3d. Note that the resonances of the pad lining plate clearly show a greater amount of damping than the other components.

![Averaged Frequency Response Functions](image)

Figure 3: Averaged frequency response functions of the a) brake calliper, b) mounting bracket, c) backplate and d) pad lining plate

### IV. Model Updating

#### A. Updating of Modal Frequency and Mode Shape

FEMtools 3.7.1 was the program used to update the finite element model. FEMtools finds the difference between the modal parameters of the test and FE model and iteratively updates the finite element model by
varying parameters set by the user. FEMtools model updating uses a least squares iterative method that is based on a sensitivity formulation using a truncated Taylor series expansion. The resulting matrix is of the form

\[
\{\Delta R\} = [S]\{\Delta P\}
\]  (4)

Where the elements of \(\Delta P\) are the unknown adjustments to the parameters required to produce the change \(\Delta R\) in the response vector. The sensitivity matrix \([S]\) contains the gradients of responses \(R\) with respect to parameters \(P\)

\[
[S] = S_{ij} = \frac{\delta R_i}{\delta P_j}
\]  (5)

The sensitivities can be found for all physical properties such as isotropic or orthotropic material properties, boundary conditions, mass and spring stiffness. The updated values of parameters \(P\) are obtained from 4 and 5

\[
\{P\} = \{P^0\} + [G](\{R^e\} - \{R\})
\]  (6)

with \([G]\) the gain matrix computed following Bayesian estimation theory as

\[
[G] = [C_p][S]^T([C_r] + [S][C_p][S]^T)^{-1}
\]  (7)

where \(P^0\) are the starting values of parameters, and \([C]\) are weighting matrices that express the analyst’s confidence in \(P^0\) and the reference responses test data \(R^e\). Iterations are continued until error functions satisfy a convergence criterion. A thorough explanation of the mathematical process used by FEMtools can be found in Dascotte (1995).

Methods of correlating experimental and numerical mode shapes are grouped into two categories: visual or numerical comparison. The modal assurance criterion (MAC) provides a numerically quantifiable degree of conformity between experimental and numerical mode shapes which gives us a better level of accuracy over visual methods of comparison (Tison, Heussaff, Massa, Turpin, Nunes & R.F 2014). The MAC was used as the response in the model updating process. The software was able to match mode shapes of the test and FE model using the MAC and iteratively reduce the frequency difference between them. The material properties were used as the variables that are optimised by the software. FEMtools successfully reduced the average frequency difference to around 2 % for all components except the brake rotor. The brake rotor had many more matched mode shapes and found more of a compromise, reducing the frequency difference to an average of 6.1%. The updated material properties of all components are summarised in table 1. In addition, tables of the matched mode shapes used during the updating process are given as Appendix B.

Problems arose as the software could not converge to a solution when attempting to update the FE models of the pads. To solve this issue, modal testing was completed on the backplate and pad lining plate as outlined in section III. FE models of the backplate and pad lining plate were created in ABAQUS and updated individually using FEMtools. Then the updated material properties were applied to the models of the entire inner and outer pad sub-assemblies for comparison with the scans of the complete brake pads.

### Table 1. Updated Material Properties of all Brake Components

<table>
<thead>
<tr>
<th>Part</th>
<th>Brake Rotor</th>
<th>Caliper</th>
<th>Bracket</th>
<th>Backplate</th>
<th>Pad Lining</th>
<th>Outer Pad</th>
<th>Inner Pad</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>Cast Iron</td>
<td>Cast Aluminium</td>
<td>Cast Iron</td>
<td>Mild Steel</td>
<td>Specialised Composite</td>
<td>Specialised Composite</td>
<td>Specialised Composite</td>
</tr>
<tr>
<td>Density (\rho) (kg/m(^3))</td>
<td>7191</td>
<td>2699</td>
<td>6912</td>
<td>7722</td>
<td>2504</td>
<td>2504</td>
<td>2504</td>
</tr>
<tr>
<td>Young’s Modulus (\epsilon) (GPa)</td>
<td>133.2</td>
<td>65.2</td>
<td>161.4</td>
<td>197.7</td>
<td>11.91</td>
<td>3.59</td>
<td>3.59</td>
</tr>
<tr>
<td>Shear Modulus (G) (GPa)</td>
<td>86.6</td>
<td>43.4</td>
<td>70.9</td>
<td>129.5</td>
<td>1.77</td>
<td>1.79</td>
<td>1.79</td>
</tr>
<tr>
<td>Poisson’s Ratio (\nu)</td>
<td>0.3</td>
<td>0.33</td>
<td>0.14</td>
<td>0.31</td>
<td>0.288</td>
<td>0.149</td>
<td>0.149</td>
</tr>
<tr>
<td>Matched Mode Shapes</td>
<td>39</td>
<td>7</td>
<td>5</td>
<td>12</td>
<td>7</td>
<td>7</td>
<td>8</td>
</tr>
<tr>
<td>Average Frequency Difference %</td>
<td>6.1</td>
<td>3.85</td>
<td>0.98</td>
<td>1.8</td>
<td>0.91</td>
<td>3.1</td>
<td>2.76</td>
</tr>
</tbody>
</table>

**B. Updating of Damping**

The next step in the model updating procedure was to update the damping of the components. This was achieved using the same method outlined by Esgandari et al. (2013). Values of modal damping were ascertained for all brake components during modal testing. However for the purposes of CEA of the brake
system in ABAQUS, the only means of including damping is to apply either structural or Rayleigh damping. As such, the most common method of including damping is to use Rayleigh damping applied to the material of each of the individual components (Elvenkemper, Wegmann, Wang, John Flint & Tseng 2006). The classic Rayleigh damping formulation as defined by Caughey (1960) is

\[ [C] = [M] \sum_{k=0}^{p-1} \sigma_k ([M]^{-1}[K])^k \]  

(8)

The simplest case is for proportional damping consisting of the first two terms from equation 8 that results in

\[ [C] = \alpha [M] + \beta [K] \]  

(9)

Where \( \alpha \) and \( \beta \) are mass proportional and stiffness proportional damping coefficients respectively. This two parameter model is the form of Rayleigh damping that is commonly used by finite element software including ABAQUS (Liu & Gorman 1995). The damping ratios corresponding to the two parameter model are

\[ \xi = \frac{\alpha}{2\omega} + \frac{\beta \omega}{2} \]  

(10)

Rayleigh damping is found by curve fitting the modal damping ratios in MATLAB with equation 10. Figure 4 shows the Modal and Rayleigh damping of the outer pad without damping shim up to 20 kHz. Rayleigh damping coefficients have been found using this method for all brake components and are summarised in table 2.

![Rayleigh Damping Curve Fitting - Outer Brake Pad](image)

Figure 4: Damping of the Outer Brake Pad fitted with the Rayleigh two parameter model

As stated in section II, one of the possible causes of brake squeal is mode coupling. Mode coupling occurs when two modes, similar in mode shape and close in modal frequency, coalesce under the influence of friction causing the system to become unstable (Ouyang, Nack, Yuan & Chen 2003). These unstable eigenmodes enter into a state of limit cycles and can be associated with squeal propensity. They can be identified using complex eigenvalue extraction with modern FE software. Any mode that has a complex eigenvalue with a positive real part indicates the presence of negative damping in the system and hence instability.

Table 2. Rayleigh Damping Coefficients

<table>
<thead>
<tr>
<th></th>
<th>Brake Rotor (no shim)</th>
<th>Outer Pad (no shim)</th>
<th>Outer Pad (with shim)</th>
<th>Inner Pad (no shim)</th>
<th>Inner Pad (with shim)</th>
<th>Backplate</th>
<th>Pad Plate</th>
<th>Brake Calliper</th>
<th>Mounting Bracket</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \alpha )</td>
<td>118.2</td>
<td>315.4</td>
<td>553.7</td>
<td>337.1</td>
<td>746.7</td>
<td>323.7</td>
<td>1075</td>
<td>493</td>
<td>124.6</td>
</tr>
<tr>
<td>( \beta )</td>
<td>2.84E-9</td>
<td>1.105E-7</td>
<td>1.128E-7</td>
<td>8.88E-8</td>
<td>1.19E-7</td>
<td>-2.93E-9</td>
<td>2.41E-7</td>
<td>8.58E-8</td>
<td>5.13E-8</td>
</tr>
</tbody>
</table>

V. Stability Analysis

As stated in section II, one of the possible causes of brake squeal is mode coupling. Mode coupling occurs when two modes, similar in mode shape and close in modal frequency, coalesce under the influence of friction causing the system to become unstable (Ouyang, Nack, Yuan & Chen 2003). These unstable eigenmodes enter into a state of limit cycles and can be associated with squeal propensity. They can be identified using complex eigenvalue extraction with modern FE software. Any mode that has a complex eigenvalue with a positive real part indicates the presence of negative damping in the system and hence instability.
ABAQUS 6.13-1 renders a relatively straightforward analysis method. It is able to model each component individually without the need to match the mesh in coincident components. In addition, the model is able to include nonlinear effects in the extraction of the complex eigenvalues (Simulia 2007).

Complex eigenvalue analysis was completed on a simplified rotor and two pads model in ABAQUS. The rotor and pads were meshed using quadratic tetrahedral elements and baseline material properties were applied to the components. The mesh was refined in the contact region of the rotor. The pads were assembled onto the brake rotor and the contact interface between the pads and rotor was defined as small-sliding. Initially the friction coefficient at this interface was set to zero. Then the boundary conditions were applied to the model. The six wheel nut holes of the brake rotor were constrained from motion in any direction. Also, the ‘ears’ of the pads were constrained to allow motion in the out of plane direction of the brake rotor only, which simulates the effect of the mounting bracket on the pads. The first step involved applying a brake pressure of 7.5 bar to the backplates of the pads. This can be seen in figure 5.

In the next step a rotation of 10 rad/s was applied to the brake rotor and $\mu$ was increased to 0.37. To ensure any comparisons are valid, the values of pressure and friction coefficient are average values of the actual conditions from the brake squeal dynamometer testing completed by CBI. The next step involved finding the natural frequencies of the system using modal analysis. Subsequently all the complex eigenvalues of the brake system were calculated. In an attempt to improve the prediction quality a mesh independence study was completed. After many runs of the CEA whilst increasing the number of elements, the largest mesh seed size that is mesh independent was determined. The subsequent mesh independent complex eigenvalue analyses utilised 99282 quadratic tetrahedral elements. The result of the CEA using baseline material properties, constant friction coefficient and a fine mesh is shown in blue in figure 6. It is plotted alongside the results from the cold temperature squeal tests from CBI. CEA is notorious for the over-prediction of unstable modes that often do not correspond to squeal and this is no exception. However squeal at 6.5 kHz and 9 kHz is successfully predicted by this initial CEA with un-updated material properties.
Subsequently the updated material properties specified in table 1 were applied to the pad lining, backplate and brake rotor and the CEA was repeated. The CEA using updated material properties (shown in black in figure 6) is an improvement over the CEA with baseline material properties as all squeal frequencies are predicted. Please note that some of the unstable modes do correspond to brake noise under 70 dB. Though, according to the SAE J2521 brake dynamometer standard, brake noise is only evident above normal cabin noise of around 70 dB(A) so this noise level is taken as the minimum for brake squeal.

The effect of adding the Rayleigh damping to the materials is demonstrated in figure 7. As expected this had the effect of reducing the number of unstable modes as well as reducing the level of instability. The addition of damping shims to the brake pads had a much greater damping effect which is also expected. A velocity dependent friction coefficient was tested and somewhat improved the prediction however none of these analyses were effective in predicting squeal in the 9 kHz range. This demonstrates how challenging it is to attain an accurate prediction from CEA. Real effects such as friction, pressure and contact distribution, pad wear, heat generation and damping shims are extremely difficult to model accurately and all have the potential to change the result significantly.

VI. Conclusions and Recommendations

Finite element modelling, modal testing and model updating was successfully completed on all brake system components. The average frequency difference between matched mode shapes from modal testing and FE modelling was reduced to 6 % or less. A new method of performing modal testing and model updating of brake pads was developed, with the resultant brake pad FE models showing very good correlation in terms of their modal parameters. Complex eigenvalue analysis was completed on a pad on disc FE model and brake squeal was predicted at the corresponding frequencies. First and foremost the updated FE model showed an improvement on the prediction of unstable modes. Further, the addition of Rayleigh damping, the damping shims on the brake pads and a velocity dependent friction coefficient was successful in thinning out some of the over-predicted unstable modes. Further modal testing and model updating is required to validate the model of the assembled brake system before the brake calliper and mounting bracket can be added to the FE model for more in depth analyses of the full brake system.

In terms of improving the prediction quality of the complex eigenvalue analysis, a thorough investigation into the material properties, hyperelastic behaviour and damping of the damping shims must be completed.
In addition, a better friction model incorporating a pressure, temperature and velocity dependent friction coefficient must be used. An investigation into the contact area and pressure distribution of the pad-rotor interface is also an important omission. Most importantly, the brake calliper and mounting bracket should be added to the model as the additional stiffness contributed by these components would further ameliorate the over-predictions.

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References

Døssing, O. (1988), Structural Testing Part 1: Mechanical Mobility Measurements, 1st edn, Bruel and Kjaer, Naerum, Denmark.,
Simulia (2007), Automotive brake squeal analysis using a complex modes approach, Technical report, Providence USA.