VIBRO–ACOUSTIC STUDIES OF BRAKE SQUEAL

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Abstract

In recent years, brake squeal has become an increasing source of customer dissatisfaction and of warranty cost. During squeal, the brake system is operating in a resonant and unstable vibration mode. Research undertaken at UNSW@ADFA to predict brake squeal propensity will be summarised. Experimental modal testing of a sample brake system was firstly used to develop and validate a numerical model based on the finite element technique. The unstable modes were then predicted by applying complex eigenvalue analysis to the finite element model. The frequencies of these unstable modes compare favourably with recorded audio signals of brake squeal. Three methods for assessing which brake components need to be modified in order to reduce brake squeal propensity are described. They are: strain energy, feed-in energy and modal participation. A practical example is given to illustrate how all these methods can be used to predict and reduce brake squeal propensity.

Nomenclature

$\boldsymbol{C}$	damping matrix
$\boldsymbol{K}$	stiffness matrix
k	spring stiffness
$\boldsymbol{K_f}$	friction coupling matrix
$\boldsymbol{M}$	mass matrix
U_i	strain energy of i^{th} component
$\boldsymbol{u}$	displacement vector
μ	coefficient of friction

Introduction

Brake squeal has been a source of customer dissatisfaction and of warranty cost. This is not only because the squeal is annoying but also because customers frequently interpret a squealing brake as an indication of a defective brake. As cited by Kinkaid et al [1], manufacturers of materials for brake pads spend up to 50% of their engineering budgets on noise, vibration and harshness issues according to Abendroth & Wernitz [2].

Brake squeal is generally considered to be caused by the excitation of an unstable vibration mode of the brake system where excitation occurs at the friction interface between the brake pads and the rotor as the brakes are applied. Under this condition, the brake rotor with large flat surfaces can act as a loudspeaker to radiate sound. The number of modes for a typical brake rotor within the human audible range can be well in excess of 80. Brake noise and vibration has been classified according to its frequency as judder, groan, hum, squeal, squelch and wire brush [3]. The occurrence of the squeal noise, usually in the frequency range of 1 kHz to 16 kHz, is sometimes non-repeatable and has been shown to be dependent on a large number of variables, such as brake pad pressure, temperature, material properties of brake pads, rotor geometry, calliper stiffness, calliper mounting bracket, pad attachment method, etc [4].

Despite considerable research into understanding, predicting and eliminating brake squeal since the 1930s [4], the noise generation mechanism is far from being fully understood and its prediction is difficult. This is partly because brake squeal is a transient phenomenon and partly because a brake system is assembled from many components with complex and varying interface conditions. Recent comprehensive reviews on brake squeal are available [1,5].

In this paper, the approach undertaken at UNSW@ADFA to predict brake squeal propensity is described. Practical examples are given to illustrate how this approach can be used to evaluate the effectiveness of treatment to reduce brake squeal propensity.

Numerical Prediction of Brake Squeal Propensity

The approach adopted in this study is to develop the capability of predicting brake squeal propensity and of evaluating options for controlling brake squeal at the design stage. As shown schematically in Figure 1, our approach consists of 4 main stages:

- Development of a finite element (FE) structural model for a brake system;
- Application of complex eigenvalue analysis to the structural model in order to identify unstable vibration modes;
- Development and use of energy methods and the modal participation method to identify components for treatment to eliminate brake squeal propensity;
- Evaluation of treatment options by incorporating changes into the FE structural model and conducting complex eigenvalue analysis to determine whether the unstable modes have been eliminated.

Each of these stages will be described in the following sections.

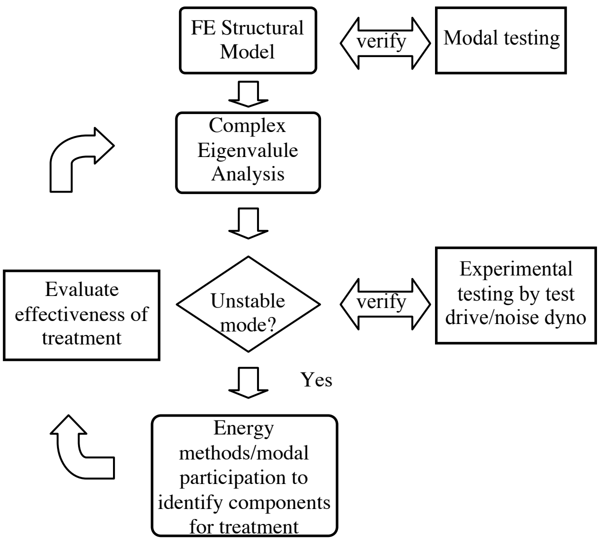


Figure 1. Schematic of approach to predict brake squeal propensity and to evaluate control options.

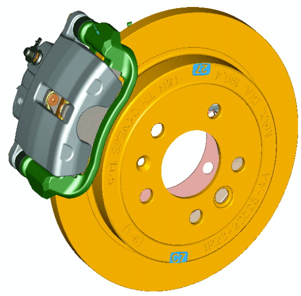


Figure 2(a) Complete brake assembly.

Figure 2(b) Individual brake components: pads, piston, pins, calliper housing, anchor bracket and rotor (clockwise from top).

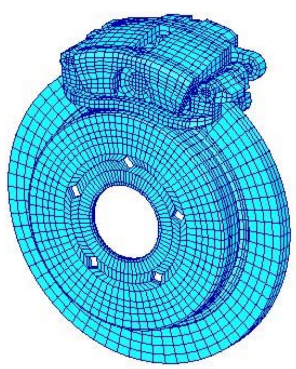


Figure 2 (c) FE model of the complete brake system.

Finite Element (FE) Structural Model

The brake system studied is fairly typical of a modern automotive disc brake system as shown in Figure 2(a). It consists of six components: pads, piston, pins, calliper housing, anchor bracket and rotor, as depicted in Figure 2(b). A FE structural model for the brake system was developed using the preprocessor MSC Patran and solved with the commercial finite element code, MSC Nastran. This FE code was selected because it has well-developed dynamical analysis capabilities that allow the implementation of the friction coupling at the pad / rotor interface, and complex eigenvalues and mode shapes for large models to be computed. This FE model was validated using experimental modal testing results of the physical brake system.

As already described in [6], experimental modal testing has been conducted on individual brake components with free boundary conditions and also on the assembly with 0 and 2 MPa brake line pressure applied respectively. These modal testing results enable the effect of coupling between various components on modal frequencies and damping to be assessed and also allow the FE structural model to be updated. It has been found that the modal characteristics of the brake rotor dominate in the assembled brake system. Significant damping and a reduction in modal peaks were observed when a typical braking pressure of 2 MPa was applied. The validated FE model of the complete brake system with 9858 8-node brake elements is shown in Figure 2(c).

Complex Eigenvalue Analysis

The equation of motion for the free vibration of a multi-degree-of-freedom system is given by

$$\boldsymbol{M}\ddot{\boldsymbol{u}} + \boldsymbol{C}\dot{\boldsymbol{u}} + \boldsymbol{K}\boldsymbol{u} = 0 \qquad (1)$$

where $\boldsymbol{M}$ is the mass matrix, $\boldsymbol{C}$ is the damping matrix, $\boldsymbol{K}$ is the stiffness matrix and $\boldsymbol{u}$ is the displacement vector.

As shown in Figure 3, the contact at the brake pad/rotor interface is simulated by connecting coincident nodes on the pad and the disc with linear springs. The springs prevent penetration of pad into the rotor and have a contact stiffness value based on the stiffness of the adjacent pad elements given by:

$$k_j = \frac{0.1}{n}\sum_{i}^{n}\frac{A_i E}{L_i} \qquad (2)$$

where k_j is the contact stiffness of the j^{th} spring, A_i and L_i are the surface area and length of the adjacent pad elements, n is the number of adjacent pad elements and E is the Young's modulus of the pad material. A static load analysis was performed to obtain the system's state under a typical braking pressure. Spring elements under tension were removed and the analysis re-run usually for several iterations before a satisfactory static solution was obtained [7].

In order to simulate the friction coupling between the pad and the rotor, a simple friction law of the following form is used for the friction force:

$$f_x = \mu f_y = \mu k_j (u_{iy} - u_{(i+1)y}) \qquad (3)$$

where f_x is the friction force, f_y is the normal force at the friction interface, μ is the coefficient of friction, k_j is the contact stiffness and u_i's are the displacements of the coincident nodes as shown in Figure 3. In equation (3), the pad has been assumed to be in constant sliding contact with the disc and that the tangential vibration velocities are small in comparison to the sliding velocity so that the friction force does not reverse direction. This friction force provides a forcing function for Equation (1), which can be rearranged as

$$\boldsymbol{M}\ddot{\boldsymbol{u}} + \boldsymbol{C}\dot{\boldsymbol{u}} + \left[\boldsymbol{K} - \boldsymbol{K}_f\right]\boldsymbol{u} = 0 \qquad (4)$$

where $\boldsymbol{K}_f$ is the friction stiffness matrix, which couples forces in the friction interface normal direction to the tangential direction and is asymmetric.

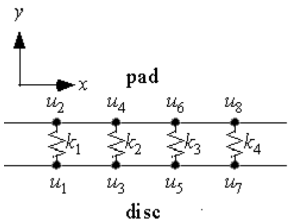


Figure 3 Connection of coincident nodes on the pad/disc interface with linear springs.

By solving equation (4) as an eigenvalue problem, complex eigenvalues can be obtained with the real part representing the modal damping and the imaginary part representing the modal frequency. A positive real part indicates the mode is unstable. There are 108 modes between 0 and 12 kHz for the system. For μ=0.5, the unstable modes with a positive real part (hence negative damping ratio) are listed in Table 1.

Table 1 Summary of unstable modes for the assembled brake system, μ = 0.5.

Mode No.	Eigenvalue	Frequency (Hz)	Damping ratio (%)
27	83.07 + 20871j	3322	-.398
43	219.0 +29283j	4661	-.748
54	27.29 + 37123j	5908	-.074
73	51.84 + 51951j	8268	-.099
79	64.46 + 55776j	8877	-.116
81	49.78 + 56426j	8981	-.088
105	215.4 + 74521j	11860	-.289

The effect of increasing friction coefficient from 0 to 0.5 on the real part of the eigenvalue for two system modes 104 and 105 is illustrated in Figure 4. As the friction coefficient is increased beyond μ=0.35, modes 104 and 105 form a stable/unstable pair. It should be noted that no structural damping is included in this model and hence the real part of the eigenvalue is exactly zero for μ less than the threshold of stability. As shown in Figure 5, in mode 104, the rotor is primarily in an out-of-plane bending mode shape, while in mode 105 the mode shape is the 2nd tangential in-plane mode. While very often it is rather difficult to predict whether squeal will actually occur, the formation of a stable/unstable pair of modes is generally used as an indication of high likelihood for squeal to occur [1]. The real part of the eigenvalue also does not correlate to the sound pressure level emitted from a brake system, but a higher magnitude may indicate that squeal is more likely. As shown in Figure 1, a common means of validating brake squeal is by brake noise dynamometer or on-vehicle test. This predicted brake squeal at 11,860 Hz (mode 105) has been validated by a brake noise dynamometer test. We have also developed a double-pulse laser holographic technique to visualise the vibration mode of the brake in a noise dynamometer when it squeals [8].

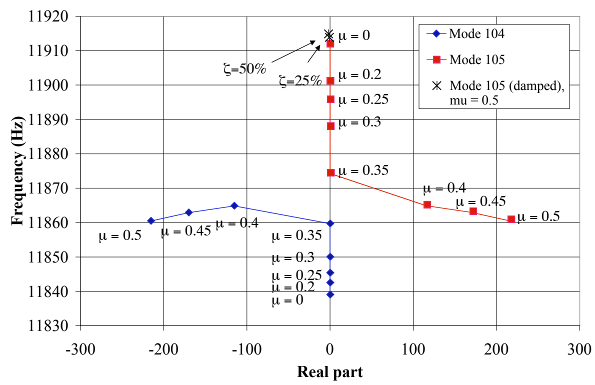
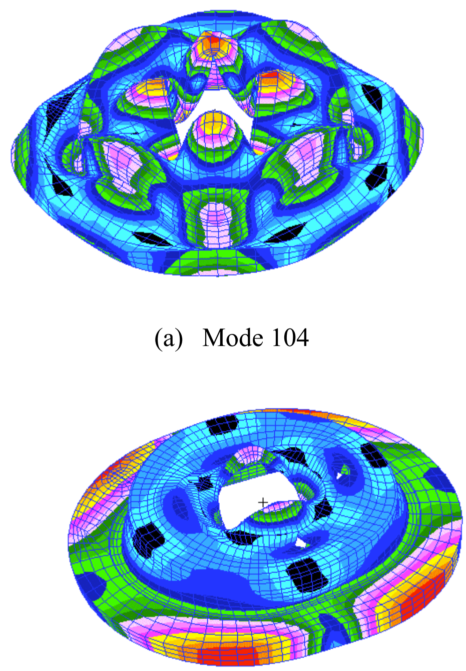


Figure 4 Effect of friction coefficient on two system modes 104 and 105.



(a) Mode 104

(b) Mode 105

Figure 5 Rotor mode shapes.

Assessment of Components' Contributions to Brake Squeal

Although unstable vibration modes can be predicted using complex eigenvalue analysis of a finite element model of a brake system, it alone cannot indicate what modifications will need to be made in order to reduce system instability. Three methods for assessing contributions of individual brake components to squeal are described here, namely strain energy, feed-in energy and modal participation.

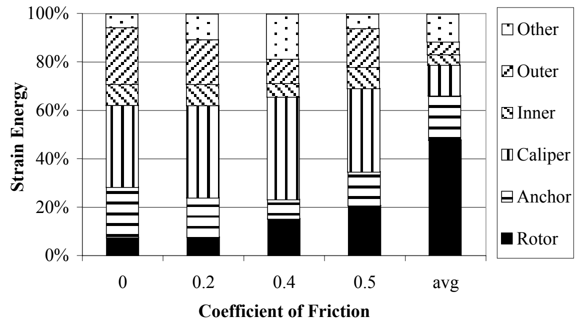
Strain Energy

Vibration of a structure involves displacement about some equilibrium position. Strain energy is the elastic potential energy of the elements of the structure due to the displacement. For a single element i in an FE model, the strain energy U_i can be calculated from the displacement vector $\boldsymbol{u}$ and the stiffness matrix $\boldsymbol{K}$ as

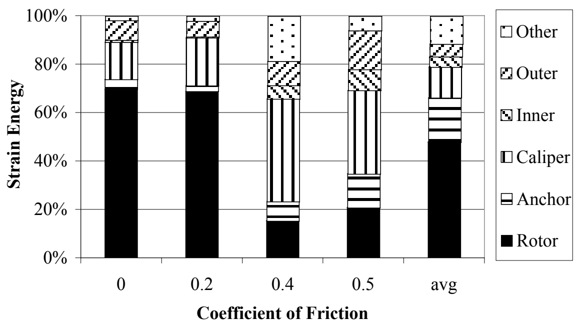
$$U_i = \frac{1}{2}\boldsymbol{u}^T\boldsymbol{K}\boldsymbol{u} \qquad (5)$$

where ^{T} indicates the vector transpose.
Hence the strain energy for a component of the system, can be obtained from the sum of the strain energies of all the elements in that component:

$$U_{component} = \sum_i U_i \qquad (6)$$



(a) Mode 104



(b) Mode 105

Figure 6 Effect of friction coefficient on strain energy distribution for modes 104 and 105.

While the strain energy for every component in a system can be calculated, the difference in size and material properties for components of the system makes it difficult to assess which components are the most active especially for an unstable mode. In order to enable a sensible comparison to be made, an average strain energy distribution for all 108 system modes between 0 and 12 kHz obtained at μ=0 was calculated. Figures 6(a) and (b) display the effect of the coefficient of friction on the strain energy distribution of the rotor, anchor, calliper, inner pad, outer pad and remaining components ("other") for modes 104 and 105 respectively. For μ=0.4 and 0.5, the strain energy distributions of the two modes are equal because they are coupled, as has already been predicted by the complex eigenvalue analysis shown in Figure 4. It can be seen from Figure 6 that for μ=0.5, the outer pad and the calliper are much more active than the average, indicating these components could be prime targets for treatment in order to reduce brake squeal propensity.

Feed-in Energy

When the system described by equation (4) enters an unstable mode, some friction work is converted into vibrational energy. This energy, called feed-in energy, is added to the system due to the relative displacement of the friction interface over a vibration cycle [9].

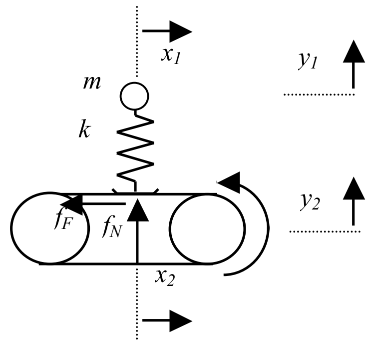


Figure 7 Two-degree-of-freedom system with sliding friction.

Consider a mass sliding on a friction surface as shown in Figure 7. Both the mass and the friction surface can move in the x and y directions. The contact stiffness of the interface is simulated by a linear spring of stiffness k. By assuming that the sliding velocity is much greater than the vibrational velocity, the friction force does not reverse direction. The friction force f_F is a function of the friction coefficient μ and the displacements y_1 and y_2:

$$f_F = \mu f_N = \mu k(y_2 - y_1) \qquad (7)$$

By assuming sinusoidal motions in both x and y directions, the feed-in energy is calculated by integrating the work done by the friction force over a cycle:

$$E_{12x} = \oint_{cycle} f_F\, dx = \mu k \int A_{12x} A_{12y} \sin(\int_{12y} \int \int_{12x}) \qquad (8)$$

where A_{12i} is the amplitude of the difference in motion between the mass and the friction surface in the i^{th} coordinate and λ_{12i} is the phase of the difference. Hence the phase difference determines whether the feed-in energy is dissipative or not. A more thorough description can be found in Guan [7]. The feed-in energy for the i^{th} mode can also be calculated from the viscous work:

$$W_{viscous} = \lambda 2\lambda\ \mathrm{Re}(\lambda_i)\,\mathrm{Im}(\lambda_i)\boldsymbol{u}_i^T \boldsymbol{M}\boldsymbol{u}_i \tag{9}$$

where λ_i is the complex eigenvalue and ^{T} indicates the vector transpose.

The feed-in energy for the 7 unstable modes of the brake system with μ=0.5 is given in Table 2. It is quite clear that the feed-in energy mode 105 is the highest, indicating that this mode is most likely to squeal and that any treatment should be aimed at reducing the feed-in energy for this mode.

Table 2 Summary of feed-in energy for the unstable modes of the assembled brake system, μ = 0.5.

Mode	Freq(Hz)	Feed-in Energy (J)		
		Inner pad	Outer pad	Total
27	3322	.323	.473	.796
43	4661	2.41	3.03	5.44
54	5908	.600	-.066	.534
73	8268	1.491	.028	1.52
79	8877	2.37	1.54	3.91
81	8981	7.44	-2.28	5.16
105	11860	-1.87	13.97	12.1

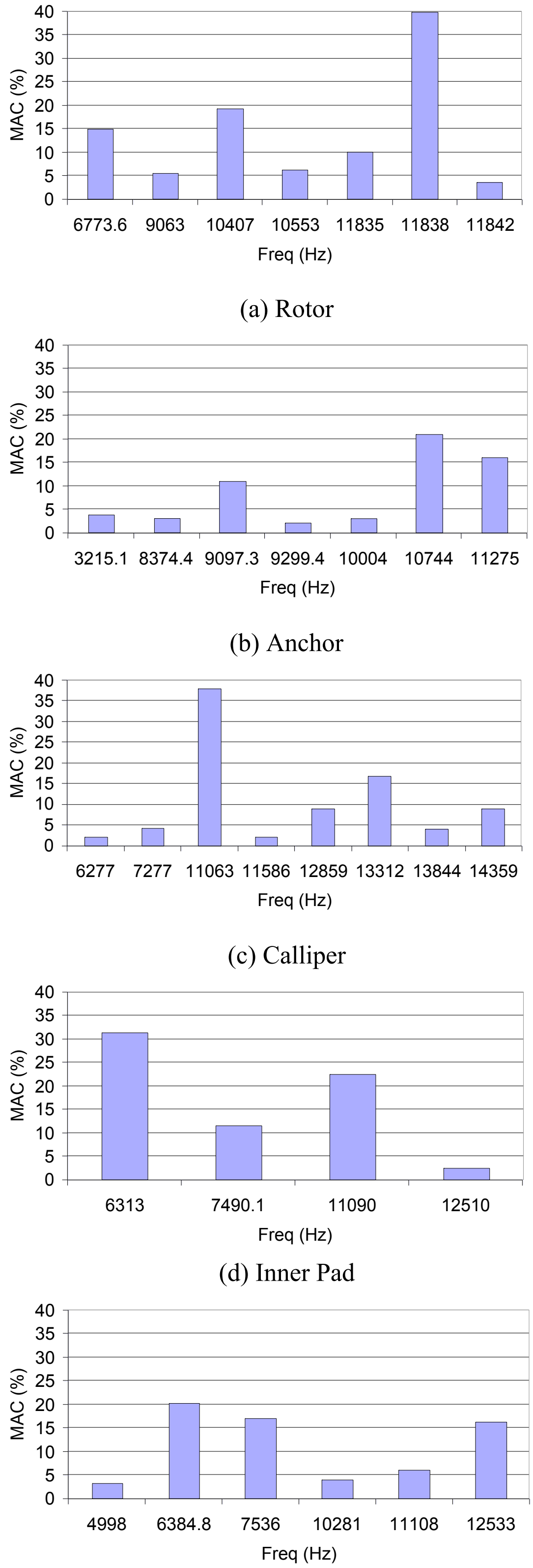
Modal Participation

The correlation between individual component modes under free boundary conditions and within the coupled system can be calculated using the modal assurance criterion (MAC). It provides a quantitative method of assessing which component modes are significant in the overall system vibration. Mathematically, the MAC is a dot product between two vectors, $\boldsymbol{u}_1$ and $\boldsymbol{u}_2$ normalised by their magnitudes:

$$MAC(\boldsymbol{u}_1,\boldsymbol{u}_2) = \frac{\left|\boldsymbol{u}_1^T \boldsymbol{u}_2^*\right|^2}{\left(\boldsymbol{u}_1^T \boldsymbol{u}_1^*\right)\left(\boldsymbol{u}_2^T \boldsymbol{u}_2^*\right)} \tag{10}$$

where ^{T} indicates the vector transpose and * indicates the complex conjugate of a complex valued vector.

The comparison of a real vector and a complex vector presents no difficulty since the conjugate of a real vector is simply the vector itself. MAC data were used primarily for determining which component modes are active in an unstable system mode. For example, a system mode is often identified by the mode shape of the rotor. The MAC allows the identification of the rotor mode shape more easily because it does not rely on visual observation and a qualitative judgement. A value of 1 indicates that two modes are identical (although they could be scaled) and a 0 indicates that there is no correlation between the modes at all. The MAC values of the rotor, anchor, calliper, outer pad and inner pad for the unstable mode 105 at 11860 Hz are given in Figure 8(a)-(e) respectively. It can be seen that there is significant contribution from the rotor, the calliper and the pads to this unstable mode.



(e) Outer Pad

Figure 8 Modal assurance criterion (MAC) for the unstable mode 105 at 11860 Hz. Only more significant modes are shown.

Evaluation of Control Options

Mode 105 with a friction coefficient μ of 0.5 has been predicted to be unstable by complex eigenvalue analysis and has been verified by brake noise dynamometer measurements to cause brake squeal. Although reducing the friction coefficient value to 0.35 would render this mode stable, this is not desirable because a lower coefficient of friction will compromise the performance of the brake.

In order to identify which brake components are to be treated to reduce squeal propensity for mode 105, MAC values in Figure 8 have to be interpreted together with strain energy distributions shown in Figure 6. While the rotor MAC values indicate that a particular mode (11838 Hz) is most dominant (Fig. 8a), its strain energy is much less than the average (Fig. 6b). On the other hand, for the calliper and pads, their MAC values also identify particular modes that are quite significant and their strain energies are much higher than the average, indicating that treatment to these components might help reducing or even eliminating brake squeal.

With respect to the pad, it is clear from Table 2 that the outer pad is solely providing the feed-in energy for mode 105 and the inner pad is actually providing friction damping to the system. This suggests that by treating the pad motion, especially of the outer pad, might help stabilise this mode. Damping shims attached to the back of the pads were simulated in the FE model with two different levels of damping applied to the model. The structural damping values for the damping shims used were 25% and 50%, representing the limits of the range specified by the manufacturer of the shims. As shown in Figure 4, the overall system damping for mode 105 is positive and increases with increase in pad damping, thus indicating that this mode is now stable. On the other hand, mode number 27 is largely unaffected by the addition of damping shims because (i) the feed-in energy is rather small in the original unstable mode; and (ii) the MAC values and strain energy distributions (not included here) for this mode indicate that pad motion is rather small.

With respect to the calliper, it has been found that by increasing the calliper stiffness by 40%, mode 105 becomes a stable mode.

Conclusions

As rightly pointed out by Kinkaid et al [1], despite considerable efforts undertaken since the early 20th century, there is still no method to completely suppress disc brake squeal. While this is a complex and difficult problem, we have attempted to develop a systematic approach based on finite element modelling to predict brake squeal propensity. Although complex eigenvalue analysis has been shown to be able to predict unstable modes for a typical brake assembly, methods based on strain energy, feed-in energy and modal participation using modal assurance criterion are required to provide insight into which individual brake components that need to be modified in order to reduce or eliminate brake squeal. While we have achieved some success, a lot more needs to be done as brake squeal is susceptible not only to change in geometry of the brake components but also to change in material properties which are also highly dependent on temperature.

Acknowledgement

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