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BRAKE NOISE IN PRACTICE

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Abstract

Squeal in disc brake systems is an on-going concern for the automotive industry. Development of countermeasures for noise problems represents a large portion of the development cycle in terms of both time and cost. In addition, brake squeal represents a significant source of customer dissatisfaction and warranty cost.

In this paper, an overview of brake noise development will be presented by way of a case study. Through rigorous testing and analysis it was determined that a noise problem could be countered by changing component dynamic behaviour, implemented through metallurgical changes, in combination with geometry modification at a key component interface. The final noise performance of the brake system was found to be a dramatic improvement compared to early testing, and well within the specification required by the vehicle manufacturer. The vehicle has since gone on to perform well in production without noise concerns arising in the field.

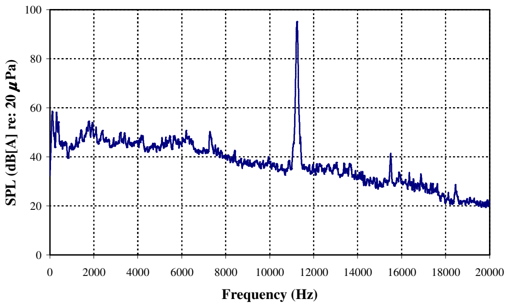
1. INTRODUCTION

Noise, vibration and harshness (NVH) is an area of significant importance in modern automobiles. Customers demand that a new vehicle will offer low interior noise levels and ride comfort. Manufacturers strive to decrease noise even further, and improvements are offered with every new model.

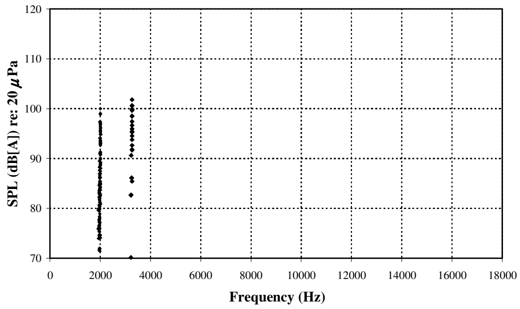
Unfortunately, brake NVH has not improved at a level commensurate with interior noise levels. Of the many types of brake NVH concerns a core focus is brake squeal, defined as brake noise above 1 kHz. Considerable research has occurred over the last several decades in understanding and controlling brake squeal, much of which is summarised in references [1-3].

Brake squeal occurs when the brake system enters into resonant, self-excited vibration. Some vibrational energy is added to the system from the friction interface between the brake rotor and pads which is then dissipated though various mechanisms including emission of sound. It is not uncommon for the sound pressure level (SPL) to exceed 100 dB(A) measured 0.5 m from the brake on a brake noise dynamometer during a squeal event. An example of a squeal spectrum, measured on a noise dynamometer during a SAE J2521 noise matrix test, is shown in Figure 1 (a) [4].

Before any new brake system enters production it has undergone an exhaustive development process that may include modal analysis of components and assemblies, FEA based simulation activities, noise dynamometer testing and on-vehicle testing. The aim of this paper is provide an overview of the activities associated with the noise development of a brake system for a typical passenger vehicle.



(a)



(b)

Figure 1. (a) Typical brake squeal spectrum (b) SPL vs Frequency for noise events over 70 dB(A)

2. DEVELOPMENT TOOLS

NVH development engineers utilise any and all necessary means to deliver a noise free brake system. While the majority of these tools are experimentally based, an increasing amount of analytical work is being undertaken during development.

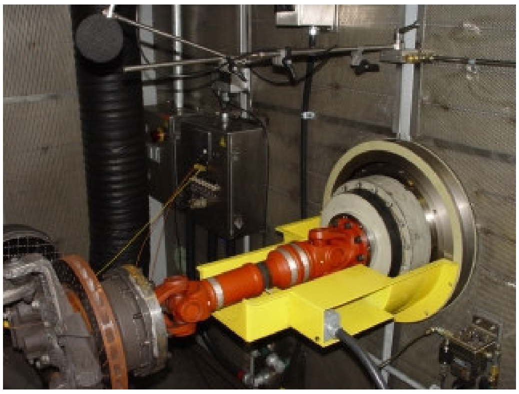
2.1 Lab Based Testing – Modal Testing

The starting point of any physical testing is to understand the dynamical characteristics of the brake system components. Experimental modal testing, using either accelerometer or laser vibrometer measurements, is used to determine the brake system modal properties. This information is put to use in a number of key ways.

- "Modal Maps" display modal frequencies across all system components and possibly vehicle suspension components. Identification of overlapping component frequencies aids identification of potential noise hot spots.
- Modal frequency distribution history is used when changing from one stage of development to another, or during a change of manufacturing process at a later date. Modal frequency distributions are used to identify variation between processes.
- Modal properties, specifically modal frequency and mode shapes, can be used for "tuning" finite element analysis (FEA) models.

2.2 Brake Noise Dynamometer

(a)



(b)

Figure 2. (a) Industrial brake noise dynamometer and environmental system. (b) Internal brake fixture

Brake noise dynamometers are widely used in both industry and research institutions. Many types are in use ranging from simple machines converted from lathes through to state of art systems able to perform a large variety of automated test procedures. Figure 2 displays a typical industrial noise dynamometer and environmental system.

Noise dynamometers are a fundamental tool for noise development activities. They allow repeatable testing to a test matrix of controlled speed, pressure, initial brake temperature (IBT) and environmental conditions while being able to accept a vehicle-representative fixture. A typical method for displaying the results from a noise dynamometer test is shown in Figure 1(b), where the SPL for individual squeal events over 70 dB(A) plotted against frequency.

2.3 On-Vehicle Measurement

Final validation of brake noise performance is based on vehicle testing. Vehicle testing is also used as a development tool, but does not offer the convenience, repeatability, controlled environment nor cost effectiveness of dynamometer testing. However, there are cases where vehicle noise concerns are not detected on the noise dynamometer, most likely due to boundary conditions that are not truly representative of the vehicle, and on-vehicle development is required.

2.4 Finite Element Analysis

FEA has become a widely used development tool in many areas of engineering including brake NVH development. FEA can be applied on a number levels including:

- Component modal analysis used to resolve modal frequency and mode shape information of parts. Indeed, it is common to forgo experimental modal analysis in preference to FEA-based modal results tuned on the basis of simple impact frequency response functions (FRFs). It is also possible to evaluate the effect of structural changes.
- Assembly modal analysis including contact modelling and complex eigenvalue squeal prediction [5,6]. While it is not completely reliable, often it is possible to identify trends and evaluate structural modifications once an adequate baseline correlation has been established.

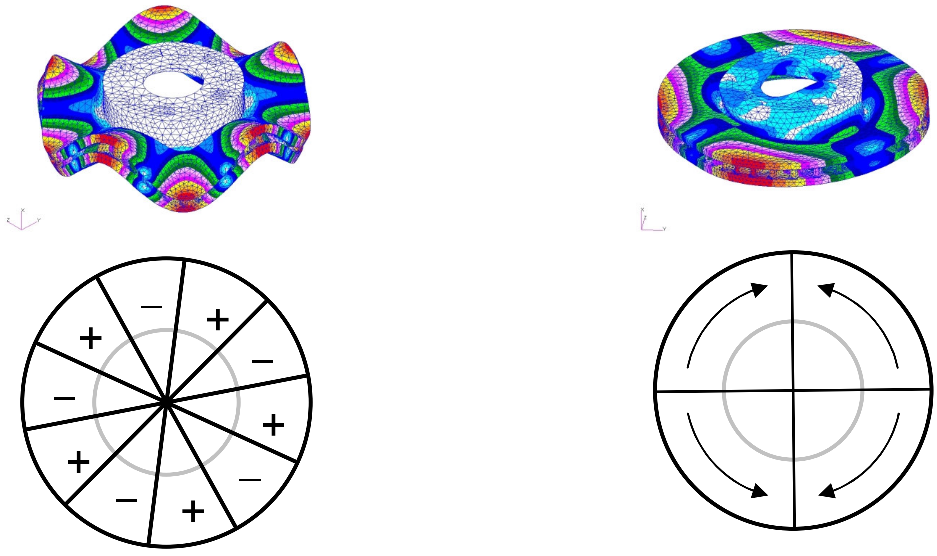
3. CONTROL STRATEGIES

A variety of commonly used noise counter measures will be presented in the current section. Chamfers are a very commonly used noise counter measure used throughout the brake industry, an example of which is shown in Figure 4(b). Chamfers change the footprint at the key rotor / pad friction interface.

Most brake systems in production today are fitted with some type of backplate shim. Two types are in common use; steel shims with grease added to the shim surfaces, and constrained layer dampers where rubber and steel layers alternate. Shims provide decoupling between components and additional system damping.

As a follow on to the modal mapping described in section 2.1, investigation of noise concerns at specific frequencies requires investigation of the modal frequencies of the components themselves. Care needs to be taken since the modal frequencies under free-free boundary conditions may change when the component is constrained in a brake assembly. However, modal frequencies can be used as measure of the variability between parts.

Of the many types of vibration modes that can occur in a brake rotor, diametrical bending modes (Figure 3(a)) and tangential in-plane compression modes (Figure 3(b)) have attracted much attention in the literature. Coupling can occur between the in-plane mode and a bending mode adjacent in frequency, with a resulting squeal frequency at the in-plane modal frequency [7-10]. Guidelines have been developed that suggest the frequency of the in-plane modes should be placed away from the adjacent bending modes. It has also been shown that noise can occur at the in-plane modal frequency even with favourable modal spacing. In one example, the noise was eliminated with rotor design modification that did not change modal frequency spacing [11].



(a) (b)

Figure 3. (a) Bending mode with 5 nodal diameters, (b) 2nd order tangential in-plane compression mode

Most of the structural components within a brake system are made of metal alloys including steel, aluminium, ductile cast iron and grey cast iron. The elastic moduli and mass densities of these alloys are usually relatively insensitive to variation in chemical composition. The exception is grey cast iron, which has an elastic modulus that can vary greatly due to changes in carbon content and other alloying elements. This, in turn, leads to major changes in the modal frequencies of brake rotors. More specific detail can be found in following references [12, 13].

Another key area in brake noise development relates to the compressibility of the brake pad, or more specifically, the compressibility of the friction material. Brake pads consist of two main sections; a steel backplate and the friction material which is molded to the backplate. The friction material is a composite structure with potentially dozen of constituents, the exact composition being proprietary information and known only to the friction supplier. Further, it is not uncommon for an underlayer to be applied between the friction material and the backplate itself, which can add beneficial thermal and mechanical properties. The sum of all these components is a pad that is relatively compressible, with a typical pad showing approximately 0.1mm compression under high brake pressures.

The compressibility of the pad can be shown to have a noise impact on the brake system. It is often seen that increased compressibility tends to reduce noise occurrences. However, increasing compressibility can have a negative impact on other aspects of brake performance such as pedal feel and drag.

4. CASE STUDY

4.1 Development Cycle

The development cycle of a brake system is composed of 3 stages: Design Verification (DV), Off-Tool Validation (OTV) and Process Validation (PV). The key features are outlined in Table 1. Ideally noise countermeasures are developed during DV, but this needs to be confirmed at each stage since changes in tooling and process can lead to changes in key components and noise performance.

Table 1. Development cycle stages.

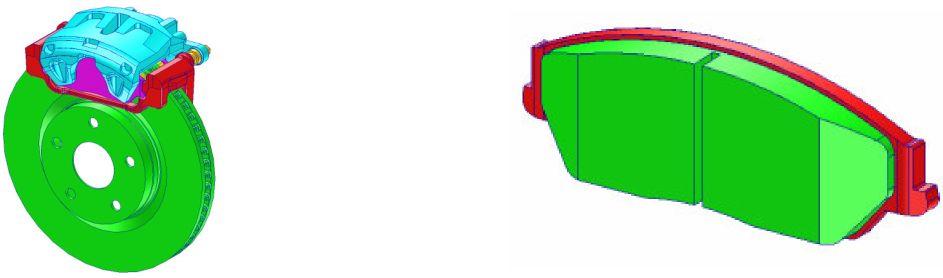
Stage	Tooling	Process
Design Verification (DV)	Prototype	Prototype
Off-Tool Validation (OTV)	Production	Prototype
Process Validation (PV)	Production	Production

4.2 Brake System Description

This discusses the design of a cast iron sliding caliper and vented disc front brake system as found on a typical passenger vehicle. The brake system that was the focus of this investigation is similar to that shown in Figure 4(a).

To develop the brake from a noise perspective two specific test matrices suitable for a brake noise dynamometer were utilized. Noise tests were performed on a shaft dynamometer with a representative vehicle suspension corner.

"Procedure A" was used in DV phase and included both a warm section (50°C to 350°C IBT) and a cold section (0°C to 50°C IBT). No ambient environmental control was used other than refrigeration during the cold section to allow 0°C IBT to be achieved.



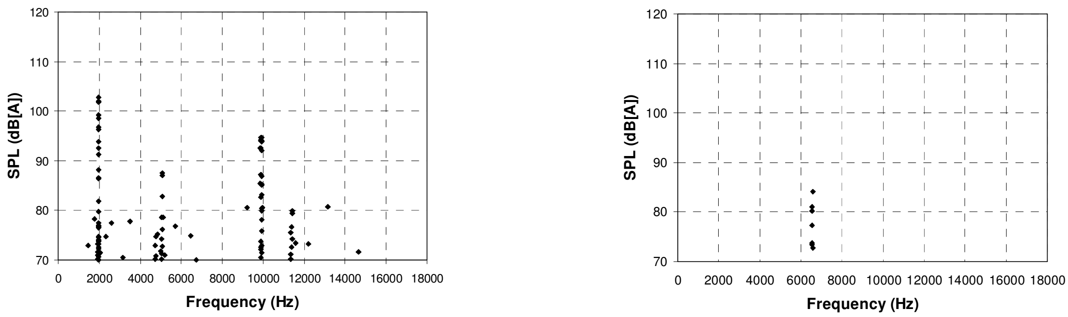
(a) (b)

Figure 4. (a) Example brake system used in the case study. (b) Brake pad with 20mm chamfer

"Procedure B" was introduced during the OTV phase to address a new customer requirement. Compared to Procedure A, the second procedure was lower duty and longer in duration, although IBT was limited to a maximum of 200°C. In addition, it was run to controlled ambient temperature and humidity conditions including tropical hot and humid conditions.

4.3 Design Verification

Brake noise development commenced by performing impact test measurements of each of the brake system components under free-free boundary conditions as a means to fingerprint their natural frequency characteristics.



(a) (b)

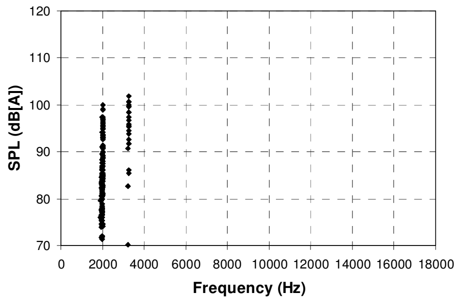
Figure 5. (a) Initial DV noise performance, (b) Final DV noise performance with modified pad chamfer and backplate shim material change.

A baseline dynamometer test on the brake system was run using Procedure A. The results of this test are shown in Figure 5(a). The noise histogram resulting from the baseline brake test is characterised by the presence of 2 kHz, 5 kHz, 6.5 kHz, 10 kHz, and 11.2 kHz squeal frequencies. The amplitude and frequency of occurrence of the squeal frequencies were in excess of the allowable limits per the specification. As such, changes to the design were required to improve the noise performance of the system.

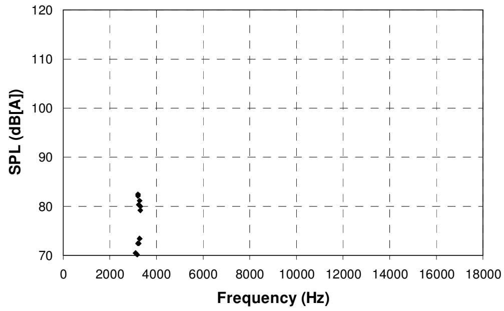
A modification to the noise insulator material and addition of a 20mm parallel chamfer to the pad geometry was made, as shown in Figure 4(b). With these changes, the noise performance of the brake system to Procedure A was improved to the level shown in Figure 5(b). Some minor occurrences of 6.5 kHz squeal were still measured, but these fall within acceptability limits set by the customer.

4.4 Off-Tool Validation

The noise histogram for the OTV system brake to procedure A is dominated by a high occurrence of 2 kHz and 3.2 kHz squeal as can be seen in Figure 6(a). This was not expected considering the brake system design was nominally the same as that which produced the 6.5 kHz squeal shown in Figure 5(b).



(a)



(b)

Figure 6. Procedure A OTV (a) Initial performance, (b) Noise with corrected rotor composition

An investigation into the brake caliper components was conducted to find the root cause of this noise increase. Individual caliper component mass and natural frequency results showed no significant difference between DV and OTV. Further, the brake pad compressibility was found to show negligible difference.

An investigation into the brake rotor followed, and immediately a difference was seen. The frequency response functions, as displayed in Figure 7, show a marked decrease in frequency of the response peaks, which in turn correspond to the modal frequencies.

This change in natural frequency can be due to either a geometry change or a material property change. Measurement of sectioned rotor samples showed little difference in the rotor geometry. However, material composition between the DV and OTV rotors is displayed in Table 2. The natural frequency shift appears to be driven by the change alloying elements copper and molybdenum.

Table 2. DV and OTV rotor composition

	DV	OTV1	OTV (2)
C	3.65	3.59	3.59
Si	2.09	1.95	1.91
Mn	0.73	0.59	0.59
P	0.02	0.02	0.02
S	0.09	0.09	0.10
Cu	0.75	0.29	0.75
Sn	0.07	0.07	0.07
Mo	0.001	0.300	0.001

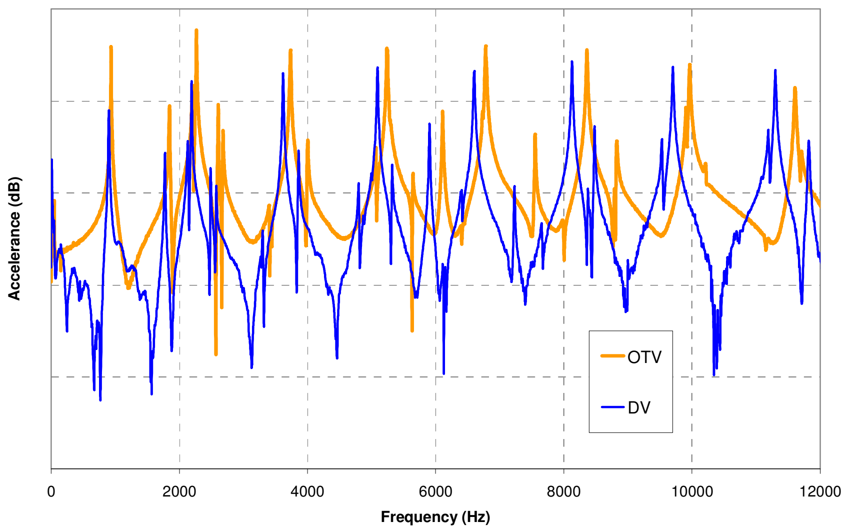
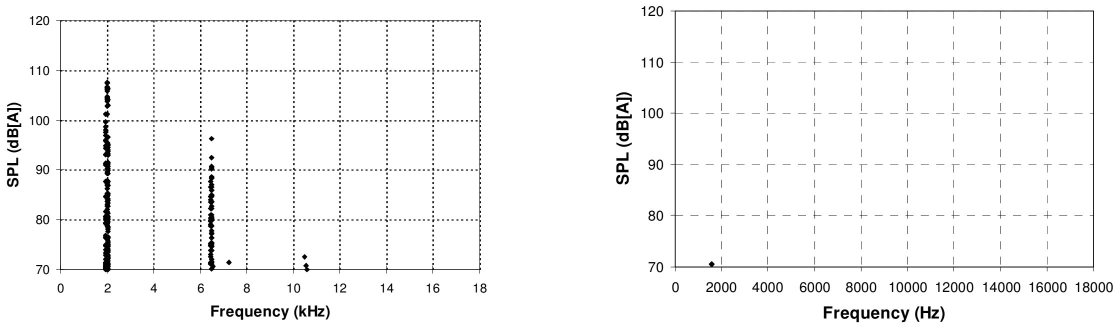


Figure 7. OTV and DV brake rotor frequency response functions.

A new batch of OTV castings were required with implementation of tighter controls on composition. Table 2 shows that the 2nd batch of OTV rotors conformed closely to the DV composition. Re-running the OTV noise test with the corrected rotors reduced the occurrence back to acceptable levels, as shown in figure 6(b).

An updated customer noise specification, procedure B, was introduced in the later stages of OTV. The noise result is displayed in Figure 8(a) and can be compared to the result in Figure 6(b). This highlights that the system was sensitive to the hot, humid condition of test procedure B, and indicated further development was required to ensure acceptable noise performance in all required conditions.

The development that followed resulted in further refinement to the pads' chamfer configuration as well as changes to the rotor geometry to lower its bending stiffness. The result of these changes can be seen in Figure 8(b), with the system performing well within acceptability limits.



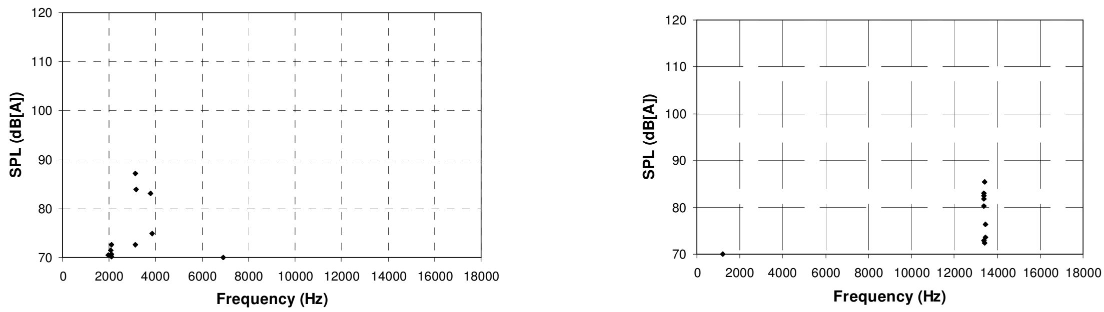
(a) (b)

Figure 8. Procedure B OTV (a) Noise performance with OTV caliper, corrected rotor, (b) Final OTV performance with modified chamfer and rotor

4.5 Process Validation

Shifting to the final stage of the development cycle required a change from prototype process to production process, so the key was to control variation due to the change to production process. Consequently, the properties of components were carefully monitored at the start of production.

The noise performance of the PV brake system is shown in Figure 9. It provides acceptable noise performance for both of the noise procedures. The brake system was also evaluated on an environmentally controlled chassis dynamometer for final sign off by the customer. It has since gone into production and has been free of noise concerns in the field.



(a) (b)

Figure 9. PV noise performance (a) Procedure A. (b) Procedure B.

5. SUMMARY

Using the example of a cast iron front caliper, a case study was presented to highlight the noise development of a brake system. Ambient temperature, humidity, test duty and test cycle affected noise performance of a brake design and produced different results and squeal frequencies

Close monitoring of rotor natural frequencies was required throughout brake noise development. Tight control of rotor material properties was recommended to ensure consistent rotor natural frequencies through development stages and across batches. Tight control of rotor core geometry was also required to minimise natural frequency variation.

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