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DISC BRAKE SQUEAL: PROGRESS AND CHALLENGES

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Abstract

Brake squeal noise has been an ongoing concern with automotive brake systems since their inception. It is generated by the vibration of an unstable vibration mode of the brake system, usually with the brake rotor acting as a loudspeaker. The squeal noise that is of concern usually falls into a frequency range from 1 to 16 kHz. Although there is often not any degradation in the braking performance, most customers tend to interpret brake squeal noise as indicative of a defective brake. Furthermore, with improvements made in interior noise and comfort levels in vehicles, brake squeal noise has become an increasing source of customer dissatisfaction and is a major contributor to warranty cost. Research into predicting and controlling brake squeal has been conducted since the 1930s and despite significant research efforts in the past 2 decades, brake squeal still remains a challenging problem that is begging for a better understanding of its generation mechanisms and better methodologies for countermeasures other than empirical approaches. Brake squeal is a transient phenomenon and is highly dependent on geometries of brake components, complex interface conditions between components and material properties that are functions of both temperature and pressure. In this paper, recent developments in understanding and controlling brake squeal noise will be reviewed and challenges that remain to be met are discussed.

1. INTRODUCTION

Brake squeal noise has been an ongoing concern with automotive brake systems. While other areas of automotive Noise, Vibration and Harshness (NVH) have seen considerable progress, brake noise continues to be a concern. Friction levels of brake materials have tended to increase and the move away from asbestos has resulted in an increasing difficulty in producing brake systems with adequate NVH performance. The net result is customer dissatisfaction and warranty cost. Commensurate with the concerns of car manufacturers about brake squeal is the increase in effort made by the industry and researchers in this area. As shown in Figure 1, over the last 20 years, there is a significant increase in the number of papers on brake squeal published in the period 2001-2005 in journals included in Thomson

ISI citation index. Similarly, Figure 2 shows that the number of brake squeal papers presented at the SAE Annual Brake Colloquium has been increasing steadily from 25% of the total number of papers in the period 1997-1999 to 37% in the period 2003-2005. With the increase in brake squeal research, the understanding of the mechanism of brake squeal has improved and numerical modeling techniques have been shown to have potential in assessing brake squeal propensity and in aiding designs. There have been a number of good reviews on issues related to brake squeal propensity and its analysis, for example, [1-3].

The objective of this paper is to provide an overview of the progress made in analysing and controlling disc brake squeal and the challenges ahead.

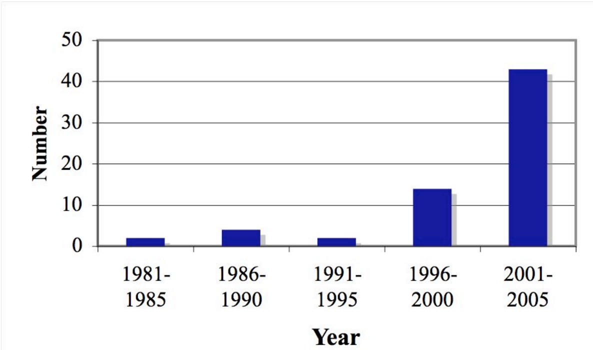

Figure 1 Number of brake squeal papers according to Thomson ISI Web of Science on 4 August 2006.

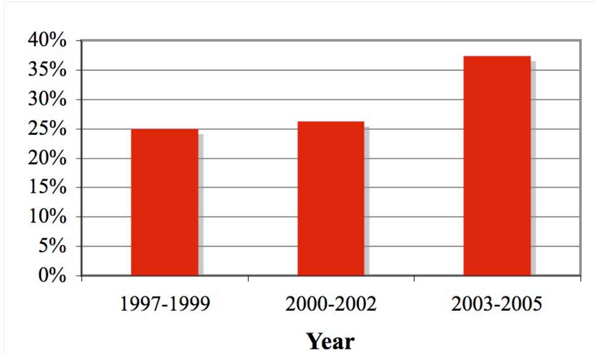

Figure 2 Number of brake squeal papers as a percentage of the papers presented at the SAE Annual Brake Colloquium.

2. CHARACTERISTICS OF BRAKE SQUEAL NOISE

Brake squeal is brake noise that occurs in the frequency range between 1 and 16 kHz. It 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 and is then dissipated though various mechanisms including emission of sound. Brake squeal is also highly non-repeatable. A small variation in speed, pressure, temperature or environmental conditions can lead to significantly different results. Furthermore, it is possible to see great differences in the frequency of squeal with the same nominal test conditions. The results of two noise dynamometer tests for components selected to be matched in terms of component natural frequencies, pad compressibility, environmental conditions and test procedure, are compared in Figure 3. It is clear that the occurrence of noise on one test was far greater than the other.

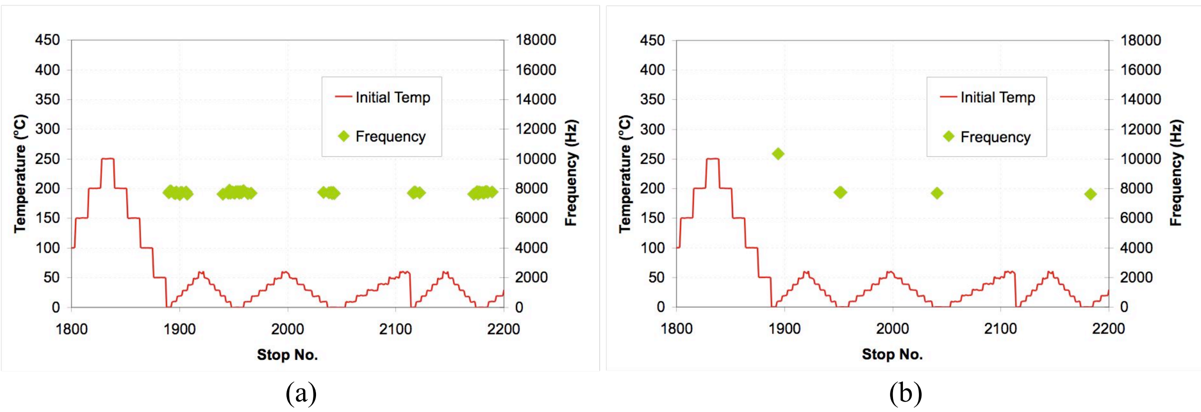

Figure 3. Comparison of the same section of two nominally identical tests on the noise dynamometer. The green dots represent noise occurrences within a single stop.

3. METHODOLOGIES

3.1 Experimental Methods

Experimental approaches to brake squeal analysis revolve about understanding the physical characteristics of the brake system, especially the behavior during a squeal event. This includes evaluating the modal properties of the brake system, both at a component level and as an assembly, investigating the nature of the friction processes and interactions within the system, and also determining the sound radiation of the characteristics of the brake system. Traditionally vibration analysis has been carried out using accelerometer based measurements. This is fine for lab based experimental modal analysis, but it is not easily applied to a squealing brake system, particularly the brake rotor.

A number of optical techniques have been developed to aid investigation of a squealing brake. The main type that has been used with practical success is the Double Pulse Holographic Interferometry (DPHI). DPHI provides excellent temporal and spatial resolution, but is somewhat specialised and suitable mostly for laboratory research [4, 5]. Scanning Doppler laser vibrometer systems offer considerable practical advantages over DPHI, and now practical, easy to use systems have been employed by several research groups, and even in industrial development [6]. While most suited to scanning artificially excited objects, fast scan systems have become available to visualise a squeal event.

The work horse in industry is the brake noise dynamometer which can provide vehicle representative test configurations in a controlled environment. This reduces the need for vehicle testing other than for validation of dyno test results and final sign off on noise performance. A variety of lab based analysis tools are also used in industry, including accelerometer and laser vibrometer based experimental modal analysis and operational deflection shape measurement of individual components, assemblies and even brake systems during squeal. However, the equipment used, and the ease of use must be of a level more suitable to an industrial environment rather than a research environment. Experimental methods still dominate development within the brake industry. Unfortunately it mostly falls back on trial and error methods, and many decades of experience in solving noise issues. An overview of brake noise development in industry is given in [7].

3.2 Brake Squeal Models

The earliest investigations into brake squeal noise employed very much simplified models with few degrees of freedom. While most of the current analysis of brake systems has shifted into the realm of finite element analysis, these simplified models have helped clarify some of the key mechanisms of brake squeal and are summarized as follows.

- Stick-Slip: This is essentially caused by the difference between static and kinetic friction coefficients. During the stick phase, energy is stored in the system and once the transition to slip occurs, the stored energy is released generating an impact excitation [8].
- Decreasing kinetic friction coefficient μ_k with increasing sliding speed: When μ_k decreases with increasing sliding speed, instability can arise due to negative damping [8].
- Sprag-slip: It has been shown that squeal can still occur even if μ_k is constant, as a result of geometrically induced instabilities or kinematic constraints [9].
- Modal coupling: When a system's modes are dynamically locked in together, energy can be efficiently transferred, resulting in instability [10,11].
- Hammering: This refers to mechanical excitations caused by disc imperfections (uneven rotor contacts during rotations), resulting in instability [12].

3.3 Finite Element Analysis

Finite Element Analysis (FEA) has grown into a widely used tool in many areas of science and engineering. Brake squeal analysis is another area where FEA has provided some useful progress. FEA is used in a number of ways including modal analysis of components and assemblies and for predictive work to assess brake squeal propensity. The physical system is usually modelled using commercially available software codes (such as NASTRAN, ANSYS and ABAQUS) to solve the steady-state equations of motion. It is not until recently that transient analysis has been attempted [13,14]. The equation of motion for the free vibration of a multi-degree-of-freedom system is given by

$$\boldsymbol{M}\ddot{\boldsymbol{u}} + \boldsymbol{C}\dot{\boldsymbol{u}} + \boldsymbol{K}\boldsymbol{u} = 0 \tag{1}$$

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.

The modelling of the friction coupling at the brake pad/rotor interface is critical to the analysis because the friction interface between the pad and the rotor is the source for self-excitation of a squealing brake. One approach is to simulate the brake pad/rotor interface by connecting coincident nodes on the pad and the rotor with linear springs [15,16] such as in NASTRAN and another approach is to use contact elements such as in ABAQUS. In the former approach, the resulting friction coupling between forces in the friction interface normal direction and tangential direction can be incorporated into equation (1) via an asymmetric friction stiffness matrix $\boldsymbol{K_f}$:

$$\boldsymbol{M}\ddot{\boldsymbol{u}} + \boldsymbol{C}\dot{\boldsymbol{u}} + \left[\boldsymbol{K} - \boldsymbol{K}_f\right]\boldsymbol{u} = 0 \tag{2}$$

3.3.1 Component modelling

Analysis of individual components is rather straight forward using FEA. Brake components, with the exception of the brake pads, are made from materials with isotropic properties that are easily updated to match experimental modal frequency data. The modeling of brake pads can be quite complex because the pad is formed from friction material, itself a composite material with approximately orthotropic material properties, and a backplate of steel. Assessing the impact of modification on a component level is one area where FEA modal analysis comes into its own.

3.3.2 Assemblies and contact modelling

Brake assemblies have been analyzed in a number of ways, firstly to understand the modal properties of the assembly in much the same way as was done for the components, and secondly for brake squeal prediction. Prediction of squeal propensity most commonly uses complex eigenvalue analysis [17, 18]. A brake system consists of 8 separate components shown in Figure 4(a), more if the steering knuckle or suspension components are included in an analysis. The interfaces between these components are a critical part of developing a brake system assembly. Connecting components with spring elements as described above is a gross simplification of the contact stiffness behaviour seen at the interface, and a suitable stiffness needs to be calculated on the basis of the surrounding elements in the components. Further to limitation of using contact springs, friction coupling needs to be applied manually. An example of this process is described in [16]. Contact elements allow for the contact modelling to be handled in an automated manner. Contact surfaces are defined on the components prior to the start of an analysis and areas of contact are determined at each increment of an analysis.

Examples of analysis using contact elements are described in [19-21].

Non-linear static analysis in itself can be useful for understanding some aspects of a brake system in operation. Many brake system modifications in practice revolve around modifying contact pressure distributions at the pad/rotor interface. Static analysis can help understand how such a modification will influence dynamic behaviour even though it doesn't directly include dynamic analysis.

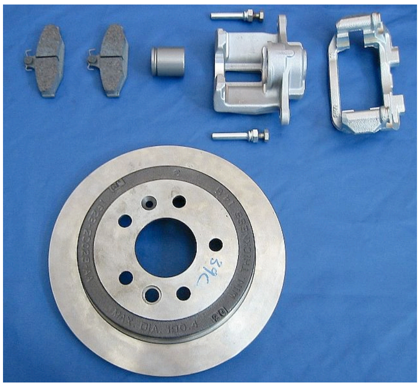

Figure 4(a) Brake system components Clockwise for the top left; pads, piston, guide pins, caliper housing, and bracket and disc rotor.

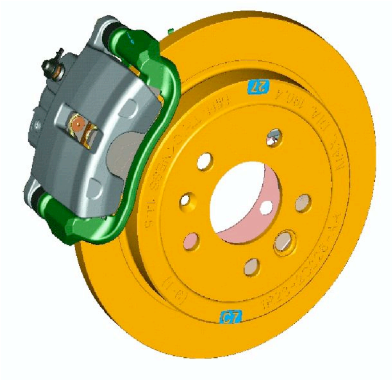

Figure 4(b) Brake system assembly.

3.3.3 Additional Analysis of FEA Models

Once a predictive model achieves good baseline correlation to a noise problem, an assembly can be further analysed with the following methods to probe the behavior of the brake system.

- The Strain Energy Method [16]: For a given vibration mode, the strain energy of each component can be calculated from the sum of the elastic potential energy of the elements of that component. This indicator can then be used to compare the role of each component between various unstable vibration modes and gives a focus on which component may be most usefully modified.
- The Feed-in Energy Method [22]: When the system described by equation (2) 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. The amount of feed-in energy can be calculated by considering the normal forces across the friction surface and the relative motions throughout a cycle.
- Modal Participation Method [16]: 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 gives an indication of which component modes are significant in a given overall unstable system vibration mode. This allows a more effective judgement to be made as to what component modes need to be modified.

3.3.4 Application of FEA to a Case Study

By solving the complex eignevalue problem in equation (2) for the complete brake assembly in Figure 4 using the complex Lanczos method in Nastran with a coefficient of friction μ=0.5 and no structural damping, 108 complex eigenvalues (hence modes) were extracted between zero and 12 kHz. This analysis identifies 7 unstable modes for which the damping is negative, as given in Table 1. Although it has been suggested by Liles [17] that modes with higher negative damping values are more likely to squeal, the relationship between squeal propensity and the level of negative damping is not clear. Furthermore, the relationship between the emitted sound level and the negative damping level is not known. The squealing frequencies identified in noise dynamometer tests fall in the range 6 – 6.5 kHz, 7.5 – 8 kHz and 11.5 – 12 kHz, corresponding respectively to the unstable modes 54, 73 and 105. On the other hand, the two modes with the highest negative damping levels are 27 and 43 occurring at 3.3 kHz and 4.7 kHz but there was no squeal for 3 – 4 kHz (mode 27) and only occasional occurrences of squeal at 4.5 – 5 kHz (mode 43) in all the noise dynamometer screening. Hence the level of negative damping alone is not a good predictor of brake squeal propensity.

The effect of increasing the 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 5. As the friction coefficient is increased beyond μ=0.35, modes 104 and 105 form a stable/unstable pair. As no structural damping was included in this model, the real part of the eigenvalue is exactly zero for μ less than the threshold of stability. Although it is not easy 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 feed-in energy results in Table 1 suggest that the unstable mode 105 is most likely to squeal. It is also clear that the feed-in energy is provided by the outer pad and the inner pad is actually dissipative. The advantage of FEA can be seen from the detailed information that can be extracted for individual component. While table 1 lists the total feed-in energy for the outer pad and the inner pad, Figure 6 displays the distribution of the feed-in energy over the outer pad and inner pad respectively. Such information could be very useful in designing shapes for the pads. The strain energy distribution (not shown here) for the unstable mode 105 indicates less rotor deformation than average, slightly higher than average level for the anchor and considerably more strain energy for the caliper housing and pads, in particular the outer pad. These results together suggest that the likely candidates for treatment to reduce/eliminate brake squeal propensity for the unstable mode 105 are the outer pad and caliper housing. Further insight can be gained by examining the MAC values. Though not shown here, the MAC results suggest that controlling the bending motion of the backplate of the outer pad, for example by adding backplate shims, would aid in reducing the instability of 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 5, the overall system damping for mode 105 is positive and increases with increase in pad damping, thus indicating that this mode is now stable.

Mode	Freq(Hz)	Damping ratio (%)	Feed-in Energy (J)		
			Inner pad	Outer pad	Total
27	3322	-.398	.323	.473	.796
43	4661	-.748	2.41	3.03	5.44
54	5908	-.074	.600	-.066	.534
73	8268	-.099	1.491	.028	1.52
79	8877	-.116	2.37	1.54	3.91
81	8981	-.088	7.44	-2.28	5.16
105	11860	-.289	-1.87	13.97	12.1

Table 1 Unstable modes for the brake system.

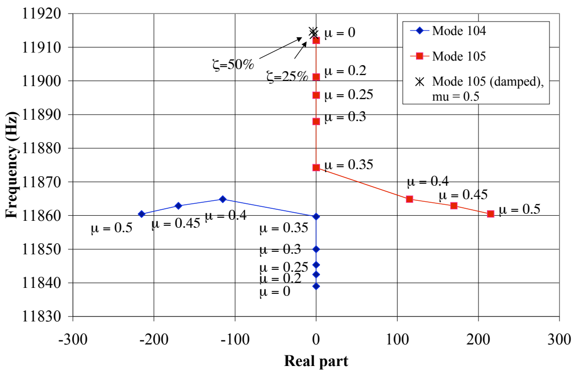

Figure 5 Effect of the coefficient of friction on modes 104 and 105.

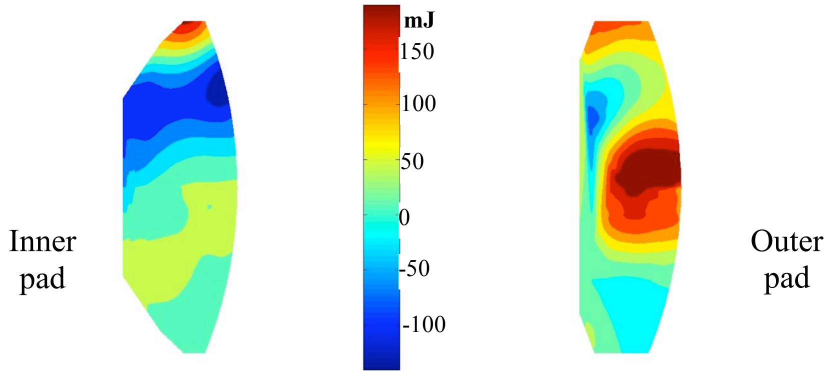

Figure 6 Feed-in energy distribution for inner pad and outer outer, ustable system mode 105.

4. FUTURE CHALLENGES AND DIFFICULTIES

Disc brake squeal noise is a problem that is far from resolved for the automotive industry. This applies to some degree to all approaches, be they experimental, analytical or numerical. Considerable progress has been made on experimental visualization techniques. DPHI has limitations in complexity and applicability to an industrial environment. Scanning laser systems have limited spatial and temporal resolution. Systems that could overcome these limitations would greatly assist in day to day brake noise development. Work also continues in correlating results between dynamometer and vehicle testing. It is not possible to fit a whole vehicle onto a shaft driven dynamometer, so inevitably some level of compromise is required with regard to the number of components that should be included.

The key limitation in understanding brake squeal and progress in numerical modelling is the complexity of boundary conditions between components. This was highlighted by the comparison in Figure 3. It is these changes on a micro scale, which may be imperceptible to normal laboratory measurements on the components and their interfaces, that represent the greatest hurdle. While current contact modelling tools do offer some scope for analysis, they appear to be inadequate for duplicating the true complexity of the interaction between components.

Modal based solution techniques by their nature require the system to be linearised about some base state. In reality the brake system is highly non-linear. The current trend of numerical modelling approaches appear to be heading towards more sophisticated modelling of all the influential interacting parameters such as contact conditions, temperature and pressures, and use of transient non-linear analysis. The FEA based predictive models, as presented here, do provide good correlation with experimental testing under some conditions, but the models are far from being a primary development tool within the brake industry.

5. SUMMARY

A brief overview of the progress made in automotive disc brake squeal noise has been presented. The main areas of current research have been highlighted and the challenges for making further progress have been discussed. Modern FEA based analysis does provide good correlation in some cases. However, experimental techniques and development on a trial and error basis still dominates the brake industry. The difficulty in brake noise analysis is the ability to capture the true complexity and non-linearity of the component interfaces. This will be a key area of development if FEA predictive models are to progress in the future. For the foreseeable future, numerical modelling and experimental testing will be used in a complementary manner to control brake squeal.

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