Review on Brake Squeal

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***Abstract*— Brake squeal noise has been under investigation by automotive manufactures for many years due to consistent customer complaints and high warranty costs. J. D. power survey from automobile industry show brake noise as being one of the most critical vehicle quality measurements. Disc brake squeal remains a complex problem in the automotive industry, since the early 20th century, many researchers have examined the problem with experimental, analytical and computational techniques. This paper provides review and bibliography of works on disc brake squeal. It also includes review on characteristic of brake squeal. A reviews of the analytical, experimental and numerical methods used for the investigation of brake squeal is given.**

***Index Terms*—Brake Squeal, Brake squeal analysis, Brake squeal mechanism.**

I. INTRODUCTION

In the past 30 years the automotive industry has seen a transition from drum brakes to disc brakes. This transition was made with the purpose of improving performance and reducing mass. The most important quality factor of brake systems can be considered brake noise. Brake squeal, which usually occurs in the frequency range between 1 and 12 kHz, has been one of the most difficult concerns associated with vehicle brake systems. Disc brake noise, in general, is one of the major contributors to the automotive industry's warranty costs. In most cases, this type of noise has little or no effect on the performance of brake system. However, most customers perceive this noise as a problem and demand that their dealer's fix it. Customer complaints result in significant yearly warranty costs. More importantly, customer dissatisfaction may result in the rejection of certain brands of brake systems or vehicles. The automotive industry is thus looking for new ways to solve this problem.

In general, brake noise has been divided into three categories, in relation to the frequency of noise occurrence. The three categories presented are low frequency noise, low-frequency squeal and high-frequency squeal. Low-frequency disc brake noise typically occurs in the frequency range between 100 and 1 kHz. Typical noises that reside in this category are grunt, groan, grind and moan. This type of noise is caused by friction material excitation at the rotor and lining interface. The energy is transmitted as a vibratory response through the brake corner and couples with other chassis components. Low-frequency squeal is generally classified as a noise having a narrow frequency bandwidth in the frequency range 1-3 kHz. The failure mode for this category of squeal can be associated with frictional excitation coupled with a phenomenon referred to as ''modal locking'' of brake corner components. Modal locking is the coupling of two or more modes of various structures producing optimum conditions for brake squeal. High-frequency brake squeal is defined as a noise which is produced by friction induced excitation imparted by coupled resonances (closed spaced modes) of the rotor itself as well as other brake components. It is typically classified as squeal noise occurring at frequency range 3-12 kHz. Since it is a range of frequency which affects a region of high sensitivity in the human ear, high-frequency brake squeal is considered the most annoying type of noise.

Brake squeal is a concern in the automotive industry that has challenged many researchers and engineers for years. Considerable analytical, numerical and experimental efforts have been spent on this subject, and much physical insight has been gained on how disc brakes may generate squeal, although all the mechanisms have not been completely understood. No precise definition of brake squeal has gained complete acceptance, but it is generally agreed that squeal is a sustained, high frequency greater than 1 kHz vibration of brake system components during a braking action resulting in noise audible to vehicle occupants or passers-by. There exists no general means for completely eliminating brake squeal. Brakes that squeal do not, in general, squeal during every braking action. Rather, the occurrence of squeal is intermittent or perhaps even random. Many different factors on both the micro- and macroscopic levels appear to affect squeal, and some of these factors (especially on the microscopic level) are not well understood. As different brake squeal experiments produce widely different and even conflicting results. Generally, a central difficulty in modeling brake squeal is one of scales. Effects on very small scales in length and time (i.e., microscopic contact phenomena and high-frequency vibrations) interact in important ways with effects on large scales (such as wear over the life of the brake and dynamics of large vehicle substructures). A number of theories have been formulated to explain the mechanisms of brake squeal, and numerous studies have tried with varied success to apply them to the dynamics of disc brakes.

II. HISTORICAL BACKGROUND

A century ago, the British engineer Frederick William Lanchester (1868–1946) patented a disc brake. In his 1902 patent he described a disc brake consisting of a disc of sheet metal which is rigidly connected to one of the rear wheels of the vehicle. To slow the vehicle, the disc is pinched at its edge by a pair of jaws. This period also evidenced many of the early developments in brake technology. For instance according to Newcomb and Spurr Mercedes (Daimler Motor Gesellschaft) and Renault both introduced variants of the modern (internal expanding) drum brake in 1903. On the other side of the Atlantic in the 1890s, according to Hughes the American inventor Elmer Ambrose Sperry (1860–1930) invented a brake featuring an electromagnetically actuated disc. The disc, which was known as a brake magnet, was placed in contact with another disc (known as a brake disc) to achieve a braking torque. Interestingly, Sperry noted that the braking torque he attained was partially due to friction between the discs and partially due to Foucault (eddy) currents. Needless to say, the early brake designs of Lanchester and Sperry were substantially modified during the 20th century. In particular, both the materials and actuation methods used have been improved. According to Harper, one of the main arenas for these developments was the aviation industry during the Second World War. Aircraft disc brakes are what are known as clutch-type. That is, the friction pads contact the disc in an annular region which extends over most of angular extent of the disc. In contrast, in spot-type disc brakes, which are now used in automobiles, the angular extent of the friction pads ranges from 301 to 501: Lanchester's 1902 design is a spot-type disc brake, whereas Sperry's is a clutch-type disc brake. The evolution of spot-type disc brakes in automobiles can be traced to developments by the Dunlop, Girdling, and Lockheed Corporations in the 1950s. Their spot-type disc brakes are substantially similar to those present in automobiles today. The widespread use of disc brakes on the front wheels of passenger vehicles can be partially attributed to increasingly stringent regulations throughout the world on vehicle braking. For example, prior to the 1970s, most Automobiles in the United States were equipped with front wheel drum brakes. This situation changed with the introduction of the Federal Motor Vehicle Safety Standard (FMVSS) No. 105. FMVSS No. 105, versions of which became effective for passenger vehicles on January 1, 1968 and 1976, imposed standards on stopping distance, brake fade and water resistance for automotive braking systems. As the brakes on the front wheels contribute 70– 80% of the braking power, the braking systems for the front wheels were crucial to satisfying FMVSS No. 105. Compared to drum brakes, disc brakes have superior water resistance and fade performance. These features, coupled with the oil crisis and the increased popularity of imported automobiles, contributed to the profusion of disc brake systems in the United States.

1) Brake noise generation mechanisms

Disc brake squeal occurs when a system experiences vibrations with very large mechanical amplitude. There are two theories that attempt to explain why this phenomenon occurs. The first theory states that brake squeal is a result of a stick–slip mechanism. Second theory is mode coupling theory. It states that high levels of vibration result from geometric instabilities of the brake system assembly. Both theories, however, attribute the brake system vibration and the accompanying audible noise to variable friction forces at the pad–rotor interface. According to the first hypothesis, the stick–slip theory, a variable friction coefficient with respect to sliding velocity between pads and rotor, provides the energy source for the brake squeal. Several studies based on this theory were conducted when disc brakes were first used on automobiles. Squeal noise was found to be more likely when the decreasing relationship between the friction coefficient and the sliding velocity become pronounced. An increase in the negative slope did not always increase the occurrence of squeal, however, the need for an alternative or accompanying theory was revealed. In the case of geometric instability, the second theory, the variable friction forces are caused by variable normal forces. Even if the coefficient of friction is constant, variable friction forces are still possible. Investigations based on this hypothesis began with Spurr and were advanced by Earles and Soar. In this case, two system modes that are geometrically matched move closer in frequency as the friction coefficient increases. These two modes eventually couple at the same frequency and become unstable.

2) Characteristics of brake squeal

One of the biggest contributors to a brake squeal is the friction material, since Squeal excitation occurs at the friction interface, and it normally takes approximately 12 months to finalize a friction material selection. This certainly makes it very difficult to predict a priori the propensity of a brake system to squeal. Also, often in the design of a brake system, priority is given to requirements such as braking performance, cost and ease of manufacture. The common practice for the different components of a brake system to be manufactured by different suppliers further complicates matters. The large number of vehicles produced means that even a low squeal propensity found during initial testing of a brake system can become a major concern once a vehicle is in production due to a much larger population size. Modifications towards the end of development phase will have two potential risks: (1) leading to production delays and increased costs to both the brake and vehicle manufacturers and (2) leading to products not fully validated with potential field warranty concern. The most significant complication in brake research is the fugitive nature of brake squeal; that is, brake squeal can sometimes be non-repeatable. There are many potential squeal frequencies (unstable modes) for a brake system. Each individual component has its own natural modes. The modal frequencies and modal shapes of the rotor, caliper, anchor and pad will change once these parts are installed. During a brake application, these parts are dynamically coupled together resulting in a series of coupled vibration modes, which are different from the component free vibration modes. The addition of the friction coupling forces at the friction interface results in the stiffness matrix for the system containing un-symmetric off-diagonal coupling terms. From the stability point of view, this coupling is considered to be the root cause of the brake squeal. A brake system may not always squeal given the ''same'' conditions. Alternatively, small variations in operating temperature, brake pressure, rotor velocity or coefficient of friction may result in differing squeal propensities or frequencies. Figs.1and 2 show the percentage occurrence of brake squeal obtained at PBR Automotive Pty Ltdusing a Rubore drag type noise dynamometer and an AK noise matrix for various brake pressures and the temperatures respectively. It can be seen from Fig.1that there is no simple relationship between the percentage occurrence and frequency of the brake squeal and the brake pad pressure. Similarly, the influence of temperature on both the occurrence and frequency of the brake squeal is quite complex (Fig. 2). Due to the above-mentioned difficulties in designing a noise free brake system, efforts to eliminate brake squeal have largely been empirical, with problematic brake systems treated in a case by case manner. The success of these empirical fixes depends on the mechanism that is responsible for causing the squeal problem. The most fundamental method of eliminating brake squeal is to reduce the coefficient of friction of the pad material. However, this obviously reduces braking performance and is not a preferable method to employ. The use of viscoelastic material (damping material) on the back of back plate can be effective when there is significant pad bending vibration. Changing the coupling between the pad and rotor by modifying the shape of the brake pad has also been found effective. Other geometrical modifications that have been successful include modifying caliper stiffness, the caliper mounting bracket, pad attachment method and rotor geometry.

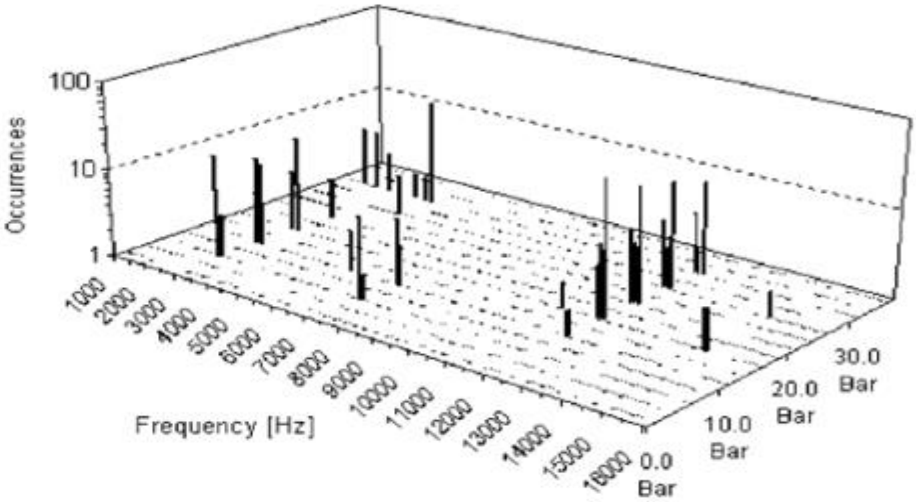


Fig.1 Variation of occurrence of brake squeal with frequency and brake pad pressure

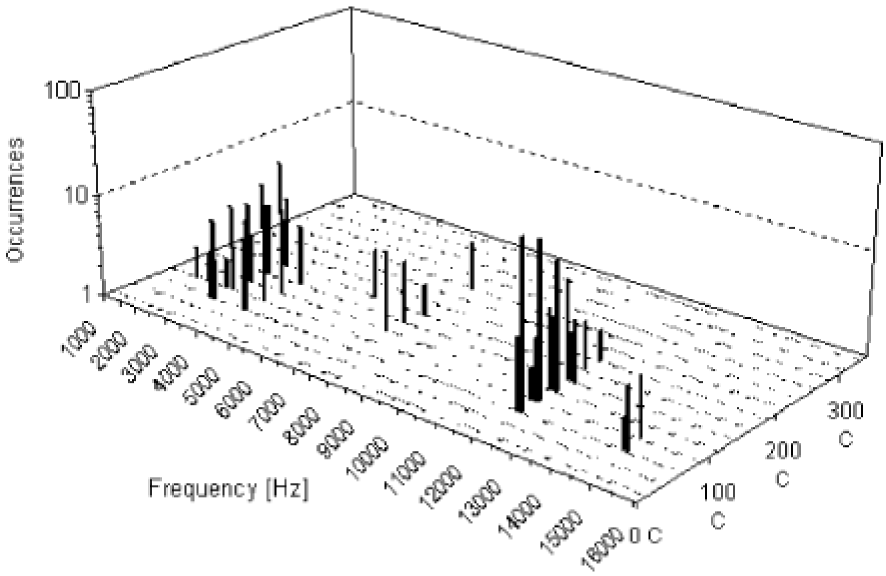


Fig.2 Variation of occurrence of squeal with frequency and temperature

III. ANALYSIS OF BRAKE SQUEAL

1) Analytical methods

The earliest research into brake squeal suggested that the variation in the friction coefficient with sliding velocity was the cause. Not only is there a difference between the static and dynamic coefficient of friction, but it was thought the drop in kinetic friction with increased sliding velocity could lead to a stick-slip condition and produce self-excited vibration. However, squeal has been shown to occur in brake systems where the coefficient of kinetic friction is constant and has led to analysis of the geometrical aspects of a brake system. Spurr proposed an early sprag-slip model that describes a geometric coupling hypothesis in 1961. Consider a strut inclined at an angle θ to a sliding surface as shown in Fig.3 (a). The magnitude of the friction force is given by

$$F = (\mu L)/(1 - tan\theta)$$

Where, μ is the coefficient of friction and L is the load. It can be seen that the friction force will approach infinity as μ approaches cot θ. When μ=cot θ the strut 'sprags' or locks and the surface can move no further. Spurr's sprag-slip model consisted of a double cantilever as shown in Fig. 3(b). Here, the arm $O'P$ is inclined at an angle θ' to a moving surface. The arm will rotate about an elastic pivot O' as P moves under the influence of the friction force F once the spragging angle has been reached. Eventually the moment opposing the rotation about O' becomes so large that $O''P$ replaces $O'P$, and the inclination angle is reduced to θ''. The elastic energy stored in O'can now be released and the $O'P$ swings in the opposite direction to the moving surface. The cycle can now recommence resulting in oscillatory behaviour. Others extended this idea in an attempt to model a brake system more completely. Jarvis and Mills used a cantilever rubbing against a rotating disc in 1963, Earles and Soar used a pin-disc model in 1971, and North introduced his eight-degree of freedom model in 1972. The culmination of these efforts was a model published by Millner in 1978. Millner modelled the disc, pad and caliper as a 6 degree of freedom, lumped parameter model and found good agreement between predicted and observed squeal. Complex eigenvalue analysis was used to determine which configurations would be unstable. Parameters investigated included the coefficient of pad friction, Young's modulus of pad material, and the mass and stiffness of caliper. Squeal propensity was found to increase steeply with the coefficient of friction, but squeal would not occur below a cut off value of 0.28. He found that for a constant friction value, the occurrence of squeal and squeal frequency depends on the stiffness of pad material (Young's modulus). Caliper mass and stiffness also displayed distinct narrow regions where squeal propensity was high. The common conclusions of these models are that brake squeal can be caused by geometrically induced instabilities that do not require variations in the coefficient of friction. Since these closed form theoretical approaches cannot adequately model the complex interactions between components found in practical brake systems, their applicability has been limited. However, they do provide some good insight into the mechanism of brake squeal by highlighting the physical phenomena that occur when a brake system squeals.

Experimental methods are expensive mainly due to hardware cost and long turnaround time for design iterations. Frequently discoveries made on a particular type of brakes or on a particular type of vehicles are not transferable to other types of brakes or vehicles. Product development is frequently carried out on a trial-and-error basis. There is also a limitation on the feasibility of the hardware implementation of ideas. A stability margin is usually not found experimentally. Unfortunately, this produces designs that could be only marginally stable.

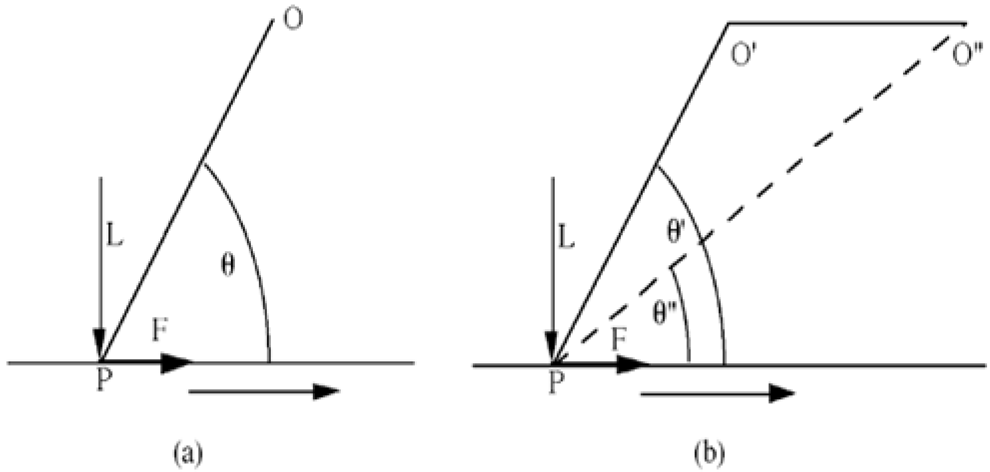


Fig 3 (a) Single strut rubbing against moving surface; (b) Sprag slip system

2) Experimental methods

The frequencies of a squealing brake are highly dependent on the natural frequencies of the brake rotor. Consequently it is of fundamental importance to be able to determine the vibration modes of the rotor. Not only will an understanding of the vibration modes of the rotor help predict how a brake system may vibrate, but it is also necessary in developing countermeasures to eliminate the problem. The existence of in-plane modes in addition to the bending modes is a further complication, and there is evidence that the in-plane modes can be the cause of some type of brake squeals as well as the bending modes. Accelerometers provide an effective tool for determining the vibration mode shapes and the forced response of a system. Fig. 4(a) shows a bending mode shape of a typical brake rotor that has been determined experimentally. A model was created using STAR MODAL software that consisted of 384 grid points over the surface of a brake rotor. Frequency response measurements were made with a B&K 2032 FFT analyser using a B&K 4374 uni-axial accelerometer and a B&K 8001 impedance head. The excitation was introduced with a B&K 4810 shaker driven by a random noise signal. Unfortunately, the contact mounting required for accelerometers limits their usage on rotating brake components. They can only be used for analysis of stationary brake components making it almost impossible to determine the mode shapes of a squealing brake rotor. Optical techniques have been used more recently. In particular, double pulsed laser holographic interferometry has been successfully applied to squealing brake systems. This has allowed the coupled mode shapes of a complete brake system to be determined while it is squealing. A holographic image is produced by triggering a laser at the maximum and minimum amplitude of a vibrating object. The difference in optical path length, caused by the deformed shape of the vibrating object, creates an interference fringe pattern on a holographic plate. The mode shape can then be determined by interpreting the fringe pattern. The advantage of holographic interferometry is that the mode shapes of a brake rotor can be determined while it is squealing. Included in the holographic image can be the rotor as well as the pads, anchor bracket and caliper. The technique can be applied to a brake system mounted on a brake dynamometer. Suspension components, such as the spindle, spring and damper, can also be included to simulate the on car performance of the brake system. An example of the value of double pulsed holography in investigating a squealing brake was work done by Nishiwaki et al. in 1989. In the brake system that was being investigated it was apparent that the mode shape of the vibrating brake rotor was stationary with respect to the brake caliper. Hence, the mode shape is also stationary with respect to the area of excitation. The rotor was modified by changing the symmetry of the rotor about its axis of rotation. The mode shapes of the modified rotor must now rotate with respect to the area of excitation, preventing the rotor from vibrating in the original vibration mode.

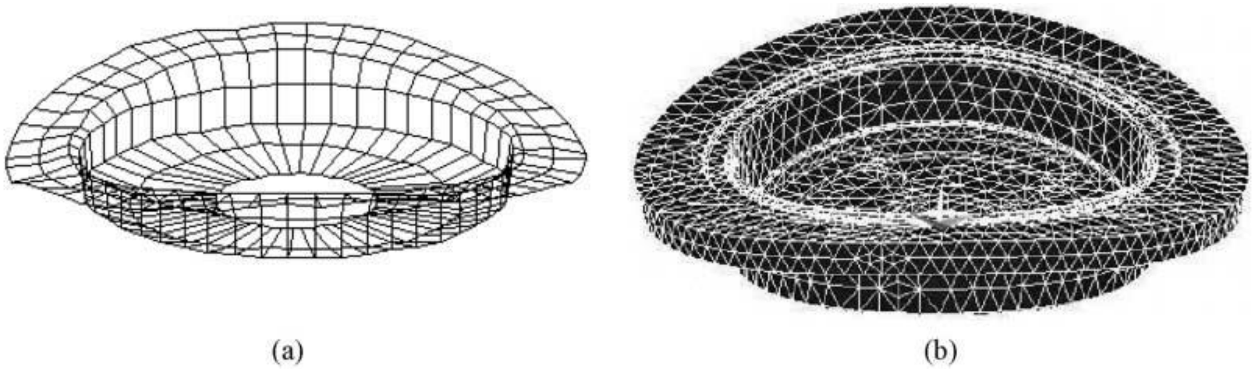


Fig. 4 (a) Experimental bending mode shape (b) FEA bending mode shape

3) Numerical methods

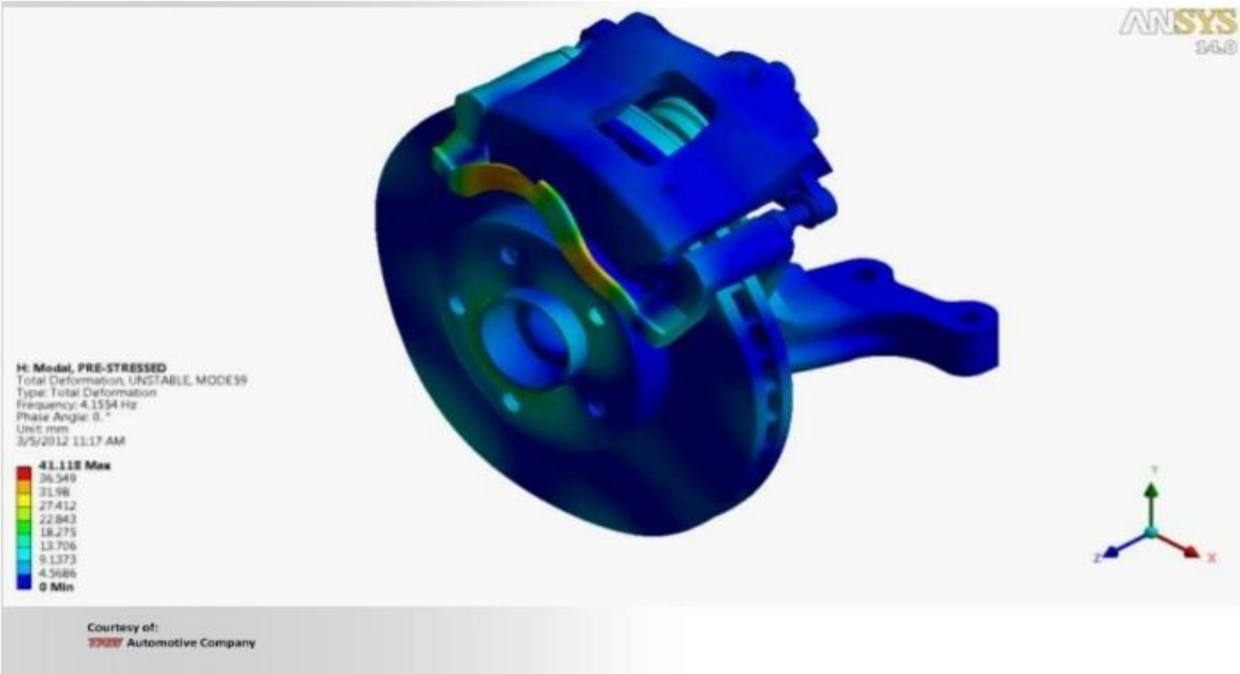
Finite element analysis (FEA) has been used in the analysis of brake squeal. CAE simulation and analysis methods play an important role in understanding brake squeal mechanism. Numerical modeling, on the other hand, can simulate different structures, material compositions and operating conditions of a disc brake or of different brakes or of even different vehicles, when used rightly. With these methods, noise improvement measures can be examined conceptually before a prototype is made and tested. Theoretical results can also provide guidance to an experimental set-up and help interpret experimental findings. It can also be used to interpret test results, prepare for upfront DoE (design of experiment), simulate structural modifications and explore innovative ideas.

Ansys 14.5 solves the brake squeal problems effectively. To predict the onset of instability a modal analysis is performed on the prestressed structure. An unsymmetric stiffness matrix is a result of the friction coupling between the brake pad and disc; this may lead to complex eigenfrequencies. If the real part of the complex frequency is positive, then the system is unstable as the vibrations grow exponentially over time. Three different methods to perform a brake squeal analysis are present in Ansys 14.5.

1) Full Nonlinear Perturbed Modal Analysis.
2) Partial Nonlinear Perturbed Modal Analysis.
3) Linear Non-prestressed Modal Analysis.

A full nonlinear perturbed modal analysis is the most accurate method for modeling the brake squeal problem. This method uses nonlinear static solutions to both establish the initial contact and compute the sliding contact. A partial nonlinear perturbed modal analysis is used when a nonlinear solution is required to establish contact but a linear analysis can be used to compute the sliding contact.

A linear non-prestressed modal analysis is effective when the stress-stiffening effects are not critical. This method requires less run time than the other two methods, as no nonlinear base solution is required. The contact-stiffness matrix is based on the initial contact status. Fig 6 shows the total deformation of unstable mode 59 brake squeal problem using Ansys 14.5



IV. CONCLUSION

Presently, research into brake squeal is focused on specific brake systems. The challenge for the future is to be able to develop general techniques and guidelines to eliminate brake squeal during the design stage. Given the complexity of the mechanisms that generate brake squeal, it appears that general guidelines are some way off in the future. For the present, the reduction of squeal noise for specific brake systems is achievable, with the additional knowledge acquired in each case adding to the overall understanding of brake squeal. Theoretical analysis of brake systems is difficult given the complexity of the mechanisms and the lack of an adequate model for the friction interface that causes brake squeal. However, this should not limit the development of simplified models as valuable insight can be gained. Understanding obtained by studying simplified models can assist in the interpretation of experimental results and the development of improved computational tools.

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REFERENCES

[1] Ma´rio Triche^s Ju´ nior, Samir N.Y. Gerges , Roberto Jordan. "Analysis of brake squeal noise using the finite element method: A parametric study". Applied Acoustics 69 (2008) 147–162.

[2] N.M. Kinkaid, O.M. O'Reilly, P. Papadopoulos. "Review Automotive disc brake squeal". Journal of Sound and Vibration 267 (2003) 105–166.

[3] Ouyang, H., Nack, W, Yuan, Y. and Chen, F. (2005) "Numerical analysis of automotive disc brake squeal: a review", *Int. J. Vehicle Noise and Vibration*, Vol. 1, Nos. 3/4, pp.207–231.

[4] Antti Papinniemi, Joseph C.S. Lai, Jiye Zhao, Lyndon Loader. "Brake squeal: a literature review". Applied Acoustics 63 (2002) 391–400.

[5] Utz von Wagner, Daniel Hochlenert, Peter Hagedorn, "Minimal models for disk brake squeal" Journal of Sound and Vibration 302 (2007) 527–539.

[6] P. Liu, H. Zheng, C. Cai, Y.Y. Wang, C. Lu, K.H. Ang, G.R. Liu. "Analysis of disc brake squeal using the complex eigenvalue method" Applied Acoustics 68 (2007) 603–615.

[7] H. Hetzler, K.Willner. "On the influence of contact tribology on brake squeal" Tribology International 46 (2012) 237–246.