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NOISE GENERATION IN VEHICLE BRAKES

ABSTRACT: Noise and vibration have become key issues in the design of automotive braking systems. The main concern over the noise problem is that it can cause discomfort to passengers and pedestrians and hence, reduce the overall acceptability of the vehicle. Good knowledge of the mechanisms involved in the generation of brake noise has thus become an important competitive factor in the design of automotive brake systems. The present paper summarizes some facts and hypotheses concerning the generation of brake noise. The different brake noise phenomena are first classified. Then several approaches, including models of various levels of detail which have been suggested to explain the root causes of brake noise generation are discussed in detail.

KEYWORDS: noise, disc, brake, generation

INTRODUCTION

Road traffic is the most widespread source of noise in all countries and the most prevalent cause of annoyance and interference. Therefore, traffic noise reduction measures have the highest priority.

Most countries encourage manufacturers to produce quieter cars and trucks by imposing noise limits on individual vehicles. These "pass-by" noise-rating limits have been reduced over the past 20 − 30 years by approximately 8 dB for cars and 15 dB for trucks.

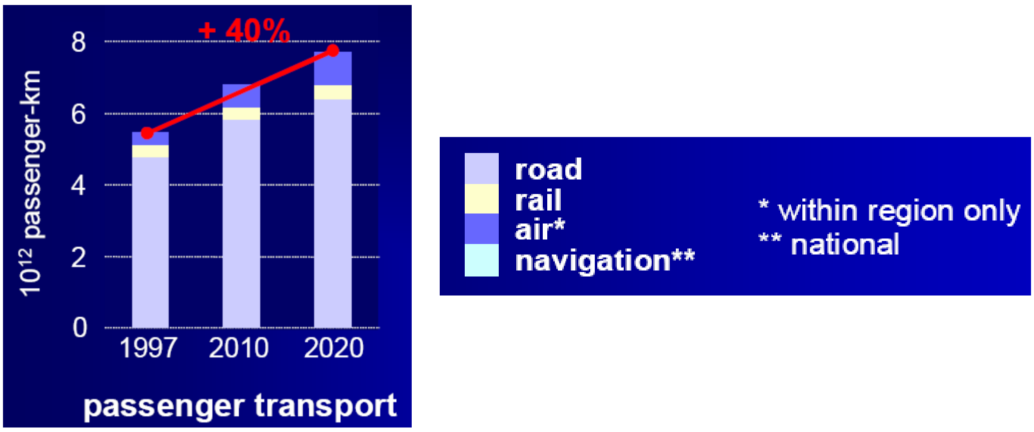

Figure 1 *Transportation Growth in Europe [1]*

More than half the population is affected by traffic noise, Figure 2. With traffic increasing, noise issue may strangle mobility and economic development.

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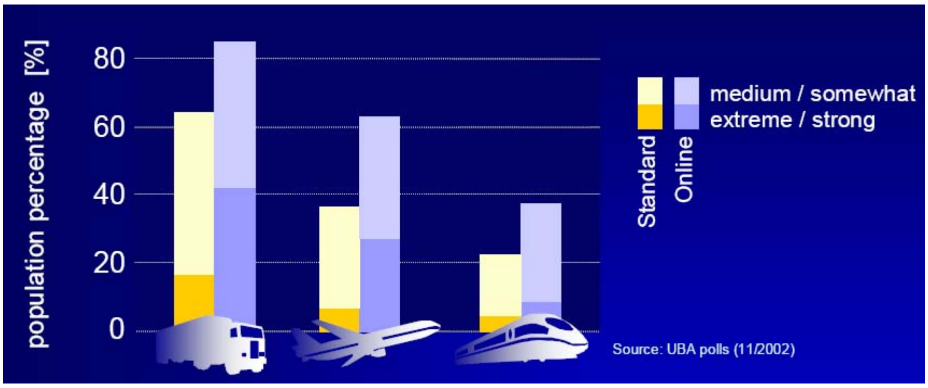


Figure 2 *Noise Exposure in Europe [1]*

Some national governments (e.g. Norway and Italy) have implemented legislation to include tests on noise emission from vehicles during normal service. These tests are usually carried out by garages as part of general tests on the condition of the vehicle; others perform spot checks. Even so, the ever-increasing number of vehicles means that the overall noise levels have not been reduced.

Road surfaces can be improved to give lower noise output. Porous asphalt and the newer "thin noise-reduced surfaces" have shown reductions of 2 − 6 dB. Railway noise can be reduced by the use of welded rail track laid on a concrete bed with elastic/resilient pads or mats [2].

Brakes are one of the most important safety and performance components in automobiles. Appropriately, ever since the advent of the automobile, development of brakes has focused on increasing braking power and reliability. However, the refinement of vehicle acoustics and comfort through improvement in other aspects of vehicle design has dramatically increased the relative contribution of brake noise to these aesthetic and environmental concerns. Brake noise is an irritant to consumers who may believe that it is symptomatic of a defective brake and file a warranty claim, even though the brake is functioning exactly as designed in all other aspects. Thus, noise generation and suppression have become prominent considerations in brake part design and manufacture. Indeed, many makers of materials for brake pads spend up to 50% of their engineering budgets on noise, vibration and harshness issues.

A wide array of brake noise and vibration phenomena are described by an even wider array of terminology. Squeal, groan, chatter, judder, moan, hum, and squeak are just a few of the names found in the literature. Of these phenomena, the one generally termed squeal is probably the most prevalent, disturbing to both vehicle passengers and the environment, and expensive to brake and automotive manufacturers in terms of warranty costs. No precise definition of brake squeal has gained complete acceptance, but it is generally agreed that squeal is a sustained, high frequency (> 1000 Hz) vibration of brake system components during a braking action resulting in noise audible to vehicle occupants or passers-by. As shall be seen later on, this squeal is often subdivided into high- and low-frequency regimes.

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 will become apparent later on in this review, 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. There are many models for squealing disc brakes: we have counted well over 15 different models in the literature. However, none of these models have attempted to include the effects at all scales mentioned above. This has led to models which capture some features of brake squeal well and ignore many others. Experimental studies also tend to have limited applicability with their results only pertaining to one brake configuration or to one automobile type.

The motivation for writing this paper was to complement, update, and expand upon several earlier reviews on the subject. This has proved to be a difficult task in part because the literature on disc brake noise is growing rapidly every year and the subject has an interdisciplinary nature [3].

CLASSIFICATION OF BRAKE NOISE PHENOMENA

The different phenomena related to brake noise span a wide range. There are vibrations of the brake system which have little interaction with the vehicle structure, but there are also vibrations showing strong interactions of brake components, vehicle chassis and body structures. Typically the types of brake noise are named according to how they sound; see Figure 3.

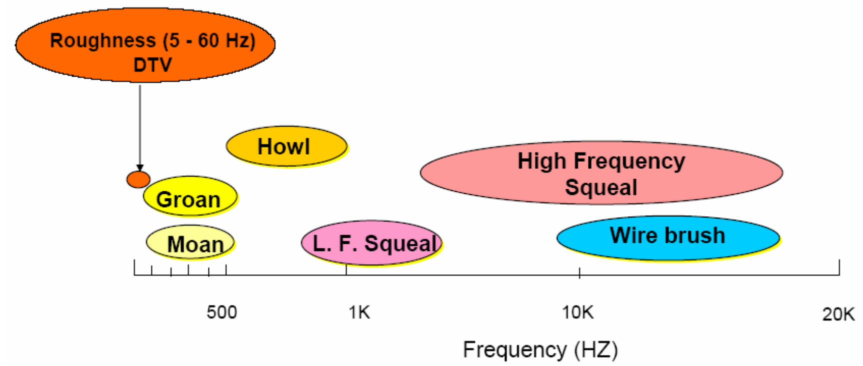


Figure 3 *Frequency Range for Brake Noise [4]*

Judder is a low frequency (< 100 Hz) forced vibration, excited by cyclic non-uniformities of the friction force. The vibration frequency is proportional to the wheel speed and decreases during a decelerating brake application. Often the amplitude of vibration varies due to the resonance amplification of suspension or vehicle structure modes.

The origins of the non-uniform friction forces causing judder are attributable to symmetry disturbances of the brake resulting from factors such as the following: manufactured or worn-out thickness variations of brake disc or drum, imperfect assembly of the caliper-piston-pad-rotor system, transient thermal deformations of brake disc or drum, or localized high-temperature regions distributed circumferentially across the brake-disc („hot-spots“).

Groan, moan and howl are vibrations in the frequency band of 100 Hz - 1 kHz. They are characterized by one or more single frequency pure tones whose frequencies are generally invariant and independent of wheel speed, temperature and pressure. The terms groan, moan and howl describe one and the same type of vibration, the difference being that groan and moan are usually associated with lower frequencies (100 Hz - 500 Hz) than howl (500 Hz - 1 kHz).

While judder is a forced vibration whose excitation and response frequency is a multiple of the wheel speed, in contrast, groan, moan and howl, as well as squeal (which will be discussed later) are results of a self excited oscillation due to a dynamic instability of the brake system. The key feature of this type of vibration is that the energy flow into the system gets synchronized in phase with a vibration mode of the structure, resulting in a positive feedback loop. The corresponding vibration mode thus becomes unstable, and the amplitude of vibration grows until it eventually is limited by nonlinearities or damping and a stable limit cycle is reached.

Low-frequency squeal is associated with frequencies between 1 kHz and 3 kHz, while **high-frequency squeal** covers the range from 3 kHz to 20 kHz. Squeal frequencies are in general invariant because they are strongly coupled to resonances of the brake system.

Several explanations for the phenomenon of squeal have been proposed. It is generally accepted that brake squeal is the result of self-excited oscillations. It is a complicated system problem which, in most cases, can not be tackled by investigation of individual components of the brake alone. From a practical point of view, squeal is one of the most important noise problems in automotive brake systems.

While all noise phenomena discussed so far are associated with discrete frequencies, there also exist vibrations of a different type. These are known as **wire-brush** and **squelch**. Wire-brush at first glance appears to be a broad band random vibration but a more detailed analysis reveals that it is a superposition of high frequency oscillations with randomly varying amplitude. It is often observed prior to the occurrence of squeal. Squelch is a superposition of several high frequency vibrations resulting in a low-frequency amplitude modulation of the envelope. While the frequencies of the carrier waves are in general time-invariant, the lower modulation frequency is continually variable and it is quite probable that it is this modulation which is heard [5].

DISC BRAKE SQUEAL

The majority of automotive products employ disc brakes with a floating caliper design. In this family of brakes, it is common to find disc brake squeal, hereby defined as a noise of 1 kHz or higher. It is accepted that brake squeal is a form of friction induced vibration of the brake system, and the vibration of the rotor is the main source of noise. Generally the frequencies and magnitudes of the noise are governed by the rotor geometry (stiffness). The driving factor is the magnitude of the friction force between the pad and the rotor cheek surface, and the overall dynamic stability of the caliper/pad/rotor system.

One of the biggest contributors to 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 number of modes for a rotor within human hearing range may be up to 80. The modal frequencies and modal shapes of the rotor, caliper, anchor and pad will change once these parts are installed in-situ. 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 unsymmetrical 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 [8].

Types of Disc Brake Squeal

All analysis and test has led to the identification of at least three different families of disc brake squeal:

- Caliper Bracket induced squeal (2-6.5 kHz). This applies to floating caliper design only. This is also known as Low Frequency Squeal.
- Pad-induced, also defined as axial, or out-of-plane squeal (4-11 kHz).
- Rotor induced, also defined as tangential, in plane, longitudinal or High Frequency squeal (7-16 kHz).

It must be noted that the frequency ranges quoted are a function of the rotor and caliper material and geometry, and therefore subject to change. Figure 4 illustrates the general distribution of the different brake squeals in the frequency domain based on the extensive testing [6].

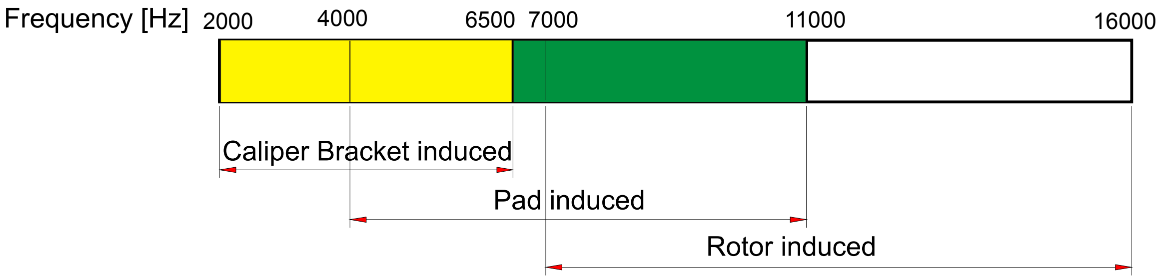


Figure 4 *Frequency Domain Illustration of the three families of Brake Squeal*

Factors influencing low frequency and high frequency brake noise are shown in Figures 5 and 6.

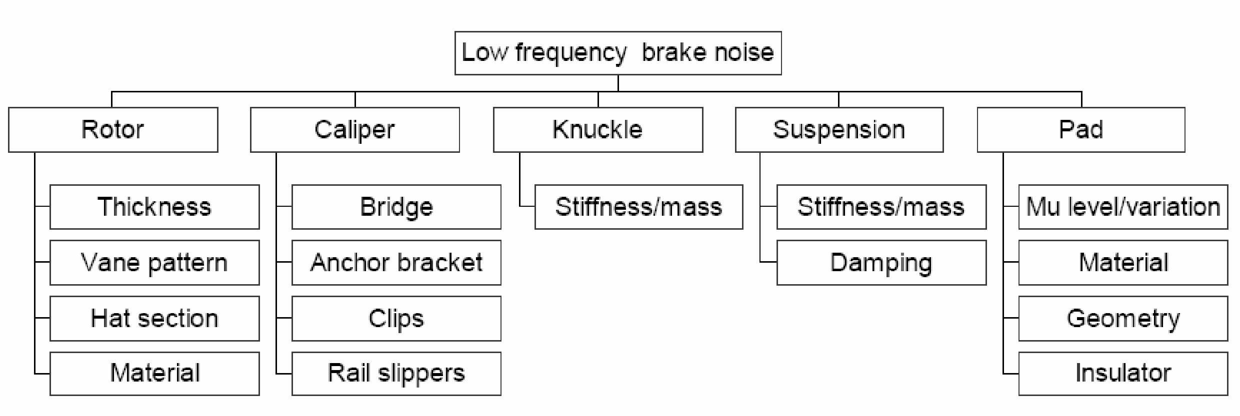


Figure 5 *Factors influencing Low Frequency Brake Noise [4]*

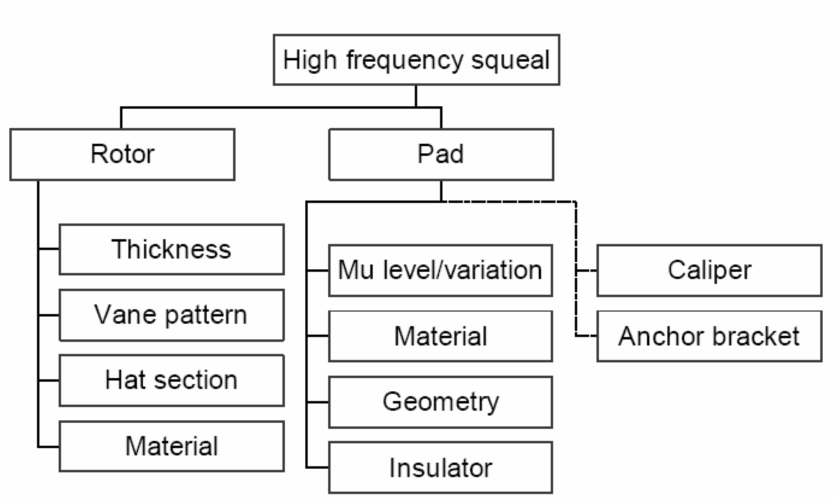


Figure 6 *Factors influencing High Frequency Brake Noise [4]*

A. Caliper Bracket induced/Low Frequency squeal

Low Frequency or Caliper bracket induced squeal is the least common form of Disc Brake Squeal. It is typically found in the 2 kHz to 6.5 kHz range, and is the result of vibration modes of the caliper bracket, similar to the one shown in Figure 7. Dual piston caliper brackets will tend to be on the lower end of the frequency spectrum, and single piston caliper brackets on the upper range. This is because dual piston brackets tend to be longer and hence have modes at lower frequencies (given similar cross-sections). There has been no evidence that caliper bracket bending modes have to be coupled with a rotor mode for this kind of squeal to occur.

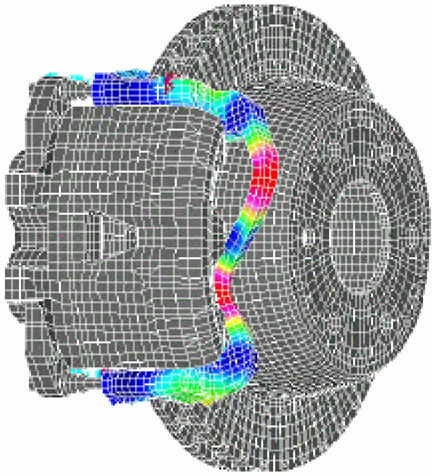


Figure 7 *Typical Bending Mode of the Caliper Bracket [6]*

B. Axial or Pad Induced Squeal

Extensive testing indicates that Axial or Pad induced squeal frequencies are aligned with the axial/out-of-plane modes (also known as nodal diametrical modes) of the rotor, and are unavoidable in the audible frequency range for the family of brake systems studied (fifteen and sixteen inch brake corners with cast iron rotors). This appears to be the most common occurring squeal. Pad induced squeal appears to be triggered by out of plane motions of the pad ends (pad-end flutter), which excite the out-of-plane vibration modes of the rotor.

C. High Frequency / Rotor Induced / Longitudinal /Tangential / In-plane Squeal

High frequency squeal is a result of the excitation of the tangential (in-plane) modes of the rotor. There are basically two types of tangential rotor modes. The first is compression; whose primary controlling parameter is the disc diameter. A typical compression mode of the rotor is illustrated in Figure 8.

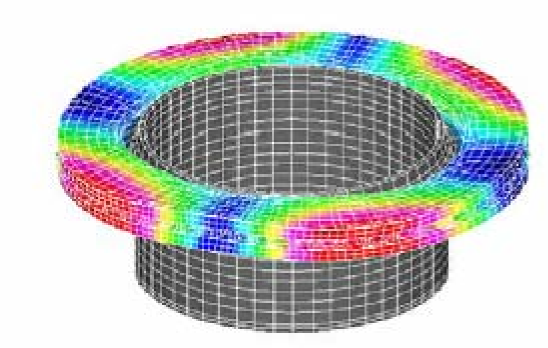


Figure 8 *Typical Compression Mode of a Rotor [6]*

For the family of rotors used in automotive disc brakes (fifteen and sixteen inch brake corners with cast iron rotors), the compression modes are as follows:
1. First - 6-7 kHz
2. Second - 9 -11 kHz
3. Third - 14-16 kHz

The second type of in-plane mode is 'racking' or 'matchboxing' (a relative sliding of the rotor cheeks, and hence only found on vented disc rotors); whose primary controlling parameters are the number of vanes, the width of the air vent, and aspect ratio (height to width) of the rotor vanes. A typical racking mode of the rotor is shown in Figure 9.

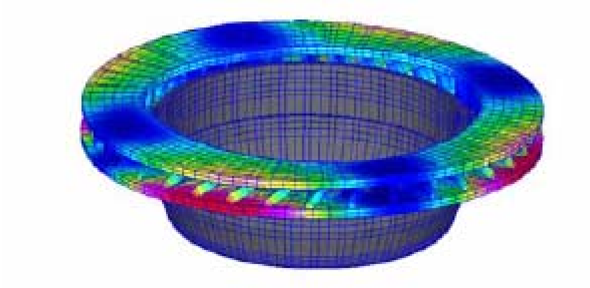


Figure 9 *Typical 'Racking' Mode of a Rotor [6]*

Squeal mechanisms and causes

As described in some of the recent review articles, hypotheses on brake squeal mechanisms can be grouped into five major categories or their combinations, in which reference [3] provides very comprehensive and detailed review. They are stick-slip (difference between static and kinetic friction coefficients), negative damping (negative slope of friction coefficient with respect to sliding speed), geometric/kinematics or GK constraint (sprag-slip, constant friction coefficient), modal coupling (flutter type instability), or mode lock-in, and 'hammering' excitation (vibration induced by uneven rotor surface variation during disc rotation). In the earlier stage of research, attempts were made to associate the brake squeal phenomenon with one of the above mechanisms in order to determine the root cause. As more test evidence and analysis results become available, it seems quite obvious that none of the above mechanisms alone can provide a complete explanation of the squeal phenomenon. In some cases, negative damping may seem more proper, but in others, modal coupling may be a better explanation. Indeed, which mechanism is the dominant one depends on both the brake system characteristics and the operational conditions. Reference [7] provides practical examples of contradictions or exceptions to some of the mechanisms listed above. That work also suggested a "hammering" mechanism from an excitation viewpoint. It is well accepted from a dynamics viewpoint that squeal is caused by brake system instability. Thus, for a brake system that tends to become unstable due to its system characteristics, when one or more proper excitation mechanisms are present, the brake may generate squeal. From a vibration energy standpoint, as long as there is a mechanism to accumulate sufficient vibration energy, the brake may yield squeal. A brake system may be highly prone to become unstable at certain frequencies due to the presence of certain types of excitation/triggering/perturbation. However, squeal may not occur in the field if no proper excitations are present at those frequencies during operations. On the other hand, even if a brake system is relatively stable at a certain frequency, squeal may still occur at this frequency if there is a very strong excitation or coupling mechanism at this frequency in the operating condition. In this paper, we will discuss the squeal mechanisms along the following aspects: triggering mechanisms (excitation source), modal coupling/lock-in mechanisms (system inherent characteristics), and noise radiation mechanisms (system response).

Analytical methods

The earliest research into brake squeal suggested that the variation in the friction coefficient with sliding velocity was the cause [9]. 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 [10], and has led to analysis of the geometrical aspects of a brake system.

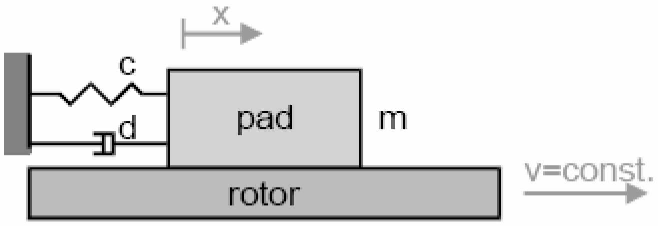


Figure 10 *Single Degree of Freedom Model for Stick-slip Motion*

Spurr proposed an early sprag-slip model that describes a geometric coupling hypothesis in 1961 [11]. Consider a strut inclined at an angle θ to a sliding surface as shown in Figure 11(a). The magnitude of the friction force is given by

$$F = \frac{\mu \cdot L}{1 - \mu \tan\theta} \tag{1}$$

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 Figure 11(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 behavior.

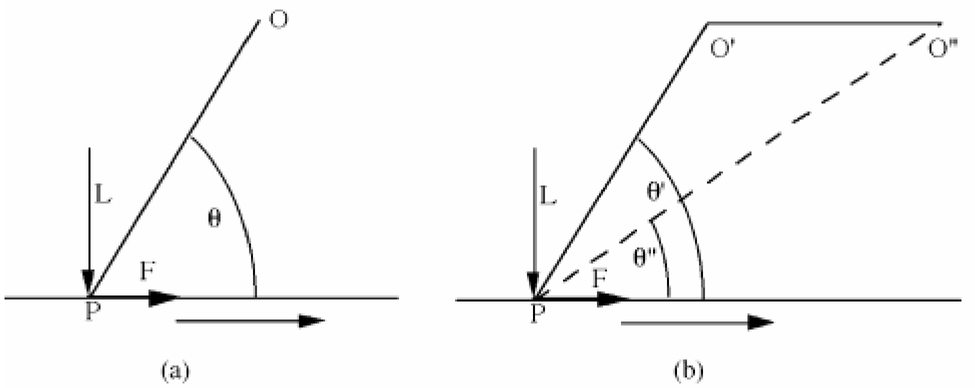


Figure 11 *(a) Single Strut rubbing against Moving Surface, (b) Sprag-slip System [8]*

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 [12], Earles and Soar used a pin-disc model in 1971 [13], and North introduced his eight-degree of freedom model in 1972 [14]. The culmination of these efforts was a model published by Millner in 1978 [15]. Millner modeled 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

Experimental investigation of brake squeal noise is non-trivial for the same reasons that brake squeals in automobiles occur seemingly randomly. Just as it is not possible yet to design a squeal-free brake, for the same reason it is not possible to design functional brakes that squeal all the time. Because every research group that experimentally investigates brake squeal has its own setup, a specific model of rotor, caliper, pads (sometimes slightly modified for instrumentation accessibility), makes it difficult to exchange and compare meaningful results.

The frequencies of a squealing brake are highly dependent on the natural frequencies of the brake rotor [16]. 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 [17].

Accelerometers provide an effective tool for determining the vibration mode shapes and the forced response of a system. Figure 12(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.

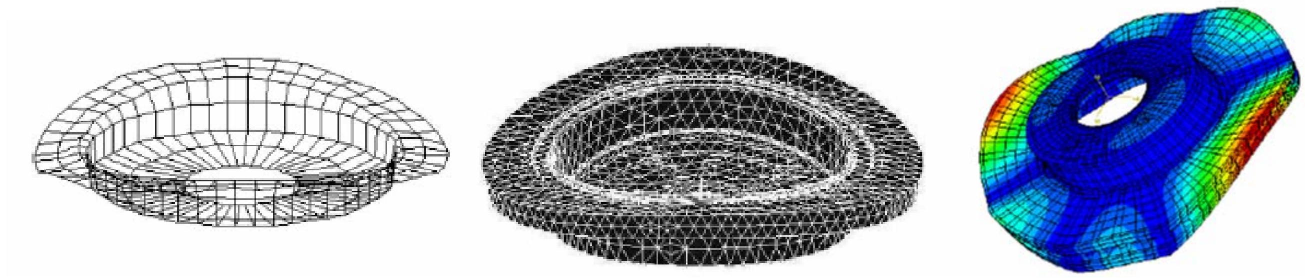
Optical techniques have been used more recently. In particular, double pulsed laser holographic interferometry has been successfully applied to squealing brake systems. This has allowed the coupled mode shapes of a complete brake system to be determined while it is squealing. A holographic image is produced 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 [16]. 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.

Numerical methods

Finite element analysis (FEA) has been used in the analysis of brake squeal. Modal analysis of brake components is an area where FEA can be readily applied. Figure 12(b) shows a finite element model of a brake rotor. The model, consisting of 8700 Tet92 solid elements, was developed using a commercial finite element code ANSYS 5.6. Unfortunately, the coupling between brake components leads to vibration modes that differ to those found for the individual components. Therefore, the real interest among researchers is to be able to model an entire brake system.



a) b)

Figure 12 *(a) Experimental Bending Mode Shape, (b) FEA Bending Mode Shape [8]*

The critical aspect in the modeling of a complete brake system is the coupling between components, particularly the rotor/pad interface. The contact stiffness itself is adjusted using experimental results, but the more difficult aspect is to introduce the tangential friction coupling. Liles included friction coupling between rotor and pad as off diagonal terms in the stiffness matrix and used a complex eigenvalue analysis to assess the stability of a brake system [18]. Once the model was developed, the effect of varying parameters such as friction coefficient, pad geometry and caliper stiffness, could be determined. Dihua and Dongying also used a similar approach to improve the design of an anchor bracket [19]. The work of these, and other, researchers has shown that it is possible to create models that incorporate the friction coupling between the rotor and the pad. However, there has been little experimental evidence to verify the accuracy of these models. They may be useful for studying the effect of varying parameters within the brake system, but their ability to model the important friction interface is limited. As small variations in operating temperature, brake pressure, rotor velocity or coefficient of friction may result in differing squeal propensities or frequencies, an accurate prediction of brake squeal using numerical methods requires an accurate determination of material properties (particularly for the friction material) under different operating conditions. Furthermore, proper modeling of the boundary conditions especially where the coupling between various components is important remains a challenge [20].

CONCLUSIONS

Despite a century of developing disc brake systems, disc brake squeal remains a largely unresolved problem. Many experimental and analytical studies have led to insight on the factors contributing to brake squeal or to the amelioration of squeal in disc brakes of a specific type or in a particular make and model of automobile. Experimental studies have accumulated a wealth of information about the nature of squeal, the vibration modes therein, the wear of brake components, and frictional interactions in brakes. Analytical studies have provided useful insights into how friction laws, geometry, and the dynamics of brake components can lead to squeal or instability in simple models of disc brakes. Finite elements have been used to try to extend these insights to more accurate brake models.

The development of a comprehensive predictive model of disc brake squeal is the ultimate goal of much of literature surveyed in this paper. This goal has been elusive, and whether or not such a model is possible remains to be seen. Indeed, given the number of factors involved and the range of designs, such a model, if developed, may be too complex to be useful. On the other hand, it is not too difficult to envisage improvements in existing models for disc brake assemblies. These improvements might include the influence of the variable decreasing speed of rotation of the rotor, the inclusion of thermal and wear effects, and the incorporation of more complex friction models and constitutive models for the pads of friction material.

Presently, research into brake squeal is focused on specific brake systems or generation mechanisms. 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.

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