ELIMINATING BRAKE NOISE PROBLEM

Jasna Glišović[1], Danijela Miloradović

UDC: 629.113-592.12: 534.322

INTRODUCTION

Noise, Vibration, and Harshness (NVH) refers to friction-induced vibrations that occur as the result of energy loss from braking. Even though it can be shown that the majority of the mechanical energy from a speeding vehicle is converted to heat, a small fraction is converted directly into vibrations that can be heard and/or felt by the operator. These vibrations are the source of NVH and are among the top warranty concerns\complaints for virtually all automotive companies.

For example, a modern midsize car weighing 1500 kg when traveling at 28 m/s (~100 km/h), has a kinetic energy of the vehicle is 600 kJ. The shortest possible distance within which the car can be stopped is about 40 m. Assuming that the retardation is constant, which is reasonable since the friction between tire and road controls the retardation force, this will take 2.9 s. As a result, the average power developed will be 206 kW and the maximum power, in the beginning of the stop, will be 412 kW. Approximately 80 % of this power is absorbed in the front brakes and these have two brake pads each. The maximum power absorbed in each pad is thus 82,000 Watts, a value similar to the maximum engine output. This power should be compared with the power level of an audible sound (e.g. car horn) which is 0.012 Watts. Thus, less than 1 part in a million of the power dissipated by a brake appears in the form of vibration.

Even though the vibrations produced by friction material do not influence either their stopping or wear performance, these friction induced vibrations have significant influence on the market acceptance of materials in both the OEM (Original Equipment Manufacturer) and aftermarket. The focus on brake NVH performance is a relatively recent trend. Before the 1980s, most motorists accepted a certain level of brake noise and roughness, even in some luxury and near-luxury models. Beginning in the 1980's customers demanded quieter brakes. This increased customer awareness of NVH also coincided with the industry push to develop non-asbestos organic (NAO) brake pads. The manufacture of low metallic and NAO friction materials formulations required modifications of traditional processing methods and addition of new ingredients. This exacerbated the noise generation problem [5]. Figure 1 shows a variety of factors contribute to brake NVH performance issues.

Nowadays, passenger cars become one of the main transportation for people traveling from one place to another. Thus, comfort issues of the passenger cars should a major concern. As a result, carmakers, brake system and friction material suppliers face challenging tasks to reduce high warranty payouts. Akay (2002) stated that the warranty claims due to the brake

[1] Corresponding author e-mail: jaca@kg.ac.rs , University of Kragujevac-Faculty of Mechanical Engineering, Sestre Janjić 6, 34000 Kragujevac, Serbia

noise, vibration and harshness (NVH) including brake squeal in North America alone were up to one billion dollars each year. Furthermore, Abendroth and Wernitz (2000) noted that many friction material suppliers had to spend up to 50 percent of their engineering budgets on the NVH issues. [1]

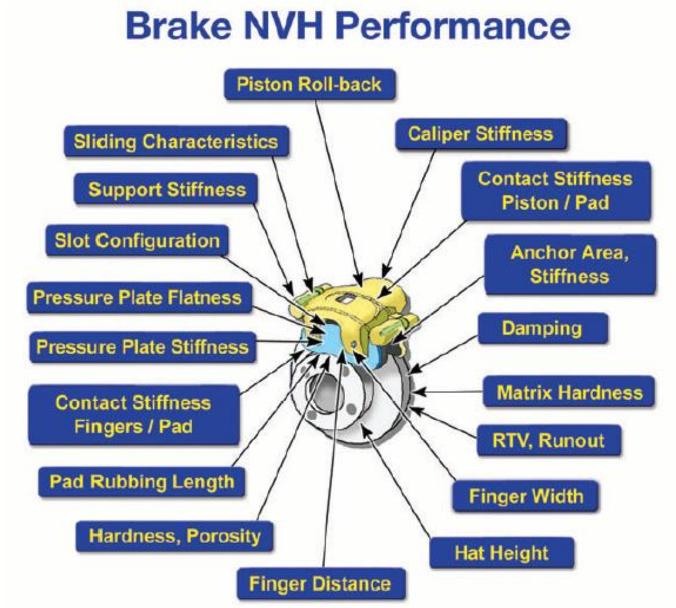

Figure 1: *A variety of factors contribute to brake NVH performance issue [2]*

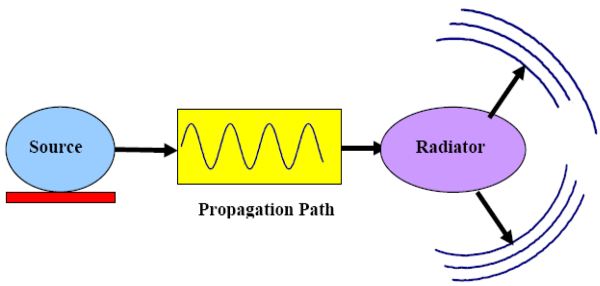

Figure 2: *Three basic attributes required to produce audible noise. Friction induced vibration is always the source. However, other brake components must contribute to the propagation and radiation in order to produce an NVH performance problem*

As illustrated in Figure 2, an NVH event involves 3 basic components, a source of noise generation, a propagation path, and a radiating member. The noise generation source is friction induced vibration at the brake pad/rotor interface. Thus, the friction material always plays a role in noise generation. However, production of NVH that is objectionable to the customer requires compatible propagation path and radiator. Thus, NVH is a systems problem, often involving not only the friction material, but also other brake system components including the rotor, caliper, steering knuckle, and suspension. NVH is a complex problem that generally involves multiple components of the braking system. Finding the root cause of an NVH performance issue is not always easy.

Brake noise tends to span the entire audible range in the frequency spectra (see Figure 3). At low frequencies, noise and vibration are characterized as creep-groan, grunt, moan, judder or shudder. At middle to high frequencies, there are any varieties of squeal noises, which occur singly, or in harmonic series and with all manners of duration or periodicity. Obviously, the complexity of the noise presentation to the driver is one reason why subjective evaluation is still so common. However, with advanced data acquisition tools it is possible to accurately capture all types of noise events and reducing the data to meaningful statistics [5].

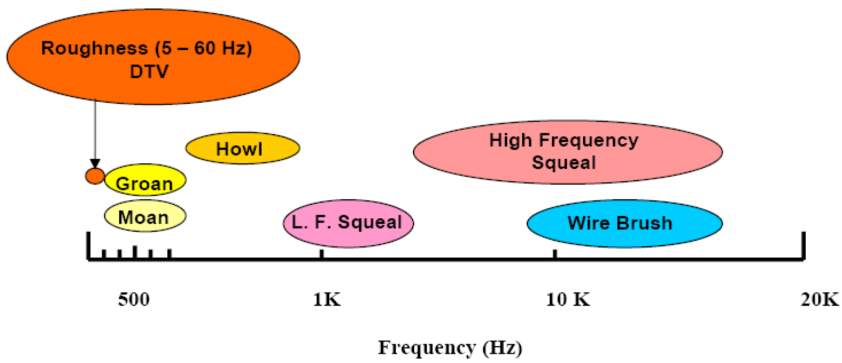

***Figure 3**: Chart showing the various types of NVH and the frequency range of these excitations*

In recent years, finite element method becomes the most popular tool in studying disc brake squeal (Ripin 1995; Tirovic and Day 1991; Abu bakar and Ouyang 2004; Mahajan et al. 1999). This is owing to the fact that experimental methods could not predict any squeal at early design stage. In addition, the finite element method is capable of simulating any changes made on the disc brake components much faster and easier than experimental methods. In order to predict the onset of squeal most researchers prefer the complex eigenvalue analysis. Discussions on such analysis in comparison with other analyses are given in details in references (Ouyang et al. 2003; Mahajan et al. 1999). The essence of the complex eigenvalue method lies in the inclusion of the asymmetric friction stiffness matrix that may be derived from contact pressure analysis. The positive real parts of the complex eigenvalues reflect the degree of instability of the (linearised) brake system and are thought to indicate the likelihood of squeal occurrence.

The contact pressure distribution in disc brakes has been investigated by a number of people. However, to date, measuring dynamic contact pressure distribution remains impossible. Tumbrink (1989) attempted to measure static pressure distribution using a ball pressure method. Contact pressure prediction by means of numerical method was studied in

(Ripin 1995; Tirovic and Day 1991; Abu bakar and Ouyang 2004). There are various models of different degrees of sophistication to predict contact pressure through numerical methods.

Although continuous investigations have been carried out over decades, so far there is still no comprehensive solution for suppressing brake squeal noise. The paper investigates effect of structural modifications on the onset of squeal. Therefore it is the authors' intention to investigate and suggest the plausible modification that could improve squeal performance and hence might help create a quieter design of the car disc brake [1].

SOLVING BRAKE NOISE PROBLEMS

In order to solve brake related noise problems, a clear insight in the fundamentals of self-exiting oscillations is of utmost importance. When a brake path is contacted to a rotating disc or drum, a non-steady friction force is generated. Under certain, usually poorly controlled, boundary conditions this reaction force will interact with the dynamics of the brake system. Typically this friction force will excite a resonance of the brake disc or caliper, amplifying the friction force, and so on.

The amplitude of this force self-sustained vibration is determined by energy considerations: during each period of vibration the amount of dissipated energy equals the amount of externally supplied energy. The dissipation of energy is determined by the damping of the system. The supply of energy is function of the disc/brake pad friction force and the system's response to this force. Several theoretical models exist that describe in very detail this energy balance. None of these models is perfect, but it is straightforward in all models that an increase of damping is required in order to control or reduce the amplitude of vibration.

However, these models also prove that by a simple addition of damping, selfexciting oscillations cannot be avoided and a more fundamental change in the dynamics of the brake systems is required. Therefore a detailed description of the behavior of the brake system under excitation (operational deflection shape and/or operational modal analysis) is necessary for solving brake related noise. The frequency of the brake noise determines the scope of this investigation. For low frequency problems not only the brake system should be considered, but also the entire wheel suspension. While for higher frequencies it can be sufficient to describe only the behavior of the disc, caliper and brake pads.

Countermeasures can then be situated at different levels: the excitation at the source level (friction force), the energy flow from the source to the noise emitter and the noise emitter itself.

This brake squeal has been tackled by several successful countermeasures at the prototype level (tuned absorbers, asymmetric disc, stiffened twist beam, surface treatment disc,…). However, the industrial solution consisted of a reinforced brake support. Stiffening ribs were added at the cast iron interface between the brake and the wheel spindle. The additional costs for this modification were almost neglectable while a 100% successful solution was obtained. [6]

Rotor

Even the metallurgy of the rotors makes a difference. Some grades of cast iron are quieter than others. That's one of the reasons why composite rotors have been used on various vehicles over the years. Besides being lighter, composite rotors can also be quieter if the right grade of cast iron is used for the rotor disk. Replacing a composite rotor with a solid cast rotor changes the harmonics and frequency of the brake system, which may increase brake noise on some applications.

Rotor finish also affects noise. The smoother and flatter the surface, the less the likelihood of the pads chattering and dancing as they ride across the surface. Rotors should be resurfaced at the proper speed and feed rate, and with sharp tool bits to achieve the smoothest possible finish. Light sanding with an abrasive disk or flexible honing brush after the rotors have been turned can improve the surface finish even more and provide an extra degree of assurance the rotors will remain noise-free. The recommended rotor finish for most applications is 60 to 80 microinches or less. A range of 20 to 50 microinches will usually guarantee quiet operation even on vehicles that are sensitive to brake noise.

Composite rotors require special care when resurfacing because they lack the rigidity of cast rotors. The rotor needs to be supported by large bell caps or adapters otherwise it may flex leaving tool chatter marks on the surface. The other alternative is to use an on-car lathe to resurface composite rotors.

Brake Pads

Equally important are the pads themselves. Some friction materials are noisier than others, just as some brands of pads are quieter than others. The sound control quality of any friction material depends on the fillers, lubricants and other ingredients that go into the mix. Some manufacturers add graphite and other materials to pads to dampen noise.

The design of the pads also influences their ability to suppress noise. If the leading edge of the pads has a sharp edge, it increases the tendency to grab and bounce more than if the leading edge is chamfered. That's why most premium grade brake pads have chamfered edges. The pads may also have a slot down the middle to increase flexibility, cooling and venting. Some pads also have integrally molded shims and a multi-layer construction to reduce noise.

Some friction suppliers use "Transfer Film Technology" (TFT) to prevent noise. TFT is not a coating on the pads, but part of the friction material itself. As the pads wear, they continuously transfer a very thin film to the rotor surface. This film, which leaves a dull gray coating on the rotors, fills in tiny imperfections in the rotor surface to make it smoother and more compatible with the pads, thus eliminating squeal-producing vibrations. This also eliminates the need for shims behind the pads to dampen vibrations (unless required by the OEM caliper design). This is because the coating prevents the pads from vibrating in the first place. And unlike spray-on noise treatments which eventually wear off the rotor surface, TFT lasts the life of the pads because it is part of the pads.

Ceramic brake pads

Automotive braking systems normally employ brake discs made of steel or grey cast iron, which are then paired with composite organic brake pads. These types of materials are suitable for use in braking systems with moderate loads, but car manufacturers are tending to design increasing numbers of prestige and sports-class vehicles that need braking systems with more braking power. As a result, new materials are being introduced into braking systems, for example, carbon–ceramic C/C SiC composites. However, much higher temperatures are generated at these ceramic surfaces, which imply some new requirements for the contact materials and the testing equipment.

One of the main differences between 'ceramic' pads and traditional pads is that 'ceramic' pads contain no steel wool or steel fibres. Steel fibres are widely recognized to provide strength to friction materials. Additionally steel fibres improve the heat conductivity from the rotor. 'Ceramic' friction materials provide braking with less fade, quicker recovery, longer pad and rotor life. Brake dust emission and brake noise can be minimized also.

A growing number of automakers, led by the Japanese OEMs, have recognized ceramic friction brake pads are often the best option for smooth, assured, quiet braking under a variety of driving conditions. In addition, it has helped them significantly reduce warranty costs by as much as 90 percent [7].

Pad Assembly Damping

In principle and in practice, additional damping in a brake system can be achieved by increasing the pad damping. Pad assemblies are a multilayer structure, with friction material, underlayer, back plate, and insulator—see Figure 4. As shown in Figure 5, designing a friction material is a challenging task. The friction material has to be optimized for performance first (to meet legal requirements), then for wear, disc thickness variation (DTV) generation, pedal feel, stopping distance, dust, and noise. Any friction material modification to improve one of those requirements usually leads to degraded performance of one or more of the other parameters.

For brake squeal, friction suppliers have learned, through the years, to design quieter friction materials. However, the need for further noise improvement is required. They are different ways to achieve that:

- Friction modification—increase pad damping and compressibility (isolate excitation and increase damping effectiveness), modify the friction level characteristics (to reduce excitation and mode coupling propensity)
- Pad geometry modification (use of chamfers and slot to reduce excitation and mode coupling)
- Underlayer modification (for increased damping, increased compressibility)
- Insulator design (add damping and isolation).

As mentioned previously, friction material suppliers are reluctant to modify the friction material. Their preferred choice is to modify the pad geometry and optimize the insulator. The addition of a damped underlayer will increase the damping/isolation, but will also change some of the characteristics of the pad assembly (compressibility, thermal conductivity, etc.). The addition of chamfers onto the pad assembly is a very effective way of reducing squeal, but will usually result in a reduction in the pad projected life.

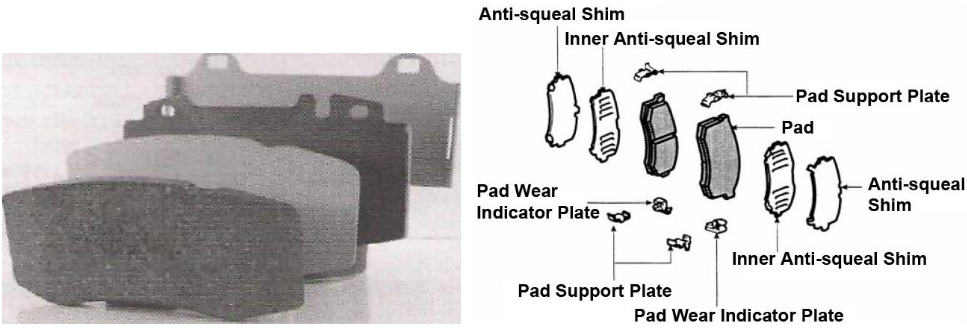

Figure 4: *Picture of a pad assembly*

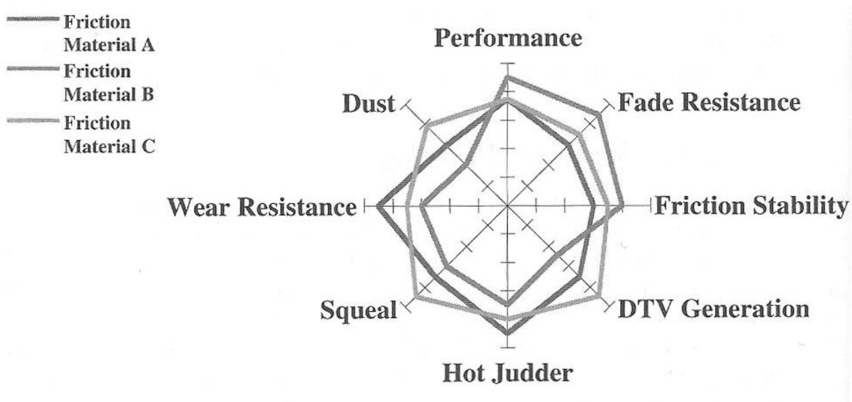

Figure 5: *Critical parameters for the design of a friction material*

A brake insulator is a sandwich of viscoelastic layers and steel layers (Figure 6). Its maximum thickness is usually around 1—1.2 mm and it is attached to the pad back plate.

There are two damping mechanisms for brake insulators:

- Material damping
- Coulomb friction damping between the insulator and the caliper piston/fingers, and the insulator and the pad back plate.

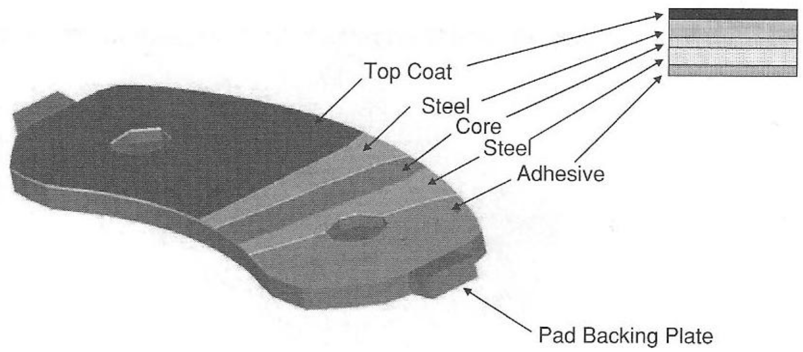

Figure 6: *Schematic of a brake insulator*

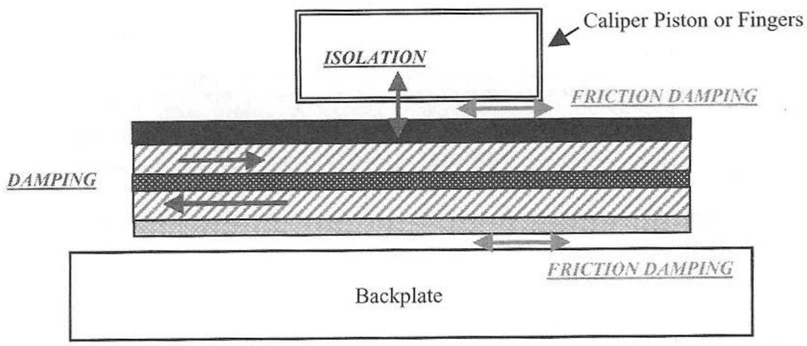

Figure 7: *Brake damping mechanisms*

In addition to those two mechanisms, the isolation effect is very important. Those mechanisms are not independent and are depicted schematically in Figure 7.

Single-Layer Insulator

A single-layer insulator consists of one metal layer coated with one viscoelastic material (Figure 8). The viscoelastic layer is usually nitrile butadiene rubber (NBR). The damping occurs as a result of the extension/compression of the viscoelastic layer. Due to the thickness limitations, the damping achieved is not very high. This insulator can be bonded or clipped on to the back plate. If it is bonded, the shearing of the bonding layer will provide additional damping and the NBR layer some isolation. If it is clipped on the back plate, the main damping mechanism will be isolation. In both cases, Coulomb friction damping will be present, and will vary with the surface finish of the NBR layer (smooth or rough).

Figure 8: *Single-layer isolator*

Constrained-Layer Insulator

Constrained layer insulators use the microshearing of a viscoelastic layer to dissipate the vibration energy. One such viscoelastic core used is shown in Figure 9. It is noted that these insulators can be bonded (for optimized damping) or clipped on the back plate (for optimized Coulomb friction damping). The level of isolation will depend on the design of the insulator.

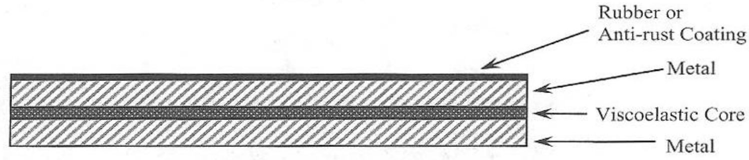

Figure 9: *Schematic of a constrained-layer insulator*

Multilayer Constrained Insulator

Multilayer constrained insulators use microshearing of the viscoelastic layers to dissipate vibration energy, but with two different cores—see Figure 10. These insulators can be bonded or clipped on the hack plate, which will either add more damping or provide Coulomb friction damping. The level of isolation will depend on the structure of the insulator.

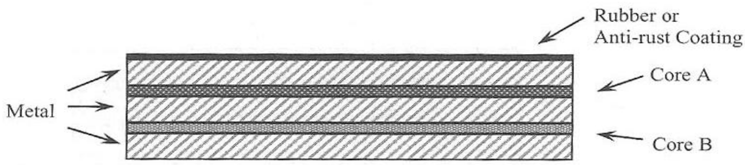

Figure 10: *Schematic of a multilayer-constrained insulator*

Double-Sticky Insulators

Double-sticky insulators have adhesives on both sides (Figure 11). They are usually used to reduce disc thickness variation (DTV) and sometimes to reduce shoe clack and caliper rattle, but they can also be used to control the contact between the caliper and the pad (control the friction damping).

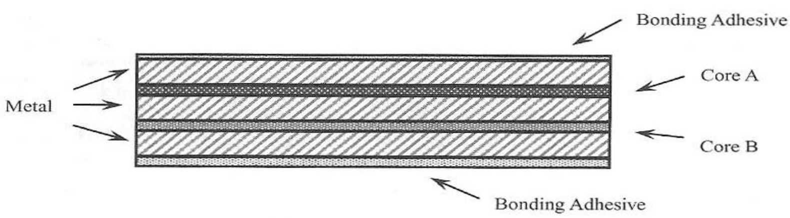

Figure 11: *Schematic of a double-sticky insulator*

Clip-on Insulators

Clip-on insulators are not bonded to the pad back plate; they are attached with clips. They are very popular in the industry because they do not require bonding equipments. They can be single-layer or multilayer and work mainly in isolation and Coulomb friction damping. This Coulomb friction damping can be modified by the addition of grease between the insulator and the back plate.

Others

Another way of solving brake squeal issues is to try to get tangential isolation (decoupling). This is done by adding rubber to the caliper abutment clips. Those clips are located between the ears of the brake pad and the caliper and are used to improve the pad sliding and reduce pad rattling/clacking (Figure 12). These rubber-coated clips have been effective in some applications, but the alteration of the rubber with time (due to the heat) leads to degraded damping performances. Another issue is the cost associated with this design (stainless steel is required to prevent any corrosion issue), and the added clearance needed for the extra rubber thickness.

Figure 12: *Picture of a caliper abutment clip*

There are significant differences in damping levels between insulators, as shown in Figure 13. Some insulators provide no added damping but can add stiffness to the pad assembly. For each project, the insulator selection process is usually based on the pad damping in free-free conditions at room temperature, but this method does not take into account the pressure, or the temperature at which the noise is occurring, nor does it measure the insulator isolation properties, or the friction damping properties. The damping has to be measured at a system level. The Coulomb friction damping, however, can only be assessed by dynamic testing, on the brake noise dynamometer or vehicle.

Figures 14 and 15 show noise data measured on a brake noise dynamometer, on pads with no brake insulator, as a function of brake pressure and pad temperature, respectively. Noise around 7-5 kHz was measured, from 25 to 225°C, and mainly above brake pressures of 15 bar.

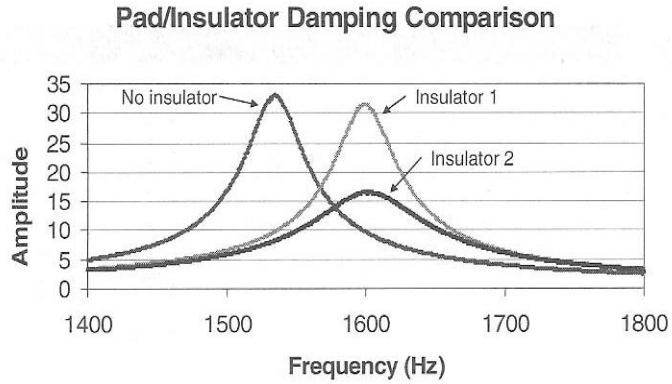

Figure 13: *Damping for various insulators*

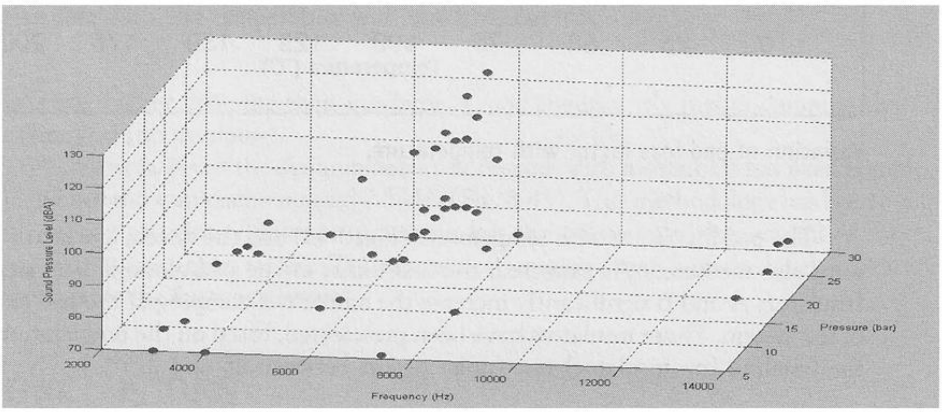

Figure 14: *Noise data with brake pressure on baseline pad configuration (no insulator)*

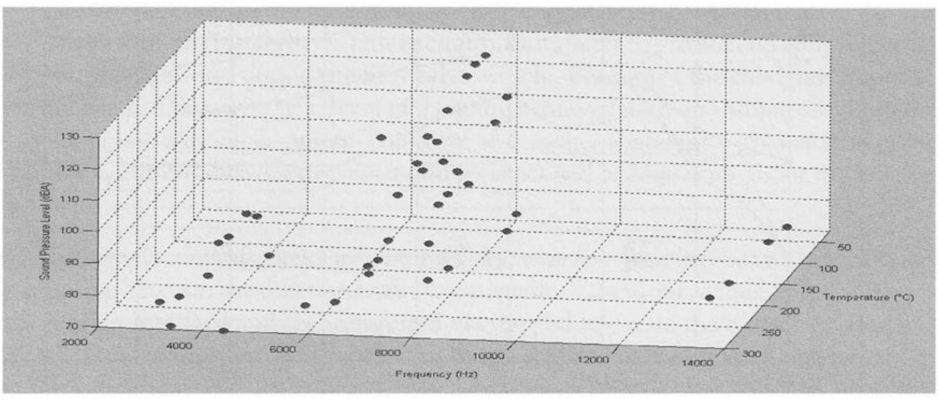

Figure 15: *Noise data with pad temperature on baseline pad configuration (no insulator)*

Insulators A and B significantly increase the amount of damping of the pad assembly and of the system. Those insulators have been preselected, based on the temperature data from the baseline (no insulator) test (noise occurs between 25 and 225°C). The insulator B is offering higher damping than insulator A, and significantly higher damping than the baseline pad (no insulator).

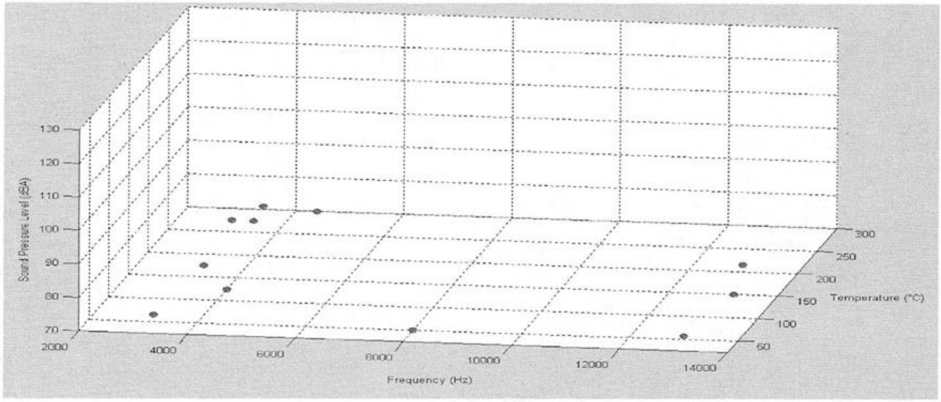

Figure 16: *Noise data with pad temperature with brake insulator B*

A confirmation of the dynamometer test on pads with insulator B has been completed and did not show any noise around 7.5 kHz (Figure 16). This methodology has been applied with success on a lot of different programs, but the unknown is the amount of damping needed to ensure a quiet system. This amount of damping depends on the level of excitation coming from the friction material. [3]

Active control of automotive disc brake rotor squeal using dither

Dither control is characterized by the application of a control effort at a frequency higher than the disturbance to be controlled. In the particular system considered here, a vibro-acoustic analysis of a disc brake system during squeal determined the acoustic squeal signature to be emanating from the brake rotor. This squeal was eliminated, and could even be prevented from occurring, through the application of a harmonic force with a frequency higher than the squeal frequency. The harmonic force was generated by a stack of piezoelectric elements placed within the brake's caliper piston. The harmonic force represented a small variation about the mean clamping force exerted bythe brake upon the rotor. The high-frequency vibration in the brake system due to the action of the control system was not heard if an ultrasonic control frequency was used. More importantly, the active control system is shown to be able to prevent squeal from even occurring. This gives rise to a possible active control system integrated into the brake system of automobiles to prevent squeal.

Dither control requires the use of an actuator to impose a fluctuating force upon the system of interest. A dither control actuator was integrated into the piston caliper, as depicted in Figure 17. The actuator was a 12.5 mm diameter, 25 mm long multi-layer piezoceramic

(PZT) stack. The actuator was positioned inside the piston with one end in contact with the inboard brake pad and the other end contacting the closed end of the caliper piston. Means were provided at each end of the stack to center it within the piston, and to ensure that only axially loads were imposed upon it. This configuration produces fluctuations in the normal component of the brake clamping force. Fluctuations in the normal force perforce induce fluctuations in the friction force.

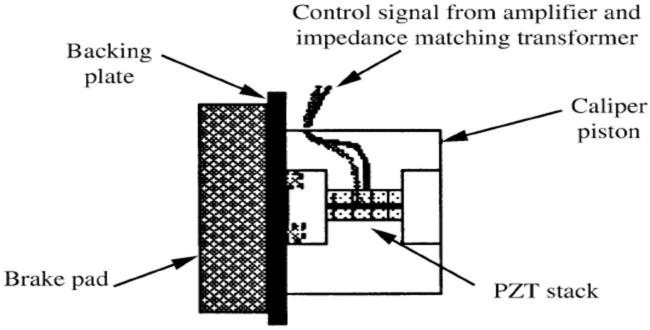

Figure 17: *PZT actuator inside caliper piston*

Generation of the signal to drive the actuator required the use of a function generator, a power amplifier, and an impedance matching transformer. The signal input to the power amplifier was generated using a function generator capability in a control program written using LabView. The impedance matching transformer is required in order to provide greater power transfer between the amplifier and the capacitive load of the PZT stack. The output of the impedance matching transformer was wired to the actuator.

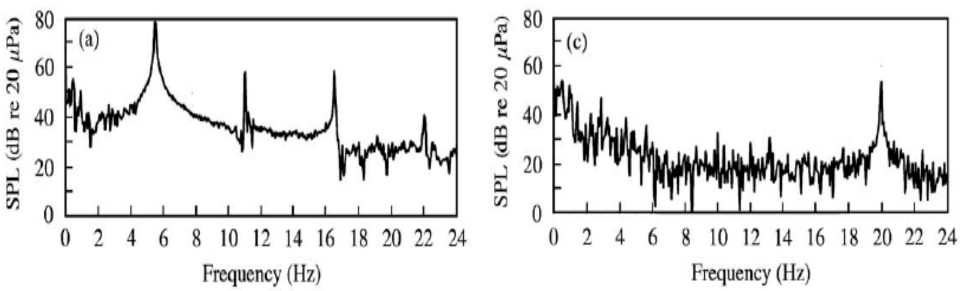

Figure 18: *Sound pressure spectrum from the brake system during stages of 20 kHz control: (a) before controlactivation, (c) after synchronization*

Figure 18(a) is the sound pressure level spectrum during squeal but without dither control. The prominent spike at 5.6 kHz is the squeal tone. Note, however, that there are clear harmonics at 11.2 and 16.8 kHz as well, though these tones are some 20 dB down from the main tone at 5.6 kHz. A weaker third harmonic is present at 22.4 kHz, some 38 dB down from the fundamental. From a qualitative perspective, only the tone at 5.6 kHz was audible.

Finally, Figure 18(c) depicts the sound pressure level spectrum once synchronization has occurred, at a control signal amplitude of 153 V r.m.s. Dither has suppressed the 5.6 kHz squeal and all of its harmonics. The only prominent tone remaining is that of the 20 kHz dither control signal. Again, since this dither signal is ultrasonic, it was not audible. [4]

CONCLUSIONS

Brake pad damping is a very important parameter for brake squeal suppression. In general, two damping mechanisms are involved in braking systems.

- Material damping: This form of damping is provided mainly by the pad assembly and in particular by the insulator. The level of damping necessary for a quiet braking system depends on the level of the friction excitation (which would drive the brake system toward instability). This excitation is difficult to predict and will change with temperature and during the life of the vehicle. Because of the thickness limitations of the pad, however, the level of damping achieved with an insulator is limited.
- Coulomb friction damping: This form of damping is present with any kind of insulator, but it is difficult to predict and its effect will change throughout the life of the car.

The introduction of a brake insulator will increase the damping, but will also influence the coupling between the different brake components. Vibrations are generated at the pad-rotor friction interface and are transferred via the pad to excite the caliper (piston[s] and fingers). This transmission will be affected by the stiffness of the contact between the pad and the caliper. Adding an insulator to the pad assembly may alter this contact stiffness and then change the pad-caliper vibration transmissibility ratio. This is what we call the isolation (decoupling) effect of brake insulators. As a rule of thumb, an insulator with a soft and thick outside layer (usually rubber) will have good isolation properties.

Brake noise is very complex in both phenomenon and mechanism. Depending on the type of braking system, different types of insulators are needed. One of the challenges for the future will be to design insulators with good damping and isolation properties and good durability.

ACKNOWLEDGMENTS

This paper was realized within the researching project “Methods for determining the dynamic characteristics of motor vehicles in the function of traffic safety” ref. no. 15014, funded by Ministry of Science of the Republic of Serbia.

REFERENCES

[1] Bakar, A. R. A., Ouyang, H., Titeica, D., and Hamid , M.K.A.: Modelling and simulation of disc brake contact analysis and squeal, Seminar on Advances in Malaysian Noise Vibration and Comfort (NVC 2005) 17-18 May 2005

[2] Brake Noise, Vibration and Harshness: Technology Driving Customer Satisfaction, Akebono braking technology, www.akebonobrakes.com/company/media_center/white_papers/ akebono_NVH.pdf, (23.09.2010)

[3] Chen, F., An Tan, C., Quaglia, R. L.: Disc Brake Squeal-Mechanism, Analysis, Evaluation, and Reduction/Prevention, SAE International, USA, 2006.

[4] Cunefare, K. A. and Graf, A. J. : Experimental active control of automotive disc brake rotor squeal using dither, Journal of Sound and Vibration (2002) 250(4), 579-590

[5] Importance of NVH in OEM and Aftermarket Friction Material, www.ceet.niu.edu/faculty/lichuanliu/491/IMS%20project/further%20information/ Pedigogical%20Automotive%20Friction%20Materials.pdf (23.09.2010.)

[6] LMS Engineering - Solving Brake Noise Problems, Application Case, http://www.lmsintl.com

[7] Belingardi G., Avalle M.: Advanced materials for automotive applications, Mobility and Vehicle Mechanics-MVM, Volume 30, Nº 2-3, pp. 51-65, 2004