

Elaborate Measuring System for Sensitivity Analyses and In-Depth Investigations of a Squealing Brake System

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ABSTRACT

Brake squeal is an elusive problem which has been the subject of investigation for many decades, but there is still a lack of knowledge regarding the excitation mechanisms. New vehicle solutions, for instance the electrical vehicle, will have a lower general noise level. Thus, silent brake systems will gain in importance. To obtain such systems, in-depth investigations of the brake disc/pad contact are required.

For these investigations a new sensor has been developed. The guide pins of the calliper are replaced by modified ones which measure the friction force. Additionally, eddy current sensors are installed for contact-free measurement of the pad movement. Furthermore, triaxial acceleration sensors are mounted in the disc vents. Thus, it is possible to evaluate the operational deflection shapes of the disc.

Next, an extensive sensibility analysis is performed. Parameters such as environmental conditions, friction coefficient and many others are thereby changed. For example Fig. 1. shows the root mean square (RMS) value of the out-of-plane vibration of one acceleration sensor mounted in the brake disc vents. Five revolutions are depicted in the polar axis plot. The sensor passes the contact area at an angle of ± 18 degrees. The significant higher value at running-in occurs only when squealing is present.

The innovative design of the guide pin sensors allows measurements with a very high resolution. Thus, the friction force and even the superposed vibration of this force can be measured. The results lead to further insights regarding brake squeal triggering mechanisms.

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INTRODUCTION

Today's automobile industry is faced with growing customer's demands. Noise levels which were previously accepted are no longer reasonable. Additionally, new vehicle designs according to the powertrain will lead to quieter vehicles. For example electric cars are significant quieter than current vehicles with combustion engines.

Consequently, all subsystems have to be optimized regarding their acoustic performance. In particular the brake systems are forced with completely new requirements in electric vehicles. Most of the brake applications are performed using the electric drive as a generator to recuperate the brake energy and charge the electric accumulator. Only shortly before standstill does the conventional brake system take over the brake application. The change of the used system for deceleration, hence from electrical generator to friction brake, should not be noticed by the customer. Therefore, brake blending is necessary, which can be implemented in different ways. Nevertheless, the friction

brake is used less and mainly at low speeds with a low brake pressure. However, precisely this operating mode and condition are critical regarding brake squeal. Moreover electric vehicles are quieter, a squealing brake is better audible. For further details on brake blending and brake systems in electric vehicles see e.g. [1, 2].

To avoid a squealing brake system, different approaches are necessary. Firstly, minimal models are used to gain a basic understanding of the principle instability mechanism such as stick-slip, sprag-slip, mode coupling, negative friction coefficient-velocity slope, etc. But these models are only minimal models and a transferability of the results to existing brake systems is hardly possible. An overview of brake squeal minimal models can be found in [3] and a discussion on their suitability in [4].

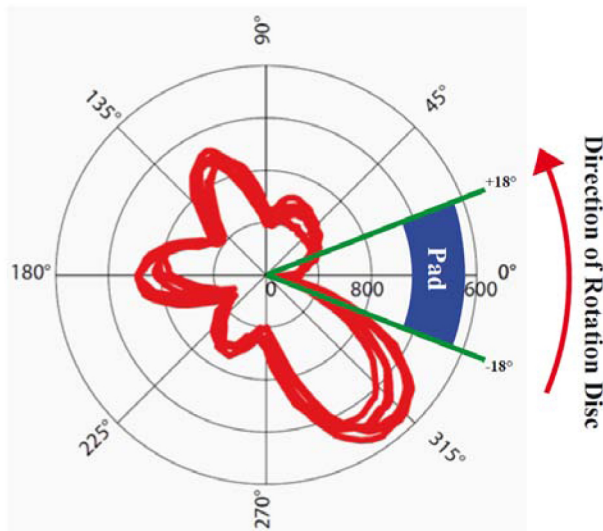


Figure 1. Polar axis plot disc of an acceleration sensor mounted in the brake disc vents, out-of-plane direction, RMS value.

Secondly, the finite element method is used for simulation and calculation. Thereby a detailed model of the brake system, especially the joints and boundaries, is required. The audible squeal is caused by self-excited vibration because of friction between brake pad and disc. A general mathematical description is given in Eq. (1).

$$M\ddot{x} + D\dot{x} + Kx = 0 \quad (1)$$

In this differential equation of an already simplified model, hence linearised and gyroscopic forces are neglected, M is the mass matrix, D the damping matrix and K the stiffness matrix of the system. The displacement vector is given by x . Because of the self-excited vibration no externally applied force is necessary, thus the right side of the equation equals zero. Because of the friction non-conservative entries are included in the stiffness matrix. Thus, the stiffness matrix is asymmetric and the system has complex eigenvalues. Eigenvalues with a positive real part are negatively damped and thus the system may become unstable.

The finite element method uses the explained characteristic in a so called “complex eigenvalue analysis”. Usually there is an overestimation of critical conditions. Hence, there are more unstable solutions in the simulation compared to unstable conditions of the real system. The second problem is that the value of the negative damping corresponds to the amplification rate. However, the more important value is the amplitude reached in the limit cycle, which corresponds to the sound intensity of the squeal noise. Thus, a complex eigenvalue with a very high negative damping value appears to be very critical. Nevertheless this unstable oscillation may result in a very uncritical limit cycle and can be neglected. [5,6,7]

Due to the above-mentioned problems, the third approach is always necessary. Experiments depict the real critical vibration modes. The operating deflection shapes can be measured and compared with the results of the simulations. Thus, the real critical unstable vibrations can be detected and treated in a targeted and controlled manner. Thereby different kinds of investigations regarding the complexity of the used measurement devices and evaluation level are performed.

For example, the standardised SAE J2521 “Disc and Drum Brake Dynamometer Squeal Noise Matrix” test [8] can be used to compare different set-ups of the brake system. The complete test consists of 2,321 brake applications at different speeds, brake pressure, brake pads/disc temperature, environmental conditions and rotation directions. Thus, a brake system can be classified according to squeal occurrence and critical operation conditions. Together with the results of the finite element method, first counter-measures could be developed.

Most of the counter-measures result in a trade-off. A damping shim and a softer brake pad material make the pedal feel softer. On the other hand, a decoupling of brake pedal and brake system is necessary for a practical implementation of systems for recuperation. Most of the current implemented systems use a “pedal simulator” to reproduce a familiar pedal feeling. A reduction of the friction coefficient would reduce the excitation mechanism, but also the performance of the brake. A softer pad material results in higher wear. But again in electric vehicles the brakes will be used less, resulting in a longer replacement period.

Consequently, also in-depth investigations are very important to gain further insight into squeal triggering mechanisms and to optimize the used counter-measures. For example, the influence of different disc surfaces on brake squeal was tested. Therefore, the disc surface was changed through shot-blasting [9]. Also the influence of different shims [10] and brake pad materials [11] were investigated.

In addition, new measurement methods are developed. In order to determine the operating deflection shape of the disc, triaxial acceleration sensors are mounted in the disc vents [12]. Therefore, also scanning laser vibrometers can be used. Basic information on this and a discussion of the results of an exemplary performed measurement can be found in [13]. For hot judder investigation, five capacitive sensors are mounted along the radial direction of the disc. Thus, the disc thickness variation after a high performance brake application can be evaluated [14]. Using twelve capacitive sensors arranged tangentially around one radius of the brake disc, the operating deflection shape of the disc can be determined [15]. Another research team modified the brake pads: four piezo-electric sensors are placed between the brake pad material and the back plate. Thus, it was possible to measure the exact friction radius; hence the effective radius of the resulting friction force acts on under the assumption of a constant friction value in the whole contact surface. Additionally, the sensors could be used as actuators and thus suppress the squealing vibration completely using active actuation [16].

METHODS

In addition to road testing, different kinds of test rigs can be used for the acoustic testing of a brake system. For the presented work it was necessary to extend the available combined brake and suspension test rig. On the brake test rig, see [Fig. 2](#), brake performance and friction coefficient tests are carried out. The suspension test rig, see [Fig. 3](#), is used for durability test runs of a vehicle corner.



Figure 2. Exemplary test set-up of the brake test rig for performance testing.



Figure 3. Exemplary test set-up of the suspension test rig for endurance testing.

For the acoustic testing, the complete vehicle axle has to be mounted on the test rig. Therefore, it was necessary to design a special rack with the same connection points that the suspension has to the car body. In addition the suspension was pre-stressed to obtain the same conditions as in reality. [Figure 4](#) shows the used configuration. Moreover, the whole

test rig is located in a climatic chamber. Thus, different environmental conditions can be simulated.



Figure 4. Test set-up for NVH investigations.

The aim was to design and construct a measuring system for in-depth investigations. An extremely noisy brake system is thereby more practicable than a quiet one. For the investigated brake system, four different brake pads were available: the original and replacement brake pads of different manufacturers which consist of organic material, and two different prototype brake pads which consist of sintered material. Usually sintered brake pads are only used in high performance application, such as rail vehicles and industrial applications. Sintered brake pads are stiffer and have a higher friction coefficient. But this results in a higher excitation and the system is more likely to squeal. The different brake pads were classified using the SAE J2521 test. One outcome of the test is the squeal index, which is defined as squealing brake applications/performed brake applications ratio. [Table 1](#) list the results of the test.

Table 1. Squeal index for all four investigated brake pads.

	Squeal Index
Original Organic	~ 0.5 %
Replacement Organic	~ 14 %
Sinter Mixture B	~ 40 %
Sinter Mixture A	~ 70 %

The original brake pads are already optimised regarding squeal and different counter-measures are used. This system squeals only very rarely. The replacement pads are tested with removed damping shims. The best brake pads for repeatable measurement are the sintered brake pads. Thereby an interesting aspect is that the only difference between the two mixtures is a slightly different amount of softener. However, such small modification in the mixture leads to a different acoustic behaviour. The sinter mixture A brake pads