# Active Control of Disk Brake Squeal

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**Abstract**

Brake squeal is a high frequency noise (1-12 kHz) of brake systems [1]. Even though it does not affect the performance of the brake, the noise decreases the passengers comfort and the subjective quality of the vehicle. Due to the fact that the source and mechanism of generating brake squeal is not completely understood yet, the design of disk brakes of modern passenger cars demands a significant amount of experiments and causes costs. Active control of brake squeal could be a helpful tool for the development of new brake systems and could simplify the experimental handling of brakes as well.

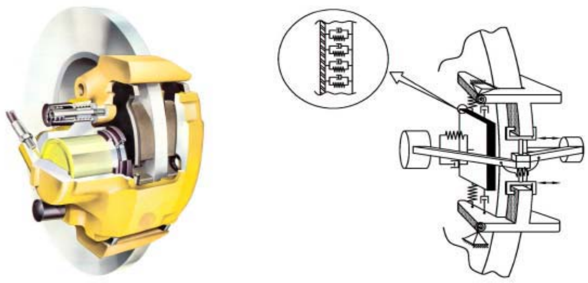
The present work is based on a nonlinear low degree of freedom model of a floating caliper disk brake. Follower loads between the brake pads and the disk cause instability of the trivial solution. The non-linear stiffness of the brake pads leads to a limit cycle behavior such that the model agrees with experimentally observed squealing. Brake pads with piezoceramic actuators are used for the manipulation of the system. Those "smart pads" can be used for both exciting and suppressing squeal. This provides new opportunities for the design process of a brake. The efficiencies of the methods are confirmed by corresponding experiments.

**Keywords:** Brake squeal, Active control, Smart pads

## 1. Introduction

Automotive brakes have greatly developed during the last decades. Engine power has increased considerably during this period and since the power to be dissipated in a brake can be several times than that of the engine, so did the forces acting at the brake. The geometric space available for the brakes in a car is however rather limited and constrained by the dimensions of the wheel, so that new forms of brake design had to be developed and some of these new designs were more susceptible to generate unwanted noises.

A typical modern floating caliper disk brake is shown in figure 1 in an artist's view, together with a mechanical model to be discussed in more detail later. The brake consists of a brake rotor (*the disk*), housing, piston, yoke and brake pads. The braking force is generated by friction between brake pads and disk. Not only the power but also the expectations for comfort have greatly increased over the last decades. This means that noise levels, and in particular brake noise, which were acceptable 20 or 30 years ago are no longer tolerated by the modern customers.



(a) (b)

Figure 1. Floating caliper disk brake, a) artist's view and b) dynamic model (schematic)

It should be noted that brake noise in general is exclusively a comfort problem, not affecting the brake function. Although there are some radical developments under way in brakes ("brake by wire"), the noise problem is not affected, as long as the brakes work with energy dissipation by dry friction. The different types of brake noise can be classified with respect to their frequencies and generating mechanisms. Brake noises in the frequency range of 100-1000 Hz are normally known as grind, grunt, moan and groan, etc. and can have different causes. The higher frequency noise in the frequency range between 1-12 kHz is normally termed squeal; it typically occurs at low speeds (0-10 km/hr), e.g. during stopping at a traffic light [1, 2, 3, 4].

The authors believe that the flutter instability, found even with a constant friction coefficient, is a more realistic cause of disk brake squeal [2, 3, 4, 6, 7, 8]. In this present paper we discuss the use of "smart pads" for the experimental investigation of brakes and the active suppression of squeal. These "smart pads" are pads with implemented piezoceramic actuators designed and manufactured at the Department of Applied Mechanics, TU Darmstadt. The authors believe that the "smart pads" should be useful in the brake designer's laboratory to establish a prototype's susceptibility to squeal, providing a tool for testing and improving brake prototypes.

## 2. Mathematical-Mechanical modeling

This section describes the mathematical-mechanical modeling of the floating caliper disk brake via the simplified model sketched in figure 2 used for formulating the control laws only. The control was then verified using the more complete nonlinear models corresponding to figure 1b and implemented in the laboratory on a standard disk brake on our test rig. The more complete mathematical models are described in [4, 5, 7, 8] and are not discussed further here.

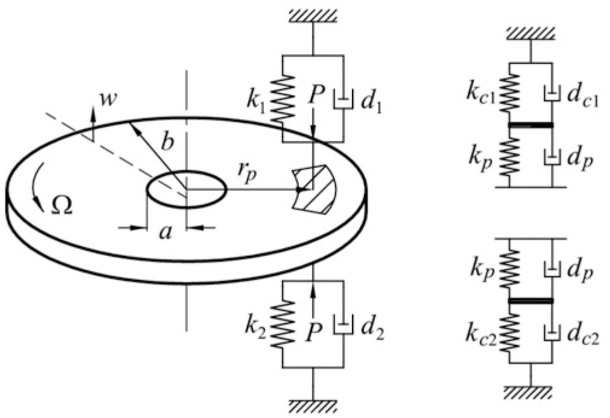



Figure 2. A simplified model of the disk brake for use in active control

In the simplified model of figure 2 the brake disk is modeled using the equations of motion of an equivalent annular KIRCHHOFF plate with corresponding boundary conditions (rigid hub and free boundary at the outer radius). The eigenvalues problem for the free vibrations can be solved analytically and the disk's eigenfrequencies and eigenfunctions or mode shapes are readily obtained.

Of course a brake disk is definitely not a simple homogeneous plate, it is actually formed by two plate-like structures with interconnecting ribs delimiting the ventilating channels and connected to a rather rigid hub. The mass density and the bending stiffness of the equivalent plate are therefore parameters which cannot directly be obtained from the material properties and from the thickness of the actual brake disk. They are the parameters of an ideal KIRCHHOFF plate dynamically equivalent to the brake disk within the frequency range under consideration [4, 7]. Generally, in brake squeal models only the first few bending modes are of importance.

Comparison with results obtained from an experimental modal analysis of an actual brake disk in the frequency range up to 5 kHz showed a good coincidence with theoretical results. With the relatively simple KIRCHHOFF plate model it was possible to correctly capture not only the eigenfrequencies and modes but also the disk's complex dynamic impedance in this frequency range by appropriately choosing only one set of parameters for the inertia term $\rho h$ and the bending stiffness $D$. This clearly shows that the simple KIRCHHOFF plate model gives an appropriate description for the dynamic behavior of the brake disk with regard to squeal within this simplified model. Of course for higher plate modes, shear stiffness and rotational inertia become important, so that a REISSNER-MINDLIN theory must be used for the plate. Figure 3 shows the second vibration mode of a brake disk clamped at a rigid inner hub during squeal, clearly exhibiting a vibration form with three nodal diameters ($m = 3$) and one nodal circle ($n = 1$).

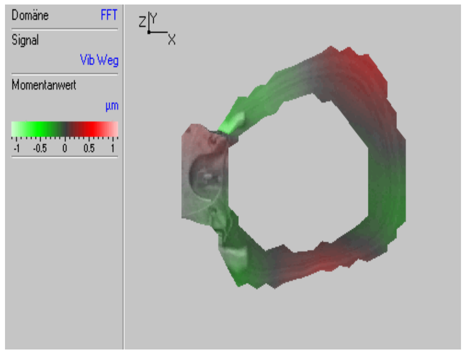



Figure 3. Laser scan image of a vibrating brake disk

The KIRCHHOFF plate is then discretized using the solutions of the eigenvalue problem. Thereby, $A_{m,n}(t)$ and $B_{m,n}(t)$ are introduced as generalized coordinates characterizing the out of plane vibrations of the discretized brake disk. In the rest of this paper we only use the (double) mode with $m$ = 3 nodal diameters and $n$ = 1 nodal circle as observed in the experiment shown in figure 3. The out of plane displacement of the disk $w$ is therefore described by the coordinates $A_{3,1}(t)$ and $B_{3,1}(t)$.

Each pad is modeled by a rigid back plate and carrying an elastic layer of negligible inertia. The forces acting on each pad in transversal direction are due to the contact pressure with the disk and the elastic deformation of the caliper.

The equations of motion of the complete system linearized for constant brake pressure $P$ can be written in matrix form as

$$\mathbf{M}\ddot{\mathbf{q}} + (\mathbf{D}+\mathbf{G})\dot{\mathbf{q}} + (\mathbf{C}+\mathbf{N})\mathbf{q} = \mathbf{f}(t)\,, \qquad (1)$$

where the vector of the generalized coordinates is

$$\mathbf{q} = [A_{3,1},\ B_{3,1}]^{\mathrm{T}}\,, \qquad (2)$$

and

$$\mathbf{f}(t) = [u_1,\ u_2]^{\mathrm{T}}\,, \qquad (3)$$

consists of the input voltages $u_1$ and $u_2$ applied to the "smart pads", which are described in the next section.

The square matrices in equation (1) have the usual properties

$$\mathbf{M}^{\mathrm{T}} = \mathbf{M},\ \mathbf{G}^{\mathrm{T}} = -\ \mathbf{G},\ \mathbf{D}^{\mathrm{T}} = \mathbf{D},$$
$$\mathbf{C}^{\mathrm{T}} = \mathbf{C},\ \mathbf{N}^{\mathrm{T}} = -\mathbf{N}. \tag{4}$$

The skew-symmetric matrices **G** and **N** are due to the rotation of the disk and the friction forces respectively. The friction forces resulting in **N** can cause instability of the trivial solution. In combination with nonlinear effects, in particular a stiffening characteristic of the pad stiffness [5] is represented in terms of linear, quadratic and cubic functions. The instability of the trivial solution results in a limit cycle behavior to be interpreted as brake squeal.

**3. Experimental set up and "smart pads"**

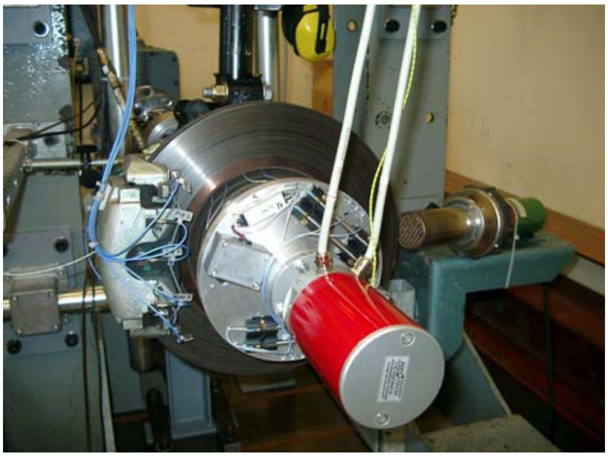


Figure 4. The test rig for brake squeal in the Darmstadt laboratory

A test rig was developed and built in the laboratory in Darmstadt with the goal to validate the models, to identify the model parameters, to test possible measures mitigating squeal and in particular also to realize active squeal control in the laboratory. In the test rig shown in figure 4, a floating caliper disk brake is shown mounted using the original suspension. Besides the actual brake and the suspension, the rig consists of an electric motor, a gearbox, a real time data acquisition and processing system ADWIN-PRO as well as sensors and their peripheries. No large power is required in the drive, since squeal typically occurs at low speeds. The data acquisition and processing system is modular and presently has 32 input channels as well as 8 output channels. Sampling rates of up to 100 kHz are possible. Programs for data processing written in ADBASIC can also run in real time for frequencies up to 100 kHz, depending on their size and efficiency. Control signals can be generated using the 8 output channels of the AD card. In addition, the ADWIN-PRO system has a MATLAB interface, through which signals can be sent to a PC for further processing, and, in the reverse direction, parameters in the real-time program can be adjusted via MATLAB. Of course MATLAB codes can not be executed in real time with such high frequencies.

Figure 5. Prototypes of "smart pads"

The sensors comprise four uni-axial accelerometers placed in the ventilating channels of the brake disk, measuring the transverse and the in-plane vibrations. Seven triaxial accelerometers can be placed at different locations of the yoke, the caliper or on the pads. A microphone is used to record the noise generated by the brake.

"Smart pads", i.e. brake pads instrumented with piezoceramic elements, were installed at the disk brake in the Darmstadt test rig, as depicted in figure 5, for brake squeal. The piezoceramic elements were placed between the pad's metallic back plate and the caliper. The "smart pads" performed very well under all conditions tested. It is of course well known that problems may occur using piezoceramic actuators subjected to high temperatures. For the piezoceramics used the Curie temperature leading to depolarization is around 350°C, which would be quite a high temperature for the back plate.

**4. Active control of brake squeal**

The active control of brake squeal was modeled and studied in [6]. Chakraborty *et al* [4] used a model similar to fig. 2 without modeling the actuators. He showed that in this model the system is not controllable in particular by a varying pre-stress realized by a varying brake pressure that however it is controllable via two control forces, which can be applied to the two brake pads. In [6] Hochlenert studied this control problem in more detail and control laws were developed for eliminating the brake vibrations leading to squeal.

In developing an appropriate control law, optimal control was used with an LQR formulation, so that the linearized equations of motion were written in the form

$$\begin{aligned}\dot{\mathbf{x}}(t) &= \mathbf{A}\mathbf{x}(t) + \mathbf{B}\mathbf{u}(t),\\ \mathbf{y}(t) &= \mathbf{C}\mathbf{x}(t) + \mathbf{D}\mathbf{u}(t),\end{aligned} \tag{5}$$

where $\mathbf{x}$ is the state vector, $\mathbf{u}$ the control vector, $\mathbf{y}$ the measurement vector, $\mathbf{A}$ the system matrix, $\mathbf{B}$ the input matrix, $\mathbf{C}$ the measurement matrix and $\mathbf{D}$ the direct transmission matrix. For chosen matrices $\mathbf{Q}$ and $\mathbf{R}$ in the cost function

$$J=\int_0^\infty [\ \mathbf{x}^{\mathrm{T}}(t)\ \mathbf{Q}\ \ \mathbf{x}(t)+\mathbf{u}^{\mathrm{T}}(t)\ \ \mathbf{R}\ \ \mathbf{u}(t)\ ]\ \mathrm{d}t, \quad (6)$$

the control law

$$\mathbf{u}(t)=-\mathbf{K}\mathbf{x}(t), \quad (7)$$

i.e. the matrix $\mathbf{K}$ was found by solving the appropriate matrix RICCATTI equation. Since the complete state vector is not measured in the experiment, with the additional difficulty that the signals of the accelerometers in the disk give the vibrations at a body fixed location in the rotating disk, a LUENBERGER observer was used to estimate the state vector. The control law so developed for the simplified low order linear model was first tested in simulations with the more complete models, where it succeeded in suppressing the self-excitation [8]. Next it was implemented in the test rig. Problems related to the real time requirements were solved developing an efficient ADBASIC code.

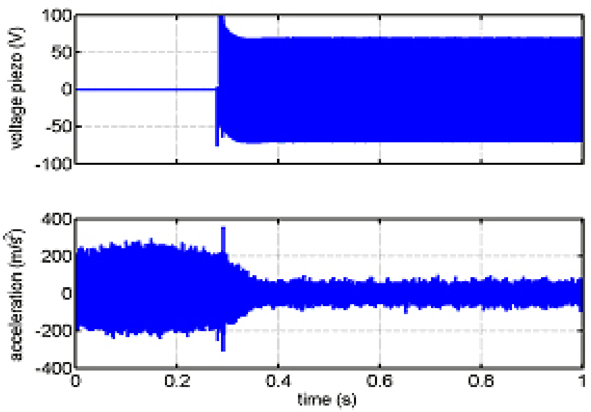



Figure 6. Active suppression of brake squeal using optimal control

Using this method, it is possible, to actively suppress squeal. Figure 6 exhibits corresponding results. The lower part of this figure shows the time signal of an accelerometer at the outer pad, which was used as an input for the observer. The initially high level of the signal indicates a squealing brake. Approximately at $t$=0.3 s, the control is switched on, as can be seen from the upper part of this figure, showing the actuator input voltage (at one of the two "smart pads"). The acceleration signal (the squealing) is suppressed within a very short time to a level, where it can no longer be heard. After switching the control off, squealing restarts (not shown in figure 6). This behavior is very reproducible. A phase shift of approximately 180° is apparent in figure 7 between the voltages $u_1$ and $u_2$ applied to the "smart pads". The resulting effect is, that while on one side the pre-stress of the pad decreases, the pre-stress of the opposite pad increases and vice-versa. Within the scope of our model, this could not be achieved using the hydraulic brake pressure as a control variable.

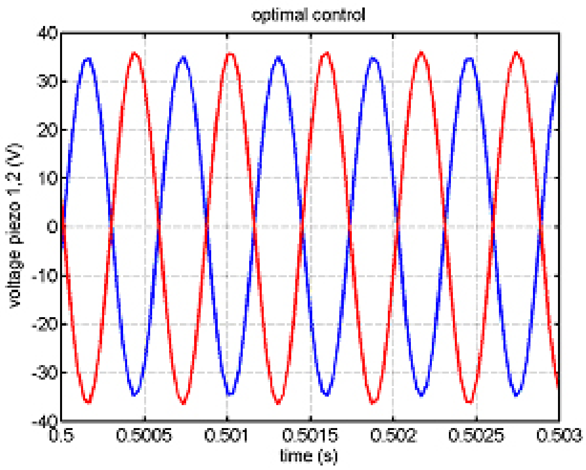



Figure 7. Voltages at the outer pad $u_1$ (blue) and the inner pad $u_2$ (red) using optimal control

**5. Conclusion**

A model was developed for a floating caliper disk brake, with particular attention being paid to modeling the self-excitation leading to brake squeal. Suppression of the self-excitation via active control has been studied. "Smart pads" with implemented piezoceramic actuators were developed. A comparably small number of degrees of freedom was chosen because the active control of the brake squeal requires an observer with real time numerical integration. Using optimal control, it was almost possible to actively suppress squeal

Anyway active control of squeal presently is not envisaged as a technique to suppress squeal in the brake of passenger car, but rather as a possible tool to be used in industrial laboratories to shorten the time for optimizing new brake designs, with high potential saving benefits. The development and laboratory implementation of the active squeal control goes along with a more profound understanding of brake squeal and a better modeling of the phenomena, ultimately leading to improvement in the design of disk brakes.

**Acknowledgments**

This research is financially supported by the Thailand Research Fund (TRF), contract no. MRG4980185. The authors would like to thank the colleagues in TU Darmstadt for their helps and incorporations.

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