

Fast Characterization of Brake Squeal Behavior

Stefan Schlagner and Utz von Wagner
Technische Universität Berlin

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1. ABSTRACT

The last decades have shown extensive efforts on the investigation of automotive disk brake squeal. The origin of brake squeal is seen in self-excited vibrations, caused by the friction forces transferring energy from the rotating disk into the brake system. Based on a very simple model, Popp et al. described in 2002 the conditions for positive work of the friction forces (i.e. excitation of squeal), which depends on the phase shift between the in-plane motion (with respect to the disk) of the brake pad and the friction forces. Experiments on active manipulation of this phase shift using pads with integrated piezoceramic actuators, performed by von Wagner et al. in 2004, resulted in successful suppression of disk brake squeal.

The authors of the present paper used a variety of models for the investigation of the origin of the excitation mechanism by observing phase relations between the friction forces and the vibrations of the pads. Several measurement techniques based on these theoretical investigations are developed in order to detect parameter regions (e.g. brake pressure, temperature, rotation speed) which are suspected to cause squeal. Compared with the state-of-the-art testing of brakes, this procedure provides a very rapid and efficient analysis of the noise behavior and has been verified by experimental results. It also provides a method of testing and modeling the efficiency of standard industrial counter measures to squeal.

2. INTRODUCTION

It is commonly acknowledged that disk brake squeal originates from self-excited vibrations caused by the friction forces, which result in unstable trivial solutions. However, literature offers different reasons for the onset of instability. As examples, a change of the friction characteristic with respect to the speed of the contact points [1-3] or a flutter instability which is found even with a constant friction coefficient [4-21], can be named. However, not all models and mechanisms in literature have been validated by physical experiments. On the one hand, it is well known that a negative slope in the friction characteristic leads to instability and self-excitation. On the other hand, laboratory experiments have shown that there may be instability and self-excitation leading to squeal even in the absence of a negative slope of the friction characteristic. Because of this, the authors of the present paper use models that also show instability with constant coefficient of friction.

Brakes are commonly modeled as multi-body systems (MBS) or using finite element methods (FEM) resulting in models with high numbers of degrees of freedom. Nevertheless, for a basic understanding of the excitation mechanism, which is seen as the origin of disk brake squeal, models with a low number of degrees of freedom are useful [4, 5]. For this purpose a very simple model by Popp is used to characterize the exciting or dissipating behavior of the friction forces. Furthermore these results will be verified by examining a model with ten degrees of freedom. Based on this examination, an experimental verification is performed.

3. MINIMAL MODEL BY POPP AND ACTIVE SUPPRESSION OF DISK BRAKE SQUEAL

The minimal model presented by Popp et al. in [6] consists of just a body sliding on a moving conveyor belt (Figure 1).

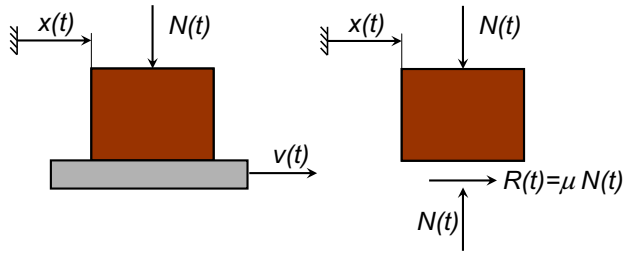


Figure 1: Minimal model by Popp: Sliding body with fluctuating normal and friction forces

The body of this model can be interpreted as a brake pad and the conveyor belt as the disk. The in-plane motion $x(t)$ of the pad is given by

$$x(t) = A \cos \omega t. \quad (1)$$

In fact, measurements at squealing disk brakes (e.g. [7]) show that there exist nearly harmonic vibrations of the pad in out-of-plane and circumferential in-plane direction (with respect to the disk) of comparable amplitudes (in the range of one μm). Compared to this, the vibration amplitudes in radial in-plane direction are much smaller (see e.g. [7]). Vibrations of the surrounding structure (e.g. the caliper) will also result in a time dependent normal force $N(t)$. This normal force, acting on the body, consists of a large constant preload N_0 (caused by the brake pressure) and a small oscillating part that is also assumed to be harmonic. In total, $N(t)$ follows to be

$$N(t) = N_0 + \hat{N} \cos(\omega t - \varphi) \quad (2)$$

and has the same frequency as the in-plane motion but a phase shift φ . For the corresponding time dependent friction force

$$R(t) = \mu N(t) \quad (3)$$

a constant friction coefficient is assumed. The conveyor belt moves with constant velocity v_0 , which is higher than $|\dot{x}(t)|$, so that the friction force acts in the same direction at all times. The mechanical work of the friction force $R(t)$ within one oscillation period

$$T = 2\pi/\omega \quad (4)$$

is calculated to be

$$W = \int_0^T \mu N(t) \dot{x}(t) dt = -\mu \hat{N} A \pi \sin(\varphi). \quad (5)$$

It is obvious that the sign of this work depends only on the phase shift φ . Hence, the phase shift φ characterizes whether the friction force induces energy into the system or if energy is dissipated by it. It characterizes correspondingly if the friction force excites or stabilizes the system. From this it follows that for a positive work the system can become unstable and for a negative work the system must be stable. In this context it should be mentioned that the work of the friction force at the pad is not the only mechanical work acting on the system. Namely there is the work due to the driving torque and due to the friction force at the disk. However, in [8] it is shown that the work of the friction force at the pad is nearly equivalent to the total work.

In comparison to a real brake it is obvious that for this model the description of the vibrating system (i. e. the disk, brake carrier and caliper) is nearly completely missing. On the other hand, the model by Popp is a simple and lucid explanation of the exciting or non-exciting nature of friction forces and therefore for the origin of squeal. In [9, 10] the results were used for the active suppression of brake squeal. For this purpose pads with integrated piezoceramic actuators (Figure 2) were used. These pads are also used in the experiments described in chapter 5. By applying a corresponding voltage to the piezoceramic actuator an additional normal force acts at the pad in such a way that a maximum dissipation is reached. Therefore the phase shift has to be $\varphi = \pi/2$. Figure 3 shows the corresponding time responses. In the upper part, the applied voltage and in the lower part the in-plane acceleration of the pad are sketched, respectively. By switching on the control at $t = 0.3\text{s}$, the voltage with the corresponding phase shift is applied to the active pad. The acceleration amplitude decreases characteristically and the squealing stops immediately. This experiment can be seen as a simple verification of the model assumption by Popp. Of course, real brakes are much more complicated in detail, still the experiment shows that the basic concept of this model is legitimate. Hence, a multi-degree-of-freedom (MDOF) model of a brake will be considered in the following, with special focus on the exciting or dissipative behavior of the friction forces.

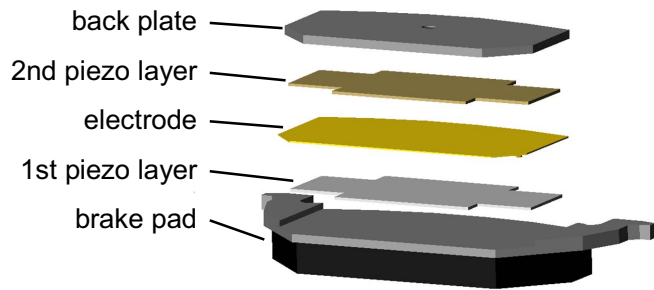


Figure 2: Pad with integrated piezoceramic actuators

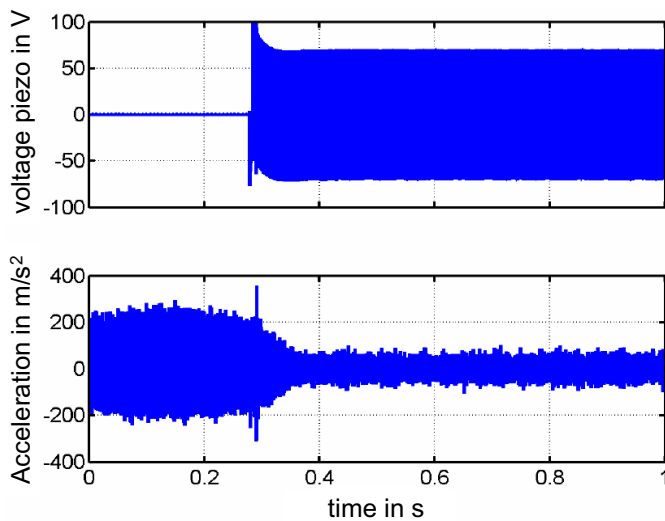


Figure 3: Active suppression of disk brake squeal using the phase relation described in [9]

4. THEORETICAL INVESTIGATIONS USING A MDOF BRAKE MODEL

The model shown in figure 4 includes a disk rotating with constant angular velocity $\dot{q}_0 = \Omega$. The model consists of a Kirchhoff plate with the shape of a circular ring, which is fixed at the inside radius. In order to get the linearized equations of motion for the model, the disk is discretized with a Galerkin procedure using two pairs of eigenmodes of the non-rotating disk which is not in contact with the pads (see also [7]). This results in four generalized coordinates $wc_1(t)$, $ws_1(t)$, $wc_2(t)$ and $ws_2(t)$. The pads can perform movements in circumferential in-plane and out-of-plane direction (with respect to the disk), specified by the rotations $\varphi_{po}(t)$ and $\varphi_{pu}(t)$ as well as by the translational motions $z_{po}(t)$ and $z_{pu}(t)$. The carrier is modeled with two degrees of freedom $y_h(t)$ and $\varphi_h(t)$, so that its typical in-plane movements as observed in experiments are possible. This results in ten degrees of freedom for the whole system. The linearized equations of motion are derived by Lagrange's equations as done in [7, 11]. The corresponding set of parameters (a nomenclature is given in the appendix) is identified at the test rig at Technische Universität Berlin. The linearized equations of motion read as

$$\mathbf{M}\ddot{\vec{q}} + (\mathbf{D} + \mathbf{G})\dot{\vec{q}} + (\mathbf{K} + \mathbf{N})\vec{q} = 0, \quad (6)$$

with mass matrix $\mathbf{M}$, damping matrix $\mathbf{D}$, gyroscopic matrix $\mathbf{G}$, stiffness matrix $\mathbf{K}$, and circulatoric matrix $\mathbf{N}$. $\mathbf{M}$, $\mathbf{D}$ and $\mathbf{K}$ are symmetric matrices and $\mathbf{G}$ and

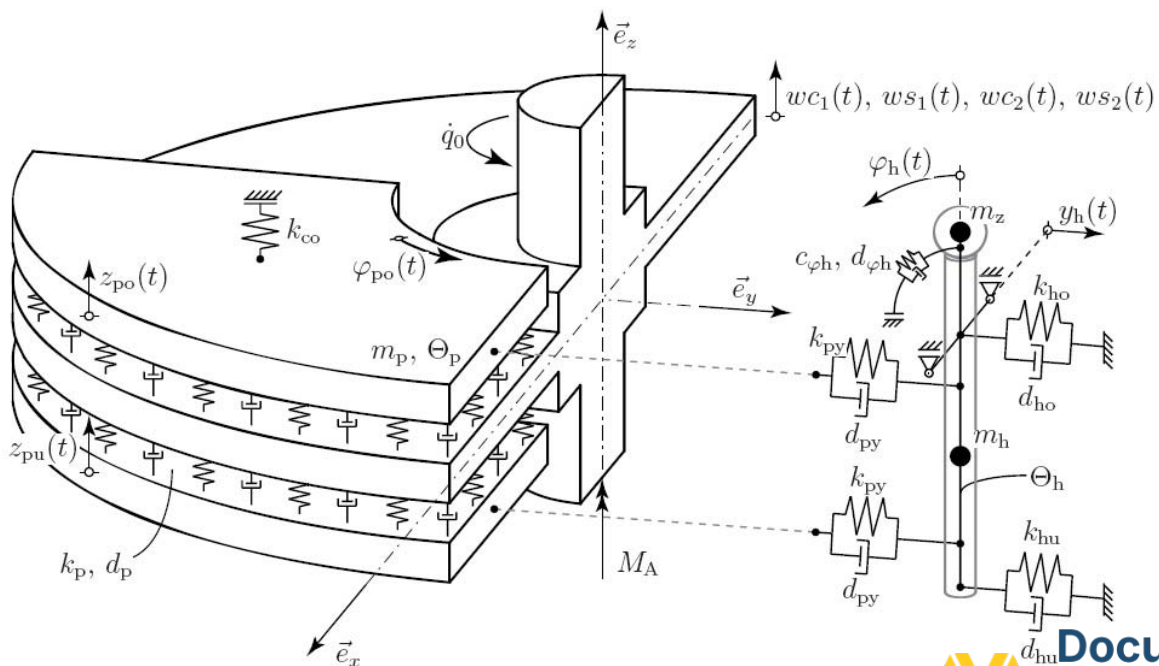


Figure 4: Brake model with ten degrees of freedom