

Steps towards Predictive Simulation and Faster Experimental Investigation of Automotive Brake Systems with Respect to Squeal

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ABSTRACT

The development process of automotive brakes is known to be challenging and time-consuming. It is an iterative process consisting of interplay between brake squeal simulation and extensive experimental investigations of the brake system at the test rig and in the vehicle. In this context, the complex eigenvalue analysis (CEA) of linearized FE models is a part of standard development process of brake systems. Nevertheless this linear analysis has not reached the status of a predictive tool yet, remaining a tool accompanying experimental investigations of the brake system only. Possible reasons may be inadequate simplifications of frictional contact, damping effects and friction material modeling on one hand and insufficiencies of the mathematical mechanical models themselves, i.e. linear vs. nonlinear stability analyses on the other hand. The extensive experimental investigations apply time consuming standard test procedures and need efficiency improvement. This paper introduces several steps towards improvement of brake development process efficiency with respect to brake squeal and gives an overview on corresponding research at the chair of Mechatronics and Machine Dynamics at TU Berlin. First it describes the refinement of friction material parameter acquisition with respect to squeal relevant loading conditions using the DCTR (dynamic compression test rig) technology. Based on this experimental identification of dynamic friction material parameters, linear as well as nonlinear stability analyses of multibody - disk brake models are performed. Pointing out limits and potentials of the particular analysis methods, the nonlinear stability analysis explains basic phenomena known from the operating experience of brake systems. Finally an experimental method for rapid identification of squeal suspicious states is introduced. The time efficiency of this method allows a fast overview on the usefulness of constructive countermeasures to avoid squeal and might contain potentials towards efficiency improvement with respect to time consuming standardized experimental investigations.

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INTRODUCTION

Brake development process implies standardized time consuming squeal noise test procedures [1] complemented by standard procedure of brake squeal simulation in terms of complex eigenvalue analyses (CEA) [2, 3]. Nevertheless the industrial application of this linear stability analysis method has not yet reached the status of a predictive tool, calling for refinement strategies. This paper is dedicated to the improvement of brake squeal simulation reliability and introduction of an experimental method for rapid identification of squeal suspicious states of brake systems. The first part deals with the refinement of friction material parameter acquisition for NVH simulations. On one hand it is well known that friction materials exhibit nonlinear

characteristics dependent on the type of loading, i.e. static, dynamic, frequency and temperature loading [4, 5, 6, 7, 8, 9]. On the other hand friction material properties influence brake squeal simulation results decisively [9,10]. For this reason it is important to reproduce the loading conditions in squealing state as close as possible in experimental setups when identifying friction material properties for brake squeal simulations. Because of this fact, the new DCTR (dynamic compression test rig) technology has been developed and tested [7, 8, 9, 11, 12]. It reproduces squeal typical out-of-plane loading situations, i.e. combined static and high frequency dynamic loading. The static preload corresponds to squeal typical brake pressures and the superimposed sinusoidal dynamic loading corresponds to the noise emitting oscillations producing displacements in micrometer range

with frequencies in kilohertz range. Finally estimated out-of-plane elastic linear and nonlinear friction material characteristics dependent on squeal typical loading parameters are presented. The second part of this paper points out qualitative differences between linear and nonlinear stability analyses results on basis of DCTR determined linear and nonlinear friction material parameters [10, 11, 12, 13]. First a linearized model of a brake is investigated. Second, assuming the friction material parameter to be the predominant source of nonlinearities a nonlinear stability analysis is performed.

The third part of the paper deals with the rapid experimental identification of squeal states [14, 15]. After the introduction of the theoretical background, the experimental set-up the processing of data and the results are presented.

DCTR MEASUREMENT AND IDENTIFICATION TECHNIQUE

Overview

The DCTR (dynamic compression test rig) measurement and identification technique enables friction material parameter acquisition for NVH simulations reproducing squeal relevant loading conditions. The DCTR technique development and its perpetual optimization is documented in [7, 8, 9, 11, 12]. This technique consists of four integral basic modules shown in Figure 1.

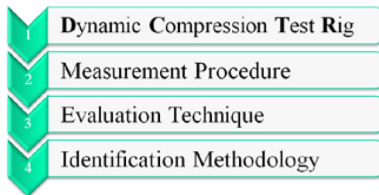


Figure 1. Integral modules of the DCTR measurement and identification technique.

In particular these modules are the dynamic compression test rig and the measurement procedure optimized upon standardized guidelines [16], an enhanced evaluation technique based on standardized evaluation procedures [17] and the identification methodology for friction material parameters out of the measured component parameters.

Dynamic Compression Test Rig and Measurement Procedure

The dynamic compression test rig is designed to measure out-of-plane friction material stiffness characteristics in a frequency range from 50 Hz up to 4000 Hz. Because of this wide frequency range also friction material parameters considering other NVH relevant issues than squeal can be determined. Additionally the DCTR is also fully capable for shim measurements. The DCTR CAE layout and the photography are shown in Figure 2. Subsequently the measurement procedure is described shortly. Details can be

found in [9]. The inner set-up is positioned in a stiff metallic block and the measurement procedure according to Figure 3 goes straight forward: The friction material specimen is subjected to squeal relevant brake pressures by adjusting the thrust screw. The preload range lies between 1 bar and 50 bar (of absolute pressure, i.e. pressure on the test specimen). The harmonic loading corresponding to squeal noise with amplitudes in micrometer range and frequencies in kilohertz range is applied by means of the piezoelectric stack actuator. Static and dynamic forces are measured by means of the load cells. From the acceleration measurements of the accelerometers placed on the lower and the upper metallic plate the relative dynamic deformation of the specimen is obtained. The successive variation of the operating condition, i.e. static preload, dynamic amplitude and frequency range determines the dependence of friction material parameters on these operating conditions.

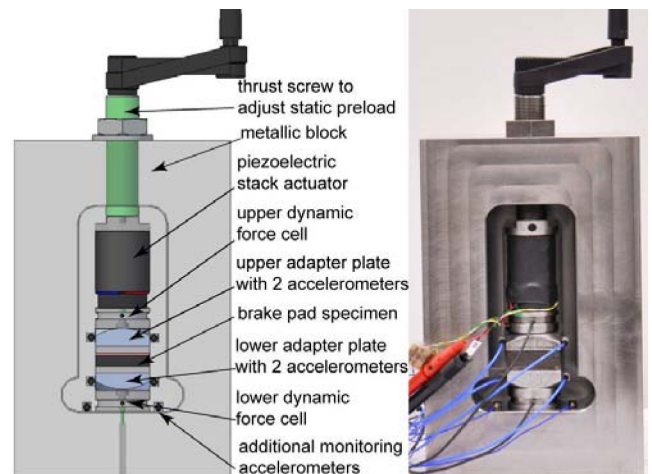


Figure 2. CAD layout (left) and photography (right) of the dynamic compression test rig.

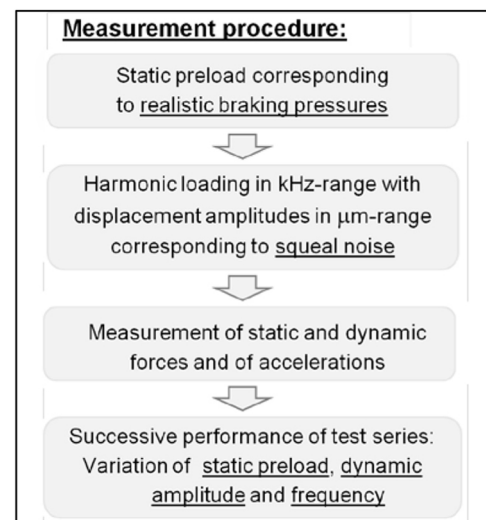


Figure 3. Measurement procedure.

Figure 4 shows the measured total dynamic force and the dynamic deformation in time domain for a constant operating condition, i.e. a constant frequency, a constant static preload and constant dynamic amplitude.

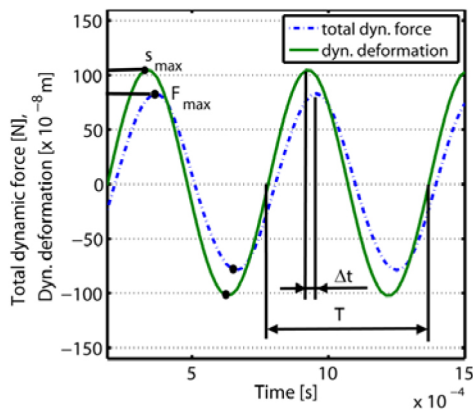


Figure 4. Total dynamic force and deformation vs. time for a constant operating condition.

Evaluation Technique and Identification Methodology

Plotting total dynamic force versus dynamic deformation obtains a linear hysteresis loop as shown in Figure 4. Considering the characteristic points $F(s_{\max})/s_{\max}$ and $F(s_{\min})/s_{\min}$ the dynamic stiffness c_{dyn} can be read

$$c_{\text{dyn}} = \frac{F(s_{\max}) - F(s_{\min})}{s_{\max} - s_{\min}}. \quad (1)$$

Determining elastic energy U_{el} and dissipative work W_d the integral damping coefficient d_{dyn} can be calculated to

$$d_{\text{dyn}} = \frac{W_d}{\pi \Omega s_{\max}^2}, \quad (2)$$

whereas Ω is the excitation frequency over a period.

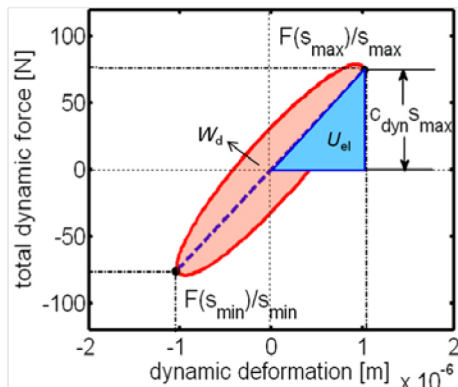


Figure 4. Determined linear hysteresis loop and calculated specimen dynamic stiffness for a constant operating condition.

Since the component dependent dynamic stiffness and integral dynamic damping values are determined, component independent friction material properties have to be identified out of these quantities. The module identification methodology takes this into account, evaluating several influence factors on measurement results, shown in Figure 5. These are in particular: specimen conditioning, contact stiffness, specimens' geometry, relaxation effects, test rig and friction effects [8, 9]. The consideration of them allows determining most accurate friction material parameter values out of the measured component values.

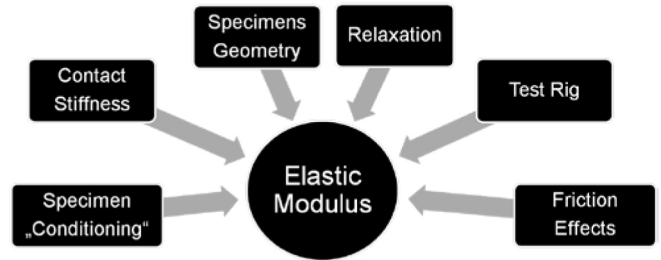


Figure 5. Influence factors on DCTR measurement results.

Dynamic Friction Material Characteristics

Figure 6 shows the measurement data of DCTR test series (points) performed for different operating conditions (different static preloads and different dynamic amplitudes) at a constant frequency for a Low Met friction material. Additionally the associated fitted nonlinear friction material characteristics are shown. Investigations performed in [7, 8, 9] show no significant dependence of the dynamic stiffness from the frequency. Contrastingly strong nonlinear stiffening dependence on static preload has to be highlighted on one hand. On the other hand the influence of the dynamic amplitude has softening characteristics and is less significant. Consequently, the strong influence of the static preload on the dynamic stiffness is expected to be one of the predominant influence factors on complex eigenvalue analysis results and is furthermore assumed to be the predominant source of nonlinearity with respect to nonlinear stability analysis.