

Active Brake Judder Compensation Using an Electro-Hydraulic Brake System

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ABSTRACT

Geometric imperfections on brake rotor surface are well-known for causing periodic variations in brake torque during braking. This leads to brake judder, where vibrations are felt in the brake pedal, vehicle floor and/or steering wheel. Existing solutions to address judder often involve multiple phases of component design, extensive testing and improvement of manufacturing procedures, leading to the increase in development cost.

To address this issue, active brake torque variation (BTV) compensation has been proposed for an electromechanical brake (EMB). The proposed compensator takes advantage of the EMB's powerful actuator, reasonably rigid transmission unit and high bandwidth tracking performance in achieving judder reduction.

In a similar vein, recent advancements in hydraulic system design and control have improved the performance of hydraulic brakes on a par with the EMB, therefore invoking the possibility of incorporating the BTV compensation feature of the EMB within hydraulic brake hardware.

In this paper, the typical characteristics of electromechanical and electro-hydraulic brake systems are presented. Based on the experimental results, the feasibility of active BTV compensation on the electro-hydraulic brake (EHB) systems is discussed. Furthermore, a BTV compensation algorithm designed for the EMB is presented and is shown to be applicable to the EHB. Using an experimentally validated model of BTV, the compensation was performed on a hardware in-the-loop EHB test rig. The preliminary results demonstrate the potential of using an EHB to compensate for brake judder.

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INTRODUCTION

Brake judder is a vibration that is felt at the steering wheel, brake pedal and vehicle body when the brakes are applied softly during mid to high speed (30 to 140 km/h) travels. It is perceived as one of the most notorious failures that can occur in a brake system and has led to recalls.

Depending on the cause and vibration frequencies, judder can be categorized into cold and hot (or thermal) judder. The majority of judder vibration is cold judder, which has one to two oscillations per wheel rotation. Cold judder is principally attributable to geometrical irregularities in rotors, such as runout, thickness variations and

friction variations on the brake lining and rotor surface [1]. On the other hand, hot judder has a higher vibration frequency (typically between six to twenty oscillations per wheel rotation) and is caused by uneven thermal deformations and expansions in rotors [1, 2]. Note that both cold and hot judders are caused by the irregularities on rotor, leading to brake torque variation (BTV) [3]. The BTV then may excite the resonance vibration modes of the suspension and steering column, causing judder vibration.

Due to the detrimental implications of judder in brake and vehicle manufacturers, major development and manufacturing resources are often spent to lower the occurrence of judder. Existing studies on the

reduction of brake judder using conventional hydraulic and electro-hydraulic brake systems mainly focus on passive approaches, such as reducing the sensitivity and the transmission of judder by modifying the brake composition [4, 5] and suspension elements [6, 7, 8]. These are costly processes, both monetarily and time-wise, as they involve multiple phases of component design, extensive testing and improving manufacturing tolerance.

Although the aforementioned activities significantly reduce the occurrence of brake judder, judder may arise due to wear during usage. When judder is detected, the after-market resolutions include replacing or resurfacing the rotor and using softer brake pads [9]. Unfortunately when action is taken at this stage, the driver has already had the undesired experience of judder vibration.

To address this issue, an active judder attenuation solution utilizing an electromechanical brake (EMB) is proposed in [10, 11, 12]. Compared to conventional hydraulic brakes, the EMB offers better fidelity in clamp force regulation, higher bandwidth in clamp force tracking and reconfigurable control software platform. Taking advantage of these strengths, novel BTV compensation algorithms are designed to regulate the EMB actuation and to produce a brake torque that is free from oscillations introduced by the geometrical irregularities in rotor. Furthermore, the proposed approach potentially relaxes the manufacturing tolerance requirements. Additionally, an early warning system can be developed to prompt the driver for servicing the brake when the rotor wear is severe, even before judder is felt. Note the proposed method eradicates judder directly from the source, as opposed to modification of the vibration transfer paths using an active suspension [13] or active steering system [14].

The proposed active BTV compensation approach imposes several performance requirements on the brake actuator which are lacking in conventional hydraulic brakes, in particular a high system bandwidth and tracking fidelity. Therefore judder can only be handled with the passive methods described previously when conventional hydraulic brakes are used. However, recently developed electro-hydraulic brake (EHB) systems [15, 16] have improved system bandwidths compared to the conventional hydraulic brake systems, therefore opening the possibility of incorporating the proposed active judder attenuation functionality in an EHB.

As alluded before, existing judder reduction methods have drawbacks in terms of high cost and bad user experience. To address these shortfalls, this paper evaluates the potential of using an EHB to actively compensate for judder. The hardware-in-the-loop test rig presented in [17] is utilized to experimentally obtain the system characteristics of a modern EHB system and to assess its potential for judder compensation.

The structure of the paper is as follows. The next section describes validated linearized models for EMB and EHB, which serve as the basis for compensator design. Then, an overview of the adaptive BTV

compensator is presented in a form applicable to both EMB and EHB. This is followed by presenting the experimental setups for both braking systems before results are presented and discussed.

LINEARIZED BRAKE MODELS

Linearized model for electromechanical brake

An EMB consists of an electric motor, planetary gears, a ball screw, a floating caliper and control electronics. An experimentally validated EMB model is presented in [18], and a procedure for identification of the system parameters is presented in [19]. Furthermore, a high-bandwidth controller design for an EMB is presented in [20, 21].

The EMB can be considered as a system that takes in clamp force reference command as input and outputs an actual clamp force. By further assuming a clamp force operating region in the vicinity of a fixed or slowly varying clamp force level, which is the operating condition corresponds to judder, the following linearized EMB model can be adopted [10, 12]:

$$\begin{bmatrix} \dot{\theta}_m \\ \ddot{\theta}_m \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ -k_p & -k_d \end{bmatrix} \begin{bmatrix} \theta_m \\ \dot{\theta}_m \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{k_p}{k} \end{bmatrix} F_{cl,ref}$$

$$F_{cl} = \bar{k} \theta_m, \quad (1)$$

where F_{cl} , $F_{cl,ref}$, $\bar{k}$ and θ_m are the brake clamp force, clamp force reference, linearised lumped stiffness coefficient, and motor position respectively. The parameters k_p , k_d are related to the system characteristics and can be modified by tuning the clamp force controller gains, where detailed tuning guidelines are provided in [12, 21].

Linearized model for electro-hydraulic brake

For the controller design purposes, a linearized model of the EHB system is utilized. Considering the system frequency response and recommendations in [22] for low pressures, it is possible to derive a complete system dynamics as a second-order transfer function with delay of the form:

$$\frac{p_i}{p_{i,ref}} = \frac{K_G \omega_n^2}{s^2 + 2\zeta \omega_n + \omega_n^2} e^{-\tau s} \quad (2)$$

where p_i and $p_{i,ref}$ respectively refer to the actual and reference hydraulic brake line pressures for caliper i . K_G , ζ , ω_n and τ are system characteristic parameters obtained experimentally.

Since judder occurs during the application of low brake pressure, cases with pressures of up to 25 bars are considered. The parameterized second-order transfer function produces step responses that match well with experimental results depicted in Figure 1.

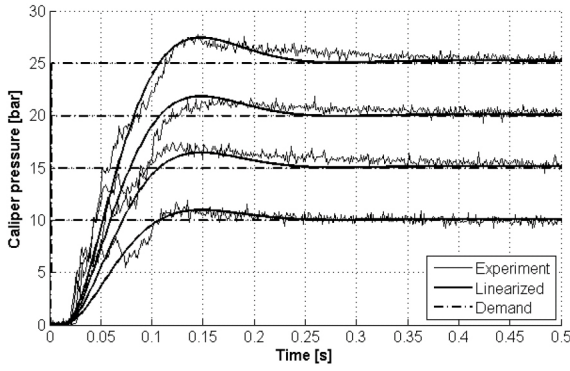


Figure 1. Validation of the linearized EHB model using low pressure references.

Note that (2) can be expressed as the following state-space equation, with clear synergies to (1):

$$\begin{bmatrix} \dot{p}_i \\ \ddot{p}_i \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ -\omega_n^2 & -2\zeta\omega_n \end{bmatrix} \begin{bmatrix} p_i \\ \dot{p}_i \end{bmatrix} + \begin{bmatrix} 0 \\ 1 \end{bmatrix} p_{i,ref} \quad (3)$$

The normal force, F_{ni} is produced by the caliper pressure between the brake pad and disk rotor on each wheel, and can be written as:

$$F_{ni} = \begin{cases} (p_i - p_0) \eta_c \frac{\pi d_{cyl}^2}{4}, & p_i > p_0 \\ 0, & p_i \leq p_0 \end{cases} \quad (4)$$

where d_{cyl} , p_0 and η_c are the diameter of brake piston, pushout pressure and wheel cylinder efficiency respectively.

OVERVIEW OF BRAKE TORQUE VARIATION COMPENSATOR DESIGN

The brake torque, T_b is generated by applying a clamp force on both surfaces of a disk rotor via brake pads. For the EMB, the brake torque can be calculated as

$$T_{b,emb} = 2\hat{\mu}r_d F_{cl} + T_d + T_{btv}(\theta_d) \quad (5)$$

where $\hat{\mu}$, r_d , T_{btv} , T_d and θ_d are the estimated coefficient of friction between brake pads and disk rotor, effective rotor radius, BTv, discrepancy in brake torque resulted from friction coefficient estimation error and disk rotational position respectively. Similarly, the brake torque generated by the EHB can be calculated as

$$T_{b,ehb} = 2\mu r_d F_{ni} + T_{btv}(\theta_d) \quad (6)$$

A brake torque profile with variations due to disk thickness variation is shown in Figure 2. It can be seen that the torque varies periodically with respect to the disk rotation, where the first harmonic has the

largest amplitude, followed by smaller amplitudes of the second and third harmonics. Taking advantage of the periodicity of the signal, a disturbance model can be constructed [10, 12]:

$$T_{btv}(\theta_d) = \sum_{k=1}^n a_k \cos(k\theta_d + \phi_k), \quad (7)$$

where a_k and ϕ_k represent the amplitude and phase-shift of the k -order harmonic respectively. The amplitude and phase-shift are unknowns, but are assumed to be time-invariant or slowly varying in the proceeding compensator design.

In the following, the structure of the adaptive BTv compensator design first proposed in [10] is briefly presented. Note that detailed derivation and analysis of the proposed compensator can be found in [12].

For the EMB, the adaptive compensator carries the form of:

$$\begin{aligned} \dot{\hat{\gamma}} &= g\varpi(\theta_d)e_b \\ F_{cl,ref} &= \Gamma_{emb}(\omega_d, \dot{\omega}_d)\Phi(\theta_d)\hat{\gamma} + \frac{1}{2\hat{\mu}r_d}T_{b,ref} \end{aligned} \quad (8)$$

Note g is the compensator gain, ω_i is the disk rotational velocity and e_b is the brake torque tracking error, given by:

$$e_b = T_b - T_{b,ref} \quad (9)$$

where $T_{b,ref}$ is the brake torque reference. Additionally, the parameterized compensator gains are given by:

$$\varpi(\theta_d) = [1 \quad \cos(\theta_d) \quad \sin(\theta_d) \quad \cdots \quad \cos(n\theta_d) \quad \sin(n\theta_d)]^T \quad (10)$$

$$\Gamma_{emb}(\omega_d, \dot{\omega}_d) = \frac{1}{2\hat{\mu}r_d k_p} \begin{bmatrix} -k_p & \omega_d^2 - k_p & -(\dot{\omega}_d + k_d \omega_d) \\ \cdots & (n\omega_d)^2 - k_p & -n(\dot{\omega}_d + k_d \omega_d) \end{bmatrix} \quad (11)$$

$$\Phi(\theta_d) = \text{diag}(1, \Phi_1(\theta_d), \cdots, \Phi_n(\theta_d)) \quad (12)$$

The components in the diagonal matrix, $\Phi(\theta_d)$ are given by:

$$\Phi_k(\theta_d) = \begin{bmatrix} \cos(k\theta_d) & \sin(k\theta_d) \\ -\sin(k\theta_d) & \cos(k\theta_d) \end{bmatrix}, \quad k = 1, \cdots, n \quad (13)$$