

Adaptive Brake Torque Variation Compensation for an Electromechanical Brake

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ABSTRACT

A novel method for attenuation of brake judder directly at the source is proposed, utilizing an electromechanical brake to actively compensate for the variation in brake torque that causes judder. Taking advantage of the high-bandwidth closed-loop clamp force tracking performance offered by an electromechanical brake, an adaptive compensator is designed to estimate the brake torque variation (BTV), and to produce a compensating clamp force command to cancel it. The compensator is tested over fixed and varying BTV frequencies by employing a production-ready prototype EMB. It is demonstrated that significant BTV attenuation is obtained using the proposed approach.

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INTRODUCTION

Brake judder is a braking-induced vibration which is perceived by the driver as a vibration at the steering wheel, brake pedal or vehicle body during light braking, typically during mid to high speed traveling (30 to 140 km/h). It is one of the most commonly occurring faults in braking systems and has resulted in major recalls, warranty claims and detrimental effects to the company reputation. Furthermore, the associated vibrations may also provoke additional wear and prompt components fracture, hence resulting in shorter service life and higher operating costs.

Brake judder is caused by the irregularities in the brake disk, and can be categorized into cold and hot (or thermal) judder [1]. Cold judder is induced by geometrical irregularities due to mounting, machining, uneven wear, uneven corrosion and uneven friction film generation [2]. Geometrically, these irregularities resemble disk runout or disk thickness variation (DTV). Disk runout may induce uneven wear and further generates DTV [3], which contributes to the brake pressure variation (BPV) and brake torque variation (BTV). For most manufacturers, the manufacturing tolerances for disk runout and DTV are less than 80 μm and 10 μm respectively [4]. However, a slight increase in DTV to 15 μm can cause brake judder in certain vehicles [2], therefore raising the need for better manufacturing tolerances. Other non-thermal related sources of judder include uneven surface film transfer.

Hot judder is induced by thermal deformation (e.g. coning and waving), uneven thermal expansion or phase

transformation of the disk material [2]. Unlike cold judder, hot judder does not require large amplitude permanent disk irregularities for its initialization. When exposed to braking pressure, permanent irregularities (both with low and high amplitudes) cause a localized rise in the contact pressure field and temperature, leading to hot spot generation [5,6,7]. The hot spots result in instantaneous DTV and spatial friction variation, causing brake pressure and brake torque pulsations during braking. Additionally, large thermal gradients may lead to disk cracking, warping and buckling [6,8], and further deteriorate the disk condition. Compared to cold judder, thermally excited judder is of higher order (with respect to the wheel speed), typically between six to twenty times the wheel rotational speed [9].

Both hot and cold judder lead to BPV and BTV, which are the primary excitation mechanisms for judder vibrations. The generated BPV are further transmitted to pedal via hydraulic systems, resulting in pedal vibration. Furthermore, the vibration excited by the BTV is transmitted to the vehicle chassis, vehicle body and the steering system through the wheel suspension system, causing noise and vibrations in the vehicle compartment and steering wheel. The vibration is also amplified when its frequency is within the resonance of the suspension and steering systems [10,11]. In particular, the front wheel suspensions of most passenger cars have fore-aft vibration mode resonance in the range of 10 to 20 Hz [2]. This frequency range corresponds to a maximal amplitude of vibrations at a critical vehicle speed of between 60 to 140

km/h for the first-order judder, and 30 to 70 km/h for the second-order judder.

Gassmann and Engel [12] investigated the excitation and transfer mechanism of brake judder experimentally in street-going vehicles, and determined the interactions between excitation and transfer mechanism. Based on this result, a holistic approach to the attenuation of brake judder was suggested in [13] by considering the total system, including the brakes, the suspension and the steering system. Brake judder attenuation strategies can be categorized into minimization of the judder excitation and modification of the transfer path as summarized in Table 1.

Table 1. Existing and proposed strategies for brake judder attenuation.

	Passive Approaches	Active Approaches
Steering	• Shift resonance frequency during design	• Judder compensation using active steering
Suspension	• Shift resonance frequency during design	• Shift resonance frequency using active steering
Brake actuator	• Reduced residual drag • Increased weight • Softer pads	• Proposed Approach: <i>Actively compensate for brake torque variation using EMB</i>
Brake disk	• Improved manufacturing tolerance • Resurface disk • Improved heat transfer	

Minimization of the judder excitation can be achieved through BPV and BTV reduction, attained by reducing disk runout, DTV and thermal effects. Initial disk runout and DTV can be improved by adopting better assembly and machining tolerances, albeit resulting in higher manufacturing costs. Subsequent disk resurfacing can be performed if the DTV level exceeds the disk specifications due to wear. However, this leads to higher operating costs. Furthermore, in order to minimize DTV generation, residual drag can be reduced by increasing clearance, optimizing sliding pin and piston seal rollback characteristics of the caliper [14,15], which may adversely affect pedal feel characteristics. Jacobsson [16] analytically showed that a heavier caliper reduced judder, but this creates conflicting design requirements for improved assembly weight and fuel economy. Other methods include: reducing thermal deformation of the disk through improved air flow and heat transfer characteristics, reducing uneven disk wear and reducing non-uniform deposition of transfer film on the disk [14,17]. Decreasing pad stiffness was also suggested in [18,19,20], although leading to the increase of pedal travel, deteriorating ABS performance and decreasing pad wear life. Lower coefficient of friction was suggested also by [21], albeit compromising braking effectiveness.

The judder transfer path can be optimized by shifting the resonance frequencies of suspension and steering systems outside of the first wheel order excitation [22]. In particular, Zhang et al. [23] reported that the suspension bush bearings characteristics greatly affect the resonance amplitude. Judder attenuation performance can be improved by increasing the

rubber bush bearing stiffness [20] and adopting a hydraulic bush bearing with higher damping [24]. Furthermore, Gruber et al. [25] suggested using an active wheel suspension to shift the resonance frequency online to dampen judder, while von Groll et al. [26] proposed using an active steering system to compensate for judder.

The current approaches suggested above are either of corrective measures (applied after the driver has experienced judder), or may affect vehicle handling performance, or are locally isolating the vibration (as the source of vibration is still exists). These approaches have been the only possible solutions under hydraulic braking system due to the low bandwidth of the closed-loop system.

On the other hand, brake-by-wire (BBW) offers high closed-loop bandwidth and fine clamp force control fidelity which provides the opportunity to eradicate judder at the source. Due to the mechanically decoupled construction of the brake pedal in BBW system using the pedal feel emulator, pedal pulsation during judder is eliminated. However, the resultant BTV that causes judder is not dealt with. The design flexibility offered by BBW through software upgrades invites investigation into value added functions, such as a brake judder attenuation.

This work proposes a novel method to actively attenuate the brake torque variation using an electromechanical brake (EMB), taking advantages of the offered high closed-loop bandwidth, and therefore reducing the judder phenomenon. This would reduce the need for a better manufacturing tolerance and extend components life.

The organization of the paper is as follows. The next section describes a linearized closed-loop EMB model that serves as the basis for the proceeding BTV compensator design. A model of the BTV is first presented, followed by the derivation of the compensator, whereby a flexible compensator structure for attenuation of multiple harmonics is proposed. Then, experimental results on a brake dynamometer will be presented and discussed. The conclusion encompasses the main results and establishes further work required for real world implementation.

LINEARIZED CLOSED-LOOP EMB MODEL

A production-ready EMB prototype shown in Figure 1 is chosen in this work to demonstrate the proposed brake torque variation compensator. The EMB consists of a three phase, permanent magnet synchronous motor, a gear reduction stage, a ball screw, a floating caliper and control electronics. An experimentally validated simulation model for the EMB is presented in [27,28]. Furthermore, the flexible clamp force controller design proposed in [28] is utilized to design the closed-loop controller, whereby the available motor capacity generates a high-bandwidth clamp force response.

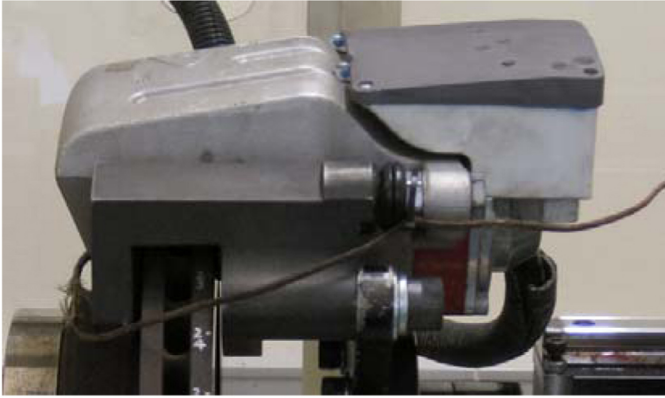


Figure 1. The production-ready prototype electromechanical brake used in this work.

The EMB takes in a clamp force command in real-time and outputs an actual clamp force. Since the brake judder occurs during light braking, it is assumed that the compensator operates in vicinity of a small clamp force region. By linearizing the closed-loop nonlinear EMB and controller models with respect to a small clamp force, the input-output relationship of the EMB is governed by the following dynamics:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ k_p & k_d \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} 0 \\ -\frac{k_p}{k} \end{bmatrix} F_{cl,r}$$

$$F_{cl} = \bar{k}x_1 \quad (1)$$

where F_{cl} , $F_{cl,r}$, k_p , k_d , $\bar{k}$, x_1 and x_2 represent brake clamp force, clamp force reference, proportional gain, derivative gain, linearized lumped stiffness coefficient, motor position and velocity respectively. By linearizing the stiffness model in the vicinity of the commanded motor position, x_{1r} , the linearized lumped stiffness coefficient, $\bar{k}$, is given by

$$\bar{k} = k_1x_{1r} + k_2x_{1r}^2 \quad (2)$$

The proportional and derivative gains can be tuned according to design specifications for the clamp force tracking performance, such as rise-time and maximum overshoot. In [28], a near-time-optimal control (NTOC) structure and tuning procedure are introduced whereby the gains are chosen to achieve high closed-loop bandwidth performance. Specifically, the proportional and derivative gains are given by

$$k_p = -\frac{bk_s}{s_{sat}}, \quad k_d = -\frac{b}{x_{2sat}} + \frac{bk_s}{s_{sat}} \xi(0), \quad b = \frac{T_{mmax}}{J} \quad (3)$$

where T_{mmax} and J are the maximum motor torque and effective moment of inertia about the EMB motor rotational

axis respectively. Additionally, k_s , s_{sat} , x_{2sat} and $\xi(0)$ are the auxiliary tuning parameters that set the region in x_1 - x_2 phase plane where the clamp force controller output is non-saturated. The reader is referred to [28] for further details of the high-bandwidth clamp force controller.

The brake torque, T generated by the clamp force is then given by

$$T = 2\mu r_d \bar{k}x_1 + d(t) \quad (4)$$

where $d(t)$, r_d and μ are BTV, effective brake disk radius and coefficient of friction between brake pad and brake disk respectively. The numerical values used in (1),(2),(3),(4) are provided in Table 2. As indicated in (4), brake torque is modeled as a static function of brake force. This implies that the mounting of the caliper is sufficiently rigid such that the movement of the caliper is fast compared to the brake torque generation. Due to this assumption, in addition to the limitation of EMB clamp force bandwidth, we only consider compensation for low order judder, which is a commonly occurring type of judder.

Furthermore, it is noted that BTV is often periodic with respect to the brake disk rotation, i.e. $d(t)=d(x_d)$, where x_d is the disk rotation. Additionally, the mean is assumed to be small using a floating type caliper. In the subsequent section, we consider a BTV model using a sum of sinusoidal signals of zero mean.

Table 2. Linearized EMB Model Parameters

J	Moment of inertia	$2.91 \times 10^{-4} \text{ kg.m}^2$
k_1	Stiffness coefficient	$2.15 \times 10 \text{ N/rad}^2$
k_2	Stiffness coefficient	$-1.32 \times 10^{-1} \text{ N/rad}^3$
k_d	Derivative gain	-232.3
k_p	Proportional gain	-5161.7
k_s	NTOC gain	2
r_d	Brake disk radius	0.127 m
x_{2sat}	NTOC gain	50
s_{sat}	NTOC gain	4
T_{mmax}	Motor torque limit	3 Nm
μ	Coefficient of friction	0.484
$\xi(0)$	NTOC gain	-0.005

TORQUE VARIATION COMPENSATOR

Disturbance Model

As eluded in the introduction, it is evident that since BTV is caused by the geometry irregularities of the brake disk, it must be periodic to the disk rotation, x_d , and may be approximated using an n -order Fourier series expansion