

Active Brake Pedal Feedback Simulator Based on Electric Drive

Michael Flad, Simon Rothfuss, Gunter Diehm, and Sören Hohmann
KIT, Karlsruhe Institute of Technology

ABSTRACT

Brake pedal feedback is important for driver's perception during the driving task as well as the pedal feel is an important factor in customer satisfaction. Therefore, a force emulation device is beneficial during the design phase to evaluate the pedal characteristic. Such a system is also needed for driving simulators. Usually, brake feedback systems in simulators rely on passive elements like springs and dampers to emulate the force. This does not allow the implementation of an arbitrary nonlinear pedal force characteristic. In this paper we propose an active pedal feedback simulator which can emulate an arbitrarily customizable and online adjustable brake pedal characteristic. The particular advantage of our pedal simulator is that the system can also emulate the exact pedal dynamics. This is advantageous compared to other active brake feedback simulators which rely on hydraulic actors. We show the mechatronic, electronic and software concept along with the required control framework and state observer. For verification the pedal simulator is parameterized with a dynamic pedal force model of a conventional middle sized car. We show that the experimental results prove accurate conformance of the measured pedal characteristics with respect to the given reference.

CITATION: Flad, M., Rothfuss, S., Diehm, G., and Hohmann, S., "Active Brake Pedal Feedback Simulator Based on Electric Drive," *SAE Int. J. Passeng. Cars – Electron. Electr. Syst.* 7(1):2014, doi:10.4271/2014-01-0325.

INTRODUCTION

The design of the brake pedal feel is an important task during the design process of new vehicles, because it is a crucial factor towards a high satisfaction of vehicle customers [1, 2]. Aside of the marketing aspects, the pedal feedback is important for the driver's perception during the driving task. Basically, there are two different perceptions while braking. The first is the deceleration of the vehicle and the other one is the feedback force of the brake pedal. In this paper we examine the brake pedal feedback force, as one important design factor for optimal pedal feel.

During the design phase of a vehicle engineers have to investigate the pedal feedback. This is especially the case for the design of vehicles with hybrid or full electrical power trains which using "Brake Blending" [3]. In such vehicle concepts the brake pedal controls also the recuperation system which does not provide an adequate feedback to the driver by itself. An additional haptic feedback which indicates the operating modes (energy regeneration or regeneration combined with the conventional friction brake system) is beneficial for economic driving. Technically this development could lead to brake systems in which the mechanical brake system no longer primarily determines the brake pedal characteristic. Regardless if a conventional or future brake pedal feel is optimized, the design of human machine interface (HMI) components requires extensive testing. Therefore, a customizable feedback simulator, which allows studies long before first prototypes

exist, is beneficial. Apart from the design task for real vehicles it is important to provide an adequate pedal feedback for advanced driving simulators [4].

State of the Art

In the HMI context brake pedal feel is primarily modeled using a static characteristic curve which describes the pedal force over the pedal travel. The curve is strongly nonlinear and there exists a hysteresis which depends on the sign of the pedal velocity. Dynamic characteristics are mainly described using an equivalent network consisting of nonlinear dampers and springs (Fig. 1). For an overview see [5, 6, 7, 8].

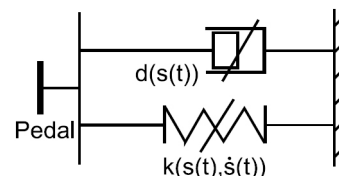


Figure 1. Equivalent network for dynamic pedal characteristic; $s(t)$ is the pedal travel

A typical brake pedal characteristic of a passenger car equipped with a conventional brake system is given for example in Fig. 2. For the example the mechanical limited maximum of the pedal travel is 100 mm. For an introduction of the mechanical brake system, which yields this characteristic, be referred to [5, 9, 10].

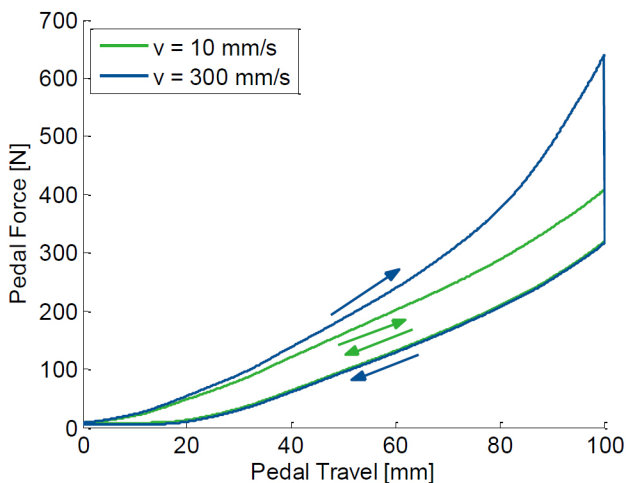


Figure 2. Typical brake pedal characteristic of a passenger car with conventional brake system for two actuation velocities

Apart from some larger systems (e.g. [11]) which use the original brake system itself, in driving simulators and also for video games the brake feedback is in general generated by using passive mechanical elements like springs and dampers. A disadvantage of this concept is the fixed brake pedal characteristic. It cannot be changed online which raises the required effort for the system design when a passive system is used. Also the implementation of a nonlinear pedal characteristic as depicted in Fig. 2 requires significant mechanical effort or respectively is not possible. In [6, 12] an active pedal force simulator is presented which is based on a hydraulic actuator. Drawback of this concept is the limitation of the dynamic potential by the hydraulic system. [13] proposes a system which can be attached to the brake pedal of a vehicle and allows the adaption of the vehicle's brake pedal characteristic. But the system allows only a very limited adaption of the static pedal characteristic. Similar work has been done on the field of active acceleration pedals (e.g. see [14, 15, 16]). The active acceleration pedals are tested in an industrial context at least since 1998. These systems allow the realization of an acceleration pedal characteristic. Naturally the maximum achievable pedal force of these systems is not sufficient for emulating a realistic brake pedal characteristic. Also these systems are only capable of emulating a quasi-static pedal characteristic.

In contrast to the concepts discussed above we propose an active pedal feedback simulator suited to emulate the full pedal feedback. The simulator uses an electrical drive and can emulate an arbitrary customizable and online adjustable brake pedal characteristic. The particular advantage is that the system can also emulate the exact desired pedal dynamic within a wide dynamic and force range. Furthermore, if an adequate control algorithm is used the systems stiffness is much higher than in an air based system. In addition the pedal characteristic of the proposed system can be freely programmed. To the best knowledge of the authors such a system was not proposed before.

This paper is organized as follows: First the mechanical and electronic concept of the brake pedal force simulator is introduced. Then a mathematical model of the pedal system is given. To determine the non-measurable system states, which are required in the control loop, a signal processing unit by means of an Extended-Kalman-Filter is presented. Based on this, a control framework is introduced. At the end we demonstrate the accuracy of the pedal simulator in an experiment.

DESCRIPTION OF THE SYSTEM

System Overview

The proposed brake pedal simulator (see Fig. 3, 4) is part of a larger driving simulator. The brake pedal can be divided into the mechanical and the electrical system, which are connected via the electric drive and the electromechanical sensors.

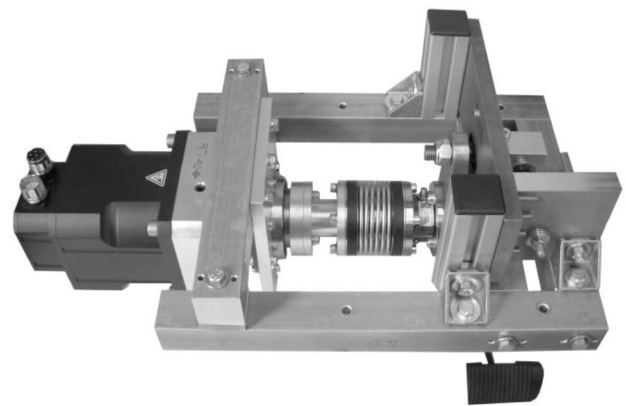


Figure 3. Top view of the mechanical components of the pedal force simulator



Figure 4. Side view with pedal's limit stop



Mechanics

The basic mechanical concept of the pedal force simulator is depicted in Fig. 5.

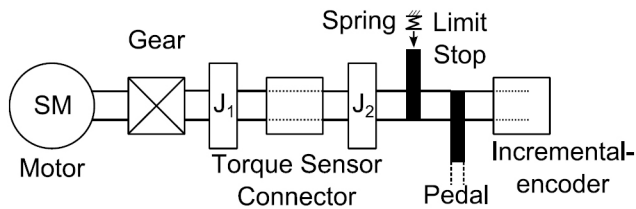


Figure 5. Basic mechanical concept of the pedal force simulator

The system consists of a synchronous motor (SM), an epicyclic gear, a torque sensor, a clutch coupling to compensate potential mechanical imprecision and to protect the torque sensor, an incremental encoder and the brake pedal of a conventional middle-size car. Furthermore, ball-bearings are used to support the shaft on which the components are attached. The pedal slewing range is limited to 18° by mechanical limit stops to protect surrounding equipment and to guarantee user's safety (Fig. 4). The range can be adapted by changing the stops. To extend the available pedal force range an additional helical compression spring can be used to support the motor. It can be configured that the additional force can set in at any point of pedal angle. The spring acts against the crank, which is also used for the pedal stops. The detailed characteristic of each component is given in the appendix.

Electronic Components

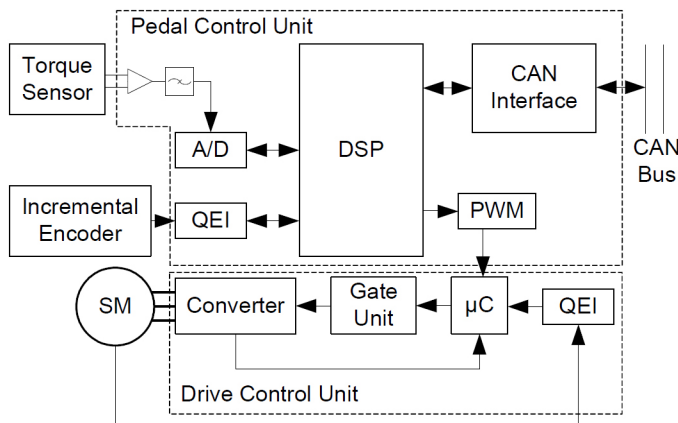


Figure 6. Block diagram of the electronic subsystem of the pedal force simulator

The electronic system (see Fig. 6) consists of two subsystems, the drive control unit and the pedal control unit. Main component of the pedal control unit is a digital signal processor (DSP) which operates at a clock rate of 150 MHz. The microcontroller manages the communication with the superior system via a controller area network (CAN) interface, controls the converter unit of the electric drive, processes the sensor data from the incremental encoder and the torque sensor and executes the control algorithm. The CAN-Bus baud rate is 1 MBaud. Note this bus is a private bus, not one of the

conventional busses of commercial vehicles. The analog output signal of the torque sensor is, after amplifying and filtering, discretized with 24 Bit. The used analog filter is a Butterworth fourth order filter. The used DSP includes a quadrature incremental encoder interface (QEI) to encode the encoder signal. The state of the art drive control unit for the synchronous motor is connected via a pulse-width modulation (PWM) command to the controller. It performs field-orientated vector control of the motor phase currents. This allows commanding the motor torque directly by setting the torque generating motor current. The drive control unit includes the converter, the gate drive electronic, a QEI to measure the position of the drive and a DSP to control the unit. The detailed characteristics of the relevant components are given in the appendix.

Software Architecture

Fig. 7 depicts the software architecture running on the DSP. The sensor input and filtering subsystems handle the calibration and digital filtering of the sensor information. This data combined with the data from the CAN-Bus is then processed in the system state estimation subsystem, where none measured quantities are calculated. The reference calculation module calculates the desired pedal force as set-point for the control algorithm. Alternatively, the reference pedal force can be set by a superior system and commanded over the CAN-Bus. Based on the set-point and the system states the controller calculates the correcting variable for the drive control unit. The failure detection and system supervision modules guarantee users safety by monitoring the system states and mastering the output modules towards the drive. The module detects failures of the mechanical limit stops, the drive and also infeasible sensor data. Dependent on the safety level classification the implementation of the system in a series vehicle may require a watchdog system which works independent to the main control unit and supervises the drive power.

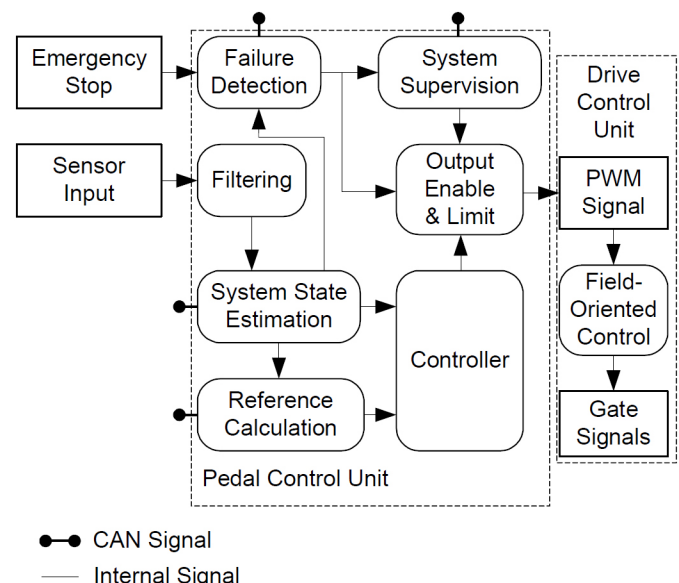


Figure 7. Block diagram of the feedback simulation system

SYSTEM MODELLING

In this section the dynamic model of the brake pedal system is given which is required for the system state estimation subsystem (see Fig. 7.) and for the control design.

Electric System

Due to the drive control unit (Fig. 6) of the synchronous motor with its field-oriented motor control, the electronic and electric system can be approximated by a dead time and a first order lag (PT1) element. The dynamic of the electronic/electric system is neglected as the resulting time constant of the electronic components are more than a factor of 100 smaller compared to the time constants of the following mechanical system.

Mechanical System

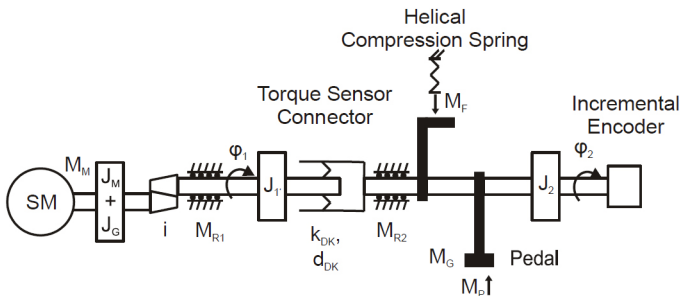


Figure 8. Mechanical model of the pedal force simulator

The mechanical system primarily consists of two shafts connected by the torque sensor and the clutch connector. The combination of the torque sensor and the connector can be modeled as a damped spring-mass system. The inertia of the drive and the gear as well as the moment of the drive is transformed to the output side of the gear box. All friction forces of the bearings in the motor, gear, incremental encoder and the pedal system itself are summarized into two friction terms, each for one side of the pedal. The gear is selected with special intention to minimize the backlash. For the selected gear the backlash is smaller than 3 minutes of angle and hence can be neglected in the model. The detailed mechanical model is depicted in Fig. 8. The quantities used in the figure are:

- shaft angle φ_1 of the left side and the shaft angle of the right side φ_2
- concentrated inertia of the gear J_G and drive J_M
- inertia of left pedal system side J_1 (included J_G and J_M transformed to the output side of the gear)
- inertia of right pedal system side J_2
- gear ratio i
- friction force on the left side M_{R1} friction force of the bearing on the right side M_{R2}
- spring constant of the combination of torque sensor and connector k_{DK}
- damping rate of the combination of torque sensor and connector d_{DK}

- torque applied by the synchronous motor M_M
- torque resulting from the applied pedal force of driver M_p
- torque induced from the additional spring M_F
- torque induced by the pedal mass M_G .

Gravity Induced Torque

As the pedal mass is not equal distributed around the rotation axis (see Fig. 4) gravity induces a torque on the pedal. By reducing the pedal to a mass point the torque M_G can be approximated by

$$M_G(\varphi_2) = m_p \cdot g \cdot \frac{l}{2} \cdot \sin(\alpha - \varphi_2) \quad (1)$$

with the pedal mass m_p , the pedal length l , and the pedal angle α of the equilibrium position of the pedal.

Additional Spring Attachment

A straight helical compression spring is used to implement the additional passive force. The spring force F_F is not always exactly aligned to the crank of the pedal system. The torque M_F is given by the equation

$$M_F(\varphi_2) = \begin{cases} k \cdot a \cdot \sin(\varphi_2 - \beta) \cdot \cos(\varphi_2 - \beta), & \varphi_2 > \beta \\ 0, & \varphi_2 \leq \beta \end{cases} \quad (2)$$

With the spring rate k , the radius a of the crank and the set-in-point β of the spring. The resulting spring system is depicted in Fig. 9.

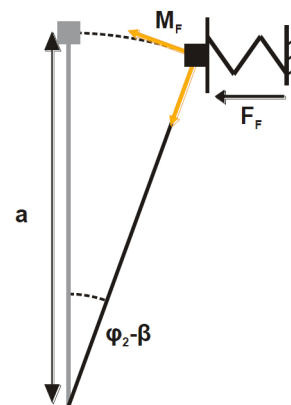


Figure 9. Model of the spring system

Friction Model

To model the friction on both sides a common friction model described in [17, 18, 19] is used. This model suffices to represent the main frictional aspects. The friction is modeled as a constant offset and a term which depends linear on the velocity, see Fig. 10.