

# Design and Testing of ABS for Electric Vehicles with Individually Controlled On-Board Motor Drives

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## ABSTRACT

The paper introduces the results of the development of anti-lock brake system (ABS) for full electric vehicle with individually controlled near-wheel motors. The braking functions in the target vehicle are realized with electro-hydraulic decoupled friction brake system and electric motors operating in a braking mode.

The proposed ABS controller is based on the direct slip and velocity control and includes several main blocks for computing of predictive (feedforward) and reactive (feedback) brake torque, wheel slip observer, slip target adaptation, and the algorithm of brake blending between friction brakes and electric motors.

The functionality of developed ABS has been investigated on the HIL test rig for straight-line braking manoeuvres on different surfaces with variation of initial velocity. The obtained experimental results have been compared with the operation of baseline algorithm of a hydraulic ABS and have demonstrated a marked effect in braking performance.

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## INTRODUCTION

Full electric vehicles with individual drive motors for each wheel belong to the most promising technologies of next-day mobility. Such powertrain architecture not only provides energy-efficient driving but also opens new embodiment variants of traditional chassis systems contributing to enhancement of active safety and driver comfort. This is particularly true for ABS, which becomes an opportunity of wheel torque modulation both through friction brakes and electric motors. In that case the ABS operation executes active safety functions and simultaneously improves the electric driving efficiency due to regenerative braking.

An analysis of state-of-the-art in ABS with electric motor actuation points to several practical solutions is worth mentioning. For instance, the works [1, 2] have introduced results of the ABS operation realized through individual in-wheel motors on full electric vehicle of executive class (vehicle mass 2100 kg; maximum vehicle velocity 200 km/h; maximum wheel power 40 kW; maximum wheel torque 550

Nm). The tests have confirmed an improved ABS performance with a higher actuation frequency in the case of blended actuation of the friction brakes and electric motors as compared with commonly-used hydraulic ABS, where modulation is usually around 9 Hz. In particular, it was mentioned that ABS function realized through electric motor actuation can reduce the braking distance up to 7 % on ice road as compared with a conventional hydraulic ABS.

Another industrial implementation of ABS with electric motors on a full-scale car is given in [3]. This variant concerns the roadster electric vehicle (mass 1250 kg) equipped with in-wheel electric motors of a maximum continuous torque 500 Nm and continuous power 73 kW. In this configuration, rear electric motors in braking mode can guarantee minimum vehicle deceleration 0.3 g on the road surfaces with the friction coefficient  $\geq 0.3$ . In the case of higher deceleration demands, the front friction brakes are activated.

Both mentioned vehicle configurations relate to the powertrain architecture with in-wheel motors. It should be pointed out that in-wheel motors have still high costs despite continuous improvement of relevant design and production technologies. However, modern techniques propose several other variants of the architecture. Of particular interest is electric vehicle configuration with near-wheel (or on-board) electric motors connected to wheels through gearbox and half-shaft, [Figure 1](#). In such case the torsional dynamics of the half-shaft can bring potential limitation in the actuation of ABS that was shown and discussed in [4, 5]. To investigate this problem in more details, the main goal of the presented study has been formulated as the development and validation of the control strategy for anti-lock brake system to be implemented on an electric vehicle with individually controlled on-board motors connected to wheels through powertrain elements with distinct torsional dynamics.

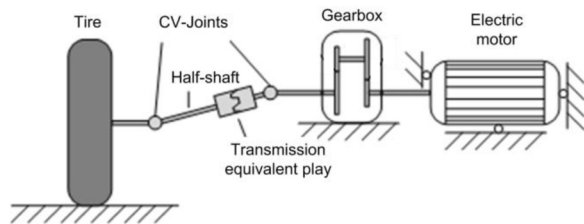


Figure 1. The powertrain architecture with on-board motor

The choice of a basic control method for the ABS has been done from analysis of different solutions presented in recent research publications. The majority of the relevant ABS investigations consider a simultaneous operation of both friction and motor variants due to strict vehicle safety requirements. The ABS architecture thus has a blended composition in which the continuous of friction braking and negative motor torques depend on operational conditions and the braking dynamics. For example, the work [6] introduces a variant of the blending using a rule-based controller. The actuation of electric motors and hydraulic brake system in this ABS is governed by a set of rules that operate with slip thresholds and four levels of specific vehicle deceleration. The study [7] has proposed to use Sliding Mode Control for the blended ABS operation. Another approach is presented in [8] in the form of a multi-layered structure in which the high-level controller realizes the blended ABS logic, and the low-level fuzzy controller determines the hydraulic actuation of the system. The high-level controller comprises feedforward and feedback compensation elements.

The analysis of the mentioned and other publications allowed to point out that many developed methods of electric motor control in ABS mode have been validated only with computer

simulation for road and operational conditions. Full-scale experiments on real vehicles are rarely presented. Most of founded research publications consider the vehicles with in-wheel motors, and there is a lack of deep studies analyzing advantages, performance and functional limitations of anti-lock braking systems with regard to electric powertrain architecture consisting of near-wheel motors, gearboxes and half-shafts.

Taking into account that the presented study is aimed at the implementation of the ABS on a real sport utility vehicle and should be subjected to experimental validation, it was decided to propose robust ABS realization using proportional-integral direct slip control with feedforward and feedback slip control parts. Next sections of the paper introduce the logic of the discussed ABS in the application to specific electric vehicle architecture and particular results of real-time validation of the ABS algorithm on the hardware-in-the-loop (HIL) test rig.

## ABS CONTROLLER

### General Structure

The proposed architecture of anti-lock brake system can be explained with [Fig. 2](#) (as applied to single wheel control).

The driver control action  $F_{driver}$  is the input of *base brake controller* generating the torque demand  $T_{dem}$  for each wheel, which is added up with the reactive torque  $T_{react}$ . The reactive torque is computed by the *reactive torque controller*, which uses the reference slip ratio  $\lambda_{ref}$ , the observed slip ratio  $\lambda$  and the vehicle velocity  $V_x$  as input parameters. The reference slip ratio is produced by the *reference slip generator*, and the parameters  $\lambda$  and  $V_x$  are computed with *slip and velocity observers* correspondingly. The sum of torques  $T_{dem}$  and  $T_{react}$  is further processed in the block defining *torque blending*. This block splits the total wheel torque demand into the torque demand for the electric motors  $T_{em\_dem}$  and for the friction brakes  $T_{br\_dem}$ . Both torques are being realized by corresponding actuators and the actuator outputs  $T_{em}$  and  $T_{br}$  are further used either in the *vehicle model* (in the case of software- and hardware-in-the-loop simulations) or the *real vehicle* (in the case of eventual experiments on the real car). The actual wheel velocity  $V_w$  and longitudinal vehicle acceleration  $a_x$  serve also as the input for the slip and velocity observers.

The main blocks of the ABS are further introduced in more details (*realization of friction, slip and velocity observers is not discussed within the framework of this paper*).

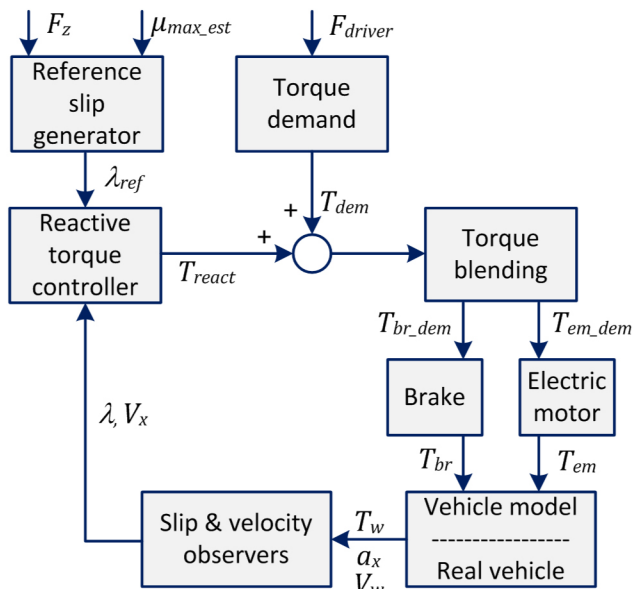


Figure 2. General structure of ABS controller

### Torque Demand

The generation of the torque demand  $T_{dem}$  is calculated from the driver control action, in particular from the brake pedal travel. In the presented study, the practical ABS realization includes a decoupled electro-hydraulic brake system with the embedded pedal unit that is able to provide the corresponding signal of the pedal position [5]. Preliminary the value of  $T_{dem\_prim}$  is processed as function of the brake pedal travel through a lookup table (LUT). Then, this value is corrected through the predictive torque in accordance with the road conditions to prevent vehicle underbraking / overbraking.

The predictive torque is defined as:

$$T_{pred} = \mu_{max} F_z r + k_{pred} (\mu_{max} F_z r), \quad (1)$$

where  $\mu_{max\_est}$  is the estimated value of maximum friction coefficient for given road conditions,  $F_z$  is the estimated normal wheel load,  $r$  is the tire rolling radius,  $k_{pred}$  is the correction coefficient. It is determined for the target vehicle that  $k_{pred}=0.25$  for front wheels and  $k_{pred}=0.5$  for rear wheels.

After the processing of the predictive torque, the torque requested by the driver is saturated as:

$$T_{dem} = \begin{cases} \min(T_{dem\_prim}, T_{pred}) & \text{if } T_{dem\_prim} \geq 0 \\ \max(T_{dem\_prim}, T_{pred}) & \text{if } T_{dem\_prim} < 0 \end{cases} \quad (2)$$

### Generation of Reference Slip

The reference slip generator is designed to determine the reference slip  $\lambda_{ref}$  by means of a tyre model. On the first step the reference slip is defined as

$$\lambda_{ref} = f(F_z, \mu_{max}). \quad (3)$$

The reference slip generator is realized in the form of a lookup table. A graphical interpretation of LUT for the target vehicle configuration is given in Fig. 3.

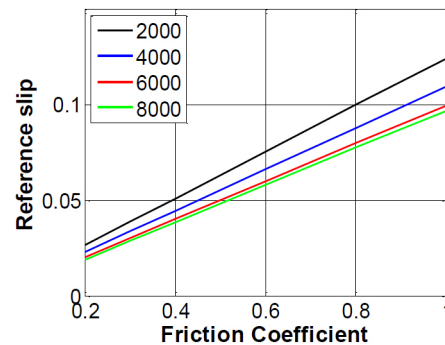


Figure 3. Generation of reference slip

On the final stage a slip target adaptation (STA) algorithm is applied to modify progressively the reference value obtained from the LUT, in order to be as close as possible to the maximum braking torque and to avoid under or overbraking caused either by inaccurate estimations of  $\mu_{max}$  or by tire-road behavior different to the one modeled by the LUT. The adaptation is realized through the correction of the reference slip in accordance with the estimation of the sign of the actual slope of the  $\mu_x$ - $\lambda$  curve. As the road friction coefficient  $\mu_x$  is directly related to the longitudinal acceleration  $a_x$ , the estimation of the road  $\mu_x$ - $\lambda$  slope can be done as follows:

$$\text{sign} \left[ \frac{d\mu}{d\lambda} \right] = \text{sign} \left[ \frac{da_x}{d\lambda} \right] = \text{sign} \left[ \frac{da_x}{dt} \frac{dt}{d\lambda} \right] \quad (5)$$

Fig. 4 shows the adaptation procedure of STA algorithm.

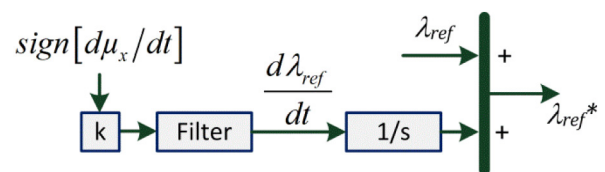


Figure 4. Adaptation procedure of STA algorithm

Positive sign indicates underbraking, whilst negative sign would mean that the maximum friction peak is overcome, so the slip reference has to be reduced. The adaptation is based in the integration of the slope estimation times a gain  $k$ .

For the real implementation, adaptation is only enabled under specific conditions: when the sign estimation is considered significant in terms of numerical stability and coherency between wheels, and when the slip tracking is accurate enough.

### Reactive Torque Controller

The general control law for reactive torque is described as:

$$V_{react} = V_{PI} \cdot \xi_{driver\_dem}, \quad (6)$$

where the correction factor  $\xi_{driver\_dem}$  is used for the saturation of the reactive torque in order to track the driver demand and to prevent generation of wrong torques. The factor  $\xi_{driver\_dem}$  is calculated depending on the driver demand.

The main component of the reactive torque controller is the proportional-integral (PI) part realising the direct wheel slip-based control. The PI controller forms the control demand  $V_{PI}$ :

$$V_{PI} = (V_p + V_I). \quad (7)$$

The proportional part of the controller is the function of the vehicle velocity

$$V_p = K_p (V_x) \min(0, e), \quad (8)$$

and the integral part is computed as:

$$\dot{V}_I = K_I (V_x, mode) \min(0, e) - \alpha \dot{T}_{dem\_wheel} sat(e) V_I, \quad (9)$$

where the coefficient  $\alpha$  is the modifying factor to define the quickness of integral part changing.

In Eqs. (8) and (9), the variable  $e$  is the control error. The saturation of the control error is effected in the following way:

$$sat(e) = \begin{cases} 1, & \max(0, e) > 0.02 \\ e, & \max(0, e) < 0.02 \end{cases}. \quad (10)$$

### Torque Blending

The blending between  $T_{em\_dem}$  and  $T_{br\_dem}$  has to fulfil three objectives: maximize energy regeneration, perform high frequency modulation of reactive torque in ABS situations taking advantage of the electric motors fast response, and do not overcome motor and battery regeneration limits for low frequency demands. The algorithm has to reach these objectives under three main working conditions:

1. Total torque demand ( $T_{dem}$ ) in the four wheels is below the maximum regenerative torque available because of battery state and individual torque demands are below available torque at corresponding motors: In this case, the brake blending allocates all torque demand to the electric motors.
2. No regenerative torque is available (e.g. full battery): The friction brakes have to provide a brake torque level, and the electric powertrain generates the modulation over and under this level by applying alternatively braking and traction torque so that the energy regeneration is null or slightly negative. To do that, the algorithm implements a filter for the reactive torque  $T_{react}$ . Then, the low frequency components of  $T_{react}$  and  $T_{dem}$  are assigned to the friction brake torque  $T_{br\_dem}$  and the high frequency components are assigned to the electric motor torque  $T_{em\_dem}$ .
3. Intermediate cases: In the rest of cases, the electric powertrain has to modulate the fast torque variations whilst the brake torque "slow" level is split between friction brakes and regeneration, taking into account motor and battery limitations but giving priority to regenerative brakes.

It should be noted that the proposed architecture has an external active vibration controller to reduce motor torque oscillations caused by powertrain dynamics, based on the feedback of motor and wheel speeds. Active vibration control algorithm is described in [10].

### Procedure of Definition of Controller Gains

The following criteria are used to find proportional and integral gains in reactive torque controller in the first approximation:

1. Recursive mean square error of longitudinal slip

$$RMSE_{\lambda} = \sqrt{\frac{1}{T} \int_0^T (\lambda_{ref} - |\lambda|)^2 dt}, \quad (11)$$

2. Integral of time-weighted-absolute-error of longitudinal slip

$$ITAE = \int_0^T t |\lambda_{ref} - |\lambda_{act}|| dt, \quad (12)$$