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.


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 ≥ 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 \( F_{\text{driver}} \) is the input of base brake controller generating the torque demand \( T_{\text{dem}} \) for each wheel, which is added up with the reactive torque \( T_{\text{react}} \). The reactive torque is computed by the reactive torque controller, which uses the reference slip ratio \( \lambda_{\text{ref}} \), 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 \( T_{\text{dem}} \) and \( T_{\text{react}} \) 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 \( T_{\text{em, dem}} \) and for the friction brakes \( T_{\text{br, dem}} \). Both torques are being realized by corresponding actuators and the actuator outputs \( T_{\text{em}} \) and \( T_{\text{br}} \) 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).
**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:

$$T_{pred} = \mu_{\text{max\_est}} F_z r + k_{pred} \left( \mu_{\text{max\_est}} F_z r \right),$$

where $\mu_{\text{max\_est}}$ 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:

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

**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

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

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.

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 $\mu_{\text{max}}$ 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:

$$\text{sign} \left[ \frac{d\mu_x}{d\lambda} \right] = \text{sign} \left[ \frac{da_x}{d\lambda} \right] = \text{sign} \left[ \frac{da_x}{dt} \right] = \text{sign} \left( \frac{da_x}{dt} \right).$$

Fig. 4 shows the 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 law is based in the integration of the sign estimation times a gain $k$. 

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*Figure 2. General structure of ABS controller*

*Figure 3. Generation of reference slip*

*Figure 4. Adaptation procedure of STA algorithm*
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:

\[ V_{\text{react}} = V_{PI} \cdot \xi_{\text{driver_dem}}, \quad (6) \]

where the correction factor \( \xi_{\text{driver_dem}} \) 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 \( \xi_{\text{driver_dem}} \) 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:

\[ V_I = K_i(V_x, \text{mode}) \min(0, e) - \alpha \dot{\lambda}_{\text{dem, wheel, sat}}(e) V_I, \quad (9) \]

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:

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

**Torque Blending**

The blending between \( T_{\text{em, dem}} \) and \( T_{\text{br, dem}} \) has to fulfill 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 \( T_{\text{dem}} \) 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 \( T_{\text{react}} \). Then, the low frequency components of \( T_{\text{react}} \) and \( T_{\text{dem}} \) are assigned to the friction brake torque \( T_{\text{br, dem}} \) and the high frequency components are assigned to the electric motor torque \( T_{\text{em, dem}} \).

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

\[ \text{RMSE}_A = \sqrt{\frac{1}{T} \int_0^T \left( \lambda_{\text{ref}} - \lambda_A \right)^2 dt}, \quad (11) \]

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

\[ \text{ITAE} = \int_0^T t |\lambda_{\text{ref}} - \lambda_A| dt, \quad (12) \]
3. Total variance of control demand

\[
TV = \int_0^\alpha \left| \frac{du}{dt} \right| dt .
\]  

The resulting cost function for the gain optimization can be proposed from the criteria 1)-3) as follows:

\[
J = 0.6 \frac{S_{dist}}{\max (s_{dist})} + 0.3 \frac{ITAE}{\max (ITAE)} + 0.1 \frac{RMSE_i}{\max (RMSE_a)} .
\]

The next step is the gain scheduling dependent on vehicle velocity. For this purpose the step braking maneuver from different initial velocities has been analyzed by means of the Differential Evolution optimization strategy. The detailed discussion of the procedures of definition and optimization of controller gains is out of scope of this paper. However, for illustrative purposes, Fig. 5 demonstrates the variant of gain scheduling for the developed reactive torque controller that has been used in subsequent ABS tests. As it can be seen, the proportional gain is being decreased and the integral gain is being increased with the brake velocity degradation. It is required to guarantee low oscillatory behavior of actual slip near to the reference value as well as adaptation of ABS frequency. The gain gradients are distinctly changed after 50 km/h to allow smooth transition of ABS operation from slip-based to velocity-based control.

![Figure 5. Gains of reactive torque controller depending on vehicle velocity (proportional gain is scaled down on 10^2; integral gain is scaled down on 10^4).](image)

### VALIDATION OF DEVELOPED ABS CONTROLLER

#### Target Vehicle Configuration

The results presented in the actual study refer to the sport utility vehicle with technical data given in Table 1. The vehicle has the powertrain configuration with four on-board electric motors, gearboxes, and half-shafts between the drive motors and the wheels, Fig. 6.

**Table 1. Technical data of the vehicle**

<table>
<thead>
<tr>
<th>Type of vehicle</th>
<th>All-wheel drive sport utility vehicle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full mass of vehicle</td>
<td>2045 kg</td>
</tr>
<tr>
<td>Maximum vehicle velocity</td>
<td>195 km/h</td>
</tr>
<tr>
<td>Tires</td>
<td>235/55 R19</td>
</tr>
<tr>
<td>Type of drive motors</td>
<td>Switched reluctance electric motors</td>
</tr>
<tr>
<td>Nominal / Peak power of one motor</td>
<td>42kW (135 Nm) / 100 kW (200 Nm)</td>
</tr>
<tr>
<td>Maximum motor speed</td>
<td>15000 min^-1</td>
</tr>
<tr>
<td>Weight of one motor</td>
<td>50 kg</td>
</tr>
<tr>
<td>Gearbox</td>
<td>Single-speed gearbox (1:10)</td>
</tr>
<tr>
<td>Battery pack</td>
<td>800 V DC</td>
</tr>
</tbody>
</table>

![Figure 6. Powertrain configuration for one axle [11].](image)

Friction brakes are actuated through decoupled electro-hydraulic system (Slip Control Booster (SCB) development of TRW Automotive company), Fig. 7.

![Figure 7. Electro-hydraulic brake system [5].](image)
Experimental Setup

Within the framework of the presented paper, the results of validation of developed ABS controller on real-time hardware-in-the-loop (HIL) test rig are discussed. Fig. 8 shows the main components of the experimental setup [12]. It is based on dSPACE modular platform with several components responsible for data input/output, the control on brake system and communication with the commercial vehicle simulator IPG CarMaker.

The friction brake part of the HIL test rig is represented by an electro-hydraulic brake system (EHB). A set of sensors measure brake pressure in each brake caliper. The brake pedal is actuated by the hydraulic brake robot providing the maximum pedal force up to 1500 N and pre-defined pedal velocity in the range from 0 to 1000 mm/s.

Real-time emulators of electric motors and powertrain components have software realization. They are parameterized through experimental data obtained by the motor producer INVERTO.

The full software simulator of the target vehicle is developed in the IPG CarMaker software environment. The road conditions, profile of vehicle maneuvers, and the driver model are also tuned through IPG CarMaker program shell.

Electric Motors and Slip Control Booster Performance

Simulation models, which have been validated on the HIL rig and vehicle demonstrator, are used to carry out actuator performance tests.

Test Program

The presented paper is concentrated on the analysis of test results for four case studies:

- ABS braking from 60, 90, and 120 km/h on low \(\mu_{\text{max}}=0.3\), middle \(\mu_{\text{max}}=0.6\) and high \(\mu_{\text{max}}=1.0\) friction road. No slip target adaptation, the system uses constant reference slip. This maneuver is used for the analysis of the system behavior at different constant reference slip ratios.
- ABS braking from 120 km/h on low \(\mu_{\text{max}}=0.3\), middle \(\mu_{\text{max}}=0.6\) and high \(\mu_{\text{max}}=1.0\) friction road. Slip target adaptation is activated.
- Comparative ABS braking from 120 km/h on low \(\mu_{\text{max}}=0.3\) and middle \(\mu_{\text{max}}=0.6\) friction road for two different values of torsional stiffness of powertrain: 9000 and 21000 Nm/rad.
- Emulation of failure of the vehicle control unit.

The corresponding results are discussed in next section.
ANALYSIS OF CASE STUDIES

1) ABS Braking without Slip Target Adaptation

For the case of deactivated adaptation of the target slip value, four variants of constant reference slip ($\lambda_{\text{ref}} = 0.04, 0.1, 0.15,$ and 0.25) were tested for all claimed braking conditions. Fig. 10, 11, 12 show examples of corresponding brake diagrams for several particular test cases. After analysis of the experimental data, the following conclusions can be done.

1.1. Tracking of reference slip. From Fig. 10, 11, 12, 13 can be concluded that the developed ABS shows smooth tracking control during steady state condition and robustness regarding the relevant range of reference slip values at different velocities on low- and middle-$\mu$ roads. On low-$\mu$ road, irrespective of the velocity at braking, the controller starts with initial phase of slip adaptation characterized by very small drop of the slip ratio (see example on Fig. 13). In the worst case (120 km/h), this drop is of 0.11 higher as the reference slip for $\lambda_{\text{ref}} = 0.25$. These initial drops are of very short duration. It should be pointed out that the system behaviour is smoother in the case of ABS with activated predictive torque block. Below 30 km/h the reference speed is tracked, instead of reference slip, due to an increased influence of the noise in the slip estimation, which would cause unstable tracking control.

Figure 10. (cont.) Velocity profile and distributions of torques on front left and rear left wheels by braking from 120 km/h on low friction road; reference slip 0.1

Figure 11. Velocity profile and distributions of torques on front left and rear left wheels by braking from 120 km/h on middle friction road; reference slip 0.1
1.2) Brake distance. Fig. 14. ABS controller ensures the reduction of the brake distance on low-μ and middle-μ road surfaces. In particular, for the given range of the initial braking velocities, the achieved reduction was from 5.4% to 8.3% on the middle-μ road and from 14% to 16.8% on the low-μ road as compared with the braking without ABS. The influence of ABS operation on the brake distance on the high-μ road can be considered as not significant, within ± 2%. Lower reference slip ratios provide reduced distances.

1.3) Torque distribution and blending. The specific character of the performed experiments has allowed to evaluate the torque blending during ABS operation with hardware friction brake system and real-time emulated electric powertrain in a braking mode. In all the tested cases, the employment of the friction brake system has been minimized to increase the regenerative braking by electric powertrain fulfilling required total brake torque. The ABS has practically minimized the operation of friction brake system at braking on low-μ road. On the high-μ road, the ABS has some torque oscillations of low amplitude with frequency of 1…3 Hz depending on the vehicle velocity.
2) **ABS Braking with Slip Target Adaptation**

In this case study, target slip values are stored in look-up tables (LUT) generated as functions of the normal wheel force and the tire-road friction characteristics. Fig. 15 displays diagrams illustrating the effect of the slip adaptation on three different road surfaces.

Analysis of the results leads to the following conclusions.

2.1. **Brake distance.** The tests of ABS with activated slip target adaptation showed a brake distance reduction in all the cases in comparison to the tests where the STA block was deactivated, Table 2.

2.2. **Tracking of reference slip.** In all the tested configurations, the system has reached the reference slip ratio within 1 sec after the start of the braking process. The single insignificant deviations from the reference slip are observed on middle- and high-friction road in the middle part of the braking process.

2.3. **Slip target adaptation** results in progressive variation of the initial reference slip target. For low mu surfaces, there is a clear reduction of target with regard to the LUT output, both in front and rear wheels.

Typical increase of slip values has taken place for all road surfaces to the end of manoeuvre. An important fact is that no wheel blocking has been also observed at low velocities (<20 km/h). The final slip values did not surpassed 0.2 on low-μ road, 0.5 on middle-μ road, and 0.65 on high-μ road.

Table 2. Comparison of brake distances for braking from 120 km/h

<table>
<thead>
<tr>
<th>Braking conditions</th>
<th>Without ABS</th>
<th>ABS without STA</th>
<th>ABS with STA</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Absolute value, m</td>
<td>Difference related to braking without ABS, %</td>
<td></td>
</tr>
<tr>
<td>120 km/h, μ=0.3</td>
<td>198.15</td>
<td>-14.6</td>
<td>-16.9</td>
</tr>
<tr>
<td>120 km/h, μ=0.6</td>
<td>101.87</td>
<td>-7.7</td>
<td>-8.3</td>
</tr>
<tr>
<td>120 km/h, μ=1.0</td>
<td>68.2</td>
<td>-1.5</td>
<td>-1.9</td>
</tr>
</tbody>
</table>

Figure 14. (cont.) Brake distances for ABS braking with different reference slips

Figure 15. Wheel slip profiles for ABS braking with slip target adaptation on different road surfaces
3) Comparative Test of ABS Braking for Different Powertrain Parameters

To investigate a possible influence of powertrain parameters on the ABS performance, additional experiments have been done for the vehicle parameters with reduced stiffness of the powertrain of 9000 Nm/rad instead of initially used stiffness of 21000 Nm/rad. The selected wheel slip profiles at ABS braking with reduced stiffness are given in Fig. 16. Intensive slip oscillations at the end of braking maneuver (by velocity below 30 km/h) are caused by changing of the control strategy (velocity control instead of slip control) and are subjected to more precise tuning of control gains.

Figure 15. (cont.) Wheel slip profiles for ABS braking with slip target adaptation on different road surfaces

Figure 16. (cont.) Wheel slip profiles for ABS braking with slip target adaptation on different road surfaces

The results have displayed no deterioration of brake performance in terms of brake distance and average deceleration. The slip dynamics of the ABS with reduced powertrain stiffness is also characterized with sufficiently smooth tracking of reference slip ratio both on front and rear values. These results are confirmed that the developed ABS control strategy can be used in the case of change of the powertrain configuration, in particular, for less stiff variant of the half-shafts between electric drive motors and the wheels.

4) ABS Failure

A special test procedure relates to the emulation of the failure of the vehicle control unit. In such situation, the internal ABS controller of the SCB system should be activated to complete the braking maneuver. Using the real-time test on the HIL test platform, the corresponding braking test has been performed for the worst-case scenario with initial velocity 120 km/h and low-μ road. As it can be seen from Fig. 17, the failure of the ABS occurred after 8 s of the braking process. The controller has recognized the critical situation and activated internal ABS of the SCB unit within 200 ms. For the given conditions, a short-term increase of the slip took place without the locking of wheels. Hence, the coordinated operation of the developed ABS and internal ABS of SCB unit is confirmed.
SUMMARY

The presented study has introduced a new ABS control strategy developed for electric vehicles with individually controlled on-board motors for each wheel. The controller is based on PI direct slip control and has been experimentally validated and verified using hardware-in-the-loop technique. A number of conclusions can be deduced from the ABS validation for straight-line braking maneuvers in different road conditions:

- The proposed control strategy for ABS actuating electric motors in a barking mode makes it possible to achieve smooth tracking of reference wheel slip as compared with conventional hydraulic ABS.
- No essential oscillations of electric motor brake torque have been observed during the ABS actuation. The highest frequency of torque oscillation, 1…3 Hz, has been taken place for the test cases on high-friction road surfaces. However the amplitude of these oscillations was insignificant.
- In the case of ABS without slip target adaptation, the reduction of brake distance as compared with the braking without ABS was insignificant for high-friction road surfaces. For low- and middle-road surfaces, the achieved brake distance reduction was in the range 5.4…16.8% depending on initial braking conditions and appointed reference slip value.
- The proposed slip target adaptation allows to reach the required reference slip value within 1 sec after the start of braking maneuver. No wheel blocking has been occurred during the adaptation process.
- The developed controller ensures stable ABS operation for powertrain configurations with different stiffness. In particular, required tracking of the reference slip has been also achieved for more critical case with low stiffness 9000 Nm/rad, where significant powertrain oscillations were initially expected.
- The proposed fail safe mechanism guarantees the switching of the ABS operation to the internal controller of SCB electro-hydraulic brake system in the case of the failure of the main ABS controller. The duration of the switching process does not exceed 200 ms, and the wheels are not locking during the corresponding transient process.

The authors plan to introduce results of ABS testing on real vehicle in further publications as next stage of the presented research.

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DEFINITIONS/ABBREVIATIONS

ABS - anti-lock brake system
FL - front left wheel
FR - front right wheel
HIL - hardware-in-the-loop
RL - rear left wheel
RR - rear right wheel
SCB - Slip Control Boost
STA - slip target adaptation
e - control error

\( F_{\text{driver}} \) - driver control action

\( F_{\text{em}} \) - brake force generated by electric motor

\( F_z \) - vertical tyre force

\( ITAE \) - integral of time-weighted absolute error

\( k \) - actuator rate

\( K_P \) - proportional gain

\( K_I \) - integral gain

\( LUT \) - Look-up table

\( r \) - tire radius

\( RMSE_A \) - recursive mean square error of longitudinal slip

\( s \) - Laplace operator

\( s_{\text{dist}} \) - brake distance

\( T_{\text{br}} \) - friction brake torque

\( T_{\text{br}, \text{dem}} \) - friction brake torque demand

\( T_{\text{dem}} \) - torque demand

\( T_{\text{em}} \) - electric motor torque

\( T_{\text{em}, \text{dem}} \) - electric motor torque demand

\( T_{\text{pred}} \) - predictive torque

\( T_{\text{react}} \) - reactive torque

\( T_w \) - wheel torque

\( TV \) - total variance

\( u \) - control demand

\( V_w \) - linear wheel velocity

\( V_x \) - longitudinal vehicle velocity

\( \lambda \) - wheel slip

\( \lambda_{\text{ref}} \) - reference wheel slip

\( \mu \) - tire friction coefficient (general)

\( \mu_{\text{max}} \) - maximum tire friction coefficient

\( \mu_x \) - longitudinal tyre friction coefficient

\( v_{\text{dem}} \) - correction gain taking into account torque demand

\( v_{\text{PI}} \) - proportional-integral gain

\( \xi_{\text{driver, dem}} \) - correction factor of driver demand