

Article

Hybrid Sliding-Mode-Based Wheel Slip Control for Brake-by-Wire Vehicles with In-Wheel Motors

Tomas Grikenis ¹, Paulius Kojis ², Valentin Ivanov ³ and Viktor Skrickij ^{2,*}

¹ Department of Mobile Machinery and Railway Transport, Vilnius Gediminas Technical University, 10105 Vilnius, Lithuania; tomas.grikenis@vilniustech.lt

² Transport and Logistics Competence Centre, Vilnius Gediminas Technical University, 10105 Vilnius, Lithuania; paulius.kojis@vilniustech.lt

³ Smart Vehicle Systems Group, Control Engineering Group, Technische Universität Ilmenau, Robert-Bosch-Ring 1, 98693 Ilmenau, Germany; valentin.ivanov@tu-ilmenau.de

* Correspondence: viktor.skrickij@vilniustech.lt

Abstract

Sliding-mode control (SMC) is valued in automotive applications for its robustness and its low computational cost, although chattering remains a practical obstacle. Higher-order and adaptive formulations reduce chattering, but each carries limitations that may restrict their use in wheel slip control. This paper presents a hybrid low-level controller that combines a first-order sliding-mode (FOSM) term with a proportional–integral (PI) term acting on the same slip error. The two branches are weighted by a hyperbolic tangent function, so that the FOSM term governs the response towards the reference slip and the PI term governs it within a narrow band around the reference. Because the weighting is continuous, the transfer between branches introduces no step change in the torque demand. The controller is embedded in a three-level brake-by-wire architecture with series brake blending between in-wheel motors and friction brakes. The study uses an experimentally validated 14-DOF vehicle model and proposes a benchmarking procedure for comparison against a velocity-scheduled PI baseline. The results suggest improved slip tracking of the braking manoeuvre.

Keywords: sliding-mode control; wheel slip control; brake-by-wire; brake blending; in-wheel motor; vehicle dynamics; anti-lock braking system

1. Introduction

Vehicle electrification is transforming the automotive industry. It eliminates exhaust emissions and lessens the need for fossil fuels for propulsion [1]. Current electric vehicles use on-board electric motors (OBM). However, corner-based architectures are gaining popularity, with in-wheel motors as a central part [2]. This shift creates scope to redesign vehicle systems, since the new actuators enable control methods previously limited by available hardware.

Among the available control approaches, sliding-mode control (SMC) is known for its robustness, which is relevant to automotive applications for safety reasons. This method does not require high computational power. However, the main drawback is chattering [3–5]. Recent work on SMC has sought to address chattering rather than merely mitigate it, particularly through barrier function adaptation [6–13]. These formulations reduce the adaptive gain as the sliding variable approaches zero, suppressing the discontinuous term near the surface but leaving little authority to keep the trajectory on it. The approach taken



Academic Editors: Jan Kalivoda and Larysa Neduzha

Received: 7 September 2026

Revised: 12 October 2026

Accepted: 14 October 2026

Published: 16 October 2026

Copyright: © 2026 by the authors.

Licensee MDPI, Basel, Switzerland.

This article is an open access article distributed under the terms and conditions of the [Creative Commons Attribution \(CC BY\)](https://creativecommons.org/licenses/by/4.0/) license.

here retains the sliding-mode term away from the surface and transfers authority to a second controller within a narrow band around it.

This paper proposes a hybrid low-level wheel slip controller in which a first-order sliding-mode (FOSM) term and a PI term operate based on the same slip error and are weighted by a continuous function of that error. The controller is evaluated within a three-level brake-by-wire architecture that includes series brake blending between in-wheel motors and friction brakes, using an experimentally validated 14-DOF vehicle model.

The paper is organised as follows: Section 2 reviews related work and presents the contribution; Section 3 presents the vehicle model, control architecture, proposed controller, and benchmarking procedure; Section 4 reports the results; and Section 5 presents the discussion and conclusions.

2. Literature Review

The braking system is a crucial part of the vehicle's active safety. The anti-lock braking system (ABS) control algorithm is widely used for braking, as well as the low-level controller for vehicle stability control (VSC) and emergency braking assist (EBA) [14,15]. Advancements in electronics and control systems, the automotive sector's adoption of these technologies, and growing interest in Advanced Driver Assistance Systems (ADAS) and automated driving vehicles have led to the creation of various x-by-wire systems that decouple control inputs from the driver and the actuators. This shift toward system decoupling lays the foundation for integrated chassis control, which may allow closer coordination between chassis subsystems and contribute to improved transport-system efficiency and sustainability objectives [16,17]. Control decoupling also includes the brake-by-wire (BbW) system, which offers multiple advantages over conventional hydraulic braking systems. The decoupled structure of BbW provides more flexibility in designing the complete vehicle dynamics control system. BbW actuators offer a wider control bandwidth than conventional hydraulic actuators, greater freedom in component packaging, fewer failure-prone mechanical components, and no need for a separate parking brake system [16,18]. If electromechanical brake callipers are used, they eliminate hydraulic lines and pumps and remove the need to regularly change brake fluid, which is susceptible to overheating and moisture absorption.

BbW enables brake blending and can use OBMs or In-Wheel Motors (IWM) to brake individually. BbW technologies have improved in recent years by introducing four-wheel independent braking [18], which, combined with friction estimation and high-frequency control, may improve braking safety. Individual braking torque control improves vehicle dynamics control, ride comfort, and energy efficiency [14].

BbW systems may use various actuators, including electrohydraulic (EHB), electromechanical (EMB), magnetorheological, electric motors, and hybrid actuators [19–29].

The main goal of the braking system is to minimise the stopping distance while maintaining steerability and directional stability. Energy recuperation and minimal particle and noise emissions are also desirable. This can be achieved by controlling each wheel's longitudinal slip ratio to the value that produces the highest friction coefficient, and therefore the highest longitudinal force.

In braking scenarios, threshold-based or slip-ratio-based feedback logic commonly controls wheel slip. The literature also calls this control logic rule-based or event-triggered [30]. These controllers rely on wheel-speed sensors and simple feedback loops, offering robustness and interpretability under nominal conditions [30,31].

However, these methods can be sensitive to parameter tuning and may not handle tyre nonlinearity or diverse road conditions effectively.

To handle this limitation, fuzzy logic controllers may be used, where a set of pre-defined logic rules creates an input–output map [32]. Fuzzy logic controllers may also estimate the road friction coefficient and other unknown parameters [33–35]. Researchers have also combined fuzzy logic with sliding-mode control [36,37]. Researchers have also applied PID, LQR, and H-infinity controllers to wheel slip control [35,38,39]. However, two approaches dominate recent publications: SMC and model predictive control (MPC). Like optimal control, MPC uses optimisation to find the optimal control input to apply to the plant [30]. Wheel slip control requires robustness; robust MPC has been developed and includes constraint-tightening methods, tubes, and chance constraints [40–43]. The principal constraint remains the required computational effort.

Another approach is SMC-based [44–46]. FOSM, integral SMC, the super-twisting algorithm, and other higher-order SMC methods are widely used. First-order sliding mode is associated with high chattering, which higher-order formulations generally reduce. Barrier function adaptation offers one route to reduce chattering. However, the gain goes to zero as the sliding variable approaches zero, which in practice prevents exact surface tracking.

This work presents a variable-structure wheel-slip controller for an electric vehicle with in-wheel motors. We propose a hybrid control approach based on FOSM and PI logic to suppress chattering without a hard switch between the two branches.

The contribution of this paper is as follows:

- We propose a hybrid wheel slip controller in which an FOSM term and a PI term act on a common sliding surface defined by the slip-tracking error.
- The two branches are weighted by hyperbolic tangent functions of the slip-tracking error rather than a hard switch, so the torque demand remains continuous during the transition phase.
- The controller is evaluated within a complete three-level brake-by-wire architecture, including series blending between in-wheel motors and friction brakes, rather than at a single friction brake in isolation.

3. Materials and Methods

We previously developed a real-time, high-fidelity 14-DOF mathematical model of a sport-utility vehicle (SUV) in IPG CarMaker and implemented its control in MATLAB/Simulink R2020a. All numerical simulations used a fixed time step of 0.001 s, and the controller inputs and outputs were held constant between samples (zero-order hold). Tyre dynamics are represented by the Magic Formula (MF) 6.1 model, parameterised from experimental data and capturing transient behaviour: lateral and longitudinal forces, self-aligning moment, combined-slip interactions, load and camber effects, and nonlinear saturation. The sprung and unsprung masses are represented as lumped mass elements. We assume a rigid vehicle body with precisely defined geometric and inertial properties, while neglecting structural deformations. The unsprung mass model separately accounts for the masses and moments of inertia of the wheel hubs and wheels. We determined the suspension kinematic and compliance (K&C) characteristics using MSC Adams 2013 software. We converted the resulting data into lookup tables (LUTs) and integrated them into the overall vehicle model. We also represented the nonlinear suspension characteristics, including damping, stiffness, elastomeric bushing, and mount properties, using LUTs and incorporated them into the overall model structure.

The vehicle model was parametrised using mass and inertia properties. The model was validated against experimental data from the proving ground. We obtained the brake-by-wire actuator demand-to-actual torque transfer functions used in the simulations through Hardware-in-the-Loop bench testing in [47]. Therefore, every subsystem active during

braking was validated using real-world or experimental test data. More details on the validation are in [48], and the field test data are in [49]. Table 1 lists the main vehicle parameters.

Table 1. Vehicle parameters.

Symbol	Parameter	Value
L	Wheelbase length	2.675 m
l_f	Distance from front axle to CoG	1.236 m
l_r	Distance from rear axle to CoG	1.439 m
h	Height of CoG	0.65 m
C_f	Front axle cornering stiffness	171,011 N/rad
C_r	Rear axle cornering stiffness	171,011 N/rad
K_{us}	Understeer gradient	0.008 rad/g
m	Mass	2189 kg
I_z	Moment of inertia of the vehicle about the z-axis	3665 kg·m ²
J_w	Moment of inertia of rotating wheel and motor assembly	3 kg·m ²
p	Baseline tyre pressure	2.5 bar
r	Dynamic tyre radius	0.3706 m

3.1. Control Architecture

In controlling a BbW system, three main control levels are considered (Figure 1). The control system follows a hierarchical architecture to ensure the desired performance at all levels.

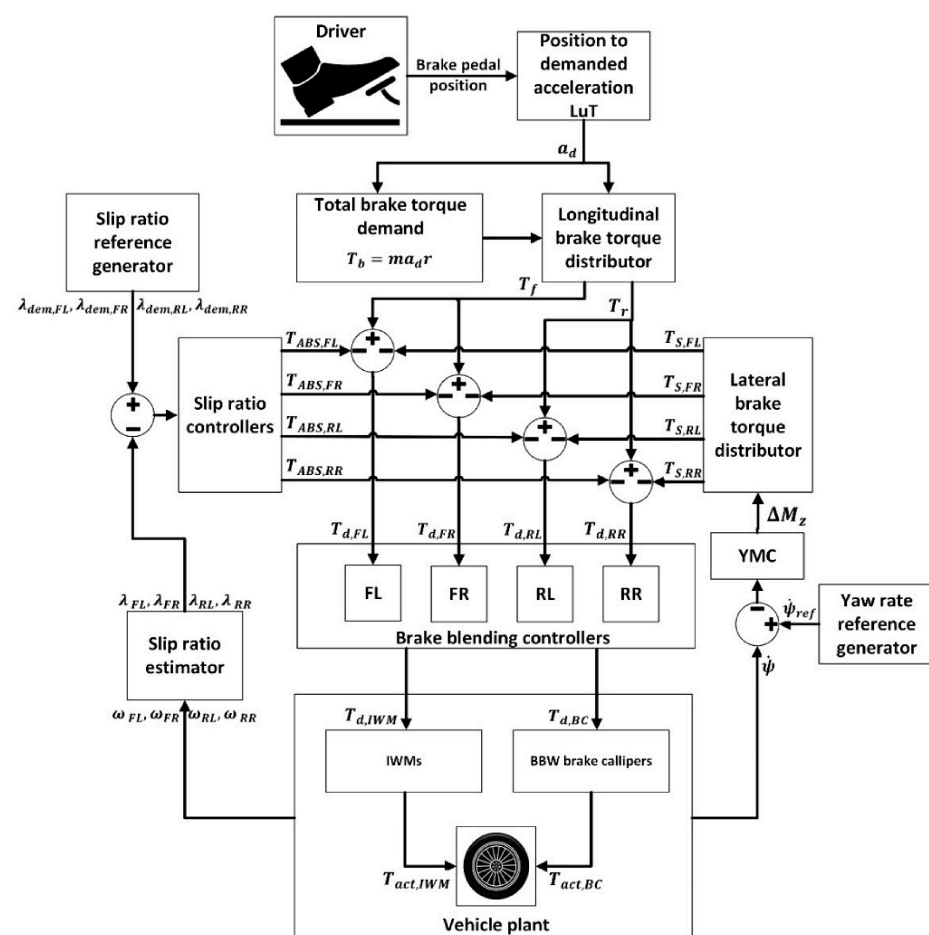


Figure 1. Control architecture of a BbW system.

The high-level controller receives the brake pedal position from a position sensor. Using a LUT, it converts the pedal position into a vehicle deceleration demand. This LUT lets the braking system designer adjust braking intensity, making the system more or less

sensitive to the driver's pedal input. The total braking torque required from all brake actuators is then calculated as

$$T_b = ma_d r; \quad (1)$$

where m —mass of the vehicle [kg]; a_d —demanded deceleration, negative during braking [m/s^2]; and r —dynamic wheel radius [m].

The mid-level controller receives the total required braking torque from the high-level controller and distributes it among the vehicle's braking actuators.

During braking, inertial forces at the CoG redistribute vertical load between the front and rear axles, commonly called longitudinal load transfer. As a result, the vertical tyre forces on the front axle increase, while those on the rear axle decrease. Applying equal braking torque to all wheels could cause the vehicle to become unstable due to wheel lock-up; therefore, this phenomenon must be considered. The ideal braking force distribution is determined using moment-equilibrium equations about the front and rear tyre-road contact points for a decelerating vehicle.

The ideal braking torque at the rear axle is expressed as follows:

$$T_r = mar \left(\frac{l_f}{L} - \frac{a}{g} \frac{h}{L} \right); \quad (2)$$

where l_f —distance between CoG and front axle [m]; L —wheelbase [m]; h —distance between road surface and CoG [m]; g —gravitational constant [m/s^2]; and a —measured longitudinal acceleration [m/s^2], which is negative during braking. Consequently, the front axle braking torque is expressed as $T_f = T_b - T_r$ and, as well as T_r , is a negative value. To ensure vehicle stability, this ideal rear-axle braking torque is reduced by 5%, thereby lowering the probability of rear-wheel lock-up.

During braking, both longitudinal and lateral stability must be ensured. When the road-tyre friction coefficient is constant, the front tyres generate equal longitudinal forces on the left and right sides. The same applies to the rear axle.

However, when the left-side wheels are on a lower-friction surface while the right-side wheels are on a higher-friction surface, the vehicle becomes unstable during braking and deviates from its straight-line trajectory (and may even spin around its axis) toward the side with the higher friction coefficient. Thus, if four low-level longitudinal slip controllers are used and all of them accurately track the same reference value, the braking force will be greater on the higher-friction side. This creates a net yaw moment about the vertical axis, destabilising the vehicle. To prevent this, the braking torque on the higher-friction side must also be controlled. Ideally, the braking torques at the front left and front right wheels should be equal, and the same should apply to the rear axle's left and right wheels.

To achieve this, a yaw moment ΔM_z must be generated about the vertical axis in the direction opposite to the resulting net (destabilising) moment. The stabilising yaw moment ΔM_z can be generated using a feedback controller. Its input is the difference between the reference and actual yaw-rate values. The reference value is calculated using the bicycle model, following the methodology presented in [50]. The generated ΔM_z is then distributed between all four wheels. This subsystem was not used in the investigation, as this paper focuses on the WSCs, but it is presented here to give the reader a full picture of the BbW braking systems.

The middle-level controller outputs negative demanded braking torques. They are calculated as $T_{d,i} = T_j - T_{ABS,i}$, where j is the front or rear axle, i is FL, FR, RL, or RR wheels, and $T_{ABS,i}$ is the negative output of the low-level controllers.

The low-level controller adjusts the braking torque demand received from the mid-level controller so that each wheel's longitudinal slip ratio remains as close as possible to

the reference value. Each wheel has an individual controller. The baseline for comparison is a PI controller, one of the most widely used control strategies in industrial applications. With appropriately selected proportional and integral gains, the controller can achieve the desired system dynamics within the selected operating range. Nevertheless, PID controllers do not inherently guarantee optimal performance and may exhibit excessive overshoot, oscillations, or steady-state deviation when gains are poorly tuned or actuator saturation occurs.

PID controllers operate solely on feedback signals and rely on observed input and output behaviour, regardless of the system's specific dynamics. This model-free approach is generally advantageous because it lets the controller work across a wide range of systems without requiring a detailed system model. However, this also makes the control strategy entirely reactive. Furthermore, PID controllers are less suitable for systems with nonlinear dynamics compared with more advanced control methods. When applying PID control, a trade-off must be made between accurate regulation and rapid response; PID controllers cannot adapt to changes in process dynamics and may respond slowly to large disturbances. Another limitation is that the derivative term can significantly amplify noise in the control signal [32–34].

Several strategies can mitigate these issues. For example, applying a low-pass filter to the derivative term can reduce signal noise, although excessive filtering may diminish the benefits of the derivative action. Therefore, limit filtering uses low-noise sensors and measurement equipment when possible. Anti-windup methods can address actuator saturation. In this investigation, we employ a PI controller with variable, velocity-dependent gains. The controller applies higher gains at higher vehicle speeds, where wheel dynamics are slower and more stable, and lower gains at lower speeds, where dynamics become faster and more sensitive. The output of the PI controllers is limited by the outputs of the high-level controller T_f and T_r . Back-calculation is used as the anti-windup method for the integral term, scaled by gains of 0.83 and 1.67 for the front and rear wheels, respectively.

The longitudinal slip is evaluated as follows:

$$\lambda_i = \frac{\omega_i r - V_x}{\max(V_x; \omega_i r; \varepsilon)}; \quad (3)$$

where V_x —longitudinal velocity [m/s]; ω_i —angular velocity of each wheel [rad/s]; r —wheel's dynamic radius [m]; and ε —small number used to prevent division by zero at low velocity, which is chosen to be 0.1. With this convention, the slip ratio is negative during braking.

3.2. Proposed Hybrid Wheel Slip Controller

The rotational dynamics of each wheel are described by

$$J_w \dot{\omega}_i = T_{d,i} - r F_{x,i}; \quad (4)$$

where J_w —rotating assembly moment of inertia (including wheel, friction brake rotor and rotor of the electric motor) [kg·m²]; $F_{x,i}$ —longitudinal tyre force at wheel i [N], which is negative during braking; and $T_{d,i}$ —applied wheel torque [N·m], which is negative during braking.

Differentiating (3) and substituting (4) gives the slip dynamics in control-affine form:

$$\dot{\lambda}_i = -\frac{F_{x,i}}{V_x} \left[\frac{r^2}{J_w} + \frac{\lambda_i + 1}{m} \right] + g(V_x) T_{d,i}, \quad g(V_x) = \frac{r}{J_w V_x}; \quad (5)$$

The first term combines vehicle deceleration and tyre force, neither of which is measured; we treat it as a bounded disturbance. The input gain $g(V_x)$ is inversely proportional to the longitudinal velocity, so the slip's sensitivity to a given torque correction increases as the vehicle slows down. This property is revisited in Section 4.

The sliding surface for each wheel is taken as the slip-tracking error, without integral or derivative terms:

$$s_i = \lambda_i - \lambda_{ref,i}; \quad (6)$$

Away from the surface, a FOSM term is applied:

$$u_{SMC,i} = k_i \operatorname{sign}(s_i); \quad (7)$$

where k_i —sliding-mode gain [N·m], selected as the maximal value of braking torque the BbW callipers can provide (different for front and rear wheels; please see Table 2), which, when multiplied by $g(V_x)$, exceeds the expected upper bound of the disturbance term in (5).

Table 2. Proposed wheel slip controller parameters.

Symbol	Parameter	Value
k_f	Sliding-mode gain, front wheels	4000 N·m
k_r	Sliding-mode gain, rear wheels	3000 N·m
K_p	Proportional gain, front/rear	36,000 V_x / 25,000 V_x
K_i	Integral gain, front/rear	1.6 V_x / 0.85 V_x
Φ	Transition band half-width, front/rear	0.03 / 0.02
β	Switching shape factor front/rear	50 / 150
λ_{ref}	Reference slip ratio	−0.11 (high friction) −0.03 (low friction)

Near the surface, the discontinuous term in (7) is the source of chattering. One response to this is to reduce its weight in a band around $s_i = 0$ and to transfer authority to a PI controller acting on the same error. The lower and upper switching boundaries are defined as $\lambda_{low,i} = \lambda_{ref,i} - \Phi$ and $\lambda_{high,i} = \lambda_{ref,i} + \Phi$, where $\Phi = 0.03$ is the half-width of the transition band for the front axle and $\Phi = 0.02$ for the rear axle [−]. We selected these values slightly higher than the chattering amplitude a pure FOSM controller produces in steady state.

The transition functions are defined as

$$w_{1,i} = \frac{1}{2} [1 + \tanh(\beta(\lambda_{act,i} - \lambda_{low,i}))] \quad (8)$$

$$w_{2,i} = \frac{1}{2} [1 + \tanh(\beta(\lambda_{act,i} - \lambda_{high,i}))] \quad (9)$$

where β —shape factor [−], which sets the width of the transition: a larger value drives it towards a hard switch at $|s_i| = \Phi$, while a smaller value widens the transition width.

The PI controller weighting factor is $w_i = w_{1,i}(1 - w_{2,i})$ and the FOSM factor is therefore $1 - w_i$.

With $\beta = 50$ and 150 for the front and rear wheels respectively, the weighting factor is close to unity at $s_i = 0$ and has decayed to below 0.01 at the band edge $|s_i| = \Phi$, so that the FOSM term is active outside $\lambda_{dem,i} \pm 0.02$ and $\lambda_{dem,i} \pm 0.03$ and the PI term is active inside it.

The corrective torque issued by the low-level controller is then the weighted combination:

$$T_{ABS,i} = [1 - w_i]u_{SMC,i} + w_i u_{PI,i}; \quad (10)$$

Both the proposed and baseline methods use exactly the same PI controller:

$$u_{PI,i} = K_P(V_x)s_i + K_I(V_x) \int_0^t s_i d\tau. \quad (11)$$

Because w_i is continuous and bounded on $[0, 1]$, the transfer between the two branches introduces no step change in $T_{ABS,i}$, which distinguishes this arrangement from a hard switch at a fixed threshold.

The PI gains are scheduled with longitudinal velocity, as described in Section 3.1, and other wheel slip controller parameters are listed in Table 2. The proposed controller uses the same PI controller and parameters as the baseline. The baseline system lacks a friction estimator that generates the longitudinal slip ratio reference. The system is described in [51]. It uses video camera data and deep neural networks to process it and output the road surface type (asphalt, cobblestone, or gravel; either can be wet or dry). In the simulation results in Section 4, the friction estimator is not part of the control loop: the model provides the road type directly, so we assume ideal classification and the correct friction coefficient. The controllers receive the slip reference at the next simulation step when the wheels enter a change in road surface. Table 2 lists the controller parameters. In all cases, the WSCs are switched off below a vehicle velocity of 3 m/s to avoid the mathematical singularity that occurs during slip estimation in (3).

3.3. Brake Blending

The torque demand at the low-level controller issues the torque demand $T_{d,i}$, which the in-wheel motor and the friction brake realise jointly at each corner. A series blending strategy is applied: the in-wheel motor is used first because it permits electrical energy recovery and responds faster, and the friction brake supplies only the remainder when the motor demand saturates.

$$T_{d,IWM,i} = \max(T_{d,i}, -T_{IWM,max}(\omega_i, SoC)); \quad (12)$$

$$T_{d,BC,i} = T_{d,i} - T_{d,IWM,i}; \quad (13)$$

where $T_{IWM,max}$ —available regenerative torque [N·m], limited by the motor characteristics and by the battery state of charge SoC , and it is negative during braking; $T_{d,BC,i}$ —friction brake torque demand [N·m], also a negative value. Both torques are negative during braking, so the $\max$ operator in (12) assigns the demand to the motor until it reaches its limit.

3.4. Benchmarking Procedure

The proposed controller is compared against the velocity-scheduled PI baseline described in Section 3.1 under identical manoeuvres, vehicle parameters and solver settings. Braking is initiated at t_a and continues to standstill at t_s .

The braking distance is as follows:

$$s_b = \int_{t_a}^{t_s} V_x dt; \quad (14)$$

Slip tracking root mean square error is employed as the main indicator of WSC's performance:

$$e_\lambda = \frac{1}{N} \sum_{i=1}^N \sqrt{\frac{1}{t_b - t_a} \int_{t_a}^{t_b} (\lambda_i - \lambda_{dem,i})^2 dt}; \quad (15)$$

where $[t_a, t_b]$ denotes the phase under evaluation, t_a is the time when the brake pedal is depressed, and t_b is the time when the WSC gets turned off (below 3 m/s). Table 3 presents the test matrix.

Table 3. Test matrix.

Test	Surface	μ	Initial Speed	Purpose
T1	Uniform	0.8	100 km/h	Nominal high friction
T2	Uniform	0.4	100 km/h	Low friction
T3	μ -transition	0.8 $\rightarrow$ 0.4	100 km/h	Reference tracking under friction change
T4	Uniform, mass +20%	0.8	100 km/h	Parameter robustness
T5	Uniform, V_x delay and noise	0.8	100 km/h	Parameter robustness

4. Results

We evaluated the proposed controller and the velocity-scheduled PI baseline under the manoeuvres listed in Table 3, using identical vehicle parameters and solver settings. Braking is initiated at $t_1 = 5$ s in all cases. Tables 4 and 5 report the indicators defined in Section 3.4. In Figures 2–5 and 7–11, the plots of the front (subscripts fl and fr) and rear (subscripts rl and rr) wheels completely coincide due to the uniform road surface friction coefficient applied across the vehicle width in the model.

Table 4. Performance indicators for low- and high-friction braking manoeuvres.

Indicator	$\mu = 0.8$ Baseline	$\mu = 0.8$ Proposed	$\mu = 0.4$ Baseline	$\mu = 0.4$ Proposed
Braking distance s_b [m]	65.8	63.5	133.1	127.1
e_λ , front axle [–]	0.0446	0.0199 (55.4% improvement)	0.0314	0.0161 (48.7% improvement)
e_λ , rear axle [–]	0.0246	0.0116 (52.8% improvement)	0.0176	0.0069 (60.8% improvement)

Table 5. Performance indicators for friction-transition braking and high-friction braking with increased-mass manoeuvres.

Indicator	Friction Transition Baseline	Friction Transition Proposed	Mass Increase Baseline	Mass Increase Proposed
Braking distance s_b [m]	73.3	68.3	67.8	65.6
e_λ , front axle [–]	0.0489	0.0221 (54.9% improvement)	0.0548	0.0229 (58.2% improvement)
e_λ , rear axle [–]	0.0222	0.0120 (46% improvement)	0.0254	0.0132 (48% improvement)

4.1. Braking on a Uniform High-Friction Surface

Figures 2 and 3 show the braking torque demand and the longitudinal slip ratio for the baseline and the proposed controller on $\mu = 0.8$. The baseline reaches standstill at $t = 9.49$ s and the proposed controller at $t = 9.26$ s, corresponding to braking distances of 65.8 m and 63.5 m, respectively, a difference of -3.5% .

Under the baseline, the front-wheel slip oscillates about the reference for the whole duration of the manoeuvre, with a corresponding oscillation in the torque demand of approximately 171 N·m peak-to-peak. With the proposed controller, the slip settles close to the reference is within 0.14 s of brake application, and the oscillation present in the baseline is largely absent between $t = 5$ s and $t = 5.8$ s. The tracking error e_λ of the front axle falls from 0.0446 to 0.0199 and from 0.0246 to 0.0116 for the rear axle, representing a decrease of 55.4% and 52.8% respectively.

From approximately $t = 5.8$ s, corresponding to a longitudinal velocity of about 24.4 m/s, the torque demand under the proposed controller develops an oscillation whose amplitude reaches approximately 144 N·m peak-to-peak. Over the same interval, the slip trace departs from the reference by up to 0.025.

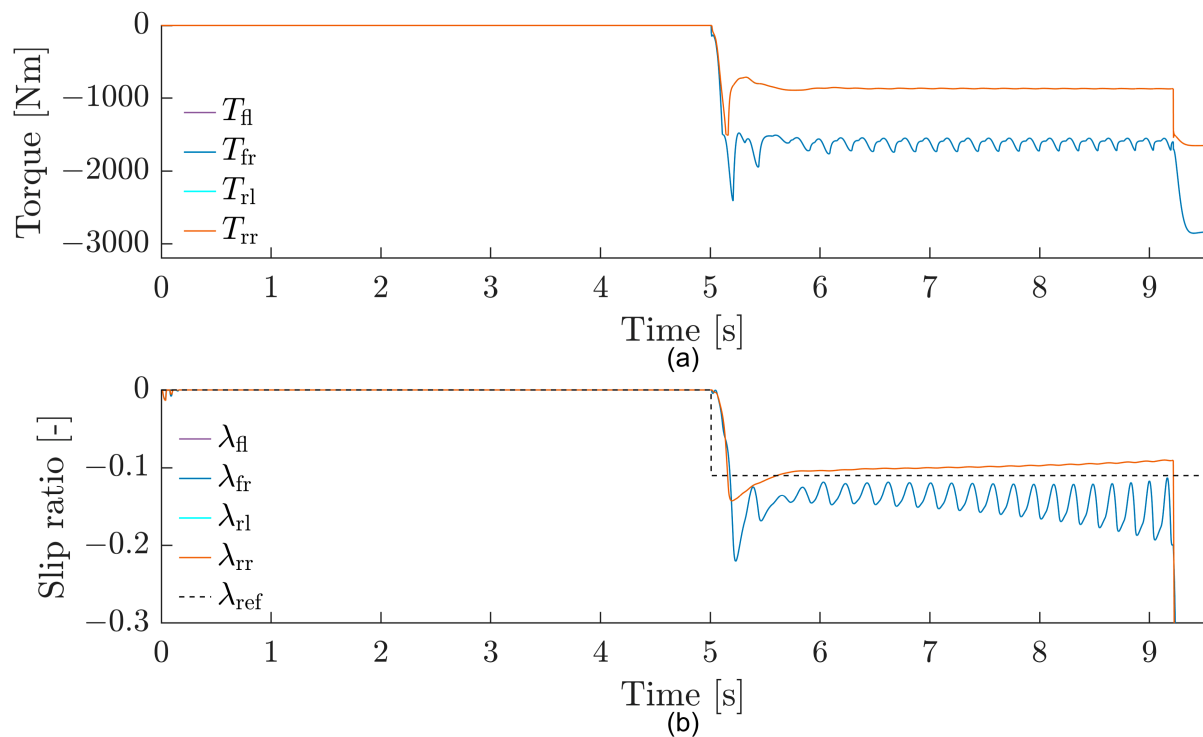


Figure 2. Baseline controller, braking from 100 km/h on a uniform high-friction surface ($\mu = 0.8$): (a) braking torque demand at each wheel; (b) longitudinal slip ratio and reference.

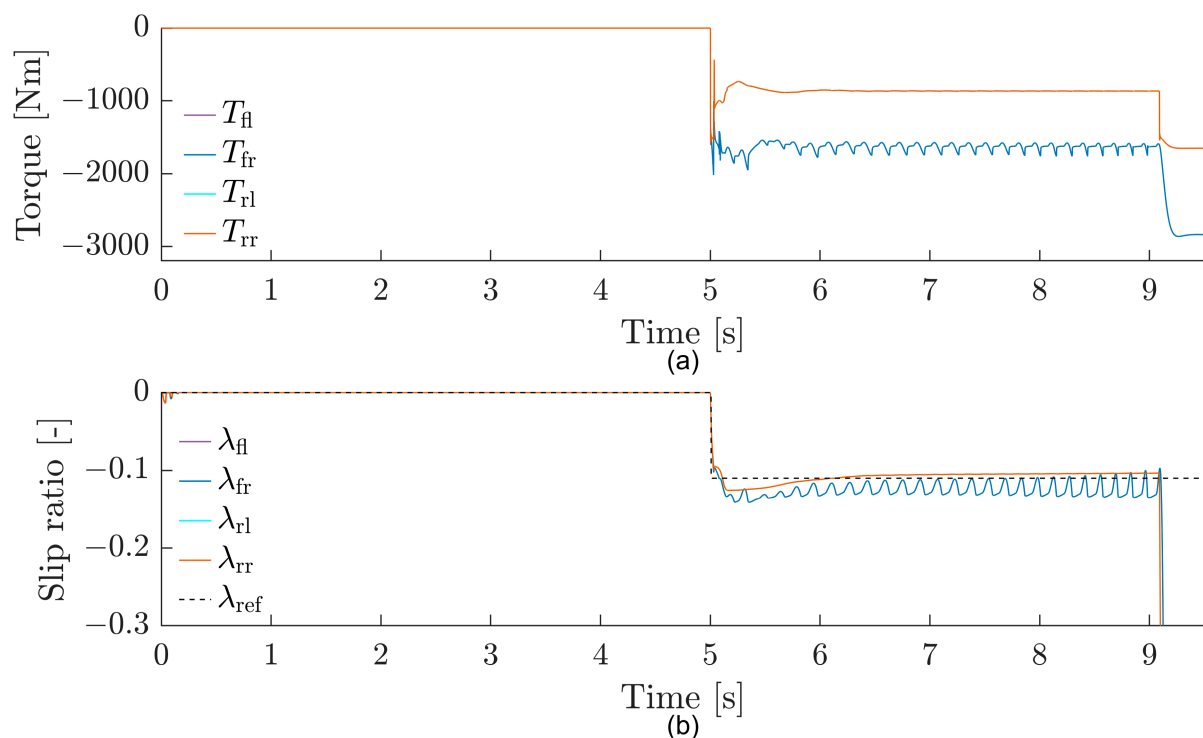


Figure 3. Proposed hybrid controller, braking from 100 km/h on a uniform high-friction surface ($\mu = 0.8$): (a) braking torque demand at each wheel; (b) longitudinal slip ratio and reference.

4.2. Braking on a Uniform Low-Friction Surface

Figures 4 and 5 show the corresponding traces for $\mu = 0.4$. Standstill is reached at $t = 13.9$ s under the baseline and $t = 13.1$ s under the proposed controller, giving braking distances of 133.1 m and 127.1 m.

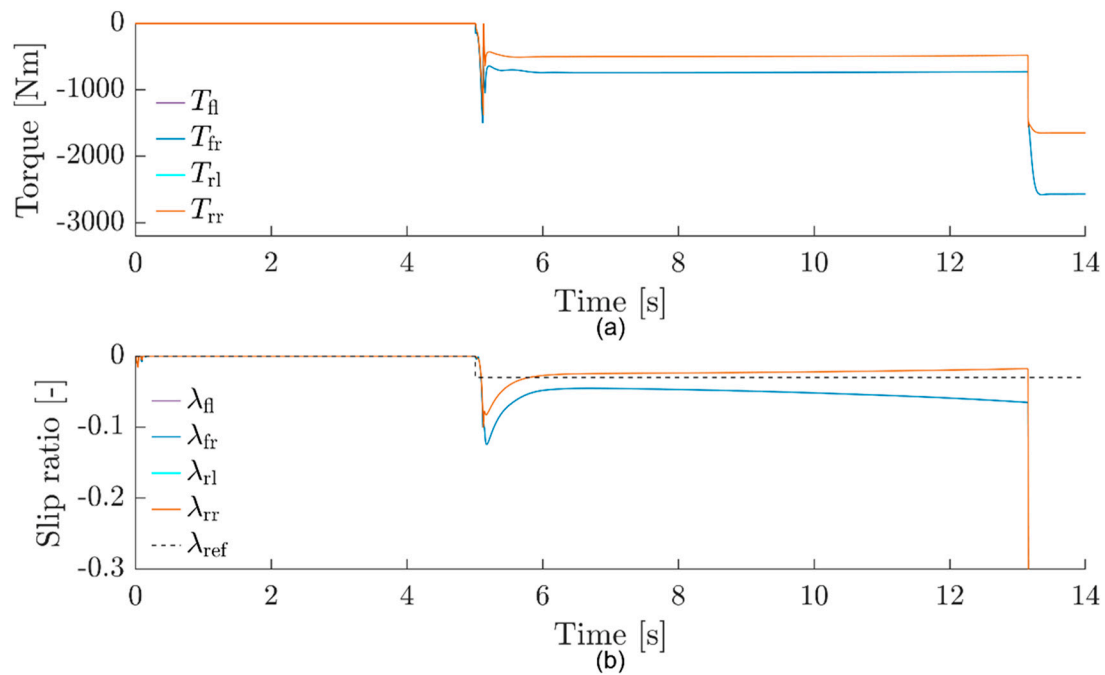


Figure 4. Baseline controller, braking from 100 km/h on a uniform low-friction surface ($\mu = 0.4$): (a) braking torque demand at each wheel; (b) longitudinal slip ratio and reference.

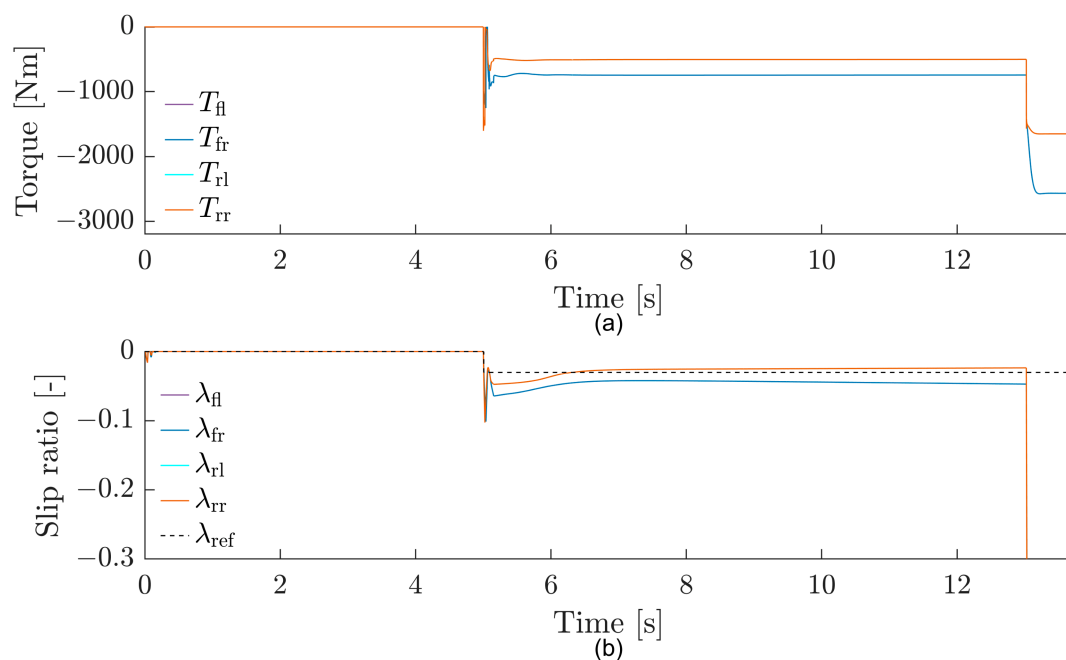


Figure 5. Proposed hybrid controller, braking from 100 km/h on a uniform low-friction surface ($\mu = 0.4$): (a) braking torque demand at each wheel; (b) longitudinal slip ratio and reference.

The pattern observed at high friction is reproduced. The proposed controller tracks the reference more closely, with e_λ decreasing from 0.0314 to 0.0161 for the front and 0.0176 to 0.0069 for the rear axle, representing a decrease of 48.7% and 60.8% respectively.

4.3. Summary of Indicators

Table 4 collects indicators for both low- and high-friction surfaces.

Figure 6 shows the longitudinal velocity traces for all four cases. The braking distance differs by 2.3 m on the high-friction surface and 6.0 m on the low-friction surface.

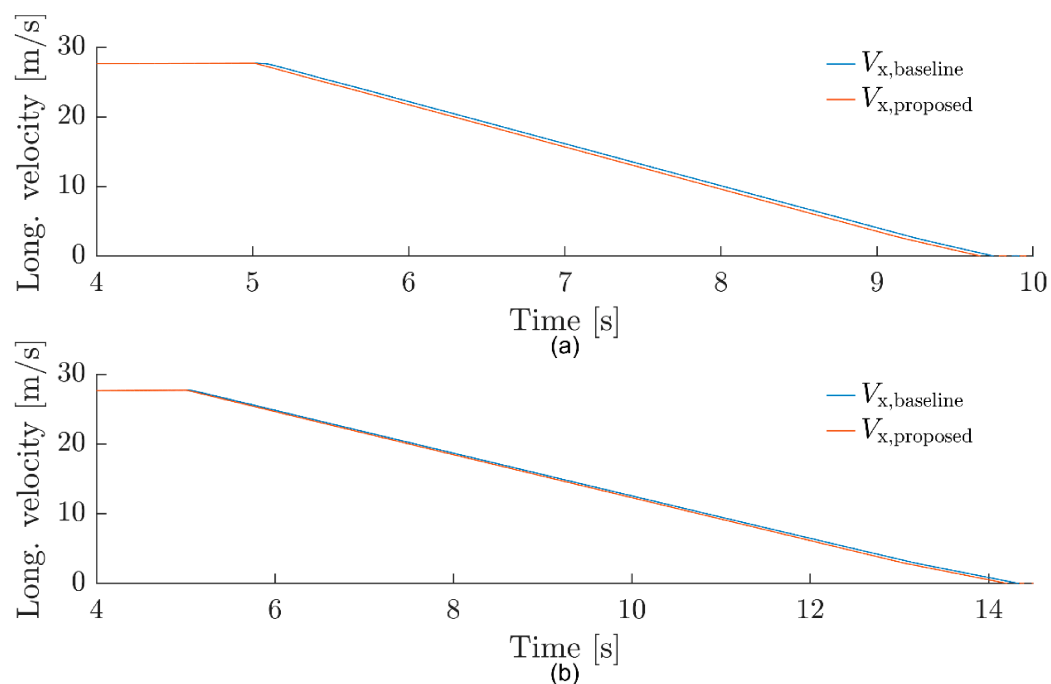


Figure 6. Longitudinal velocity during braking from 100 km/h: (a) high-friction surface, $\mu = 0.8$; (b) low-friction surface, $\mu = 0.4$. The baseline and proposed controllers are shown for each surface.

4.4. Braking Across a Friction Transition

Figures 7 and 8 show braking across a transition from $\mu = 0.8$ to $\mu = 0.4$ occurring at $t = 8.04$ s for the baseline and at $t = 8.23$ s for the proposed controller. After the transition, the recovery time to within 0.02 of the reference slip is 0.81 and 0.32 s under the baseline and 0.13 and 0.12 s under the proposed controller, with peak slips of -0.14 and -0.06 for high and low friction, respectively.

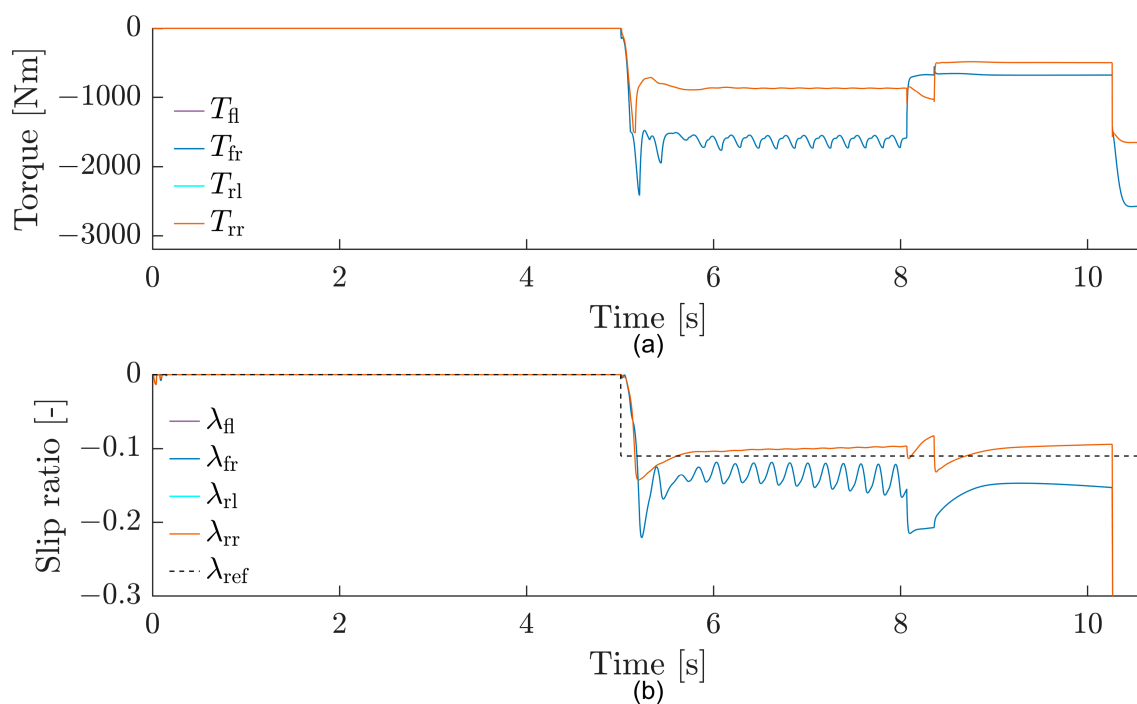


Figure 7. Baseline controller, braking from 100 km/h across a friction transition from $\mu = 0.8$ to $\mu = 0.4$: (a) braking torque demand at each wheel; (b) longitudinal slip ratio and reference.

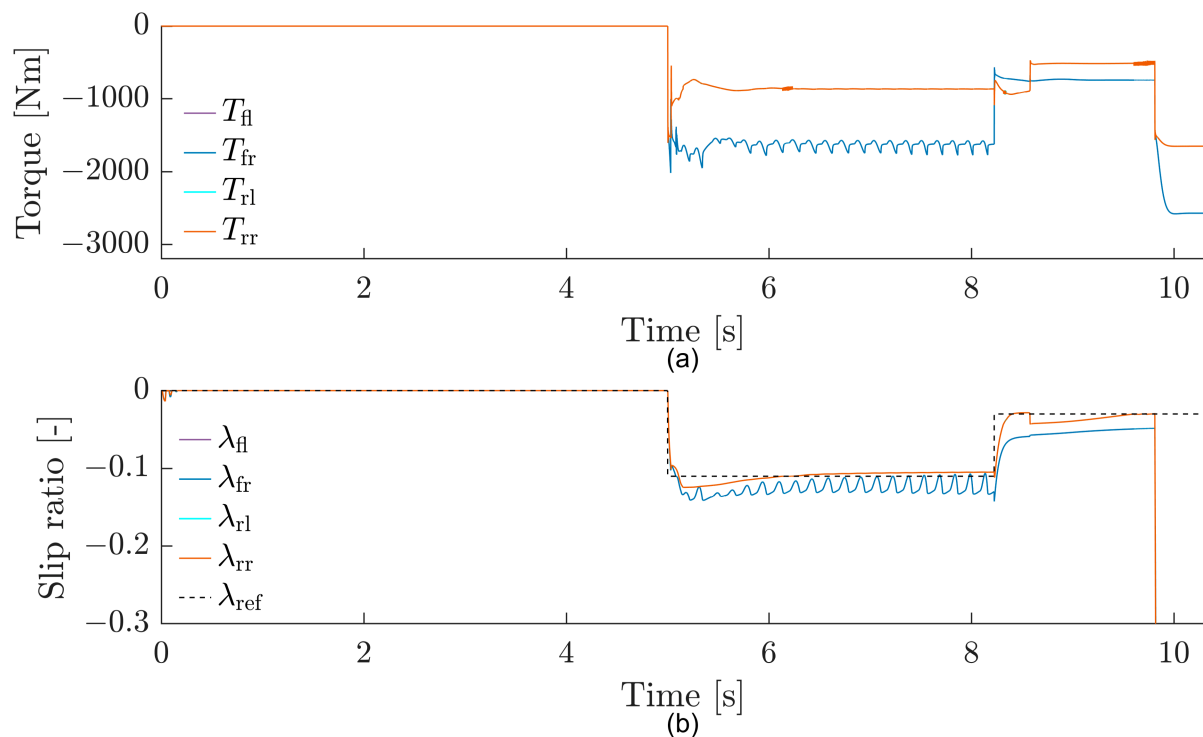


Figure 8. Proposed controller, braking from 100 km/h across a friction transition from $\mu = 0.8$ to $\mu = 0.4$: (a) braking torque demand at each wheel; (b) longitudinal slip ratio and reference.

4.5. Sensitivity to Vehicle Mass

Figures 9 and 10 show braking on $\mu = 0.8$ with the vehicle mass increased by 20% relative to Table 1, with controller parameters unchanged for the baseline and proposed controller, respectively.

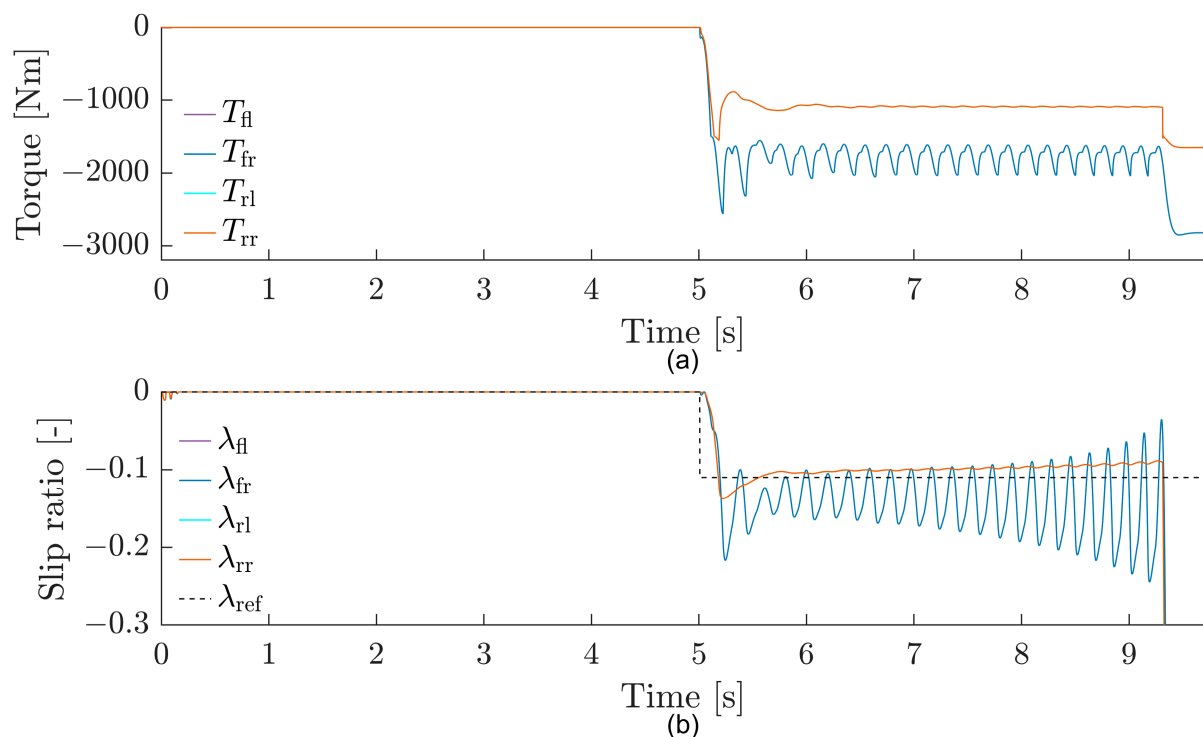


Figure 9. Baseline controller, braking from 100 km/h on $\mu = 0.8$ with vehicle mass increased by 20%: (a) braking torque demand at each wheel; (b) longitudinal slip ratio and reference.

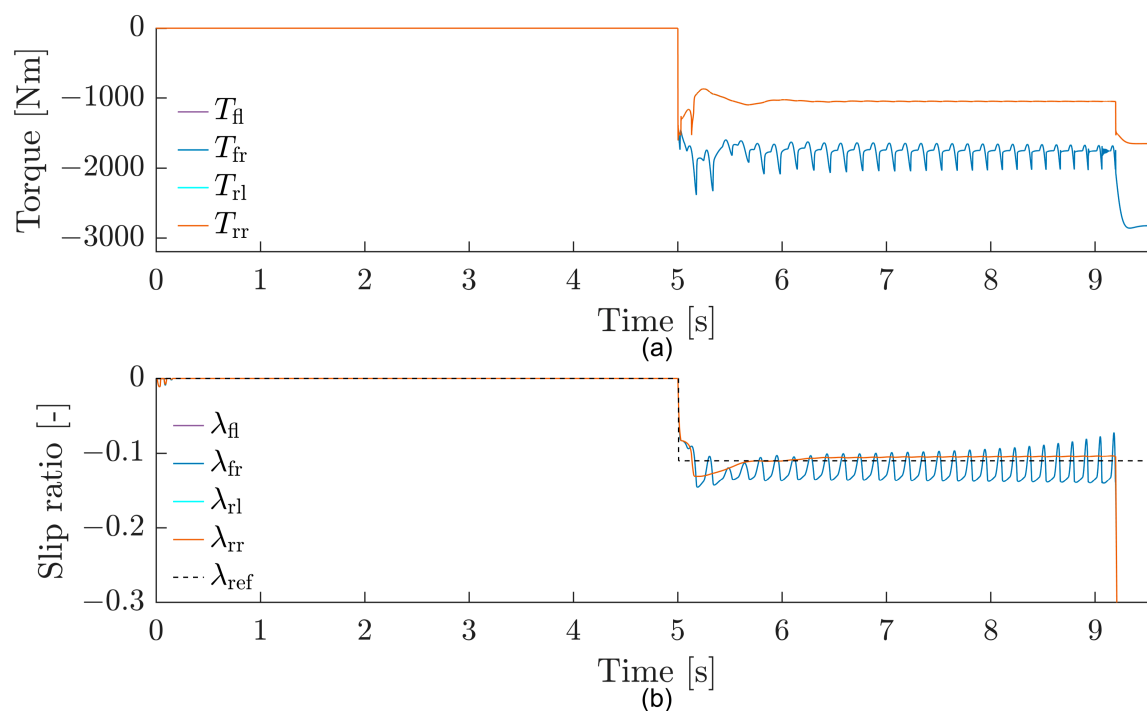


Figure 10. Proposed controller, braking from 100 km/h on $\mu = 0.8$ with vehicle mass increased by 20%: (a) braking torque demand at each wheel; (b) longitudinal slip ratio and reference.

Table 5 summarises the performance indicators for the friction-transition tests and the increased-mass tests. Again, all indicators show improved performance for the proposed controller compared to the baseline controller.

4.6. Sensitivity to Sensor Signal Delays

In a real-world application, sensor signal delays and noise are inevitable, and filtering is used to counteract noise. However, filtering introduces delay, which can disrupt the feedback control loop because the controller acts on a delayed estimate of the system's actual state. As part of the sensitivity analysis, we tested the proposed controller's response to sensor noise and signal filtering delays. Band-limited white noise with a power spectral density height of $10^{-7} \frac{\text{m}^2/\text{s}^2}{\text{Hz}}$, sampled at 0.001 s, was added to the measured longitudinal velocity signal, which was then filtered using a median filter with a sliding window of 10 samples, introducing a delay of 4.5 samples (4.5 ms). The results shown in Figure 11 suggest that the proposed controller performs similarly to the test without the signal noise and delays. The braking distance remains the same, 63.5 m, and e_λ changed very slightly from 0.0199 to 0.0168 for the front and 0.0166 to 0.0110 for the rear wheels.

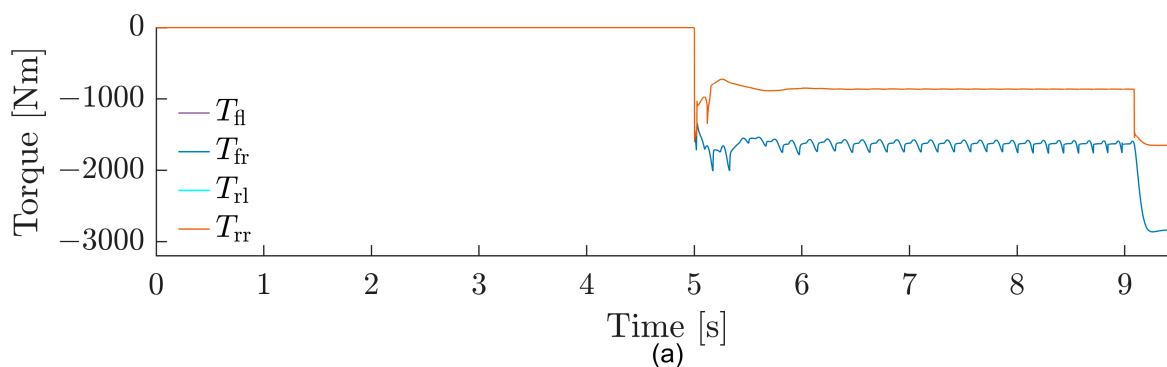


Figure 11. Cont.

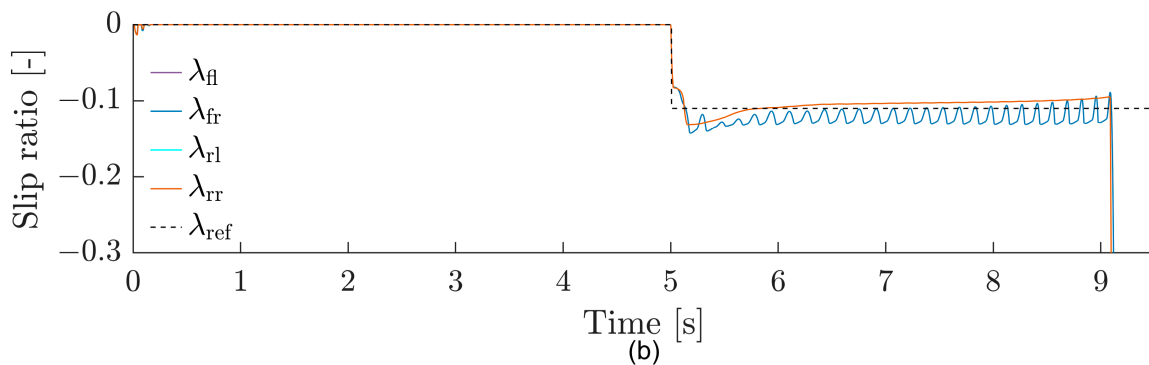


Figure 11. Proposed controller, braking from 100 km/h on $\mu = 0.8$ with sensor noise and delay: (a) braking torque demand at each wheel; (b) longitudinal slip ratio and reference.

5. Discussion and Conclusions

This paper presented a hybrid wheel slip controller in which a FOSM term and a PI term act on a common sliding surface defined by the slip-tracking error. The two terms blend using a weighting factor based on the hyperbolic tangent of the slip-tracking error, rather than switching at a fixed threshold. The controller was evaluated within a three-level brake-by-wire architecture that blends in-wheel motors and friction brakes, using an experimentally validated 14-DOF vehicle model that includes actuator dynamics (modelled as transfer functions).

Compared with the baseline controller, braking distances decreased by 2.2–6.0 m on the high- μ and low- μ surfaces, with the proposed controller achieving shorter distances. On this evidence, the proposed law should not be presented as improving stopping performance. Instead, the results show differences in longitudinal slip tracking quality and torque-demand smoothness.

The approach is closely related to hard-switching hybrid SMC/PI schemes, which change controller structure near the reference slip. Here, the transition is continuous rather than discrete, removing the discontinuity in torque demand at the switching instant.

Three conclusions follow. First, continuous weighting between a sliding-mode and a PI branch offers a workable alternative to a hard switch for wheel slip control: it retains the sliding-mode term's tracking behaviour without introducing a discontinuity in torque demand. Second, the benefit is slip tracking and torque smoothness, not braking distance. RMS slip error was reduced from 46% to 60.8% on different surfaces. Third, the controller remained robust to variations in vehicle mass, sensor delay, and noise.

Author Contributions: Conceptualization, V.I. and V.S.; Modelling, T.G.; Methodology, V.I. and V.S.; Software, T.G., P.K. and V.S.; Validation, P.K. and V.S.; Formal analysis, T.G., P.K., V.I. and V.S.; Investigation, T.G., P.K., V.I. and V.S.; Resources, P.K., V.I. and V.S.; Data curation, V.S.; Writing—original draft, T.G., P.K., V.I. and V.S.; Writing—review & editing, V.I. and V.S.; Supervision, V.I. and V.S. All authors have read and agreed to the published version of the manuscript.

Funding: This research received no external funding.

Institutional Review Board Statement: Not applicable.

Informed Consent Statement: Not applicable.

Data Availability Statement: The original contributions presented in this study are included in the article. Further inquiries can be directed to the corresponding author.

Conflicts of Interest: The authors declare no conflicts of interest.

Abbreviations

The following abbreviations are used in this manuscript:

ABS	Anti-lock braking system
ADAS	Advanced driver assistance systems
BbW	Brake-by-wire
BC	Brake caliper
CoG	Centre of gravity
DOF	Degrees of freedom
EBA	Emergency braking assist
EHB	Electrohydraulic brake
EMB	Electromechanical brake
FL, FR, RL, RR	Front left, front right, rear left, rear right
FOSM	First-order sliding mode
IWM	In-wheel motor
K&C	Kinematics and compliance
LQR	Linear quadratic regulator
MPC	Model predictive control
OBM	On-board motor
PI	Proportional–integral
PID	Proportional–integral–derivative
SMC	Sliding-mode control
SoC	State of charge
SUV	Sport utility vehicle
VSC	Vehicle stability control
WSC	Wheel slip control
YMC	Yaw moment control

References

- Oubelaid, A.; Belkhier, Y.; Bouddou, R.; Khosravi, N. A State-of-the-Art Review of Sustainable Electrified Vehicles: Architectures, Hybrid Energy Storage Integration, Energy Management and Key Challenges. *Energy Rep.* **2026**, *16*, 109493. [\[CrossRef\]](#)
- Kojis, P.; Danilevičius, A.; Šabanovič, E.; Skrickij, V. The Second Generation of Electric Vehicles: Integrated Corner Solutions. In *TRANSBALTICA XII: Transportation Science and Technology*; Prentkovskis, O., Yatskiv, I., Skačkauskas, P., Junevičius, R., Maruschak, P., Eds.; Springer International Publishing: Cham, Switzerland, 2022; pp. 87–100. [\[CrossRef\]](#)
- Oliveira, T.R.; Hsu, L.; Nunes, E.V.L. Smooth Sliding Control to Overcome Chattering Arising in Classical SMC and Super-Twisting Algorithm in the Presence of Unmodeled Dynamics. *J. Frankl. Inst.* **2022**, *359*, 1235–1256. [\[CrossRef\]](#)
- Boiko, I.; Fridman, L. Analysis of Chattering in Continuous Sliding-Mode Controllers. *IEEE Trans. Autom. Control* **2005**, *50*, 1442–1446. [\[CrossRef\]](#)
- Fridman, L.M. An Averaging Approach to Chattering. *IEEE Trans. Autom. Control* **2001**, *46*, 1260–1265. [\[CrossRef\]](#)
- Obeid, H.; Fridman, L.M.; Laghrouche, S.; Harmouche, M. Barrier Function-Based Adaptive Sliding Mode Control. *Automatica* **2018**, *93*, 540–544. [\[CrossRef\]](#)
- Obeid, H.; Laghrouche, S.; Fridman, L.; Chitour, Y.; Harmouche, M. Barrier Function-Based Adaptive Super-Twisting Controller. *IEEE Trans. Autom. Control* **2020**, *65*, 4928–4933. [\[CrossRef\]](#)
- Laghrouche, S.; Harmouche, M.; Chitour, Y.; Obeid, H.; Fridman, L.M. Barrier Function-Based Adaptive Higher Order Sliding Mode Controllers. *Automatica* **2021**, *123*, 109355. [\[CrossRef\]](#)
- González, A.; Ovalle, L.; Fridman, L. Final Set Adjustment in Barrier Function Adaptation Exploiting Properties of Signed Power-Based Controllers. *IEEE Trans. Autom. Control* **2024**, *69*, 5486–5491. [\[CrossRef\]](#)
- González, A.; Ovalle, L.; Fridman, L. Is It Reasonable to Implement Barrier Function Adaptation Under Sensor Noise? *IEEE Trans. Control Syst. Technol.* **2025**, *33*, 1627–1639. [\[CrossRef\]](#)
- Ovalle, L.; Gonzalez, A.; Fridman, L.; Laghrouche, S.; Obeid, H. Analysis of Barrier Function Based Adaptive Sliding Mode Control in the Presence of Deterministic Noise. *Automatica* **2025**, *171*, 111946. [\[CrossRef\]](#)
- Ovalle, L.; Gonzalez, A.; Fridman, L.; Haimovich, H. Discrete Implementations of Sliding-Mode Controllers with Barrier-Function Adaptations Require a Revised Framework. *Automatica* **2026**, *185*, 112797. [\[CrossRef\]](#)

13. Grikenis, T.; González, A.; Kojis, P.; Fridman, L.; Skrickij, V. Sliding-Mode Controllers Implementation for Direct Yaw Moment Control in Four-Wheel Steered Ground Vehicles. *Control Eng. Pract.* **2026**, *175*, 107069. [\[CrossRef\]](#)
14. Savitski, D. Robust Control of Brake Systems with Decoupled Architecture. Ph.D. Thesis, Technische Universität Ilmenau, Ilmenau, Germany, 2019.
15. Babyak, M.; Keršys, R.; Neduzha, L. Improving the dependability evaluation technique of a transport vehicle. In Proceedings of the Transport Means, Palanga, Lithuania, 30 September–2 October 2020.
16. Skrickij, V.; Kojis, P.; Šabanovič, E.; Shyrokau, B.; Ivanov, V. Review of Integrated Chassis Control Techniques for Automated Ground Vehicles. *Sensors* **2024**, *24*, 600. [\[CrossRef\]](#)
17. Bondarenko, I.; Severino, A.; Olayode, I.O.; Campisi, T.; Neduzha, L. Dynamic Sustainable Processes Simulation to Study Transport Object Efficiency. *Infrastructures* **2022**, *7*, 124. [\[CrossRef\]](#)
18. Ji, Q.; Zheng, L.; Bi, Y.; Pang, H. Review of Brake-by-Wire Technology for Low-Speed and Autonomous Vehicles. *World Electr. Veh. J.* **2024**, *15*, 581. [\[CrossRef\]](#)
19. Heydrich, M.; Ricciardi, V.; Ivanov, V.; Mazzoni, M.; Rossi, A.; Buh, J.; Augsburg, K. Integrated Braking Control for Electric Vehicles with In-Wheel Propulsion and Fully Decoupled Brake-by-Wire System. *Vehicles* **2021**, *3*, 145–161. [\[CrossRef\]](#)
20. Jonner, W.-D.; Winner, H.; Dreilich, L.; Schunck, E. Electrohydraulic Brake System—The First Approach to Brake-By-Wire Technology. In Proceedings of the International Congress & Exposition, Detroit, MI, USA, 26 February 1996. [\[CrossRef\]](#)
21. Savaresi, S.M.; Tanelli, M. *Active Braking Control Systems Design for Vehicles*; Advances in Industrial Control; Springer: London, UK, 2010.
22. Ganzel, B. Slip Control Boost Braking System. U.S. Patent 9,221,443, 29 December 2015.
23. Soomro, M.; Hassan, M.K.; Mustafa Abro, G.E.; Marimuthu, M. Electronic Wedge Brakes: A Review of Design, Modeling, Control Strategies, and Testing Methodologies. *IEEE Access* **2025**, *13*, 67378–67399. [\[CrossRef\]](#)
24. Li, Y.; Shim, T.; Shin, D.-H.; Lee, S.; Jin, S. Control System Design for Electromechanical Brake System Using Novel Clamping Force Model and Estimator. *IEEE Trans. Veh. Technol.* **2021**, *70*, 8653–8668. [\[CrossRef\]](#)
25. Choi, C.; Lee, K.; Lee, W. Observer-Based Phase-Shift Fault Detection Using Adaptive Threshold for Rotor Position Sensor of Permanent-Magnet Synchronous Machine Drives in Electromechanical Brake. *IEEE Trans. Ind. Electron.* **2015**, *62*, 1964–1974. [\[CrossRef\]](#)
26. Yu, L.; Ma, L.; Song, J.; Liu, X. Magnetorheological and Wedge Mechanism-Based Brake-by-Wire System with Self-Energizing and Self-Powered Capability by Brake Energy Harvesting. *IEEE/ASME Trans. Mechatron.* **2016**, *21*, 2568–2580. [\[CrossRef\]](#)
27. Cheon, J.S.; Kim, J.; Jeon, J. New Brake By Wire Concept with Mechanical Backup. *SAE Int. J. Passeng. Cars—Mech. Syst.* **2012**, *5*, 1194–1198. [\[CrossRef\]](#)
28. Chen, P.; Wu, J.; Zhao, J.; He, R.; Liu, H.; Zhang, K. Design and Position Control of a Novel Electric Brake Booster. *SAE Int. J. Passeng. Cars—Mech. Syst.* **2018**, *11*, 389–400. [\[CrossRef\]](#)
29. Gong, X.; Qian, L.; Ge, W.; Wang, L. Research on the Anti-Disturbance Control Method of Brake-by-Wire Unit for Electric Vehicles. *World Electr. Veh. J.* **2019**, *10*, 44. [\[CrossRef\]](#)
30. Pretagostini, F.; Ferranti, L.; Berardo, G.; Ivanov, V.; Shyrokau, B. Survey on Wheel Slip Control Design Strategies, Evaluation and Application to Antilock Braking Systems. *IEEE Access* **2020**, *8*, 10951–10970. [\[CrossRef\]](#)
31. Ivanov, V.; Savitski, D.; Shyrokau, B. A Survey of Traction Control and Antilock Braking Systems of Full Electric Vehicles with Individually Controlled Electric Motors. *IEEE Trans. Veh. Technol.* **2015**, *64*, 3878–3896. [\[CrossRef\]](#)
32. Qasem, G.A.A.; Abdullah, M.F.; Farid, M.; Bakhuraisa, Y.A. Enhancing Anti-Lock Braking System Performance Using Fuzzy Logic Control Under Variable Friction Conditions. *Symmetry* **2025**, *17*, 1692. [\[CrossRef\]](#)
33. Kanungo, A.; Kumar, P.; Gupta, V.; Salim; Saxena, N.K. A Design an Optimised Fuzzy Adaptive Proportional-Integral-Derivative Controller for Anti-Lock Braking Systems. *Eng. Appl. Artif. Intell.* **2024**, *133*, 108556. [\[CrossRef\]](#)
34. Abreu, R.; Botha, T.R.; Hamersma, H.A. Model-Free Intelligent Control for Antilock Braking Systems on Rough Roads. *SAE Int. J. Veh. Dyn. Stab. NVH* **2023**, *07*, 269–285. [\[CrossRef\]](#)
35. Ivanov, V.G.; Algin, V.B.; Shyrokau, B.N. Intelligent Control for ABS Application with Identification of Road and Environmental Properties. *Int. J. Veh. Auton. Syst.* **2006**, *4*, 44. [\[CrossRef\]](#)
36. Zhu, S.; Fan, X.; Wang, P.; Chen, X. Design of ABS Fuzzy Sliding Mode Control System Based on Pavement Recognition. *Trans. Can. Soc. Mech. Eng.* **2023**, *47*, 436–447. [\[CrossRef\]](#)
37. Zhang, H.; Zhang, C.; Xu, L.; Jia, Z. Slip Ratio Control Based on Adaptive Fuzzy Sliding Mode for Vehicle with an Electromechanical Brake System. *Mech. Ind.* **2024**, *25*, 15. [\[CrossRef\]](#)
38. Johansen, T.A.; Petersen, I.; Slupphaug, O. Explicit Sub-Optimal Linear Quadratic Regulation with State and Input Constraints. *Automatica* **2002**, *38*, 1099–1111. [\[CrossRef\]](#)

39. Toumi, N.; Aldeeb, A.N.; Magtoun, A.M.A. H-Infinity Framework with Nonlinear Validation: A Mixed-Sensitivity Approach to Robust Wheel Slip Regulation Using H-Infinity Control for Variable Road Conditions. In *Proceedings of the 2026 IEEE 5th International Maghreb Meeting of the Conference on Sciences and Techniques of Automatic Control and Computer Engineering (MI-STA)*; IEEE: Sebha, Libya, 2026; pp. 1–6.
40. Wang, H.; Yan, Z.; Pi, D.; Yan, Y.; Wang, Y.; Fan, D.; Zhang, C.; Wang, Y. Hierarchical ABS Control Based on MPC Slip Rate Tracking. *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.* **2026**, *240*, 7339–7348. [[CrossRef](#)]
41. Arricale, V.M.; Genovese, A.; Tomar, A.S.; Kural, K.; Sakhnevych, A. Nonlinear Model of Predictive Control-Based Slip Control ABS Including Tyre Tread Thermal Dynamics. *Appl. Mech.* **2022**, *3*, 855–888. [[CrossRef](#)]
42. Mayne, D.Q.; Kerrigan, E.C.; Van Wyk, E.J.; Falugi, P. Tube-based Robust Nonlinear Model Predictive Control. *Int. J. Robust Nonlinear Control* **2011**, *21*, 1341–1353. [[CrossRef](#)]
43. Oldewurtel, F.; Jones, C.N.; Morari, M. A Tractable Approximation of Chance Constrained Stochastic MPC Based on Affine Disturbance Feedback. In *Proceedings of the 2008 47th IEEE Conference on Decision and Control*; IEEE: Cancun, Mexico, 2008; pp. 4731–4736. [[CrossRef](#)]
44. Amadini, M.; Milivinti, M.; Mastinu, G.; Gobbi, M. ABS Based on Sliding-Mode Control and on Force and Moment Measurement. *IEEE Access* **2025**, *13*, 70581–70597. [[CrossRef](#)]
45. Munphano, E.B.; Tewa, A.K.S.; Jacques, B.E.J. Flexible Super Twisting Adaptive Robust Control for Fractional Antilock Braking System. *Discov. Veh.* **2026**, *2*, 18. [[CrossRef](#)]
46. He, L.; Ye, W.; He, Z.; Song, K.; Shi, Q. A Combining Sliding Mode Control Approach for Electric Motor Anti-Lock Braking System of Battery Electric Vehicle. *Control Eng. Pract.* **2020**, *102*, 104520. [[CrossRef](#)]
47. Heydrich, M.; Kellner, B.; Ivanov, V. Methodologic Assessment of Brake-by-Wire System Modelling with Regard to Accuracy, Model Complexity and Optimisation Efforts. In *Proceedings of the WCX SAE World Congress Experience 288442*, Detroit, MI, USA, 11 April 2023; p. 2023–01–0666.
48. Šabanovič, E.; Kojis, P.; Šukevičius, Š.; Shyrokau, B.; Ivanov, V.; Dhaens, M.; Skrickij, V. Feasibility of a Neural Network-Based Virtual Sensor for Vehicle Unsprung Mass Relative Velocity Estimation. *Sensors* **2021**, *21*, 7139. [[CrossRef](#)]
49. Vaseur, C.; Van Aalst, S. *Test Results at Ford Lommel Proving Ground ESR 11 (Version Final) [Data Set]*; Interdisciplinary Training Network in Multi-Actuated Ground Vehicles: Geneva, Switzerland, 2019.
50. Park, K.; Joa, E.; Yi, K.; Yoon, Y. Rear-Wheel Steering Control for Enhanced Steady-State and Transient Vehicle Handling Characteristics. *IEEE Access* **2020**, *8*, 149282–149300. [[CrossRef](#)]
51. Šabanovič, E.; Žuraulis, V.; Prentkovskis, O.; Skrickij, V. Identification of Road-Surface Type Using Deep Neural Networks for Friction Coefficient Estimation. *Sensors* **2020**, *20*, 612. [[CrossRef](#)]

Disclaimer/Publisher’s Note: The statements, opinions and data contained in all publications are solely those of the individual author(s) and contributor(s) and not of MDPI and/or the editor(s). MDPI and/or the editor(s) disclaim responsibility for any injury to people or property resulting from any ideas, methods, instructions or products referred to in the content.