

Estimation of Brake Friction Coefficient for Blending Function of Base Braking Control

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ABSTRACT

The brake architecture of hybrid and full electric vehicle includes the distinctive function of brake blending. Known approaches draw upon the maximum energy recuperation strategy and neglect the operation mode of friction brakes. Within this framework, an efficient control of the blending functions is demanded to compensate external disturbances induced by unpredictable variations of the pad disc friction coefficient. In addition, the control demand distribution between the conventional frictional brake system and the electric motors can incur failures that compromise the frictional braking performance and safety. However, deviation of friction coefficient value given in controller from actual one can induce undesirable deterioration of brake control functions. The main objective of the presented study is to propose a method to compensate disturbances induced by variations of brake linings friction coefficient through modifications of the brake torque demand for the enhancement of both brake performance and active safety.

The achievement of a compensation mechanism requires the estimation of relevant vehicle states. Hereunto, a novel technique based on a linear Kalman observer is proposed for the online estimation of the brake friction coefficient by relying upon the wheel speed sensors and inertia measurement unit (IMU). Such a tool enables a more efficient use of the frictional brakes aimed at minimizing losses of friction coefficient by keeping them in the optimal operational conditions. A simulation analysis will be carried out using the commercial vehicle dynamics simulation software IPG CarMaker to test the functionality of the developed estimator in the real-time mode. Experimental results from brake dynamometric test rig will be considered in the vehicle dynamics simulation software to reproduce the real behaviour of brake linings friction coefficient. The resulting improvements in brake control functions will be analysed against longitudinal base braking cases involving blending functions also in presence of failure of the electric motors.

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INTRODUCTION

In the recent years, the decoupled electrohydraulic brake system (EHB) has had a wide application in automotive brake design [1] [2] [3]. The lack of a direct hydraulic connection between the brake pedal and brake callipers guarantees additional degrees of freedom in the brake pressure actuation and reveals necessary for the proper enabling and control of the brake blending function in electric vehicles [4]. Whilst the development of strategies for brake torque distribution in electric vehicle is a widely known research topic [5] [6], the compensation of disturbances during base brake manoeuvres is still a poorly investigated function. The external disturbances induced by variations of brake linings coefficient of friction can lead to unpredictable vehicle behaviour, which in turns affects the brake blending control process. Therefore, it is expectable that the disturbance compensation through the variation of the brake pressure actuation leads to the enhancement of braking performance and vehicle safety.

Indeed, in braking system control algorithms, the knowledge of brake linings coefficient of friction plays a crucial role. Deviation of its value given in controller from actual one can induce undesirable deterioration

of brake control functions. A large set of phenomena occurring between the brake pad and disc affects the brake effectiveness and the dynamics of the pad-disc contact causing a wide variation of the friction coefficient: brake fading [7], [8], brake bedding [9], brake hysteresis against the pressure [10], brake hysteresis against the speed [11], brake wear [12] and aging [9], dependence against the environmental conditions [13] etc. The knowledge of the actual brake friction coefficient results in an accurate control of the frictional braking torque and allows the enhancement of both base brake functions, brake blending and regenerative braking control. In addition, the knowledge of the brake friction coefficient reveals also useful to guarantee failsafe operation and maintain the brake in the safe operating region by avoiding the fading.

Unfortunately, the influence of the previously listed phenomena on frictional interaction between brake disc and pad cannot be evaluated in real-time mode due to the complexity of known model-based approaches. Therefore, the present work proposes an estimator of the brake linings coefficient of friction that draws upon a linear Kalman filter of the wheel torque. The Kalman filter is widely used as system state observer because it ensures good robustness for tolerating

differences between models and real dynamics of the vehicle, variations of model parameters and signal errors [14]. In its intent, the state observer outperforms classical analytical methods and reveals suitable for online applications by guaranteeing robustness of the estimation against changing road conditions and reliability against changes of the plant characteristics due to brake wear and aging, or in case of replacement with aftermarket brake linings. Afterwards, a compensation control function is developed in order to mitigate the external disturbances that affect the brake blending performance and driving safety of electric vehicles during base braking control.

The developed observer and estimator are assessed against a reference vehicle model from the commercial software IPG CarMaker. Owing to the absence of an embedded model for brake friction generation, experimental results from brake dynamometric test rig are considered in the vehicle dynamics simulation software to reproduce the real behaviour of the brake linings coefficient of friction. The simulations are performed for an electric vehicle equipped with an electrohydraulic brake system and individual in-wheel electric motors. The transition from the friction braking to the electric braking or vice-versa has a remarkable impact on the brake performance. Its analysis reveals necessary in case of disabled regenerative braking function (e.g. fully charged battery) or failure occurrence. It is expectable that the knowledge of the brake friction coefficient enables an optimal intervention control of the conventional brake and a better monitoring of its working conditions. The obtained results represent functionality of the developed estimator in real-time mode.

The paper comprises the following parts. In the next section the controller and its functional blocks are presented. In the second section the selected method for state estimation is put forth and validated against the vehicle dynamics simulator for different driving manoeuvres. In the third section, the estimation method for the brake linings coefficient of friction is put forth and validated. At last, the procedures for compensation of disturbances induced by variations in the coefficient of friction are illustrated for different blending configurations.

CONTROL STRUCTURE

The employed controller is suited to a full electric vehicle equipped with individual in-wheel electric motors (IWM) and an electrohydraulic brake system (EHB) that cooperate to perform braking actions. The structure of the employed control is schematically reported in the Figure 1. From now on, the i index will be used to identify the four tyres: fl, fr, rl, rr . The hat symbol "^" will be instead used to represent an estimated or observed quantity. A reference vehicle deceleration map [15], as a function of the brake pedal travel s_{pedal} , is used in the Base Brake Controller in order to derive the demanded vehicle braking torque T_{dem} . The Brake Blending block is liable for the coordination between the friction brake system and electric motors by providing the demanded torque signals, respectively $T_{brake_FR_i_dem}$ and $T_{brake_EM_i_dem}$, to achieve the demanded deceleration. Thanks to the feedforward configuration of the control system, once the load distribution is defined, it results trivial to evaluate the effective regenerative brake component $T_{brake_EM_i}$ and the frictional brake torque $T_{brake_FR_i}$ for each wheel. Afterwards, a linear Kalman filter is liable for the observation of the wheel torque $\hat{T}_{wheel_i}$ that in turns is necessary to estimate the brake linings coefficient of friction $\hat{\mu}_{brake_i}$. The observer of the wheel torque relies on measurement coming from the vehicle sensors, viz. wheels speed ω_i , longitudinal acceleration a_x and lateral acceleration a_y , and on-board estimates of the tyres vertical loads $\hat{F}_{z_i}$, longitudinal slips $\hat{s}_{L_i}$ and side slips $\hat{s}_{S_i}$.

Brake friction losses owing to brakes warming up and fading occurrence can arise after a set of brake applications. The compensation of disturbances induced by friction coefficient variation is realized by updating the brake torque demand. The estimated brake linings friction coefficient is used in the calculation procedure for the conversion of the brake torque demand of frictional brakes to the pressure demand, which is then sent to the EHB system. The inputs of the controller are signals from the set of sensors measuring the wheel speeds, electric motor current, brake pedal position, vehicle longitudinal and lateral acceleration and the brake pressure received from the EHB control unit. An accurate description of each functional block is provided in the following sections.

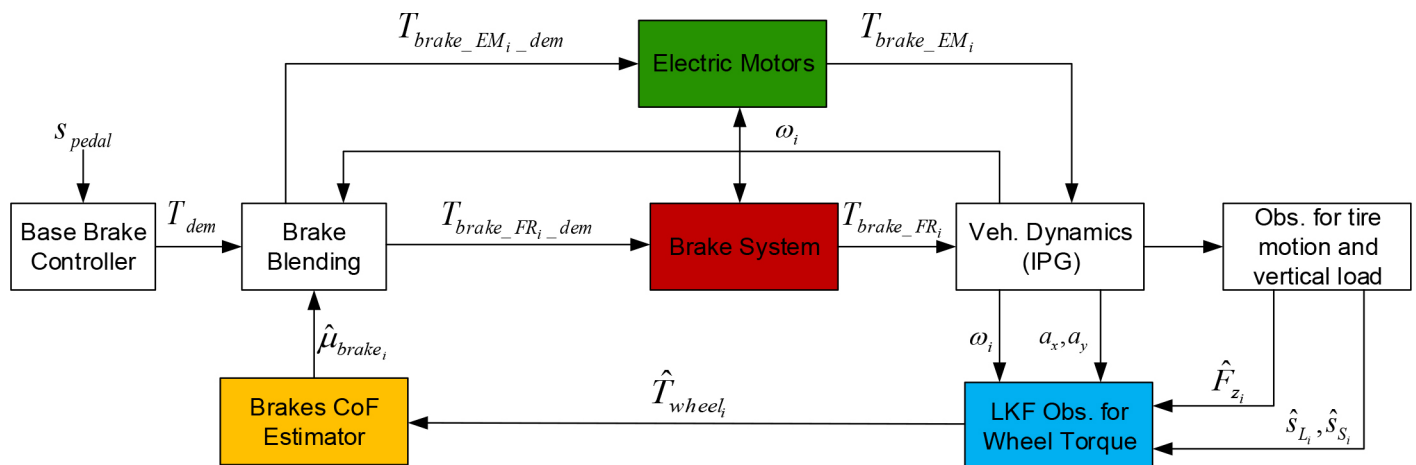


Figure 1. Overall scheme of the implemented controller with enabled blending function and friction compensation during base braking. The main blocks are reported in colours and associated detailed descriptions can be found in the text.

Blending Function of Base Brake Control

The base-brake controller generates the torque demand to achieve the reference deceleration. The value of reference deceleration can be extracted from the brake pedal displacement sensor in accordance with the [Figure 2](#), so that the torque demand can be calculated in accordance with [eq. \(1\)](#).

$$T_{dem} = m_v \cdot a_x^{ref}(s_{pedal}) \cdot r_w \quad (1)$$

Wherein, m_v represents the mass of the vehicle, a_x^{ref} is the reference deceleration as function of the brake pedal displacement s_{pedal} , and r_w represent the wheel radius. The brake blending factor α is defined in order to allocate the demanded intervention level between the conventional brake system $T_{brake_FR_dem}$ and the electric motors $T_{brake_EM_dem}$, in compliance with the [Equation \(2\)](#).

$$T_{dem} = \alpha \cdot T_{brake_EM_dem} + (1 - \alpha) \cdot T_{brake_FR_dem} \quad (2)$$

The defined torque demands are used to generate the control signals, whose actuation must undergo the physical system limits of both the frictional brakes and electric motors. It is worth noticing that limitations in the gradient of generation of wheel brake torque for both the electrohydraulic and electric brake system must be considered. The rate limit of pressure build-up towards the brake callipers is simulated with a second order transfer function as reported in [\[16\]](#). Instead, the rate limit affecting the generation of the regenerative torque is rendered by means of a first order transfer function similarly to [\[4\]](#). The knowledge of the estimated brake linings coefficient of friction allows calculating a brake pressure control signal. The torque exerted by the electric motors, generally equal to the demanded value, must comply a threshold value depending on the motor rotational speed and battery state of charge (SOC).

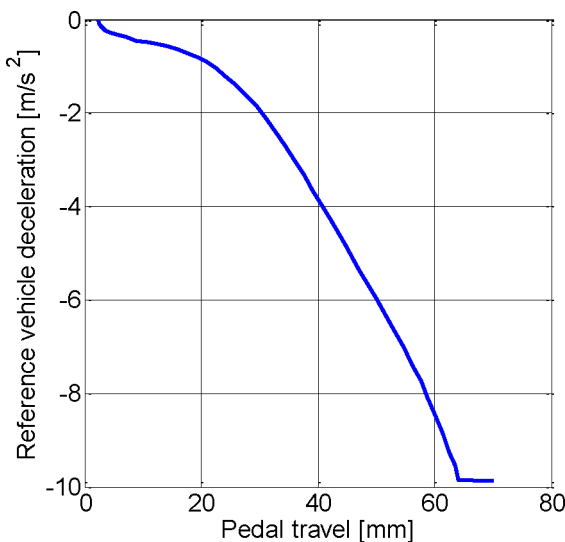


Figure 2. Reference vehicle deceleration map implemented in the Base Brake Controller

MODEL SIMULATION

In the present paper, the wheel torque observer and the friction coefficient estimator are assessed against a baseline vehicle simulated in IPG CarMaker. The Pacejka Magic formula tire model [\[17\]](#) is employed to depict the road-vehicle contact forces with effectiveness and low computational burden. The model includes also aerodynamic forces, nonlinear suspension model, steering system, decoupled electrohydraulic brake system. Owing to the absence of an embedded model for brake friction generation, the dynamic model [\[18\]](#) is used to reproduce the real behaviour of brake linings friction coefficient in the vehicle dynamics simulation software, previously calibrated against the experimental results obtained from brake dynamometric test rig at the Technische Universität Ilmenau.

Vehicle Model

The simulation results concern a full electric vehicle with all-wheel drive powertrain including individual electric in-wheel motors and decoupled electrohydraulic brake system. The vehicle parameters are reported in the [Table 1](#).

Table 1. Vehicle configuration

Vehicle type	Passenger full electric vehicle
Total vehicle mass	1463 kg
Powertrain type	AWD
Drivetrain type	Individual in-wheel type motors without reduction gear
Brake System type	Decoupled electrohydraulic
Tyre size	195/65 R15

Model of Electric Vehicle Powertrain

For the purpose of the study here presented, the electric powertrain includes individual in-wheel electric motors with possibility of enabling the regenerative braking mode with controllable power split between conventional and electric systems. The behaviour of the electric motors during the braking phases is modelled through a look-up table emulating the steady-state behaviour and a first-order transfer function for reproducing the transient behaviour.

Model of the Brake System

The real behaviour of brake linings friction coefficient in the vehicle dynamics simulation software is modelled through the Ostermeyer model [\[18\]](#). This latter is based on two differential equations in the friction and temperature states and captures the dynamical behaviour of the friction and its dependence against the temperature. This encompasses the fading and the hysteretic effect of the brake system. As reported in [\[18\]](#), the model relies upon the following set of differential equations

$$\dot{\mu} = -[b \cdot (Nv + \varepsilon)\mu - a \cdot T] \quad (3)$$

$$\dot{T} = -c[(T - T_0) - \gamma \cdot Nv] \quad (4)$$

For a more detailed description of the Ostermeyer model, please refer to [18]. Herein it is worth remarking that: (i) the group (Nv) represents the magnitude of the combination between the normal load and the speed; (ii) $a, b, c, \gamma, \varepsilon$ are constant parameters linked to the pad chemical composition to be calibrated against experimental data; (iii) μ represent the pad-disc friction coefficient; (iv) T stands for the disc contact temperature, where T_0 is its initial state.

An assessment of the Ostermeyer's model presented here, consisting in its identification and validation against the experimental data acquired on the brake rig at the Technische Universität Ilmenau [19]. The acquired data correspond to the two tests reported in the Figure 3 for fixed constant sliding speed and oscillating input brake pressure.

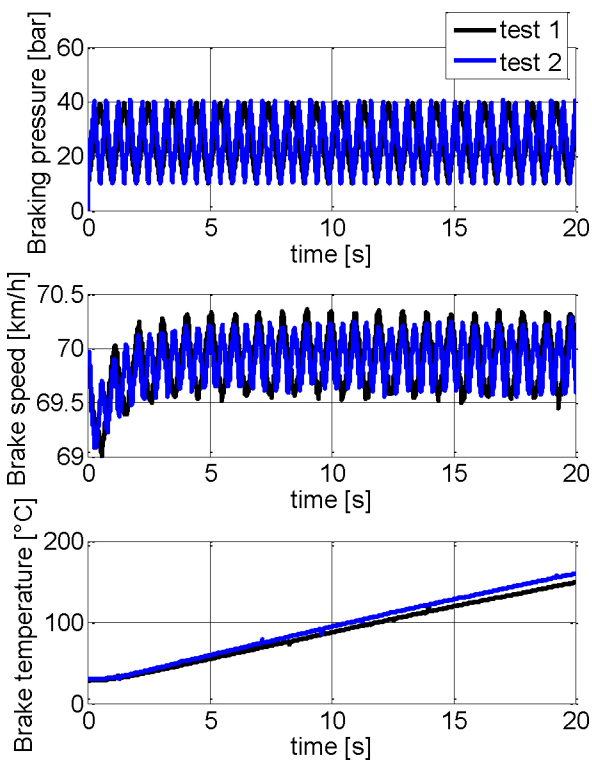


Figure 3. Experimental test conducted on the brake test rig at the Technische Universität Ilmenau whereby the test 2 features a double operating frequency compared to test 1.

During such a testing procedure, the brakes undergo increasing temperature, which significantly affects the brake performance. The dynamic model represented by the Equation (3) and (4) has been firstly identified against the first set of experimental data (Test 1) and then validated against the second data set (Test 2). The fitting was achieved by solving an optimization problem whose input variables are the model parameters $a, b, c, \gamma, \varepsilon$. To this regard, a minimization algorithm based on the Newton's method has been applied [20].

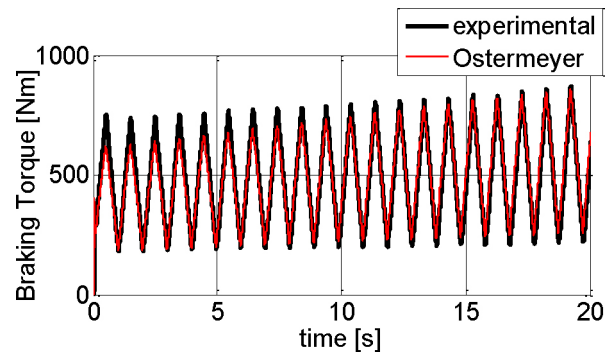


Figure 4. Identification result of the Ostermeyer's model against the Test 1 data set.

In Figure 4, the comparison between the experimental and the model predicted braking torque is reported. It is noticeable that, even in presence of increasing temperature value, the simulated profile does not depart from the reference. This confirms that the model under discussion takes into account the friction dependence against the temperature.

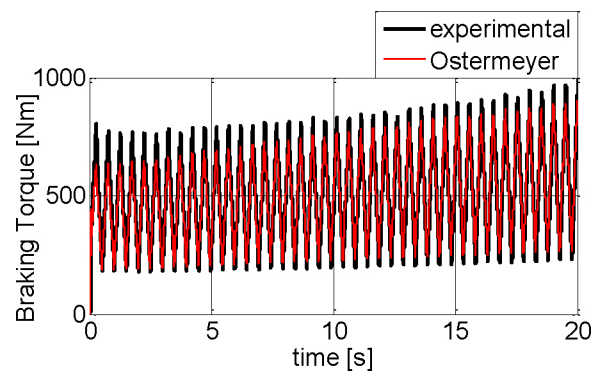


Figure 5. Validation result of the Ostermeyer's model against the Test 2 data set.

In the Figure 5 is instead reported the validation test performed against the second data set. Although a lack of accuracy is noticeable in correspondence of low temperature values (first half of the profile in diagram), the model exhibits a good overall prediction capability even in presence of increased pressure actuation frequency. Moreover, the model reveals capable to catch an increase of the average brake torque induced by the brake warming up.

WHEEL TORQUE OBSERVER

The literature research effort of recent years has demonstrated that the state observation technique in all the engineering fields represents a valid substitute to the model based approach. The observation of the tire-road friction coefficient [21], [22], [23] and tire forces [24], [25] has been topic of great interest among the vehicle dynamics control community. What is common among these methods is that available vehicle sensors are used to estimate tire/road information based on the state variables of the wheels and the vehicle. In comparison to the above listed methodologies, the represented observer features an augmented state vector based on the wheels dynamics and a tyre model. Hence, the state space formulation associated with the observer is rearranged in order to handle the wheel torque as an augmented state [26] and guarantee the observability condition [27].

Estimation Process

The observer state-space formulation stems from a quarter-car model for the vehicle longitudinal and lateral dynamics with fixed parameters. With reference to Figure 6, the proposed tool relies on the signals coming from the chassis accelerometer and the wheel speed sensors, reported as blue blocks, and on three units, represented as orange blocks. The KF combines information coming from models (prediction step) and sensors (update step), in presence of uncertainty, to provide a minimum variance estimation of the system state variables.

The filter state, measurement and input vectors are defined in the eq. (5), (6), (7) for the generic time step k . In addition, the dynamics of the state variables is derived in compliance with the discrete state space set of equations (8a)-(8d) as follows:

$$x_k = \{\omega_{i,k}, \dot{\omega}_{i,k}, F_{L_{i,k}}, T_{wheel_{i,k}}\}_{16 \times 1}^T \quad (5)$$

$$z_k = \{\omega_{i,k}^S, F_{L_{i,k}}^{VS}\}_{8 \times 1}^T \quad (6)$$

$$u_k = \{\dot{F}_{L_{i,k}}\}_{4 \times 1}^T \quad (7)$$

$$\begin{cases} \omega_{i,k+1} = \omega_{i,k} + \Delta t \cdot \dot{\omega}_{i,k} \\ \dot{\omega}_{i,k+1} = \frac{T_{wheel_{i,k}} - F_{L_{i,k}} r_{\omega}}{I_{\omega}} \\ F_{L_{i,k+1}} = F_{L_{i,k}} + \Delta t \cdot \underbrace{\dot{F}_{L_{i,k}}}_{input} \\ T_{wheel_{i,k+1}} = T_{wheel_{i,k}} \end{cases} \quad (8a) \ (8b) \ (8c) \ (8d)$$

The aforementioned system of equations constitutes the observer process model in discrete time formulation, wherein Δt is a small discrete time step. In compliance with the latter set of eq., the hub speed of each wheel $\omega_{i,k}$ is updated through (8a) once its derivative $\dot{\omega}_i$ is known from the wheel dynamics of eq.(8b); wherein I_{ω} is the wheel inertia and r_{ω} is the wheel radius, T_{wheel_i} is the torque exerted on the wheel hub, F_{L_i} represents the wheel longitudinal force. For the sake of convenience, the rolling resistance is included in the longitudinal tyre force in order to keep a time-invariant system. The longitudinal wheel force (F_{L_i}) is updated by providing its derivative as system input u_k to the eq. (8c). This latter relies on the Burckhardt tyre model with fixed parameters as a function of the wheel slips ($\hat{S}_{Li}, \hat{S}_{Si}$) and the vertical forces ($\hat{F}_{Zi}$). For a better understanding of the Burckhardt model, please refer to [28]. The vertical wheel loads and the slip ratios, in turn, are observed in accordance with the methodologies reported respectively in [29] and [28]. At last, in (8d), a random walk approach is used for the wheel torque state $T_{wheel_{i,k}}$; it will be then corrected in the update phase.

With regard to the update part, the virtual sensor estimates the chassis longitudinal (F_x^{vs}) and lateral (F_y^{vs}) forces by drawing upon the measured longitudinal (a_x) and lateral (a_y) accelerations in accordance with the following eq. (9) and (10):

$$F_x^{vs} = m_v a_x - F_{R_x} \quad (9)$$

$$F_y^{vs} = m_v a_y \quad (10)$$

where, m_v is the mass of the vehicle and F_{R_x} is the sum of the forces opposed to the vehicle motion, namely the rolling resistance and the air drag force. The output vector z_k leading the observer update phase is constituted by the wheel speed sensors measurement ω_i^S , and the virtually estimated longitudinal force for each wheel $F_{L_i}^{vs}$.

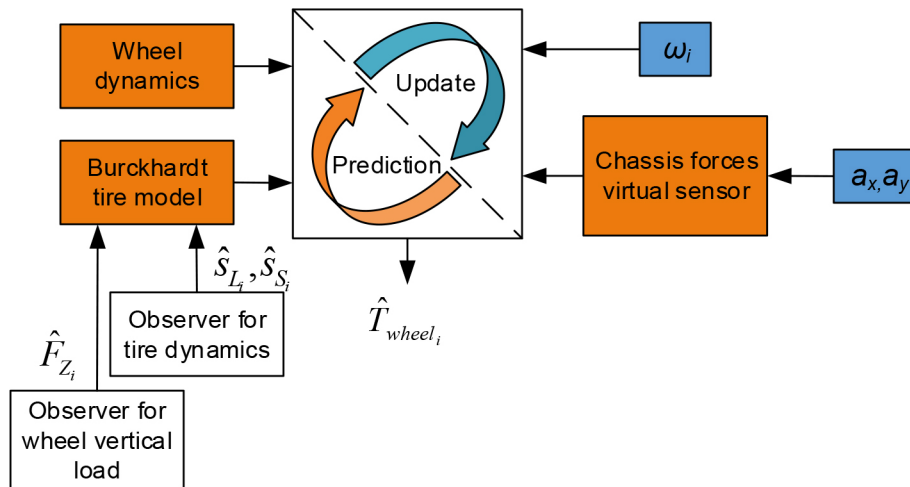


Figure 6. Scheme of the proposed linear wheel-torque observer: the update part is based on the measurements coming from the wheel speed sensors (ω_i) and chassis accelerometers (a_x, a_y); the prediction part relies on the wheels dynamics equations and on the Burckhardt tire model [28] that, in turn, requires the knowledge of tire slip coefficients ($\hat{S}_{Li}, \hat{S}_{Si}$) and the vertical load ($\hat{F}_{Zi}$).