

Article

A Minimally Intrusive Methodology for Power Loss Identification in Electric Powertrains for Physics-Based Analytical Modeling

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Abstract

This paper presents a minimally intrusive experimental methodology for identifying and modeling power losses in the electric powertrain of a battery electric vehicle, including the inverter, electric motor and speed reducer. Measurements are performed on a roller test bench equipped with an eddy current brake, using two complementary approaches to determine the mechanical power at the wheel: (i) a direct measurement based on an onboard rotary torque sensor integrated into a driveshaft; (ii) an indirect estimation derived from brake power measurements corrected for bench losses and tire longitudinal slip. The two approaches are systematically compared in order to quantify the accuracy loss associated with brake-based measurements and to identify the operating conditions under which they can reliably substitute direct torque measurements. The experimental results show that brake-based estimations provide acceptable accuracy at moderate–high torque levels, while significant deviations occur at low torque. Based on the experimental dataset, an overall power loss model is identified using a polynomial function of motor torque and speed. Two fitting strategies are investigated: an unconstrained least-squares approach, allowing all coefficients to vary freely, and a constrained formulation enforcing physically admissible (non-negative) loss terms; while the unconstrained method slightly improves the numerical fit, it may lead to non-physical coefficients and invalid efficiency predictions. In contrast, the constrained approach preserves physical interpretability and ensures consistent loss and efficiency maps. Finally, a step-by-step practical guide is provided to facilitate the implementation of the proposed methodology for powertrain loss identification on electric vehicles without extensive mechanical disassembly.

Keywords: power losses; roller test bench; on board torque meter; efficiency map; electric vehicle; power losses modeling; experimental methodology; analytical model



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1. Introduction

The transition of the automotive industry to fully electric vehicles (EVs) is a key component in global efforts to combat climate change. In recent years, sales have seen a significant and steady increase, reaching 20 million EVs sold in 2025, accounting for one fourth cars sold worldwide [1]. This trend is expected to continue and even accelerate in the coming years [2]. As a result, numerous vehicle prototypes are being developed, and those already in production will require rigorous maintenance tracking to ensure long-term performance and reliability.

For these studies to be carried out, the staff in charge of EV energy performance must have access to the most efficient quantification method possible, which typically involve the use of efficiency or loss data maps [3–5].

The first approach for generating data maps relies on physical modeling of the system. Losses in AC electric machines can be modeled using the d-q axis method (Park transformation) [6–8]. Losses in the inverter, arising from conduction and/or switching phenomena, can also be modeled using approaches of varying complexity, ranging from simplified formulations [6] to more detailed models [7,8]. Finally, losses in the transmission—including gearing losses, sealing losses, and bearing losses—can likewise be modeled by accounting for the effects of load and rotational speed [9]. These approaches generally rely on manufacturer data, complemented by experimental measurements, which are at least used for model validation. In this case, the data maps are directly generated by the models and do not require additional data-fitting technique.

A second approach relies exclusively on experimental data. In this case, the generation of data maps requires the application of data-fitting technique, along with validation methods to mitigate the risks of under-fitting and over-fitting. This latter approach can be further divided into two sub-categories.

The first consists of isolating the components to be characterized on a dedicated test bench, specifically designed for this purpose. This methodology requires dismantling the vehicle under study. The measurement techniques employed are generally conventional and, as a result, are often only briefly described in the literature. Examples of such approaches can be found in [6,10].

The second sub-approach aims to avoid dismantling the vehicle by performing measurements on a complete one, either under real driving conditions [11] or on a roller test bench [7,12]. In this case, in addition to the previously mentioned losses, tire power losses, mainly due to longitudinal slip, must be considered. When these losses are not intended to be included in the analysis, it may be relevant to directly measure the torque—and thus the power—transmitted upstream of the tire, by integrating a torque measurement device on the driveshaft. Such instrumentation can be implemented using various techniques [13] and represents an attractive compromise between measurement accuracy, vehicle intrusiveness, and test system development time. The authors of [14] describe such an approach applied to the measurement of transient torque in a hybrid electric vehicle.

However, to the authors' knowledge, no publication has reported power loss measurements using an onboard torque sensor on a battery electric vehicle. The primary objective of this paper is to address this gap. The second objective is to present and detail polynomial analytical model fitting and validation techniques based on a physics-based approach, an aspect that is rarely addressed in the existing scientific literature. Finally, the third objective is to assess whether the addition of an on board torque sensor is strictly necessary, or whether power measurements performed downstream of the tire, combined with a tire slip loss model, can provide a reliable estimation of the power losses upstream of the tire.

Consequently, the experimental measurements will be conducted on a roller test bench, as it offers a practical and economical option commonly used in vehicle testing. The experimental vehicle underwent two minor modifications: the electrical circuit at the battery output was opened to insert a shunt for current measurement, and one of the final drive shafts was modified to incorporate a commercial '*ready to use*' rotary torque sensor.

The paper is organized as follows: First, the considered power loss are presented together with the associated notation. The materials and methods are then described, including the experimental vehicle, the rotary torque meter, the roller test bench and the procedure used to calibrate mechanical friction losses in the latter. The issue of the equivalent mass of the vehicle, rotating components, and rollers is subsequently discussed. Finally,

the theoretical framework for quantifying longitudinal slip and its compensation in the measurements is introduced and analyzed. Next, a loss modeling approach based on a two-variable polynomial function—torque and motor speed—will be proposed and discussed in detail, with the aim of considering only the dominant first-order physical effects.

The loss model coefficients will be identified using the least-squares method applied to both complete measurement datasets (torque meter and brake-based measurements). The method is briefly reviewed, and the risks of underfitting and overfitting are assessed using a cross-validation procedure.

The measurements and maps obtained from both measurement methods will be compared to assess how complementary they can be and to quantify the loss of accuracy associated with using the brake-based measurement method.

Finally, a step-by-step summary of the method will be provided to guide the reader through its implementation.

2. Power Losses of Electric Vehicle

Figure 1 gives a Sankey diagram of the considered power losses for this study.

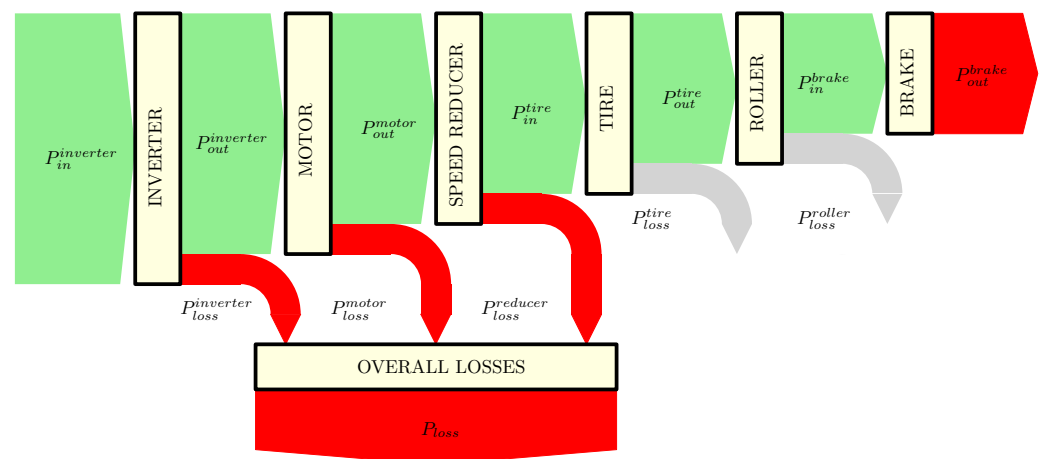


Figure 1. SANKEY Diagram of the power losses for an electric vehicle.

Firstly, it should be noted that losses in the battery pack are beyond the scope of this paper and are not considered. Consequently, the input power is the electrical power supplied to the inverter by the battery pack, denoted as $P_{inverter_in}^{in}$. The output power is the mechanical power delivered by the transmission shafts to the driven wheels, denoted as P_{in}^{tire} .

The overall power loss of the vehicle transmission, denoted as P_{loss} is the sum of power losses in inverter, electric motor and speed reducer, respectively, denoted as $P_{loss}^{inverter}$, P_{loss}^{motor} and $P_{loss}^{reducer}$.

$$P_{loss} = P_{loss}^{inverter} + P_{loss}^{motor} + P_{loss}^{reducer} = P_{in}^{inverter} - P_{in}^{tire} \quad (1)$$

The measurement of electrical power $P_{in}^{inverter}$ has been performed by current and voltage measurements or using the CAN messaging from the Battery Management System, as discussed later in this paper.

The measurement of P_{in}^{tire} has been performed with two different approach. First method, which gives the best accuracy, uses both an embedded rotary torque meter (quantity T_s) and an angular speed measurement (quantity ω_s) on the transmission shaft. By assuming that the power is the same at both driven wheel, it can be written as

$$P_{in}^{tire} = 2T_s\omega_s \quad (2)$$

The second method, in a case where no torque meter is used, estimates the tire power input from power measurement at the eddy current brake of the roller test bench, denoted as P_{out}^{brake} . In this case, two additional losses must be accounted for: tire longitudinal slip loss and roller test bench loss, respectively, denoted as P_{loss}^{tire} and P_{loss}^{roller} . The total friction losses in the test bench (including rollers and brake losses) will be determined through a calibration process, while the tire losses will be determined from measurements of the tire's longitudinal slip. With knowledge of these last two quantities, it is then possible to estimate the input power to the wheel. A quantity estimated from brake-based power measurements will be denoted with a star superscript (*), in contrast to the same quantity without a star when obtained from direct torque meter measurements. Thus, P_{in}^{tire*} will represent the estimated value, while P_{in}^{tire} will refer to the measured one. Using this convention, P_{in}^{tire*} can be written as:

$$P_{in}^{tire*} = P_{out}^{brake} + P_{loss}^{tire} + P_{loss}^{roller} \quad (3)$$

3. Materials and Methods

3.1. Experimental Vehicle

For validation reason only, the vehicle chosen for this study, the Renault Twizy 80, was deliberately selected for its limited motor torque. This characteristic ensures that the torque at the output of the gearbox remains moderate, making it compatible with the use of a reasonably sized commercial torque meter that can be integrated into the driveshaft. This setup enables a comparison, across the full operating range of the vehicle, between the mechanical power measured using the torque meter and the power estimated from brake measurements, corrected for tire longitudinal slip and mechanical losses in the test bench.

Figure 2 illustrates the electric powertrain of the Renault Twizy 80.

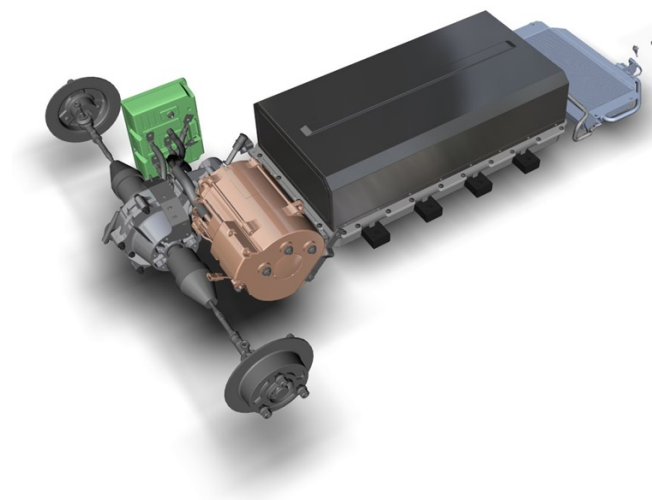


Figure 2. Electric powertrain of Renault TWIZY 80.

This vehicle was also chosen because of the availability of extensive technical documentation. Indeed, the “Twizy Open Source P.O.M.” project [15], initiated by the vehicle's manufacturer, has made a large amount of information publicly available. Additionally, several educational institutions have developed practical exercises based on this vehicle and documented their results [16,17]. As a consequence, high-quality data are accessible:

- Vehicle CAN Messagery: BMS, inverter, cluster.
- Electric motor efficiency versus torque and speed (Figure 3a).
- Inverter efficiency versus torque and speed (Figure 3b).

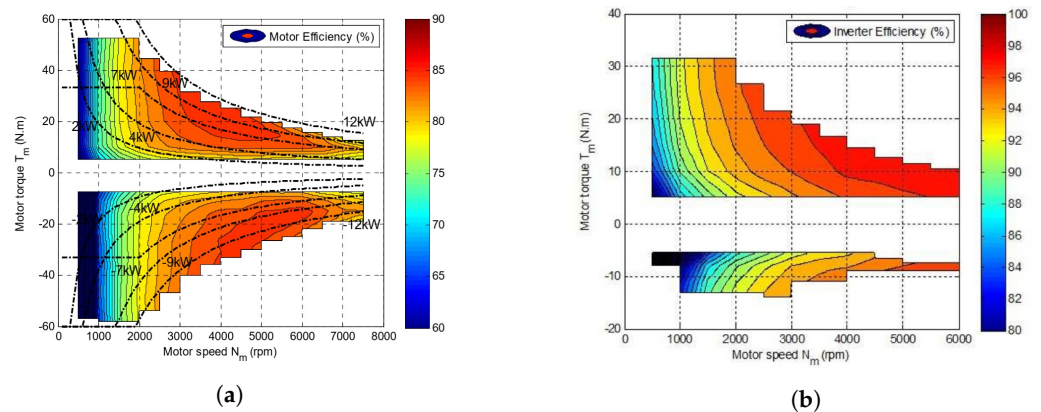


Figure 3. Manufacturer efficiency maps. (a) Motor efficiency map. (b) Inverter efficiency map.

Table A1 (Appendix B) gives the main characteristics of the Renault TWIZY 80.

3.2. Torque Meter

Figure 4 and Table A2 (Appendix B), respectively, gives a view of the selected torque meter and its specifications. Because it was not feasible to embed a rotary torque meter between the Induction Motor (IM) and the speed reducer, it has been chosen to place the torque meter between the speed reducer and the rim, on the right transmission shaft of the rear-driven car.

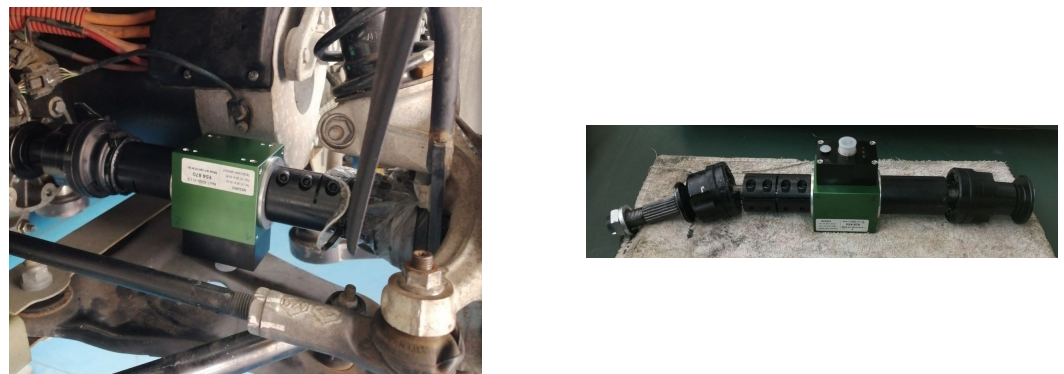


Figure 4. Rotary torque meter installed on drive shaft.

3.3. Roller Test Bench

Figure 5 shows a simplified diagram of the roller test bench. The torques are measured at two different location: first at one driveshaft using the embedded torque meter, and second at the eddy current brake using a load cell. Moreover, it will be measured: battery voltage, battery current, and the several angular velocity sources—motor, drive shaft, and front and rear rollers.

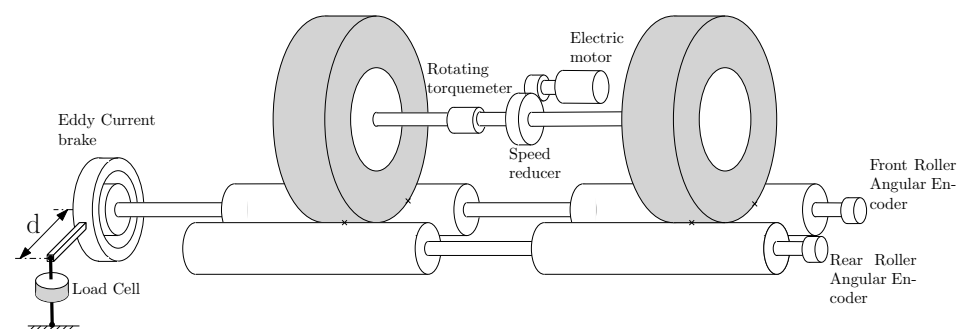


Figure 5. Overview diagram of the roller test bench.

Integrating a commercial rotary torque sensor into a drive shaft (see Figure 4) presents two main challenges: space constraints and potential mechanical weakening. The torque sensor requires sufficient space to accommodate the device itself along with its clamping systems at both ends. If space is limited, a smaller torque sensor may have to be used, which might not be able to cover the full expected torque range. Furthermore, this modification can lead to reduced mechanical strength or grip in the drive shaft, potentially preventing the transmission of the maximum expected torque. For these reasons, complementary measurements using a dynamo-meter at eddy current brake of the bench may be necessary to compensate for potential gaps in shaft-based torque measurement.

Table A3 gives the specifications of roller test bench used for this work.

The test rig is equipped with four rollers of the same radius R_{roll} . Each pair of front and rear rollers is rigidly coupled and connected to an angular encoder which will be used to determine the speed of rotation.

The eddy current brake acts only on the front roller line. The braking torque is measured in steady state condition, on the eddy current brake stator using a load cell placed at a distance d from the axis of rotation.

Figure 6 illustrates the forces acting to the vehicle when it is on the roller test bench.

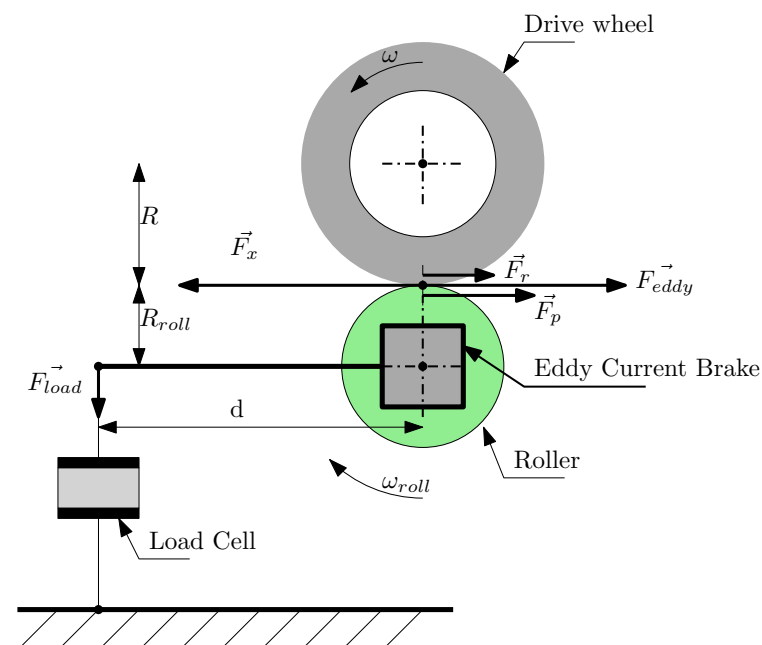


Figure 6. Force applied on drive wheel.

Equation (4) gives the vehicle law of motion on the roller bench. m_{eq} is the equivalent mass simulated by the rollers inertia and all rotating parts. This is discussed in a dedicated subsection below.

$$m_{eq} \frac{dv_x}{dt} = F_x - F_{eddy} - F_p - F_r \quad (4)$$

v_x is the tangential speed on rollers or vehicle speed. F_x is the driving force created by tire, F_{eddy} is the resistant force created by the eddy current brake action, F_r is the rolling resistance mainly due to the tire and finally F_p is the intrinsic total resisting force of the rollers due to the frictional forces generated by the bearings supporting the rollers and the aerodynamic eddy current torque due to its inner ventilation.

F_{eddy} is determined directly from the measurement of the applied force on load cell, denoted as F_{load} (Figure 6):

$$F_{eddy} = F_{load} \frac{d}{R_{roll}} \quad (5)$$

Calculating F_p correctly is a delicate operation, which is usually repeated several times. It depends on many factors, including temperature and rolling friction.

In the case where the roller test bench do not use an eddy current brake but a reversible electric motor, two methods are commonly used [12,18]: the constant resistance force method and the stationary points method. In both cases, the rollers are unloaded, meaning the weight applied on driven axle is not supported by the rollers. The calibration of frictional losses on the test bench is therefore not accurate: when the car rests on the rollers, the radial load generated by the weight of the axle alters bearing's behavior.

Since the test bench used in this study is not equipped with an electric motor and considering the objective of calibrating it under realistic load conditions, a deceleration-based method was adopted: The car is driven to the maximum speed allowed by the inverter with eddy current brake deactivated, ensuring that no resistant torque is applied to the rollers. The vehicle is then switched to neutral mode using the DNR selector in order to prevent any regenerative braking. The roller's deceleration is then recorded without any action from the driver on the brakes or driving wheel. This method is known as the 'coast down' method [19].

During deceleration of the car, because $F_x = F_{eddy} = 0$ the law of motion can be simplified as:

$$m_{eq} \frac{dv_x}{dt} = -F_p - F_r = -F_{res} \quad (6)$$

F_{res} , defined as the total resistance force, is the sum of the resisting force of the roller due to friction forces F_p and the rolling resistance force F_r due to tire and transmission loss.

The drawback of the coast down method is that it is not possible to isolate F_p and F_r from F_{res} . However, given the objective of the paper presented here, the authors consider this limitation to be inconsequential, as energy losses in tires due to rolling resistance are not taken into account in the efficiency mapping of a powertrain. Therefore, this method has the advantage of naturally excluding these losses.

Finally, the mechanical power dissipated by eddy current brake is given by:

$$P_{out}^{brake} = F_{load} \cdot d \cdot \omega_{roll} \quad (7)$$

3.3.1. Equivalent Mass

m_{eq} is the equivalent mass of the bench that provides a translation kinetic energy equal to the rotational kinetic energy of the rollers and rotating components in the vehicle's transmission (such as the driveshaft, motor rotor, etc.).

$$\frac{1}{2} m_{eq} v_x^2 = \frac{1}{2} J_{roll} \omega_{roll}^2 + \frac{1}{2} J \omega^2 \quad (8)$$

J_{roll} is the rotational inertia of the rollers and J is the inertia of the rotating parts of the vehicle.

The tangential speed v_x can be evaluated from both the roller and the wheel perspectives:

$$v_x = \omega_{roll} R_{roll} = \omega R_e \quad (9)$$

By introducing the ideal no-slip kinematic laws given by (9), (8) can be rearranged:

$$m_{eq} = J_{roll} \left(\frac{1}{R_{roll}} \right)^2 + J \left(\frac{1}{R_e} \right)^2 \quad (10)$$

Determining J is not feasible without knowing the weight and dimensions of each rotating component, which may require complete disassembly of the vehicle. An alternative method for evaluating J is based on the work of [20,21]:

$$m_{eq,v} = m + J \left(\frac{1}{R_e} \right)^2 \quad (11)$$

with m being the vehicle true static mass and $m_{eq,v}$ the equivalent mass of the vehicle when taking into account the inertia of the rotating parts. The authors of Ref. [21] assume that $m_{eq,v}$ is proportional as m and propose:

$$m_{eq,v} = \delta \cdot m \quad (12)$$

with δ a rotational inertia factor defined as:

$$\delta = 1 + \frac{I_w}{mR_e^2} + \frac{i_0^2 i_g^2 I_p}{mR_e^2} \quad (13)$$

where I_w and I_p are, respectively, the moment of inertia of the wheel associated with all rotating parts that moves with it (transmission shaft) and all rotating components associated with the power plant. i_g and i_0 are, respectively, the gear ratio of the transmission and final drive. Finally the authors of Ref. [20,21] propose an empirical relation, valid for a passenger car, that takes into account the effect of wheels and the rest of transmission:

$$\delta = 1 + 0.04 + 0.0025G^2 \quad (14)$$

with G is the total gear reduction given by:

$$G = i_0 i_g \quad (15)$$

Using the previous equations, it is now possible to estimate J . The equivalent mass of the TWIZY was calculated to be 593 kg, which leads to a rotating inertia J of 8.48 kg·m².

J has to be added to the roll inertia of the Rollers, according to Equation (10), to calculate the bench equivalent mass. Finally, a bench equivalent mass value m_{eq} of 970 [kg] is found.

3.3.2. Bench Calibration

Figure 7 gives the roller bench calibration results for the Twizy driven up to 80 km/h and then set in NEUTRAL mode. The resistant force F_{res} has been classically fitted with a second degree polynomial function:

$$F_{res} = C_2 v_x^2 + C_1 v_x + C_0 \quad (16)$$

Table 1 gives the obtained coefficient, and correlation coefficient R^2 .

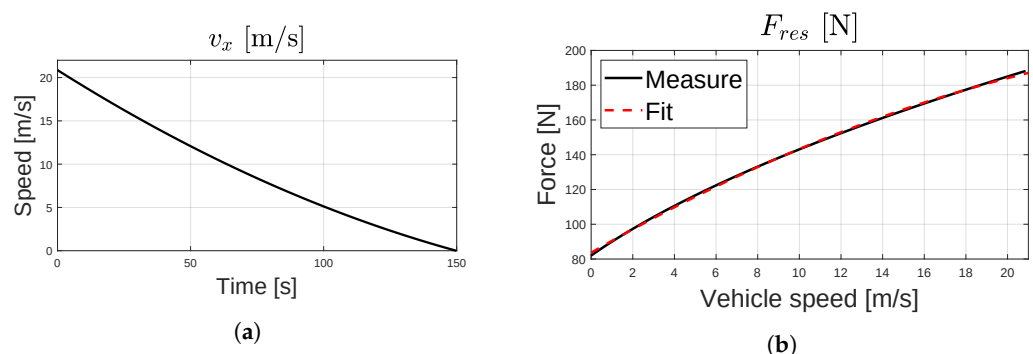


Figure 7. Rolling resistance bench calibration. (a) Vehicle speed v_x versus time. (b) Resistant force versus v_x .

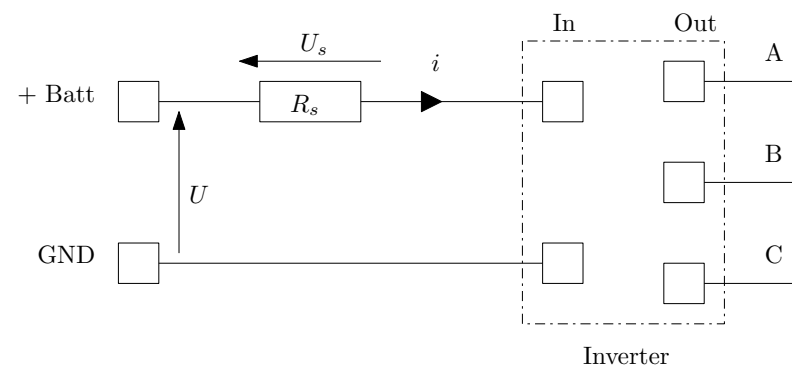
Table 1. Resistant force fit results.

C_2 [N·s ² /m ²]	−0.0947
C_1 [N·s/m]	6.9139
C_0 [N]	83.64711
R^2	=0.9996

3.4. Electrical Power at Inverter Input

A second modification was made to the vehicle: the electrical circuit upstream of the inverter was opened to integrate an electric shunt required for current measurement.

Figure 8 depicts the electric power measurement at inverter input. R_s is the shunt between battery and inverter input. U and U_s are, respectively, the voltage at battery and shunt.

**Figure 8.** Electric power at inverter input.

Because the current is pulsed, the inverter input power $P_{in}^{Inverter}$ is calculated using acquisition data over n samples of the battery voltage and the shunt voltage, denoted U_i and U_{s_i} , respectively. For each sample, the instantaneous current is computed as $i_i = U_{s_i}/R_s$.

$$P_{in}^{inverter} = \frac{1}{n} \sum_{i=1}^n U_i i_i = \frac{1}{n} \sum_{i=1}^n U_i \frac{U_{s_i}}{R_s} \quad (17)$$

This power is also evaluated using the data coming from the CAN messages emitted by the Battery Management System (BMS). These values are mean values calculated by the BMS, and are emitted each 30 ms: U_{bat} and I_{bat} , the mean battery voltage and mean battery current, respectively. This power is also evaluated using data obtained from CAN messages transmitted by the Battery Management System (BMS). These values are mean quantities computed by the BMS and transmitted every 30 ms: U_{bat} and I_{bat} , corresponding to the mean battery voltage and the mean battery current, respectively. Using these two quantities, the inverter input power can be calculated as:

$$P_{in}^{inverter} = U_{bat} I_{bat} \quad (18)$$

These two methods have been used for a calibration process, and the results obtained have been very similar. For convenience, the experimental dataset was created using CAN messaging data.

3.5. Tire Longitudinal Slip

On a conventional roller test bench, the tire is part of the power transmission chain to the eddy current brake. Therefore, when estimating the wheel power from a power

measurement taken at the brake, it is necessary to account for the power losses caused by longitudinal tire slip.

The phenomenon of longitudinal slip of a tire is well known and is a consequence of its viscoelastic nature. It results from the combination of two main effects. The first effect is the reduction in wheel's effective radius when a torque is applied. This phenomenon is caused by the change in orientation of the carcass ply cords under motor torque. As a result, the cords tilt and are no longer aligned radially. This reduction in effective radius induces a virtual slip, meaning that it is not caused by an adhesion issue. The wheel's rotational speed increases to compensate for the reduced radius, creating an apparent slip. The second effect is a real slip that occurs in the contact patch due to adhesion limit. This phenomenon is described, for instance, in Pacejka's brush model [22–24].

Figure 9 illustrates a driven wheel under vertical load. T_s represents the torque applied to the rim by transmission shaft. F_x is the tractive longitudinal force generated by the tire in reaction to the applied torque. ω denotes the angular velocity of the rim, while v_x corresponds to the wheel's longitudinal forward velocity.

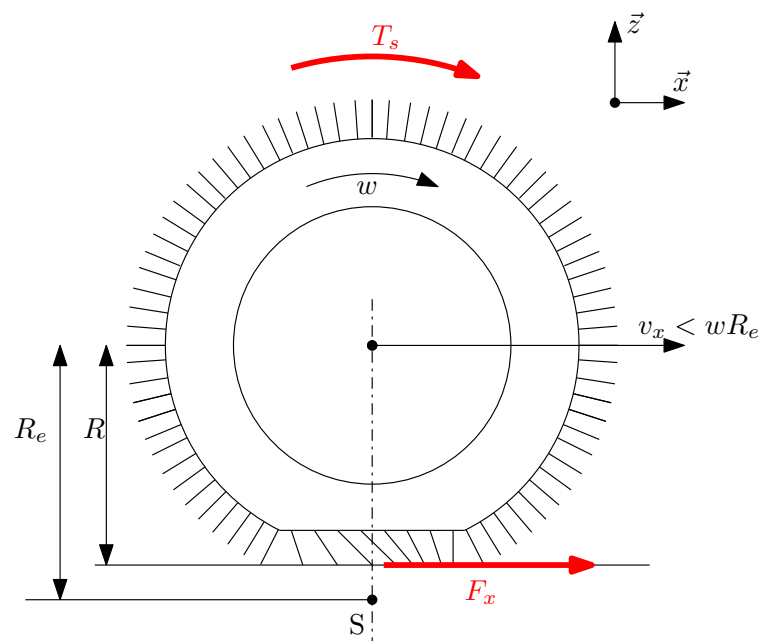


Figure 9. Tire in pure longitudinal slip conditions.

S is the slip point: it is a virtual point associated with the rim, located at a distance R_e from the center of rotation. R_e is the effective loaded radius of the tire. It is measured in a free-rolling condition, meaning no torque is applied to the rim. Finally, R is the effective radius of the wheel when a torque T_s is applied. Due to two primary effects, R is smaller than R_e : the radial deflection of the tire carcass and the deformation of tread blocks under applied forces (as described in the brush model).

ω_0 is then defined as the angular velocity that would be observed if the wheel were in free-wheel condition.

$$\omega_0 = \frac{v_x}{R_e} \quad (19)$$

The real angular velocity of the wheel ω is higher than ω_0 due to tire deformation and real slip. It is posed as follows:

$$\omega = \frac{v_x}{R} \quad (20)$$

Next step is to define the sliding velocity v_s as:

$$v_s = R_e \omega - v_x = R_e (\omega - \omega_0) \quad (21)$$

One can note that this sliding velocity is calculated from the one that should be observed in the free-wheel condition. The result is a higher slip speed than it would be calculated using the true radius of the wheel under torque. This overestimation of slip is performed for taking into account the real slip that occurs within the contact patch, in addition to the slip solely due to the tire's geometric deformation.

The definition used for longitudinal slip, denoted as λ , differs from the one proposed by Pacejka [22]. It is an alternative definition employed by various authors, such as Sina et al. [23], for instance.

$$\lambda = \frac{v_s}{R_e * \omega} = \frac{\omega - \omega_0}{\omega} \quad (22)$$

The latter definition will be used in the present paper, as it requires fewer assumptions in tire power modeling, as detailed below.

The longitudinal tire slip is determined on a roller test bench by comparing two angular velocity measurements: wheel ω and roller where the eddy current brake acts as ω_{roll} . In the case illustrated in Figure 5, these are the front rollers. Typically, the angular velocity of the front rollers is lower than that of the rear rollers due to the tire relaxation occurring after passing over the resisting roller. The ideal velocity ω_0 can then be calculated by considering the roller radius, R_{roll} :

$$\omega_0 = \frac{v_x}{R_e} = \frac{\omega_{roll} R_{roll}}{R_e} \quad (23)$$

Since the wheel is in equilibrium, it can be written, assuming that $R_e \approx R$:

$$T_s^* = F_x R_e \quad (24)$$

Let T_s^* note the torque applied at the rim. The superscript * indicates that this quantity is estimated, in accordance with the notation conventions adopted in this paper.

Using these quantities, it is possible to quantify the efficiency and power losses caused by the tire's longitudinal slip phenomenon. The power supplied to the rim is given by:

$$P_{in}^{tire*} = T_s^* \omega_s \quad (25)$$

The previous equation is defined for a single tire. If it is assumed that the same torque and angular velocity are applied at the second tire of the driven axle, then the total power transmitted to the tires is twice P_{in}^{tire*} . In the remainder of this subsection, only one wheel is considered.

The power at tire output is given by:

$$P_{out}^{tire} = F_x v_x \quad (26)$$

The power loss in the tire, denoted P_{loss}^{tire*} , is expressed as:

$$P_{loss}^{tire*} = P_{in}^{tire*} - P_{out}^{tire} = F_x R_e (\omega - \omega_0) = \lambda P_{in}^{tire*} \quad (27)$$

The tire efficiency η^{tire} can then be expressed as:

$$\eta^{tire} = \frac{P_{out}^{tire}}{P_{in}^{tire*}} = \frac{P_{in}^{tire*} - P_{loss}^{tire*}}{P_{in}^{tire*}} = (1 - \lambda) \quad (28)$$

Equation (27) allows the calculation of the power supplied to the rim, P_{in}^{tire*} , from the measurements of the brake power P_{out}^{brake} and longitudinal tire slip λ . Then, the estimated Torque T_s^* is calculated using (25).

3.6. Motor Torque

Since the torque at the electric motor output shaft, T_e , is not directly measured, it is estimated from the known values of T_s and the angular velocity ω_s by applying the principle of energy conservation and assuming that the total power is equally distributed between the two wheels of the driven axle.

$$P_{out}^{motor} = T_e \omega_e = 2P_{in}^{tire} = 2 \cdot T_s \omega_s \quad (29)$$

The speed reducer imposes:

$$\omega_e = G \omega_s \quad (30)$$

The motor torque is:

$$T_e = \frac{2T_s}{G} \quad (31)$$

This simple scaling transformation using the gear ratio G , from (T_s, ω_s) to (T_e, ω_e) is necessary to express the maps in their conventional form—that is, it is referred to the motor side.

Nevertheless, the measured and modeled losses include those occurring in the speed reducer and therefore represent the total system losses, i.e., inverter + motor + speed reducer.

3.7. Loss Modeling

The graphical representation of losses and efficiency in an electric motor or drivetrain is commonly illustrated using a map based on two variables: torque T_e and angular velocity ω_e .

It is typical to express the losses in an electric power plant as a polynomial function of the form [5,25]:

$$P_{loss}(T_e, \omega_e) = \sum_{i=0}^{n_1} \sum_{j=0}^{n_2} k_{ij} \cdot T_e^i \cdot \omega_e^j \quad (32)$$

$k_{i,j}$ are the coefficients of the polynomial model, n_1 is the maximum degree of the polynomial with respect to the torque T_e , and n_2 is the maximum degree of the polynomial with respect to the angular velocity ω_e .

This approach enables a physics-based modeling of losses and account for a broader range of physical factors. Each coefficient has one or several physical justification has explained by [20,26].

Authors such as Roshandel [5] suggest that a 3×3 matrix is sufficient for the k_{ij} coefficient to account for the various losses in an electric machine, including converter losses, conductive losses, iron losses, permanent magnet (PM) losses, mechanical losses, and transmission system losses.

In the following section, we will discuss and justify the dominant physical effects associated with each term of this matrix.

k_{00} represents a constant loss of power not depending on T_e or ω_e .

k_{01} is physically justified by a constant dry friction force.

k_{02} takes into account the viscous damping force and the iron losses [25].

k_{10} typically takes into account the loss in the semiconductors of the inverter [26,27].

The coefficient k_{11} is not commonly found in the literature. Firstly, it can be understood as a proportional loss to the output mechanical power $T_e \omega_e$. Roshandel [5] explains that this term can be introduced to account for second-order effects on additional Copper (Joule)

losses in the constant torque region for an Induction Motor (IM). The term k_{12} has no known first order physical basis, either in the literature or according to the authors's understanding. This term can be introduced to account for second-order effects [5] in copper (Joule) loss and iron loss for an IM motor. According to author's aim to take into account only first order factor, both coefficients k_{11} and k_{12} will not be used.

The coefficient k_{20} corresponds to the well-known copper (Joule) losses. Since torque T is proportional to current i , and copper losses scale with i^2 , this term has a strong physically justification.

The term k_{21} may be used to account for longitudinal tire losses, if necessary. The power loss in the tire can be expressed as $P_{loss}^{tire} = \lambda P_{in}^{tire}$, with λ that is proportional to torque T_e or T_s , and $P_{in}^{tire} = T_s \omega_s$. Thus, tire power losses can be modeled as being proportional to $T_e^2 \omega_e$. However, in this paper, because tire losses are out of the scope of the efficiency map, this term will not be used.

Finally, the coefficient k_{22} is not referenced in the literature, excepted to account for second order factors. The authors propose to retain it in order to represent copper losses occurring upstream of the inverter. Indeed, if we assume that the mechanical power is equal to the electrical power (with unit efficiency), one can write: $i_{bat} = \frac{T_e \omega_e}{U_{bat}}$ where i_{bat} and U_{bat} are, respectively, the supply current to the inverter and the battery pack voltage.

Assuming the battery voltage U_{bat} is almost a constant, it follows that the current i_{bat} is proportional to the product $T_e \omega_e$. Now, the copper (Joule) losses upstream of the inverter are scaled to i_{bat}^2 . Therefore, these losses are proportional to the product $T_e^2 \omega_e^2$.

Figure 10 summarizes this discussion, with a color convention as follows:

- A green background indicates a factor with a physical justification and a first-order effect on losses. Therefore, it is included in the least-squares fitting process.
- A yellow background indicates a factor with either a negligible first-order effect on losses or one that falls outside the scope of losses considered by the authors.

	j index →		
i index ↓	k_{00} Constant losses	k_{01} Dry friction losses	k_{02} Iron losses + Viscous Friction
	k_{10} Semi-conductors losses	k_{11} No first order justification	k_{12} No first order justification
	k_{20} Copper losses downstream of the motor controller	k_{21} Tire longitudinal slip losses	k_{22} Copper losses upstream of the motor controller

Figure 10. Physical representation of k_{ij} polynomial parameters for power loss modeling.

3.8. Unconstrained and Constrained Least-Squares Fitting

It is considered an experimental data triplet consisting of the three quantities $(P_{loss,i}, T_{e,i}, \omega_{e,i})$, or alternatively $(P_{loss,i}^*, T_{e,i}^*, \omega_{e,i})$ if the measurements are obtained, respectively, from the torque meter or from the brake load cell. The index i refers to the triplet under consideration within the complete set of n triplets. In the remainder of this paper, the methodology will be illustrated using the experimental triplet $(P_{loss,i}, T_{e,i}, \omega_{e,i})$.

As a first step, we construct a triplet of dimensionless values, denoted $(\tilde{P}_{loss,i}, \tilde{T}_{e,i}, \tilde{\omega}_{e,i})$, defined as follows:

$$\begin{aligned}\tilde{P}_{loss,i} &= \frac{P_{loss,i}}{\max_{1 \leq i \leq n} (P_{loss,i})} \\ \tilde{T}_{e,i} &= \frac{T_{e,i}}{\max_{1 \leq i \leq n} (T_{e,i})} \\ \tilde{\omega}_{e,i} &= \frac{\omega_{e,i}}{\max_{1 \leq i \leq n} (\omega_{e,i})}\end{aligned}$$

The least-squares method can then be applied using matrix notation [28]:

$$\tilde{P}_{loss} = \hat{P}_{loss} + \epsilon = QK + \epsilon \quad (33)$$

where $\hat{P}_{loss}$ denotes the vector containing the dimensionless estimated power losses obtained by least-squares method, Q is the regressor matrix, K is the vector of parameters to be estimated, and ϵ is the vector of residual errors:

$$Q_{n \times 9} = \begin{bmatrix} 1 & \tilde{\omega}_{e,1} & \tilde{\omega}_{e,1}^2 & \tilde{T}_{e,1} & \tilde{T}_{e,1}\tilde{\omega}_{e,1} & \cdots & \tilde{T}_{e,1}^2\tilde{\omega}_{e,1}^2 \\ 1 & \tilde{\omega}_{e,2} & \tilde{\omega}_{e,2}^2 & \tilde{T}_{e,2} & \tilde{T}_{e,2}\tilde{\omega}_{e,2} & \cdots & \tilde{T}_{e,2}^2\tilde{\omega}_{e,2}^2 \\ \vdots & \vdots & \vdots & \vdots & \vdots & \ddots & \vdots \\ 1 & \tilde{\omega}_{e,n} & \tilde{\omega}_{e,n}^2 & \tilde{T}_{e,n} & \tilde{T}_{e,n}\tilde{\omega}_{e,n} & \cdots & \tilde{T}_{e,n}^2\tilde{\omega}_{e,n}^2 \end{bmatrix}$$

$$K_{9 \times 1} = \begin{bmatrix} k_{00} & k_{01} & k_{02} & k_{10} & k_{11} & \cdots & k_{22} \end{bmatrix}^t$$

According to least-squares theory, the K vector can be determined as:

$$K = (Q^t Q)^{-1} Q^t \tilde{P}_{loss} \quad (34)$$

When applying this basic method, a vector K containing 9 values is produced, some of which may potentially be negative. This technique will hereafter be referred as the “unconstrained” method.

A second approach, referred to as the “constrained” method, consists of enforcing all k_{ij} coefficients to be strictly positive or zero. Consequently, any fit that results in any k_{ij} negative coefficient is rejected. For example, one can perform a new fit after removing the column associated with the negative coefficient in the Q regressor matrix and associated row the K vector, or use a dedicated constrained least-squares function.

This constraint is not systematically applied in the literature: some authors allow negative coefficients to enhance the quality of the fit, although this affects physical interpretability. It will be discussed in detail in Section 4.2.

3.9. Verification of Underfitting and Overfitting

In any data-fitting technique, the model may be subject to overfitting or underfitting.

In order to ensure the quality of the fit obtained using the least square method, a classical technique known as K -fold cross-validation was employed [29].

The experimental dataset, consisting of 78 data points, was randomly divided into 6 subsets of 13 values each. For each of the 6 combinations, a least-squares fit was performed using 5 subsets as the training set, while the remaining subset was used as the validation set.

The absence of underfitting is assessed by verifying that the mean of the residuals, denoted as $\bar{\epsilon}$ is close to zero, indicating that no systematic information remains unmodelled.

The risk of overfitting is evaluated using a accuracy indicator, chosen here as the RMSE (Root Mean Square Error). It is verified that RMSE values obtained for each of the

6 cross-validation combinations are comparable, indicating good generalization capability of the model.

Once these two conditions are satisfied, a final fit is performed on the entire dataset (78 data points). The overall RMSE and the final mean of the residuals are then computed to characterize the final performance of the model.

4. Results

4.1. Raw Measurements

Figure 11 shows the 78 recorded measurement points, corresponding to the electrical parameters at the battery output: U_{bat} and I_{bat} . It can be observed that the battery output current can reach very high values, up to 263 A, which is fully representative of the conditions encountered in an standard electric vehicle under full load. Here, the battery voltage ranges between 54.8 V and 51 V.

Figure 12 shows the 78 recorded measurement points, corresponding to the torque and motor speed pairs (T_e, ω_e) as well as the (T_e^*, ω_e) values. These measurements effectively cover the entire operating range of the motor. Additionally, the evolution of the maximum torque versus motor speed is typical of an electric motor. It is observed that the maximum measured torque is 48 N·m, which is lower than the manufacturer's specified value of 57 N·m, likely due to an inverter limitation.

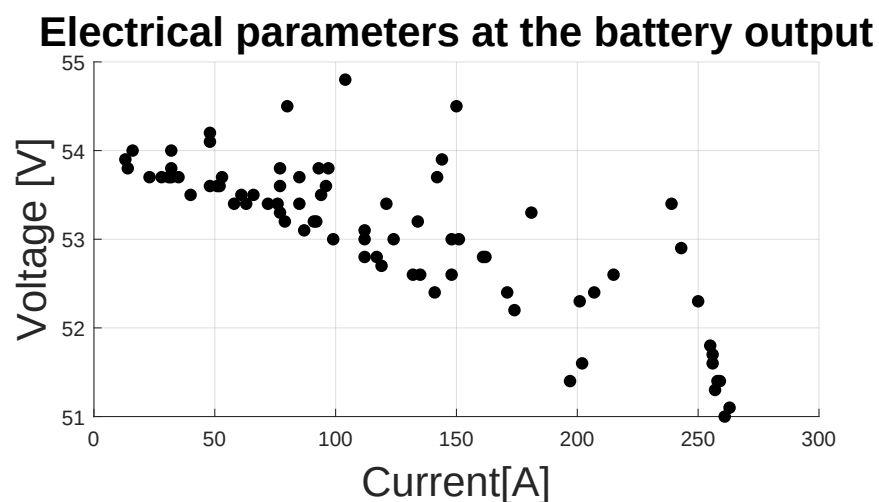


Figure 11. Battery voltage versus current.

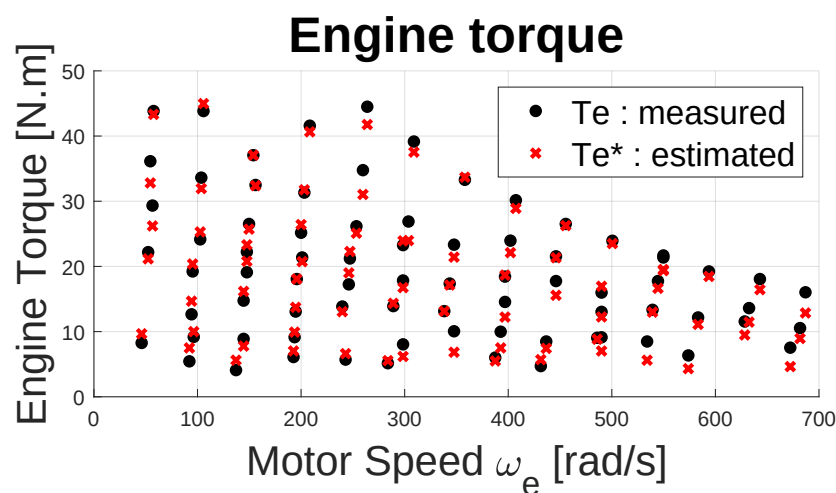


Figure 12. Torque and RPM dataset.

Figures 13 and 14, respectively, depict the total motor force transmitted by both tires and motor torque as a function of longitudinal slip. As expected, the force increases proportionally with slip, despite a relatively low correlation coefficient. This linear behavior suggests that the adhesion limit has not been approached or reached. Consequently, the use of the slip-induced power loss model defined in Equation (27) remains valid.

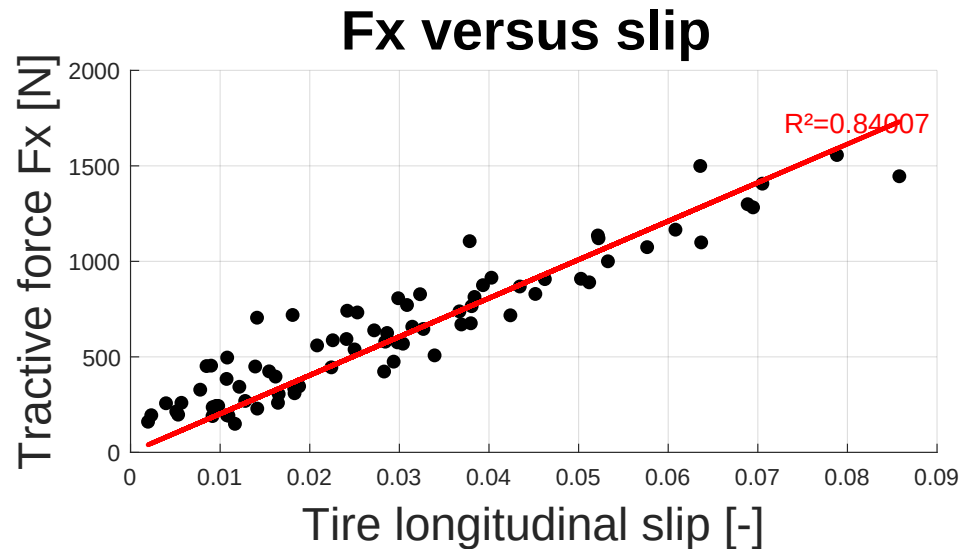


Figure 13. Motor force as function of λ .

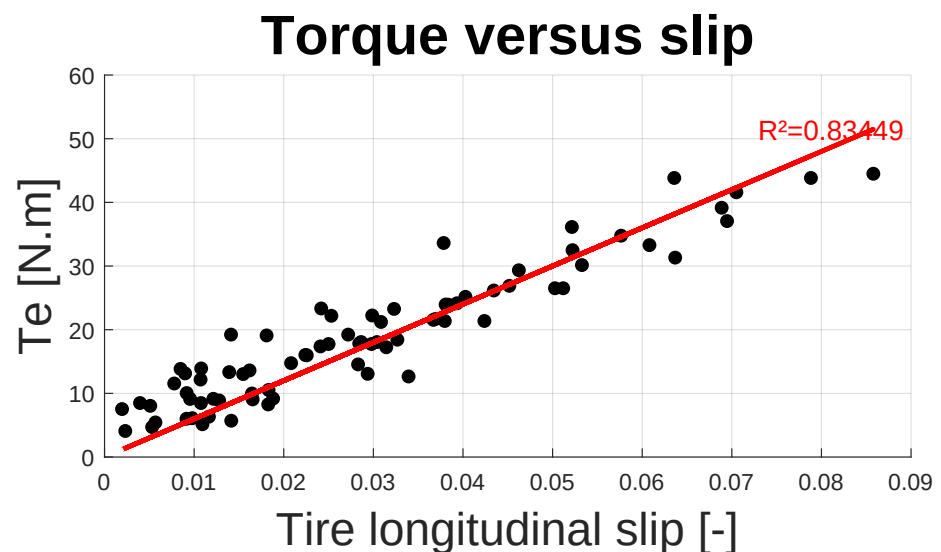


Figure 14. Torque, T_e , as function of λ .

Figure 13 highlights a variation of approximately ± 250 N around the value estimated by linear regression. Such a level of uncertainty is commonly observed in experimental vehicle measurements [30,31] or is adopted as white noise in simulations aiming to replicate realistic behavior under real-world conditions [24].

Figure 15 shows the estimated motor torque T_e^* as a function of the measured motor torque T_e . Figures 16 and 17 present, respectively, the torque error $(T_e^* - T_e)$ and the relative torque error $\frac{(T_e^* - T_e)}{T_e}$, both plotted against T_e .

Torque estimated versus measured

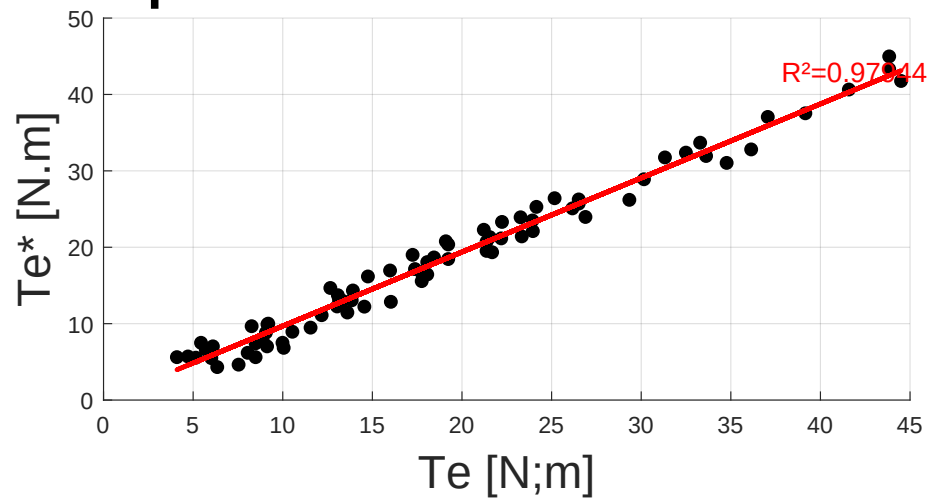


Figure 15. T_e^* as function of T_e .

Torque error

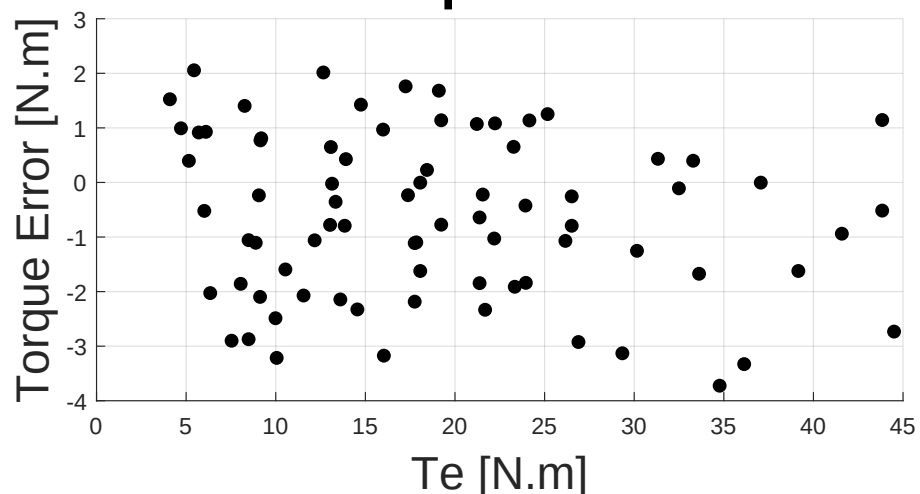


Figure 16. Torque error as function of T_e .

Figure 15 shows that the correlation between the estimated torque T_e^* and the measured torque T_e is almost satisfactory. Figure 16 indicates that the torque error remains within ± 2 N.m, regardless of the value of T_e . More importantly, this figure suggests that there is no significant bias, such as improper calibration of the test bench's resistant force F_{res} or inaccurate estimation of the tire's unloaded radius R_e . The observed error is therefore primarily attributed to measurement uncertainties, notably those related to the brake load cell and the estimation of tire longitudinal slip.

As a result, Figure 17 shows that the relative error, for torque levels above 20 N.m, remains within $\pm 10\%$, which is generally considered acceptable for this type of measurement.

This is an important result as it indicates that if the torque and consequently the tire slip is too low (in this case, $\lambda = 0.035$, as shown in Figure 14), then the torque estimation without a rotary torque meter is not reliable.

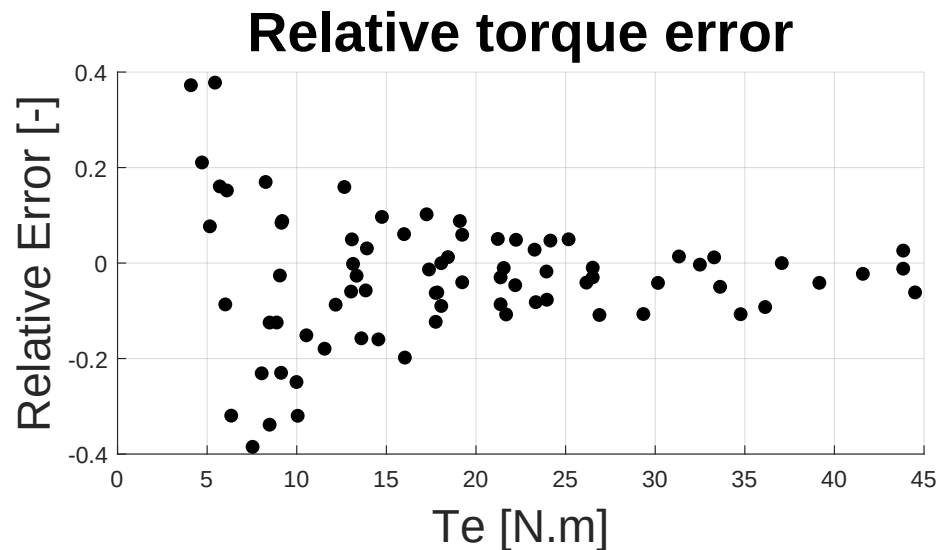


Figure 17. Relative torque error as function of T_e .

4.2. Data Fit Results

A least-squares fitting procedure was applied to the dataset of 78 measurement triplets $(P_{\text{loss}}, T_e, \omega_e)$ and $(P_{\text{loss}}^*, T_e^*, \omega_e)$. The model was validated using the K-fold cross-validation method, as previously described.

For each measurement dataset (torque meter and brake-based), an initial fit—referred to as ‘unconstrained’ fit—was performed using all 9 coefficients k_{ij} of the matrix, allowing the coefficients to take on negative values.

Subsequently, a second fit—referred to as ‘constrained’ fit—was carried out, in which undesired terms ($k_{2,1}$) or those lacking first-order physical justification ($k_{1,1}$, $k_{1,2}$) were excluded from the fitting process. Any negative coefficient resulting from the fit triggers a new iteration, during which the associated regressors (corresponding to the negative k_{ij}) are removed, until only positive coefficients remain.

Table 2 summarizes the fitted coefficients k_{ij} , the Root Mean Square Error (RMSE) and the mean value of the residuals $\bar{\epsilon}$ obtained for both models and both measuring techniques.

Table 2. Data Fitting Results.

	Torque Meter	Brake-Based
unconstrained	$\begin{bmatrix} -0.0867 & 1.1673 & -1.1881 \\ 1.5402 & -6.5156 & 7.7955 \\ -0.4443 & 4.3750 & -4.5748 \end{bmatrix}$	$\begin{bmatrix} -0.1656 & 0.7409 & 0.0068 \\ 1.2977 & -5.4861 & 4.5260 \\ -0.7870 & 5.2867 & -4.2140 \end{bmatrix}$
RMSE	0.092237	0.11573
$\bar{\epsilon}$	4.4646×10^{-14}	1.0896×10^{-13}
constrained	$\begin{bmatrix} 0.0464 & 0 & 0.3145 \\ 0.6609 & 0 & 0 \\ 0.1396 & 0 & 0.5502 \end{bmatrix}$	$\begin{bmatrix} 0.0116 & 0 & 0.5839 \\ 0.0961 & 0 & 0 \\ 0.3544 & 0 & 0 \end{bmatrix}$
RMSE	0.10539	0.11951
$\bar{\epsilon}$	2.9179×10^{-17}	-1.8504×10^{-17}

Figure 18 presents the predicted power loss maps, $\hat{P}_{\text{loss}}$ and $\hat{P}_{\text{loss}}^*$, obtained using both fitting techniques (“constrained” and “unconstrained”) for both datasets (torque-meter-based and brake-based). Figure 19 gives the predicted efficiencies.

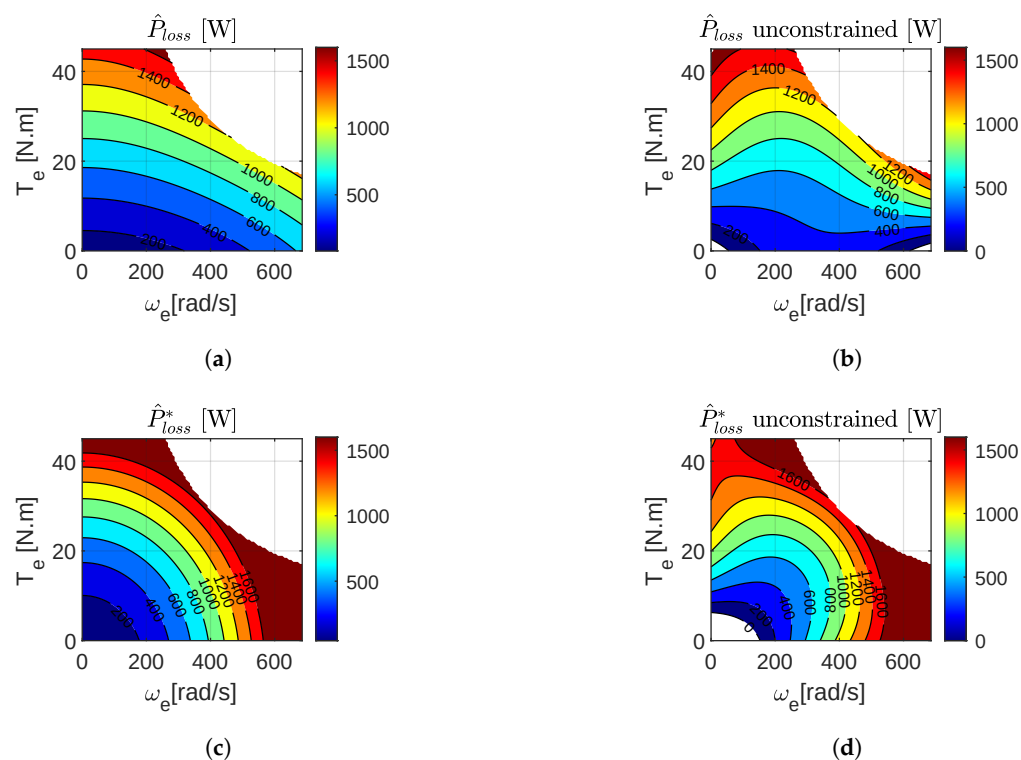


Figure 18. Predicted overall losses. (a) Constrained fit for torque meter measurements. (b) Unconstrained fit for torque meter measurements. (c) Constrained fit for brake-based measurements. (d) Unconstrained fit for brake-based measurements

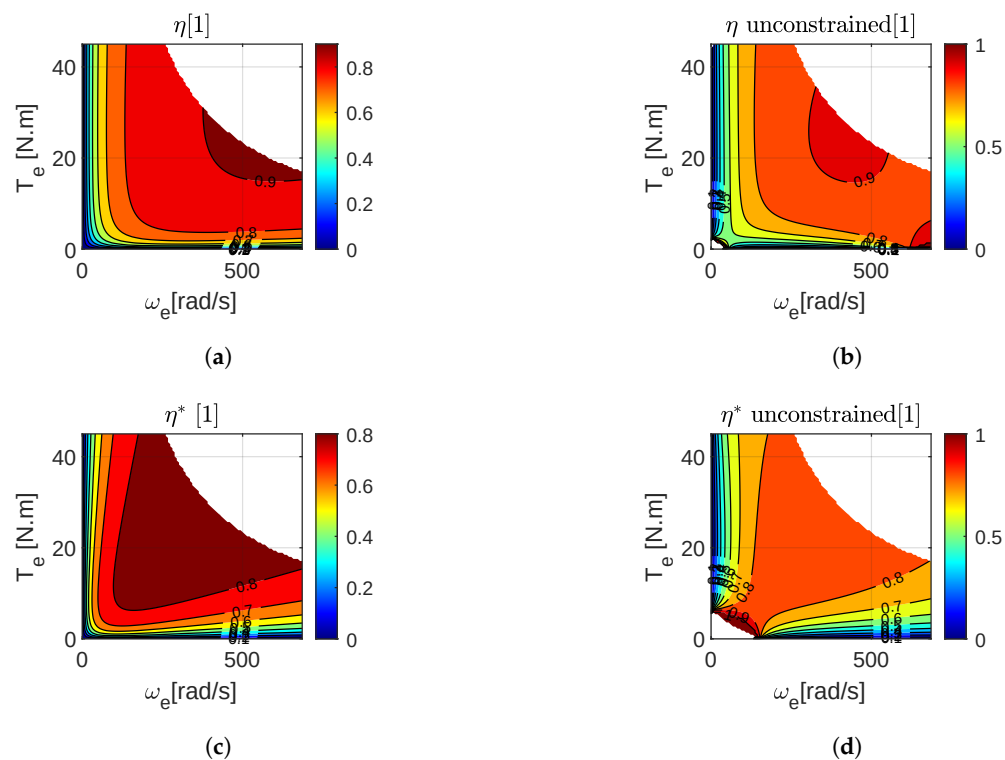


Figure 19. Predicted efficiencies. (a) Constrained fit for torque meter measurements. (b) Unconstrained fit for torque meter measurements. (c) Constrained fit for brake-based measurements. (d) Unconstrained fit for brake-based measurements.

5. Discussion

5.1. Discussion on the ‘Constrained’ and ‘Unconstrained’ Fitting Techniques

First, the ‘unconstrained’ fitting technique must be assessed. The fact that this method produces negative k_{ij} coefficients raises two major issues.

The first issue is that, in order to improve the fit quality, the ‘unconstrained’ method tends to overestimate certain physical losses associated with positive coefficients, while compensating by introducing negative values. As a result, the k_{ij} coefficients become difficult to interpret, making it impossible to compare different powertrain configurations.

The second issue is that these negative coefficients can lead to negative power loss values or efficiency values greater than 1 in certain areas of the maps. These invalid points must therefore be removed, which creates gaps in the maps. Such areas are clearly visible in Figure 18 in the lower left and right parts of plot (b), and in the lower left area of plot (d).

This problem does not occur with the ‘constrained’ method, whose maps are much smoother and more continuous. The RMSE analysis (Table 2) also shows that applying the ‘constrained’ method only slightly reduces the fit accuracy. For these reasons, the authors consider the ‘constrained’ technique to be the only suitable approach and it will be used for the analysis presented in the remainder of this paper.

5.2. Discussion on the Results

First, it should be noted that the obtained efficiency maps show good agreement with the expected results for AC motors, as described by [32,33], as well as with the data provided by the vehicle manufacturer (Figure 3).

A difference can be observed between the two fitted models (Table 2): the dataset obtained using brake measurements yields a zero value for the coefficient $k_{2,2}$, whereas the dataset based on torque sensor results in a non-zero value for the same coefficient. Additionally, the coefficients obtained from each fitting procedure differ significantly.

Figure 20 gives the graphical representation of the absolute power losses error defined as $|\hat{P}_{loss}^* - \hat{P}_{loss}|$. Figure 21 gives the graphical representation of the relative power loss error defined as $\frac{|\hat{P}_{loss}^* - \hat{P}_{loss}|}{\hat{P}_{loss}}$.

These figures further highlight that the model based on brake measurements predicts greater power losses at high rotational speeds compared to the model derived from torque meter. Figure 21 clearly shows that the relative error remains acceptable for high torque values. This observation is consistent with the findings discussed earlier in the section on raw measurements.

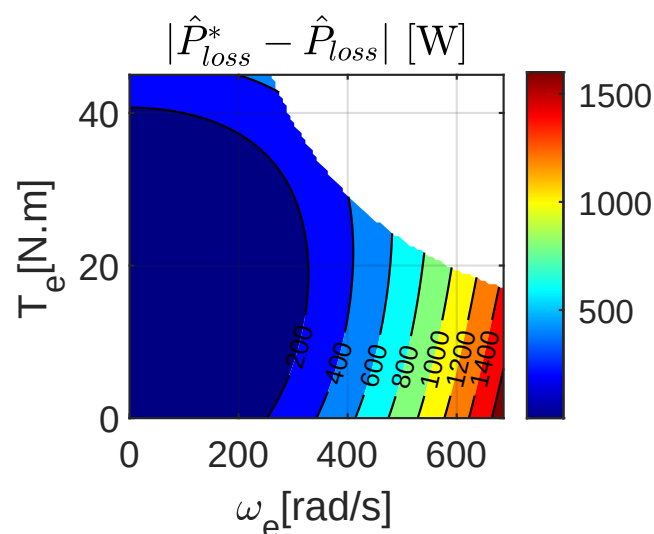


Figure 20. Absolute power loss error.

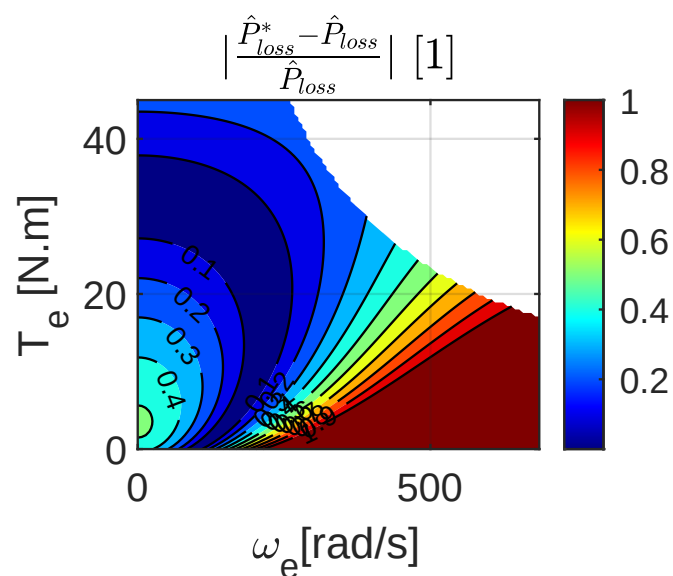


Figure 21. Relative power loss error.

Finally, Figures 22, 23, and 24, respectively, show the efficiencies computed from both measurement sets, along with the absolute and relative errors between the two.

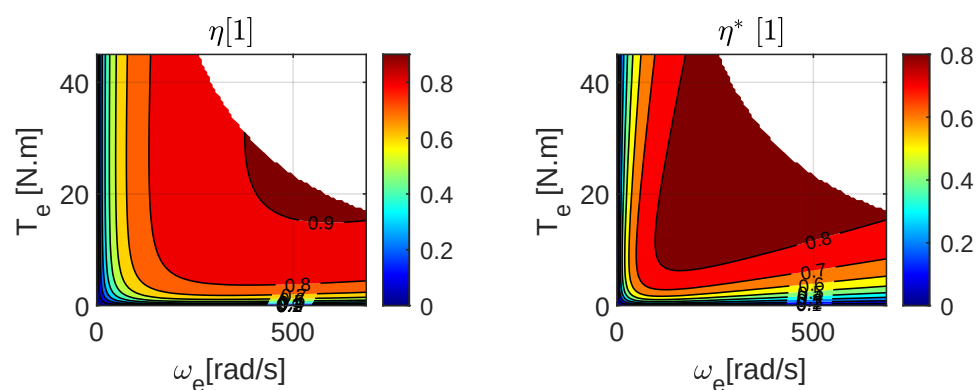


Figure 22. Overall efficiency.

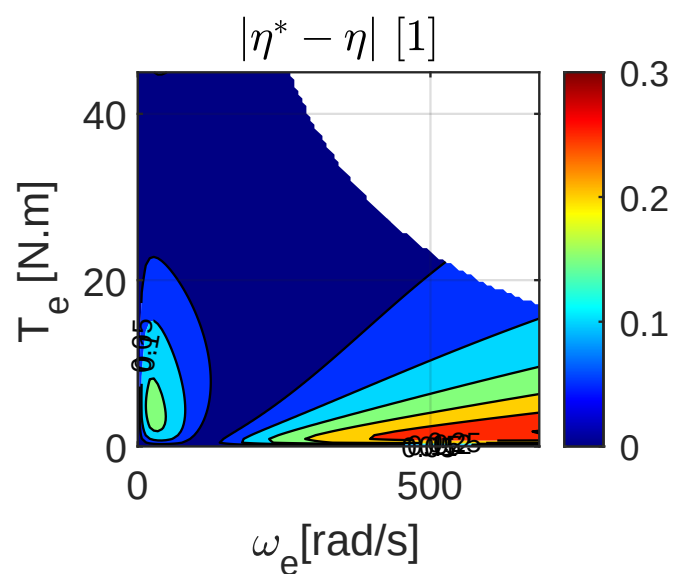


Figure 23. Absolute efficiency error.

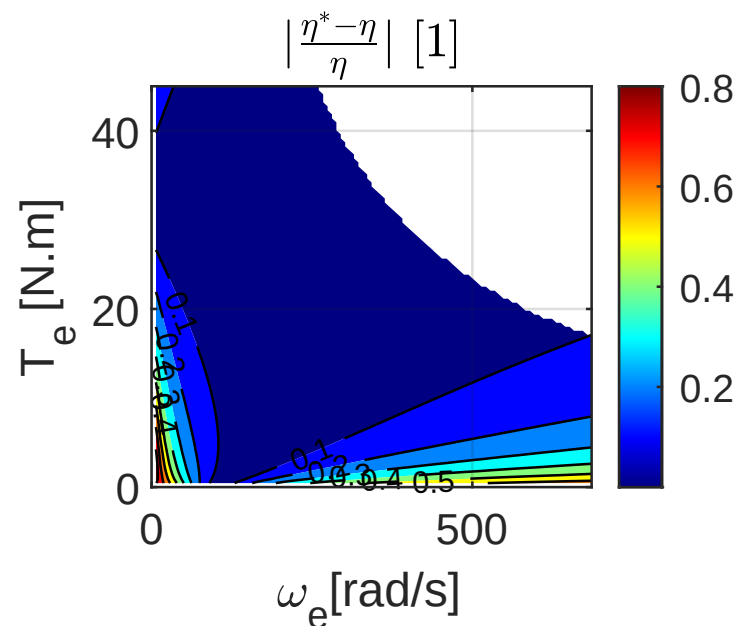


Figure 24. Relative efficiency error.

Firstly, it can be observed that the model based on brake measurements predicts the same maximum efficiency value, but over a broader operating region than the prediction coming from torque meter datasets. Finally, the analysis of the absolute and relative error plots confirms the previous observations regarding power losses: the discrepancies between the two models remain small or within acceptable limits at high torque levels.

6. Conclusions

6.1. Summary of Work

In summary, this paper addressed three main topics: the methodology for measuring power losses in electric vehicles, analytical loss modeling, and the use of an on-board torque sensor compared with raw brake-based measurements, after compensating for longitudinal tire slip and test bench friction losses.

Methodology: The first contribution of this paper is the synthesizing and reviewing of the theoretical and practical elements required for experimental implementation, including calibration procedures and the compensation of losses in the tire. It also addresses the modeling aspect by applying the least-squares method and K-fold cross-validation. Appendix A provides a detailed, step-by-step guide to implementing the method proposed in this paper.

Modeling: The second contribution of this paper is the modeling of losses using a two-factor polynomial function, namely torque and rotational speed. The origin and physical justification of first-order losses are reviewed using a second-degree polynomial function, summarized in the form of a 3×3 coefficient matrix. This formulation shows that certain terms can be deliberately excluded based on physical considerations. A ‘constrained’ fitting method, enforcing strictly positive loss coefficients, is then implemented and compared with an ‘unconstrained’ approach that may yield negative coefficients. Although the ‘constrained’ method results in a limited loss of accuracy, it offers the advantages of improved coefficient interpretability and the generation of complete efficiency maps, without the need to remove regions where efficiencies may exceed unity—an issue that may occur with the ‘unconstrained’ method.

On-board torque meter: The third contribution of this paper focuses on the measurement of losses in an electric vehicle using an on-board torque sensor installed on the

final drive shaft at the wheel. This experimental approach addresses a gap in the existing literature. A comparative analysis between results obtained from the torque sensor and those derived from brake-based measurements shows that the accuracy of the latter remains acceptable when the motor torque—and consequently the longitudinal tire slip—is sufficiently high. Conversely, when the motor torque or tire slip becomes too low, the resulting errors increase significantly and become clearly unacceptable, thereby making the use of an on-board torque sensor essential.

This analysis was enabled by the use of an experimental vehicle featuring a relatively low maximum motor torque, thereby remaining within the measurement range of a standard, commercially available, ready-to-use torque sensor.

In contrast, typical electric vehicles generally feature motor torque levels that are an order of magnitude higher than those of the vehicle considered in this study. With comparable gear reduction ratios, the resulting driveshaft torque would therefore require a sensor with a much wider measurement range, making the integration of a conventional commercial torque sensor into the transmission shaft particularly challenging. One possible solution to this limitation consists of using a dedicated transmission shaft incorporating an MEMS-based torque measurement system with wireless data transmission [13]. Another approach is to combine measurement methods in a complementary manner, as follows:

- At low motor torque levels, the use of a moderately sized embedded torque sensor ensures sufficient measurement accuracy.
- At high motor torque levels, the torque sensor can be removed, and measurements can instead rely on data obtained from the brake system.

Switching measurement methods during a test campaign inherently entails a risk of systematic error propagation in the resulting loss or efficiency models and maps. To mitigate this risk, it is essential to compare data obtained using both methods within their overlapping operating range and to verify that any observed discrepancies remain within acceptable limits, as performed in the present study.

6.2. Limitations and Future Work

The first limitation of this study lies in the proposed two-variable polynomial function (torque and rotational speed), which does not account for temperature effects. Tire slip behavior, as well as losses in the motor, inverter, and gearbox, are dependent on this parameter. It would therefore be valuable to extend this work by investigating this aspect further, both from an experimental and a modeling perspective.

The second limitation concerns the development of an analytical model applied solely to the motoring region of the powertrain. The regenerative region has not been investigated. This aspect also represents a promising avenue for further extending the present work.

Author Contributions: Conceptualization, P.B., G.C. and P.H.; methodology, P.B.; validation, P.B., G.C. and P.H.; investigation, P.B.; writing—original draft preparation, P.B.; writing—review and editing, G.C. and P.H.; supervision, P.B.; project administration, G.C.; funding acquisition, G.C. All authors have read and agreed to the published version of the manuscript.

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

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

Nomenclature

The following abbreviations are used in this manuscript:

Notation	Units	Designation
δ_f	[1]	Rotational inertia factor
ω_f	[rad/s]	Front roller angular speed
ω_r	[rad/s]	Rear roller angular speed
ω_e	[rad/s]	Electric motor angular speed
ω_s	[rad/s]	Tire/rim/transmission shaft angular speed
ω_0	[rad/s]	Ideal angular speed of tire
λ	[1]	Tire longitudinal slip
η, η^*	[1]	Overall efficiency from torque meter or brake-based measurement
η_{tire}	[1]	Tire efficiency
F_x	[N]	Motor force developed by tire
F_{load}	[N]	Load cell force
F_{eddy}	[N]	Eddy current brake force
F_p	[N]	Friction force
F_r	[N]	Rolling resistance force
F_{res}	[N]	Total resistant force
G	[1]	Total gear reduction ratio
J	[kg.m ²]	Inertia of vehicle rotating parts
J_{roll}	[kg.m ²]	Inertia of rollers
m	[kg]	Mass of the vehicle
m_{eq}	[kg]	Equivalent mass simulated by rollers and rotating parts of the vehicle
$m_{eq,v}$	[kg]	Equivalent mass of the vehicle including rotating parts
$p_{in}^{inverter}$	[W]	Electric power at battery pack output/Inverter input
$p_{out}^{inverter}$	[W]	Electric power at inverter output/motor input
p_{out}^{motor}	[W]	Electric power at electric motor output/speed reducer input
p_{in}^{tire}	[W]	Electric power at Speed reducer output/tire input
p_{out}^{tire}	[W]	Electric power at tire output/roller test bench input
p_{out}^{brake}	[W]	Effective power dissipated at eddy current brake
P_{loss}, P_{loss}^*	[W]	Overall power loss from torque meter or brake-based measurement
$\hat{P}_{loss}, \hat{P}_{loss}^*$	[W]	Predicted overall power Loss from torque meter or brake based measurement
$p_{loss}^{inverter}$	[W]	Inverter power loss
p_{loss}^{motor}	[W]	Electric motor power loss
p_{loss}^{tire}	[W]	Tire power loss due to longitudinal slip
p_{loss}^{roller}	[W]	Roller test bench power loss due to internal friction
R_{roll}	[m]	Rollers radius
R_e	[m]	Effective loaded radius of tire in free-wheel condition
R	[m]	Effective loaded radius of tire under torque
T_s, T_s^*	[N.m]	Torque at drive shaft from torque meter or brake-based measurement
T_e, T_e^*	[N.m]	Torque at electric motor shaft from torque meter or brake based measurement
v_x	[m/s]	Tangential speed on rollers/Vehicle speed

Appendix A. Step-by-Step Guide to Implementing the Proposed Method

Step 1: Design the modified drive shaft to integrate the most suitable torque meter.

Step 2: Open the electrical circuit upstream of the inverter to insert a current measurement device (shunt for example).

Step 3: Measure the free-rolling radius of the tire, R_e , on the roller bench using the roller encoders and (23). For example, perform ten full manual revolutions of the wheel while recording the rollers' angular displacement.

Step 4: Calculate the vehicle's equivalent mass m_{eq} on the test bench. (Refer to Section 3.3.1).

Step 5: Perform several coast-down tests in neutral gear to determine the resistive force $F_{res} = f(v_x)$. (Refer to Section 3.3.2 and 4.)

Step 6: Calibrate the eddy current brake load cell. Furthermore, calibrate the current measurement system, whether using the shunt resistor and/or data retrieved from Battery Management System CAN messages. (Refer to Section 3.4.)

Step 7: Conduct the experimental campaign using the torque meter, covering the widest possible range of steady-state operating points. Each point should be repeated, for example, five times. Record the following measurements: U_{bat} , I_{bat} , T_s , F_{eddy} , ω_f , ω_r and ω_s or ω_e .

Step 8: If necessary, remove the torque meter equipped shaft and complete the experimental dataset by measuring: U_{bat} , I_{bat} , F_{eddy} , ω_f , ω_r and ω_s or ω_e .

Step 9: Compute the electrical input power at the inverter $P_{in}^{inverter}$ Equation (17) or (18), then the mechanical power at the wheel P_{in}^{tire} using Equation (2), and finally the motor torque T_e using Equation (31).

Step 10: At each operating point, calculate the total resistant force F_{res} using (16), then deduce the traction force $F_x = F_{eddy} + F_{res}$ (according to (4) in steady state condition) and the output power at the wheel P_{out}^{tire} using (26). Compute the tire's longitudinal slip λ using (22), and from this, determine from (25) the corrected input power P_{in}^{tire*} and finally the estimated torques T_s^* and T_e^* .

Step 11: Calculate the total power losses P_{loss} using (1) as well as P_{loss}^* . Compute the efficiencies η et η^* .

Step 12: Filter the measurement data by excluding clearly erroneous points (e.g., $F_x < 0$, $P_{loss} < 0$, $\eta > 1$ etc.). For each operating point, retain the median value of overall losses from the remaining valid samples.

Step 13: Plot $F_x = f(\lambda)$ and verify that the tire operates within its linear region. Exclude any data points that fall outside this range.

Step 14: Plot the following graphs:

- $T_e^* = f(T_e)$.
- Absolute error: $|T_e^* - T_e| = f(T_e)$.
- Relative error: $\frac{T_e^* - T_e}{T_e} = f(T_e)$.

Check for the presence of any systematic bias. If a significant bias is observed, first verify the tire radius R_e , the friction loss calibration of the test bench and the calibration of the brake load cell.

Step 15: Merge the two datasets (torque-meter- and brake-based) to produce a single measurement set covering the widest possible operating range.

Step 16: Perform an unconstrained least-squares fit using all nine coefficients k_{ij} . Check overfitting or underfitting using a K-fold cross validation process. Compute the RMSE and mean value of residuals of this 'optimal' but non-physical model. Refer to Section 3.8.

Step 17: Carry out a 'constrained' iterative fit, starting with the six physically meaningful coefficients. Ensure that the final set of coefficients should be strictly positive or zero. Check overfitting or underfitting using a K-fold cross validation process. Compute the RMSE and mean value of residuals of the 'constrained' fit and compare it to that of the 'unconstrained' fit to assess its acceptability.

Step 18: Generate loss and efficiency maps based on the identified model.

Appendix B. Technical Information

Table A1. Renault TWIZY 80 technical specifications.

Chassis	
Wheelbase [m]	1686
Front track [m]	1094
Rear track [m]	1080
Weight	
Kerb weight [kg]	474
Kerb weight on the front [kg]	206
Kerb weight on the rear [kg]	268
Speed Reducer	
Gearbox type	Automatic
Type	two Gear stages reduction
Reduction ratio	9.23
Number of forward gears	1
Efficiency [1]	0.96
Tires	
Front tires	125 × 80 R13
Rear tires	145 × 80 R13
Effective loaded radius Re [m]	0.266
Battery	
Battery type	Lithium ion
Voltage [V]	46–58
Battery weight [kg]	98
Capacity [kWh]	6.1
Max current [A]	350
Motor	
Motor type	MB L7e
Model version	M75
Motor number	3CG
Type approval	ACVYB0
Motor features	AC Induction Motor
Peak power 20 s [kW]	13
Nominal Power [kW]	8
Maximum torque	57 Nm
Motor RPM at max torque	From 0 to 2100
Propulsion type	Electric

Table A2. Torque Sensor specifications from https://www.burster.com/fileadmin/user_upload/redaktion/Documents/Products/Data-Sheets/Section_8/8661_EN.pdf (accessed on 8 February 2026).

Model	Burster 8661-5200
Measuring range	±200 [N·m]
Max. rotary speed	15,000 [RPM]
Sensitivity	±0.1 % F·S
Mass	1.5 [kg]

Table A3. Roller test bench technical specifications.

Mechanical design	Bem Muller
Electronic control	Rotronics
Brake type	Eddy current
Roll radius R_{roll} [m]	0.16
Load Cell distance d [m]	0.194
Simulated Vehicle mass [kg]	850
Total Inertia of rotation J_{roll}	21.755 [kg·m ²]
Angular coders resolution [deg]	0.1

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