

Design of an Advanced Traction Controller for an Electric Vehicle Equipped with Four Direct Driven In-Wheel Motors

Kiumars Jalali
University of Waterloo

Kai Bode
Technical University of Braunschweig

Steve Lambert and John McPhee
University of Waterloo

Copyright © 2008 SAE International

ABSTRACT

The vision for the future automotive chassis is to interconnect the lateral, longitudinal, and vertical dynamics by separately controlling driving, braking, steering, and damping of each individual wheel. A major advantage of all wheel drive electric vehicles with four in-wheel motors is the possibility to control the torque and speed at each wheel independently. This paper proposes a traction controller for such a vehicle. It estimates the road's adhesion potential at each wheel and adjusts each motor voltage, such that the longitudinal slip is kept in an optimal range. For development and validation, a full vehicle model is designed in ADAMS/View software, in co-simulation with motor and control elements, modeled in MATLAB/Simulink.

INTRODUCTION

Recently, electric vehicles (EV) have attracted a great deal of interest as an elegant solution to environmental and energy concerns. Thanks to substantial improvements in electric motor and battery technologies, EVs now have driving performance and efficiency metrics that are comparable to those of internal combustion (IC) engine vehicles. EVs have no tailpipe emissions because they have no fuel, combustion, or exhaust systems. Further benefits of EVs include a reduction in noise pollution as well as the noise, vibration, and harshness (NVH) of the vehicle itself, due to the elimination of the engine and other powertrain components. In addition, EVs are the most exciting platforms on which to apply advanced motion control techniques, since not only the torque but also the motor speed can be generated and controlled quickly and precisely. The torque response of an electric motor is on the order of a few milliseconds and, therefore, 10 to 50 times faster than that of the IC engines and hydraulic braking systems in use today [1]. Hence, the most advanced traction control system can be implemented

by controlling the electric motors' torques to maintain the optimal traction of each tire. This applies in particular to vehicles with independently driven in-wheel motors, since the ratio of drive torque between the wheels can be adjusted quickly to meet the requirements of the current driving situation.

As part of an ongoing Auto21 (a Canadian Network Center of Excellence) project, Collaborative Design Tools for Distributed, Multi-disciplinary Design Projects, an electric vehicle (AUTO21EV) is being designed. It is equipped with four direct drive in-wheel motors, an active steering system, and passive suspension systems on the front and rear axles (Figure 1).

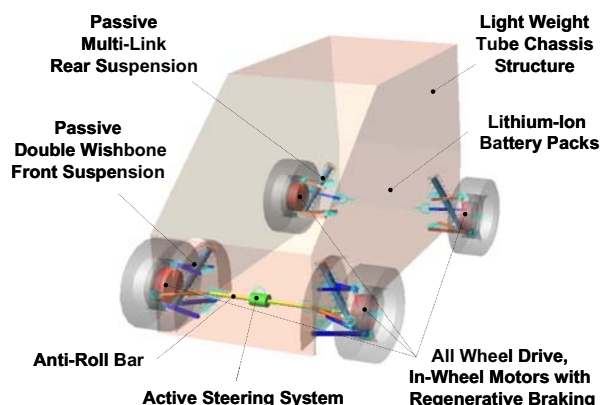


Figure 1: AUTO21EV concept

Table 1 demonstrates the relevant vehicle parameters of the AUTO21EV used in the simulations. Based on this vehicle concept, a novel traction control system is introduced, which estimates the road's adhesion potential at each tire and adjusts the motor voltage on each wheel to maintain the longitudinal slip within an ideal range. The performance of the controller has been verified based on a full vehicle model, using ADAMS/View and Pecejka tire models. Electric motors, as well as the control system elements, are modeled in a

MATLAB/Simulink environment. Traction controller and road condition estimator are validated using straight-line acceleration from rest on μ -split and wet road conditions.

parameter	value	unit
Vehicle mass (M)	1075	kg
Moment of inertia in yaw direction (I_{zz})	1171	Kg m ²
Distance from CG to front axle (a)	0.82	m
Distance from CG to rear axle (b)	0.98	m
Front track width (t_f)	1.275	m
Rear track width (t_r)	1.350	m
Front tire cornering stiffness (C_f)	45570	N/rad
Rear tire cornering stiffness (C_r)	45570	N/rad
Tire size	175/55 R15	-
Tire unloaded radius (r)	0.2905	m

Table 1: Vehicle parameters used in the simulations

SLIP RATIO CONTROL

The principal task of the control system is to accelerate the vehicle with the maximum possible traction force in the tire contact patch. Major influencing parameters of the contact force are the road's longitudinal adhesion coefficient $\mu_x = F_x / F_z$, which is calculated through dividing the tire's longitudinal force F_x by its vertical force F_z , and the longitudinal slip ratio λ_x . As a general rule, the longitudinal slip ratio is calculated for accelerating and decelerating situations using the following definitions, respectively:

$$\lambda_{x,accel} = \frac{\omega_w \cdot r_{dyn} - v_x}{\omega_w \cdot r_{dyn}}, (v_x \geq 0, \omega_w \geq 0, \omega_w \cdot r_{dyn} \geq v_x) \quad (1)$$

$$\lambda_{x,decel} = \frac{v_x - \omega_w \cdot r_{dyn}}{v_x}, (v_x \geq 0, \omega_w \geq 0, \omega_w \cdot r_{dyn} \leq v_x) \quad (2)$$

where v_x is the vehicle chassis velocity, ω_w is the wheel angular velocity and r_{dyn} is the dynamic tire radius. In this way, the slip ratio always remains a positive number in the interval of $0 \leq \lambda_x \leq 1$.

Figure 2 illustrates μ_x for three different road conditions. The curves have been simulated with the full vehicle model in ADAMS and represent different tire/road interfaces as they might appear on a dry, wet, and icy road, respectively. The maximum adhesion coefficient occurs at 17% slip ratio on a dry road, 10% slip ratio on a wet road, and 5% slip ratio on an icy road. Therefore, the mission of the traction controller is to adjust the longitudinal slip ratio at each individual wheel to the value that provides the highest adhesion coefficient between the tire and the road. In this way, the vehicle is able to negotiate a μ -split road, a road with two different adhesion coefficients on its surface, with the least

possible velocity lost and with the highest possible traction.

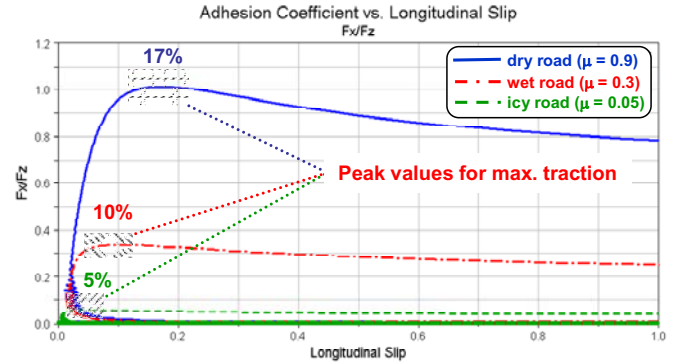


Figure 2: Adhesion coefficient versus longitudinal slip for a specific Pacejka tire model

CONTROL PATH

To fully exploit the advantages of the in-wheel motor concept, each tire is equipped with its own control system. The principal technique is, however, the same at each wheel. Figure 3 shows the control loop, consisting of a PID controller, the electric motor model, and the ADAMS dynamic model of the vehicle as a Simulink sub-block.

The control variable is the wheel's longitudinal slip λ_x , however, the wheel speed n is fed back internally in order to determine the induction voltage in the motor coils. The saturation element limits the applied voltage to ± 200 V, which is a typical voltage range for in-wheel motors.

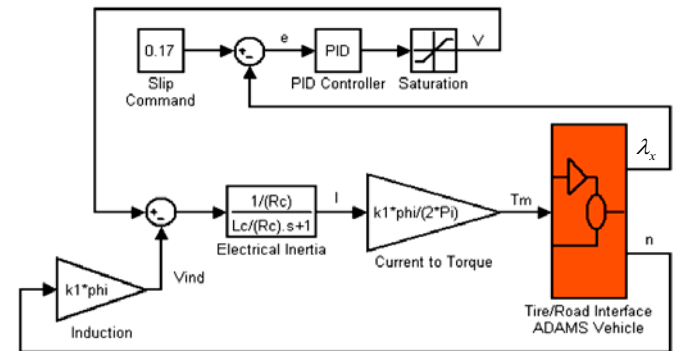


Figure 3: Traction control loop for a single wheel

MOTOR MODEL

In-wheel motors on the market are usually designed as direct drive permanent magnet synchronous machines (PMSM). No gear reduction is used between the motor and the wheel drive shaft. This reduces weight and size of the system, but it also requires that the speed-torque characteristic directly meets the requirements of a vehicle. These machines are characterized by having permanent magnets in the rotor, usually rare-earth magnets to increase the power density. They follow a continuously rotating magnetic field in the stator, which

they are brushless and robust. These motors are designed 'upside down', which means that the rotor is the rotating external part, which can also be attached to the rim, and the stator is the stationary part attached to the wheel shaft. This is primarily done to reduce the moment of inertia of the rotating parts so that less energy is required to stop or accelerate the wheels. In addition, PMSMs have the capability to increase performance and efficiency of motors by eliminating the magnetization loss in the field of the motors. The flux weakening characteristics of the PMSM not only improves the efficiency of the motors, but also prevents overheating of the tires by the rotor [2].

As a first attempt and for simplification purposes, a permanent magnet DC motor (PMDC) is modeled, to provide a good approximation of the system dynamics. PMDC motors have permanent magnets in the stator and coils in the rotor, through which an alternating current flows [3]. The dynamic behavior can be expected to be similar to a synchronous machine of the same size. In the future, the PMDC motor will be replaced by an appropriate PMSM.

The dynamic behavior of the PMDC motor is expressed as a Laplace Transform in the following, giving a relationship between input voltage $\hat{V}$, rotor speed $\hat{n}$ and output torque $\hat{T}_m$:

$$\hat{T}_m = \frac{k_1 \phi}{2\pi (L_c / R_c) s + 1} (\hat{V} - k_1 \phi \hat{n}) \quad (3)$$

For a simplified controller design, the induction of the coils is ignored, which results in a first order lag element (PT₁) as the transfer function from voltage $\hat{V}$ to output torque $\hat{T}_m$:

$$TF_{T_m}(s) = \frac{\hat{T}_m}{\hat{V}} = \frac{k_1 \phi}{2\pi (L_c / R_c) s + 1} \quad (4)$$

The motor parameters, being the motor resistance R_c , the motor inductance L_c , the magnetic flux ϕ and the machine constant k_1 , are adapted such that the model reflects the behavior of the PMSM as closely as possible. They are obtained with the simple PMDC motor equations and the following assumptions:

- The maximum mechanical power $P_{mech,max}$ of each in-wheel motor under steady-state condition is 25 kW.
- The maximum torque $T_{m,max}$ of each in-wheel motor is 300 Nm.
- The maximum voltage available is 200 V.

DRIVE TORQUE TO SLIP TRANSFER FUNCTION

The transfer function from torque to slip, required to fully describe the block diagram in Figure 3, is approximated from simulation results of the vehicle model on a dry road. The technique, as common in dynamic systems analyses, is to apply a torque step input at all four wheels without any steering input, and to monitor the slip response. The output will be the same between left and right sides. After steady-state has been reached, the longitudinal slip ratio at the front will be higher than at the rear, due to weight transfer caused by the acceleration. For slip controlled braking the reverse will hold. For a step input of 100 Nm, the simulation result is presented in Figure 4.

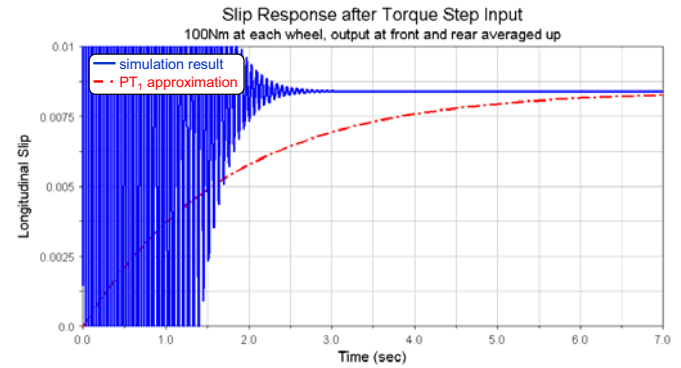


Figure 4: Actual and approximated slip response

The observed oscillations in the first two seconds are caused by the Pacejka tire model. The high amplitudes occur due to the fact that the rotational speed ω_w , which appears in the denominator of equation (1), is close to zero immediately after starting from rest and the fact that it takes some time for the vehicle model to find its steady-state position. The relation between motor torque and slip ratio is approximated as a first order lag element, which can be written as follows:

$$TF_{\lambda}(s) = \frac{\hat{\lambda}_x}{\hat{T}_m} = \frac{K_{sl}}{1 + \tau_{sl}s} \quad (5)$$

Due to the oscillations in the slip response, it seems contradictory at first glance that a PT₁ element is not a good approximation to capture the slip response. This is, however, consistent with the considerations in [4], where the transfer function from motor torque to traction force for an electric vehicle were modeled as a first order lag element. Assuming that the slip ratio remains in the range of zero and the peak friction, where it is approximately linearly related to the adhesion coefficient (Figure 2), and assuming no lag between traction force and slip, the estimation of the transfer function from motor torque to slip ratio is justifiable. This is shown as the dot-dash line in Figure 4. The parameters K_{sl} and τ_{sl} are derived from the step response, using the slip value at steady-state, divided by the applied test torque, and the time constant of the first order lag element.