

The Application of Control and Wheel Torque Allocation Techniques to Driving Modes for Fully Electric Vehicles

Andrew Pennycott, Leonardo De Novellis, Aldo Sorniotti, and Patrick Gruber
Univ. of Surrey

ABSTRACT

The combination of continuously-acting high level controllers and control allocation techniques allows various driving modes to be made available to the driver. The driving modes modify the fundamental vehicle performance characteristics including the understeer characteristic and also enable varying emphasis to be placed on aspects such as tire slip and energy efficiency.

In this study, control and wheel torque allocation techniques are used to produce three driving modes. Using simulation of an empirically validated model that incorporates the dynamics of the electric powertrains, the vehicle performance, longitudinal slip and power utilization during straight-ahead driving and cornering maneuvers under the different driving modes are compared.

The three driving modes enable significant changes to the vehicle behavior to be induced, allowing the responsiveness of the car to the steering wheel inputs and the lateral acceleration limits to be varied according to the selected driving mode. Furthermore, the different driving modes have a significant impact on the longitudinal tire slip, the motor power losses and the total power utilization. The control and wheel torque allocation methods do not rely on complex and computationally demanding online optimization schemes and can thus be practically implemented on real fully electric vehicles.

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INTRODUCTION

Electric vehicles are an important area of automotive technology due to their potential for lower emissions outputs than cars powered by fossil fuels [1]. The transportation sector produces around one quarter of all carbon dioxide emissions [2], which are an area of particular focus due to their connection with global warming [3]. Electric vehicles have also come into focus due to concerns about the price and availability of fossil fuels [4]. Nevertheless, limited range is still a major shortcoming of electric vehicles [5]. Measures such as regenerative braking and energy efficient control allocation that aim to enhance energy efficiency have the potential to increase range and thereby alleviate one of the major drawbacks of the technology.

The handling and fun-to-drive elements of fully electric vehicles can be enhanced using continuously-acting control systems [6]. For instance, the understeer characteristic which governs the steady-state lateral acceleration response of the car to steering wheel inputs can be modified. Electric motor actuators that are individually controllable offer greater improvements in

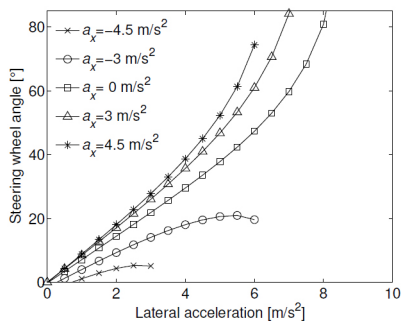
vehicle performance over alternative technologies such as torque vectoring differentials which typically have limitations in torque transfer and bandwidth.

Control systems produce reference levels for the overall longitudinal force to satisfy acceleration and deceleration requirements and a corrective yaw moment for cornering maneuvers. For electric vehicles equipped with four electric motors, these two targets can be realized via an infinite number of combinations in the individual wheel torques; the assignment of the four wheel torques to meet the high level targets is referred to as the control allocation problem [7]. Although each wheel torque combination will produce the same net traction and yaw moment, the combinations will differ with regard to other aspects such as power utilization and tire slip.

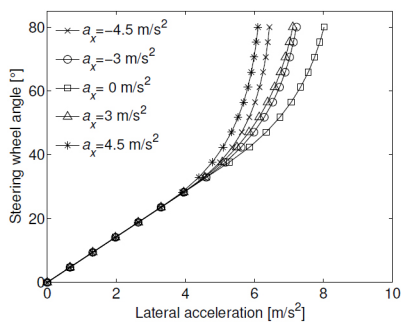
In this paper, three driving modes are defined which differ in terms of their understeer characteristic and secondary cost function. Simulations are used to demonstrate the effects of the various driving modes on the dynamic behavior of the vehicle, on the induced longitudinal slip ratio and the power losses incurred during various types of maneuver.

OBJECTIVES AND DRIVING MODES

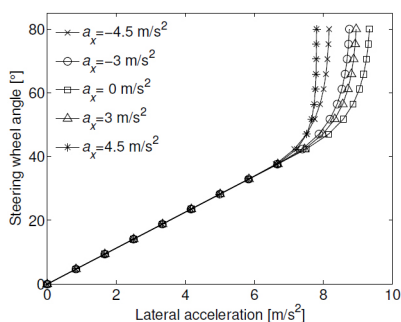
The main objective of the control allocation is to alter the vehicle dynamics performance. A target understeer characteristic - the steady state relationship between the steering wheel angle and the lateral acceleration - is to be realized through continuously producing a corrective yaw moment. For the passive, uncontrolled vehicle, the understeer characteristic changes with the longitudinal acceleration. The goal of the control system presented in this paper is firstly to reduce the impact of the longitudinal acceleration on the yaw behavior, secondly to extend the linear region of the understeer characteristic and thirdly to alter the gradient of the relationship in this linear region. Two different understeer characteristics - A and B - are used in this study and are shown in [figure 1](#), along with the understeer characteristic of the passive vehicle to which no corrective yaw moment is applied.



(a) Passive Vehicle



(b) Understeer Characteristic A



(c) Understeer Characteristic B

Figure 1. Understeer characteristic of passive vehicle and the two target understeer characteristics (A and B) of control allocation.

A key feature of the system is over-actuation: there are two targets (traction force and yaw moment) to be realized through the action of four wheel torques. Therefore, since the targets can be met using numerous combinations of the four wheel torques, in addition to meeting the reference traction and yaw moment, a secondary cost function can be minimized. Such a cost function can reflect various aspects of vehicle behavior. Two secondary cost functions are used in this paper. Cost function 1 aims to reduce the overall level of tire slip by allocating the wheel torques according to the vertical loads on the tires. On the other hand, cost function 2 allocates the wheel torques such that the motor power losses are minimized. Mathematical details of the two cost functions are provided in later sections.

Three driving modes are utilized in this paper: normal, eco and sport. Each combines a specific understeer characteristic with a secondary cost function; the different combinations are included in [table 1](#).

Table 1. Different combinations of understeer characteristics ([figure 1](#)) and cost functions yield the three driving modes.

Driving Mode	Understeer characteristic	Secondary cost function
Normal	A	1
Eco	A	2
Sport	B	1

In order to realize the two different understeer characteristics, a reference yaw rate, r_{ref} , is generated according to vehicle states such as the longitudinal speed and longitudinal acceleration; details of this calculation procedure are given in [6]. A target traction force, F_x^{ref} , is set according to input from the driver in order to maintain the required vehicle speed. Moreover, a high level controller, which comprises a combination of feedforward and proportional-integral-derivative controllers, drives the actual yaw rate, r , to the target value by imposing a reference yaw moment, M_z^{ref} . The reference yaw moment is produced through setting appropriate values of the four wheel torques contained in vector τ_w . The overall control structure is shown in [figure 2](#).

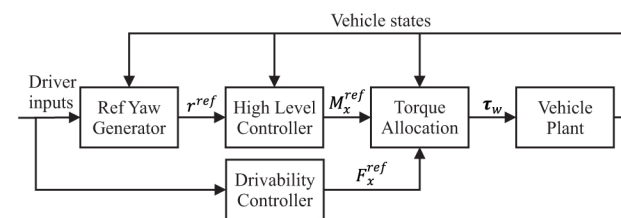


Figure 2. Overall control structure.

CONTROL ALLOCATION EQUATIONS

Relating Wheel Torques with Control Allocation Targets

The front-left, front-right, rear-left and rear-right wheels are referred to using the subscripts 1, 2, 3 and 4, respectively. Using the basic vehicle geometry (figure 3) including the halftrack, t_h , front- and rear-semi wheel bases, b_f and b_r , and the wheel radius, r_w , the wheel torques at the four wheels of the car, $\tau_{w,1-4}$, can be related to the traction force, F_x , and yaw moment, M_z .

The net traction force and yaw moment are

$$F_x = \frac{1}{r_w} (\tau_{w,1} \cos \delta_{w,1} + \tau_{w,2} \cos \delta_{w,2} + \tau_{w,3} + \tau_{w,4}) \quad (1)$$

$$M_z = \frac{1}{r_w} (\tau_{w,1} (-t_h \cos \delta_{w,1} + b_f \sin \delta_{w,1}) + \tau_{w,2} (t_h \cos \delta_{w,2} + b_f \sin \delta_{w,2}) - \tau_{w,3} t_h + \tau_{w,4} t_h). \quad (2)$$

$\delta_{w,1}$ and $\delta_{w,2}$ represent the counter-clockwise wheel angles at the front-left and front-right wheels (not shown in the figure). Assuming small wheel angles:

$$F_x \approx \frac{1}{r_w} (\tau_{w,1} + \tau_{w,2} + \tau_{w,3} + \tau_{w,4}) \quad (3)$$

$$M_z \approx \frac{t_h}{r_w} (-\tau_{w,1} + \tau_{w,2} - \tau_{w,3} + \tau_{w,4}). \quad (4)$$

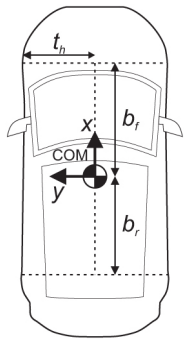


Figure 3. Vehicle geometry used for control allocation formulation.

Let γ_l and γ_r be the front-to-total wheel torque ratios on the left- and right-hand sides of the car, i.e.

$$\gamma_l = \frac{\tau_{w,1}}{\tau_{w,1} + \tau_{w,3}} \quad (5)$$

$$\gamma_r = \frac{\tau_{w,2}}{\tau_{w,2} + \tau_{w,4}}. \quad (6)$$

Solving equations (3), (4), (5), (6) gives an approximate solution to the wheel torque allocation problem.

$$\tau_{w,1} = \frac{r_w \gamma_l}{2} \left(F_x^{ref} - \frac{M_z^{ref}}{t_h} \right) \quad (7)$$

$$\tau_{w,2} = \frac{r_w \gamma_r}{2} \left(F_x^{ref} + \frac{M_z^{ref}}{t_h} \right) \quad (8)$$

$$\tau_{w,3} = \frac{r_w (1 - \gamma_l)}{2} \left(F_x^{ref} - \frac{M_z^{ref}}{t_h} \right) \quad (9)$$

$$\tau_{w,4} = \frac{r_w (1 - \gamma_r)}{2} \left(F_x^{ref} + \frac{M_z^{ref}}{t_h} \right) \quad (10)$$

The wheel torque solutions depend on the traction force and yaw moment demands. The sums of the left- and right-hand wheel torques are

$$\tau_{w,l}^{tot} = \frac{r_w}{2} \left(F_x^{ref} - \frac{M_z^{ref}}{t_h} \right) \quad (11)$$

$$\tau_{w,r}^{tot} = \frac{r_w}{2} \left(F_x^{ref} + \frac{M_z^{ref}}{t_h} \right). \quad (12)$$

The wheel torques can now be expressed in the form

$$\tau_{w,1} = \tau_{w,l}^{tot} \gamma_l \quad (13)$$

$$\tau_{w,2} = \tau_{w,r}^{tot} \gamma_r \quad (14)$$

$$\tau_{w,3} = \tau_{w,l}^{tot} (1 - \gamma_l) \quad (15)$$

$$\tau_{w,4} = \tau_{w,r}^{tot} (1 - \gamma_r). \quad (16)$$

Secondary Cost Functions

Both the secondary cost functions applied in this paper determine target values for the front-to-total wheel torque ratios, denoted as γ_l^{ref} and γ_r^{ref} . The actual ratios, γ_l and γ_r , are driven to these values by online quadratic programming. The two cost functions differ in terms of how the reference ratios are defined: cost function 1 defines the reference ratios according to the vertical loads so that the overall tire slip is reduced, while cost function 2 sets γ_l^{ref} and γ_r^{ref} according to the efficiency characteristics of the motor as detailed in the sections below.