

Comprehensive Optimization of Dynamics Performance and Energy Consumption for an Electric Vehicle via Coordinated Control of SBW and FIWMA

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ABSTRACT

This paper presents a coordinated controller for comprehensive optimization of vehicle dynamics performance and energy consumption for a full drive-by-wire electric vehicle, which is driven by a four in-wheel motor actuated (FIWMA) system and steered by a steer-by-wire (SBW) system. In order to coordinate the FIWMA and SBW systems, the mechanisms influencing the vehicle dynamics control performance and the energy consumption of the two systems are first derived. Second, the controllers for each subsystem are developed. For the SBW system, a triple-step control technique is implemented to decouple the yaw rate and sideslip angle controls. The FIWMA system controller is designed with a hierarchical control scheme, which is able not only to satisfy the yaw rate and sideslip angle tracking demands, but also to deal with actuation redundancy and constraints. Finally, coordinated control is proposed to obtain the comprehensive optimization of the vehicle dynamics control performance and energy consumption. The developed coordinated control is convenient for implementation as the structures and control algorithms of the subsystems remain unchanged. In this way, the control algorithms of the subsystems can be developed independently. Simulations are carried out with a CarSim software full-vehicle model under two typical driving scenarios, and with different road conditions. The results show the effectiveness of the developed coordinated control.

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1. INTRODUCTION

Electric vehicles enjoy several advantages in terms of energy efficiency, driving comfort, handling performance, and stability compared with internal combustion engine vehicles [1,2,3]. Among all types of electric vehicles, the electric vehicle that uses x-by-wire technology on all of its actuators, namely, a full drive-by-wire vehicle in which four-wheel torque and steering angles are independently actuated by electric motors, is an emerging configuration, thanks to both fast and precise electric motor torque responses and actuation flexibility [4].

As a full drive-by-wire vehicle is a highly over-actuated system, the actuator redundancy makes the coordinated control of this type of vehicle challenging but rewarding. Numerous studies have been conducted to improve vehicle dynamics control performance through coordination of the subsystems, by such means as active steering, braking, and active suspension systems. In [5], a four-wheel-distributed traction, braking, and steering vehicle was studied. It was proven that the stability performance was enhanced by coordinated control of the steering system and braking system; A holistic model predictive control (MPC)-based yaw stability control system via coordination between active front steering and motor torque distribution is discussed in [6]. A hierarchical control scheme, which

enjoys excellent real-time performance without losing control accuracy, has been widely adopted for vehicle dynamics control [7, 8]. Redundant degrees of freedom provide additional flexibility in control allocation design [9, 10], thus, various objective functions can be formulated to obtain additional performance. A sliding-mode based control algorithm is proposed in [11], in which an analytic method is utilized to solve the tire force coordination problem. In [12], a quadratic optimization based control allocation problem is formulated to coordinate the tire forces to each wheel, minimizing not only the tracking error between the desired and the actual yaw rate, but also the tire working load and tire energy dissipation. The authors of [13] propose a method to optimize vehicle handling performance and energy efficiency by considering in-wheel motor efficiencies. However, only the coordinated control of the four in-wheel motors is taken into account; steering control is not considered in [14].

In the above studies, the coordination of the subsystems is achieved by adjusting the relative weighting between each system during control allocation. This causes difficulties at the practical level, since the control algorithms of the subsystems, which are already well developed, have to be modified, as motion control is no longer carried out inside the subsystem layer. Thus, an upper coordinated layer is developed in [13]. In this way, the control algorithms of the

subsystems remain unchanged. However, only vehicle dynamics control performance is considered in [15]; analysis and optimization of energy consumption is not included.

We aim to deal with the coordinated control of multiple subsystems, namely the four in-wheel motor actuated (FIWMA) system and the steer-by-wire (SBW) system, in order to simultaneously enhance the vehicle handling control performance and energy efficiency in normal driving conditions. First, the dynamics control performance and energy consumption properties of the subsystems need to be analyzed. Then, the control algorithms of the subsystems are developed. Based on the control algorithms of these subsystems, a simple but effective rule-based coordinated control scheme is proposed to obtain the comprehensive optimization of both the vehicle dynamics performance and energy consumption. Unlike the coordinated control conducted in the control allocation layer in [6,7,8,9,10,11,12,13,14], the developed rule-based coordinated control scheme leaves the control algorithms of each subsystem unchanged which is easy at practical level.

2. VEHICLE DYNAMICS MODELING

In this paper, a linear model is utilized for the SBW controller design, and a nonlinear model is developed for high-level motion controller design for the FIWMA system. Additionally, to control the wheel slip ratio and steering angles, the wheel dynamics and SBW system are modeled.

2.1. Vehicle Dynamics Model for SBW Controller Design

A steering control-oriented vehicle dynamics model with added rear steering was modified from the common linear bicycle model [7]. The state space form of the linear vehicle model is derived as the following:

$$\dot{\Omega} = A\Omega + Bu \quad (1)$$

where $\Omega = [\beta \ \gamma]^T$ and $u = [\delta_f \ \delta_r]^T$. In addition,

$$A = \begin{bmatrix} -\frac{2(C_f + C_r)}{MV_x} & -(1 + \frac{2C_f l_f - 2C_r l_r}{MV_x^2}) \\ -\frac{2(C_f l_f - C_r l_r)}{I_z} & -\frac{2(C_f l_f^2 + C_r l_r^2)}{I_z V_x} \end{bmatrix} \text{ and } B = \begin{bmatrix} \frac{2C_f}{MV_x} & \frac{2C_r}{MV_x} \\ \frac{2C_f l_f}{I_z} & -\frac{2C_r l_r}{I_z} \end{bmatrix},$$

where M is the vehicle mass, V_x is the longitudinal speed, I_z is the yaw moment inertia, and C_f and C_r are the cornering stiffness of the front and rear tires, respectively. The distances from the center of gravity to the front and rear axles are denoted as l_f and l_r , respectively. γ is the yaw rate, β is the vehicle sideslip angle, and δ_f and δ_r are the steering angles of the front and rear axles, respectively.

2.2. Vehicle Dynamics Model for FIWMA Controller Design

A three degrees of freedom vehicle dynamics model is used for FIWMA controller design. The vehicle motion equations are given in state space form as follows:

$$\dot{\mathbf{x}} = \begin{bmatrix} x_2 x_3 - \frac{\rho}{2M} C_D A_f x_1^2 \\ -x_1 x_3 \\ 0 \end{bmatrix} + \begin{bmatrix} \frac{1}{M} & 0 & 0 \\ 0 & \frac{1}{M} & 0 \\ 0 & 0 & \frac{1}{I_z} \end{bmatrix} \begin{bmatrix} F_x \\ F_y \\ M_z \end{bmatrix} \quad (2)$$

where V_y is the lateral speed, ρ is the air density, C_D is the coefficient of drag, and A_f is the front area of the vehicle. When the system state is defined as $\mathbf{x} = [x_1 \ x_2 \ x_3]^T = [V_x \ V_y \ \gamma]^T$ and the input vector as $\mathbf{v} = [F_x \ F_y \ M_z]^T$, the generalized longitudinal forces, lateral forces, and yaw moment constituted by the longitudinal and lateral forces of each tire are calculated as follows:

$$F_x = F_{xlf} \cos \delta_{lf} - F_{y lf} \sin \delta_{lf} + F_{xrf} \cos \delta_{rf} - F_{y rf} \sin \delta_{rf} \\ + F_{xlr} \cos \delta_{lr} - F_{y lr} \sin \delta_{lr} + F_{xrr} \cos \delta_{rr} - F_{y rr} \sin \delta_{rr} \quad (3)$$

$$F_y = F_{y lf} \cos \delta_{lf} + F_{x lf} \sin \delta_{lf} + F_{y rf} \cos \delta_{rf} + F_{x rf} \sin \delta_{rf} \\ + F_{y lr} \cos \delta_{lr} + F_{x lr} \sin \delta_{lr} + F_{y rr} \cos \delta_{rr} + F_{x rr} \sin \delta_{rr} \quad (4)$$

$$M_z = \frac{d_f}{2} (F_{y lf} \sin \delta_{lf} - F_{x lf} \cos \delta_{lf} + F_{y lr} \sin \delta_{lr} - F_{x lr} \cos \delta_{lr}) \\ + \frac{d_r}{2} (F_{x rf} \cos \delta_{rf} - F_{y rf} \sin \delta_{rf} + F_{x rr} \cos \delta_{rr} - F_{y rr} \sin \delta_{rr}) \\ + l_f (F_{x lf} \sin \delta_{lf} + F_{y lf} \cos \delta_{lf} + F_{x rf} \sin \delta_{rf} + F_{y rf} \cos \delta_{rf}) \\ + l_r (-F_{x lr} \sin \delta_{lr} - F_{y lr} \cos \delta_{lr} - F_{x rr} \sin \delta_{rr} - F_{y rr} \cos \delta_{rr}) \quad (5)$$

where F_{xij} , F_{yij} , and δ_{ij} ($ij \in \{lf, rf, lr, rr\}$) represent the longitudinal forces, lateral forces, and steering angles of the left front, right front, left rear, and right rear tires, respectively. The sideslip angle of each wheel α_{ij} can be calculated from the following equations:

$$\alpha_{lf} = -\delta_{lf} + \arctan \frac{V_y + l_f \gamma}{V_x - \frac{d_f}{2} \gamma} \quad (6a)$$

$$\alpha_{rf} = -\delta_{rf} + \arctan \frac{V_y + l_f \gamma}{V_x + \frac{d_f}{2} \gamma} \quad (6b)$$

$$\alpha_{lr} = -\delta_{lr} + \arctan \frac{V_y - l_r \gamma}{V_x - \frac{d_r}{2} \gamma}$$

$$\alpha_{rr} = -\delta_{rr} + \arctan \frac{V_y - l_r \gamma}{V_x + \frac{d_r}{2} \gamma} \quad (6d)$$

where d_f and d_r are the vehicle treads of the front and rear axles, respectively.

2.3. Wheel Dynamics and Tire Modeling

In this paper, Pacejka's magic formula tire model [15] is used to generalize the tire longitudinal force, lateral force, and self-aligning moment. The rotational dynamics of each wheel is given as follows:

$$J_w \dot{\omega}_{ij} = T_{wij} - RF_{xij} \quad (7)$$

where J_w is the rotational moment of inertia of the wheel, T_{wij} is the driving or braking torque applied to the wheel, R is the tire rolling radius, and F_{xij} is the tire longitudinal force from the ground.

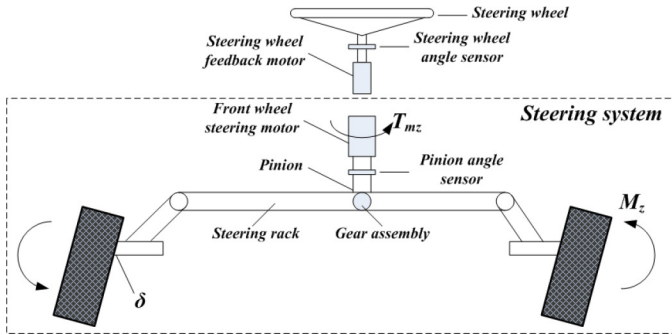


Figure 1. Configuration of front axle steer-by-wire (SBW) system.

2.4. SBW System Modeling

The architecture of the SBW system discussed in this paper is illustrated in Fig. 1. The dynamics of the steering system are modeled as a second order system, as follows:

$$J_{wz} \ddot{\delta}_{ij} + b_{wz} \dot{\delta}_{ij} + M_{zij} = T_{mzij} i_s \quad (8)$$

where J_{wz} is the moment of inertia of the Z axle at the road wheel, b_{wz} is the steering system damping at the road wheel, M_{zij} is the self-aligning moment calculated from magic formula tire model, i_s is the steering ratio, and T_{mzij} is the steering motor torque, which can be written in terms of the motor constant, k_m , motor current, i_m , motor efficiency, η , and motor transmission ratio, i_p :

$$T_{mzij} = k_m i_m i_p \eta \quad (9)$$

3. CONTROLLER DESIGN FOR SUBSYSTEMS

The controllers of each subsystem should be designed first, as the main concern about this paper is to develop the coordinated control aims at improving overall vehicle performance, while leaving the

controllers of the subsystems unchanged for implementation convenience. In this paper, the linear bicycle model is used as the reference model as the same in [13].

3.1. Controller Design for SBW System

The structure of the SBW system controller is illustrated in Fig. 2. A triple-step control technique is used [16, 17] to generate the steering angle command and to decouple the yaw rate and sideslip angle controls. A combined feed forward and feedback scheme is developed to calculate the steering motor torque command.

The linear model from Section 2.1 is utilized because the SBW system is effective only in linear tire working regions. The steady-state-like control $\delta_s = [\delta_{fs} \ \delta_{rs}]^T$ is derived by setting $\dot{\beta} = 0$ and $\dot{\gamma} = 0$ in Eq. (1). The explicit forms of u_{1s} and u_{2s} are shown in Eq. (10).

$$\delta_{fs} = \beta + \frac{1}{2C_f} \left(\frac{mV_x}{l_r + l_f} + 2C_f \frac{l_f}{V_x} \right) \gamma, \quad \delta_{rs} = \beta + \frac{1}{2C_r} \left(mV_x - 2C_r \frac{l_r}{V_x} - \frac{mV_x}{l_r + l_f} \right) \gamma \quad (10)$$

To offset the negative effects caused by reference variations in the control system, feed forward control is used. The control law is constituted by combining steady-state-like control and feed forward-like control together as follows:

$$\delta_f = \delta_{fs} + \delta_{ff}, \quad \delta_r = \delta_{rs} + \delta_{rf} \quad (11)$$

By substituting Eqs. (10) and (11) into Eq. (1), the feed forward-like controls δ_{ff} and δ_{rf} , which are related to reference model variations, are determined as follows.

$$\delta_{ff} = \frac{I_z \dot{\gamma}^*}{2C_f(l_f + l_r)}, \quad \delta_{rf} = -\frac{I_z \dot{\gamma}^*}{2C_r(l_f + l_r)} \quad (12)$$

To eliminate the tracking error, corresponding feedback control needs to be designed. To this end, adding feedback terms to Eq. (11) allows the control law to be generalized as follows:

$$\delta_f = \delta_{fs} + \delta_{ff} + \delta_{fe}, \quad \delta_r = \delta_{rs} + \delta_{rf} + \delta_{re} \quad (13)$$

Substituting Eqs. (10), (12), and (13) into Eq. (1) leads to

$$mV_x \dot{\beta} = 2C_f \delta_{fe} + 2C_r \delta_{re}, \quad I_z (\dot{\gamma} - \dot{\gamma}^*) = 2C_f l_f \delta_{fe} - 2C_r l_r \delta_{re} \quad (14)$$

To make e_β and e_γ exponentially stable, the feedback control law is chosen as the form described in Eq. (15).

$$\delta_{fe} = \frac{k_\beta mV_x l_r e_\beta + k_\gamma I_z e_\gamma}{2C_f(l_f + l_r)}, \quad \delta_{re} = \frac{k_\beta mV_x l_f e_\beta - k_\gamma I_z e_\gamma}{2C_r(l_f + l_r)} \quad (15)$$

By combining Eqs. (10), (12), and (15), the control law is a whole given as follows: