

A Novel Hierarchical Global Chassis Control System for Distributed Electric Vehicles

Yugong Luo, Kun Cao, Yifan Dai, Wenbo Chu, and Keqiang Li
Tsinghua Univ.

ABSTRACT

The current global chassis control (GCC) frequently makes use of decoupled control methods which depend on driving condition partition and simple rule-based vertical force distribution, and are insufficient to obtain optimal vehicle dynamics performance. Therefore, a novel hierarchical global chassis control system for a distributed electric vehicle (DEV), which is equipped with four wheel driving/steering and active suspension systems, is developed in this paper. The control system consists of three layers: in the upper layer, the desired forces/moments based on vehicular driving demands are determined; in the middle layer, a coordinated control method of longitudinal/lateral/vertical tire forces are proposed; in the lower layer, the driving/steering/suspension control is conducted to realize each distributed tire force. As the most outstanding contribution of this paper, a non-convex optimization problem with multiple constraints for coordinated control of longitudinal/lateral/vertical tire forces is solved, in which (1) tire force distribution problem is theoretically concluded as a constrained non-convex optimization problem, (2) a unique objective function that combines the tire workload and the dynamic ratio of the vertical forces is designed to evaluate tire force distribution, (3) 14 constraints including vehicular driving demands, tire friction limitations and actuator natures are involved to bound each tire force reasonably, and (4) an algorithm that combines constrained optimization and feasible region planning is proposed to solve the constrained non-convex optimization problem. Simulation results based on Matlab/Simulink and CarSim show that the proposed hierarchical global chassis control system effectively achieves better vehicle attitude and handling stability during the accelerated double lane change scenario compared with the other GCC methods.

CITATION: Luo, Y., Cao, K., Dai, Y., Chu, W. et al., "A Novel Hierarchical Global Chassis Control System for Distributed Electric Vehicles," *SAE Int. J. Passeng. Cars – Electron. Electr. Syst.* 7(2):2014, doi:10.4271/2014-01-0091.

INTRODUCTION

In the last decades, many kinds of electronic control systems, including the traction control system (TCS), the direct yaw moment control (DYC), the active front/rear wheel steering (AFS/ARS) and the active suspension system (ASS), have been studied and developed for the vehicle chassis control. These systems expand the controllability of longitudinal, lateral or vertical tire forces to improve vehicle dynamics performance, respectively [1]. However, stand-alone systems may lead to contradictory control objectives due to the coupling of tire forces and vehicle motions. Therefore, global chassis control (GCC) which coordinates driving/braking/steering/suspension subsystems has been widely researched in the recent years [2].

Among different methods for the coordinated control of braking/driving and suspension systems, a typical strategy was to adjust the active suspension forces along with the change of the desired tire braking forces and reduce braking the distance as a result [3]. Based on this idea, vehicle pitch control was also used to improve the vehicle attitude and the ride comfort in specific driving conditions when braking safety is not the critical concern [4]. Another coordinated control method was to

design different braking and suspension control strategies based on the partition of a slip ratio index [5]. Other methods also discussed coordination of the braking and suspension systems which applied braking to reduce the vehicle velocity and improve the roll safety with the aid of active suspension forces in critical driving conditions [6].

For coordination of the steering and suspension systems, a common idea was to enlarge the lateral tire force limitation and enhance the vehicle lateral stability by better adjustment of the wheel steering angles and the vertical tire forces by using active steering and active suspension systems [7,8,9]. Control methods including fuzzy logic, PID control and simple empirical rules were used to design the coordination strategies. There was also research that focused on the roll stability by controlling steering-by-wire systems and active suspension systems based on a roll index [10].

A lot of research has been performed for coordination of driving/braking and steering systems to enhance vehicle dynamics performance on the horizontal plane. In these studies, optimal distribution of longitudinal and lateral tire

forces was identified as the key problem and different objective functions for tire force distribution evaluation were proposed. The typical objective functions included minimizing the weighted sum of tire workloads [11, 12], minimizing the maximal tire workload [13], maximizing the remainder tire friction [14], and minimizing weighted sum of the workload average and variance, which could simultaneously reduce the total tire workload and tire workload variance [15]. Other research restricted the four tire workloads to be equal as constraints for the longitudinal/lateral tire force distribution problem [16, 17].

Although coordinated control from two directions of longitudinal/lateral/vertical vehicle dynamics were widely discussed as above, global chassis control which indeed coordinated driving/braking, steering and suspension systems were far from satisfactory. W. Cho et al. discussed coordinated control of the four wheel independent driving, the active front steering and the active suspension system, but vertical tire forces were just adjusted by simple distribution of active suspension forces between the front (left) and rear (right) tires [18]. J. Yoon et al. designed different control strategies by dividing the driving condition into three kinds of control modes (roll control, yaw control and sideslip angle control) [19]. S. Lu et al. also developed control strategies by different combination of braking/steering/suspension systems according to different driving conditions [20]. S. Fergani et al. developed a method to enhance tire grip by controlling vertical tire forces, braking forces and steering angle, but vertical tire force control was separated and decoupled from longitudinal and lateral tire force control, which might lead to compromises in driving and cornering performance [21].

To conclude existing GCC methods, optimization methods were only developed for coordinated longitudinal/lateral tire force control, while coordinated longitudinal/lateral/vertical tire force control was mostly manually decoupled by means of driving condition partition and non-rigorous empirical rules, which was not able to improve both vehicle handling stability and vehicle attitude simultaneously. Therefore, it is necessary to apply optimization methods into coordination of longitudinal/lateral/vertical tire forces to obtain optimal tire forces theoretically. In the meanwhile, to provide more freedom and potential for tire force controllability, distributed electric vehicles (DEVs) equipped with four wheel driving, active steering and active suspension systems have been widely studied [22, 23, 24].

In this paper, a novel GCC system for a distributed electric vehicle (DEV) with four wheel driving/steering and active suspension systems is developed. Firstly, the architecture of the proposed hierarchical global chassis control system is presented, and then each layer of the control system is explained in detail for clear demonstration of our ideas. Afterwards, simulations using Matlab/Simulink and CarSim are conducted to validate the proposed GCC system.

ARCHITECTURE OF HIERARCHICAL GLOBAL CHASSIS CONTROL SYSTEM

As an overview, the proposed hierarchical GCC system for a DEV which consists of three layers is shown in [figure 1](#).

In the upper layer, the desired external forces and moments are determined regarding the vehicular driving demands. In this layer, vehicle state parameters are assumed to be known from sensor-based measurement or observation. On this basis, vehicular driving demands including longitudinal accelerating, steering, and pitch/roll/yaw motions are discussed and desired external forces and moments are determined respectively.

The desired forces and moments are then optimally distributed to each longitudinal/lateral/vertical tire force by formulating and solving a constrained non-convex optimization problem in the middle layer. In this layer, the focus is on three areas: first, a suitable objective function is designed, which is effective to evaluate longitudinal/lateral/vertical force distribution; second, constraints of each tire force are constructed, considering vehicular driving demands, frictional limitations between tires and ground, and also actuator characteristics; third, the optimization problem constituted by the objective function and constraints is solved by combination of constrained optimization algorithm and feasible region planning method.

As each optimal longitudinal/lateral/vertical tire force is obtained, the lower layer deals with the actuator control problem to guarantee that each tire force can be precisely achieved. In this layer, each motor driving torque, each tire steering angle and each active suspension force are respectively controlled to realize each distributed longitudinal/lateral/vertical force.

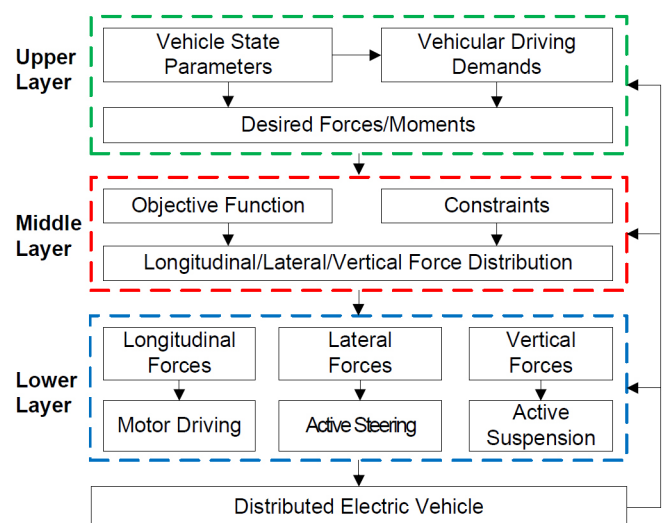


Figure 1. Architecture of the hierarchical global chassis control system for distributed electric vehicles

Upper Layer: Desired Forces/Moments Determination

Above all the other efforts for optimal vehicle dynamics control, the desired vehicular driving demands should be figured out. During vehicle driving, longitudinal acceleration and steering should be essentially satisfied for desired longitudinal/lateral and yaw motion. For good vehicle handling and stability, large vertical acceleration and severe pitch/roll angles of vehicle sprung mass should be prevented as well.

Generally, the desired longitudinal acceleration can be determined from the acceleration pedal and vehicle velocity, as has been widely discussed by other researchers. As the desired longitudinal acceleration is determined, the desired longitudinal force can be determined by [equation \(1\)](#).

$$F_{x,des} = ma_{x,des} \quad (1)$$

where, $F_{x,des}$ is the desired longitudinal force, $a_{x,des}$ is the desired longitudinal acceleration, and m is the vehicle mass.

To follow the ideal lateral trajectory, steering control is introduced and yaw motion is produced as a result. In order to realize desired lateral and yaw motion, the desired lateral force and the desired yaw rate must be determined. Using the information of the desired turning radius, the longitudinal vehicle velocity and the vehicle mass, the desired external lateral force during steering is determined by [equation \(2\)](#).

$$F_{y,des} = mV_x^2 / R \quad (2)$$

where, $F_{y,des}$ is the desired external lateral force, V_x is the vehicular longitudinal velocity, and R is the desired turning radius.

In real conditions, deviation always exists between the actual vehicle position and the ideal trajectory as shown in [figure 2](#). Therefore, the desired vehicle yaw angle has to be dynamically corrected according to the horizontal vehicle attitude. As shown in [figure 2](#), two kinds of corrections can be applied: (i) correction of the angular error between the longitudinal vehicle axis and the ideal trajectory; (ii) correction of the distance error between the vehicle mass center and the ideal trajectory. These angular corrections are given by [equation \(3\)](#) and [equation \(4\)](#) respectively. Besides, as the ideal vehicle yaw angle can be determined by [equation \(5\)](#) and the geometrical relationships between these three angles can be clearly described by [equation \(6\)](#), the corrected vehicle yaw angle is given by [equation \(7\)](#). And then the desired turning radius can be determined by [equation \(8\)](#).

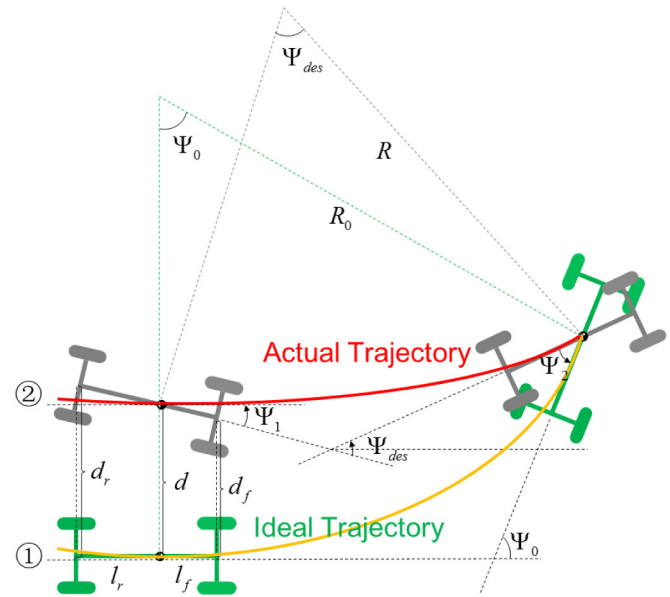


Figure 2. Schematic demonstration for deviation between the actual vehicle position (deep grey vehicle and red trajectory) and the ideal trajectory (green vehicle and yellow trajectory)

$$\Psi_1 = -(d_f - d_r) / l \quad (3)$$

$$\Psi_2 = -d / L \quad (4)$$

$$\Psi_0 = L / R_0 \quad (5)$$

$$\Psi_{des} = \Psi_0 + \Psi_1 + \Psi_2 \quad (6)$$

$$\Psi_{des} = \frac{L}{R} \quad (7)$$

$$R = \frac{R_0}{1 - \frac{R_0}{L} \left(\frac{d_f - d_r}{l} + \frac{l_r d_f + l_f d_r}{lL} \right)} \quad (8)$$

where, Ψ_1 is the angular correction due to the angular error, Ψ_2 is the angular correction due to the distance error, Ψ_0 is the ideal vehicle yaw angle, Ψ_{des} is the corrected vehicle yaw angle, d_r (d_f) is the distance from front (rear) axle midpoint to the ideal trajectory, d is the distance between the vehicle mass center to the ideal trajectory, l is the wheelbase, L is the previewed distance, and R_0 is the radius of ideal trajectory.

Furthermore, by using the single-track vehicle model with front/rear wheel steering as shown in [figure 3](#), [equation \(9\)](#) is obtained and then the desired yaw rate is determined by [equation \(10\)](#).

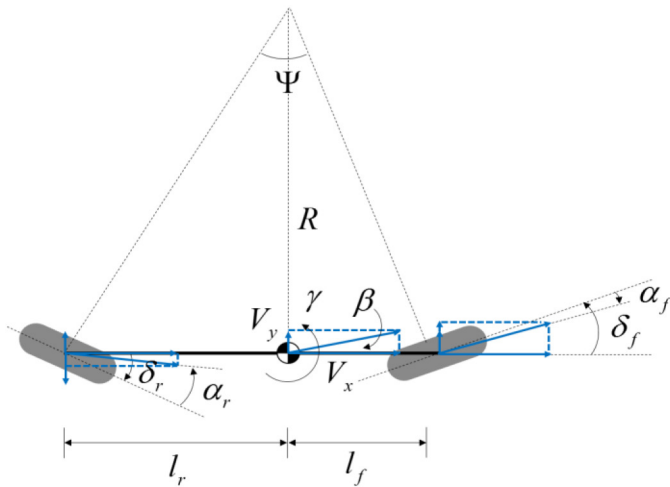


Figure 3. Single-track vehicle model with front/rear wheel steering

$$\Psi = \frac{l}{L} \Psi_{des} = \frac{l}{R} = \delta_f - \delta_r = -\alpha_f + \alpha_r + \frac{\gamma \cdot l}{V_x} \quad (9)$$

$$\gamma_{des} = \left(\frac{l}{R} + \alpha_{f,des} - \alpha_{r,des} \right) \frac{V_x}{l} \quad (10)$$

where, Ψ is the yaw angle respect to the length of wheelbase, δ_f (δ_r) is the desired front (rear) tire steering angle, $\alpha_{f,des}$ ($\alpha_{r,des}$) is the desired front (rear) tire sideslip angle feedback from equation (46), and γ_{des} is the desired yaw rate.

In order to follow the desired yaw rate and control the body roll/pitch angle due to longitudinal/lateral acceleration, desired roll/pitch/yaw moments are determined from errors of roll/pitch angle and yaw rate using PID method, as shown below:

$$M_{z,des} = (K_{p1} + K_{i1} \cdot s^{-1} + K_{d1} \cdot s)(\gamma_{des} - \gamma) \quad (11)$$

$$M_{x,des} = (K_{p2} + K_{i2} \cdot s^{-1} + K_{d2} \cdot s)(\rho_{des} - \rho) \quad (12)$$

$$M_{y,des} = (K_{p3} + K_{i3} \cdot s^{-1} + K_{d3} \cdot s)(\theta_{des} - \theta) \quad (13)$$

where, K_{p1} , K_{p2} and K_{p3} are proportion parameters, K_{i1} , K_{i2} and K_{i3} are integration parameters, K_{d1} , K_{d2} and K_{d3} are differential parameters, s is the Laplace operator, $\rho_{des} = 0$, $\theta_{des} = 0$.

To avoid large vertical acceleration which leads to worse ride comfort and increasing suspension displacement, the sum of each vertical tire force should approximately equal to the vehicle weight, as given below:

$$F_{z,des} = mg \quad (14)$$

where, $F_{z,des}$ is the desired external vertical force, and g is the gravitational constant.

Middle Layer: Longitudinal/Lateral/Vertical Tire Force Distribution

As the desired external longitudinal/lateral/vertical forces and moments determined in the upper layer are implemented by all the tire forces together, it becomes a key problem to distribute these forces and moments into each tire force optimally. In the discussion of the middle layer, optimal distribution of longitudinal/lateral/vertical tire force is emphasized.

Objective Function for Tire Force Distribution

As previously mentioned in [11, 13, 15], different kinds of objective functions for longitudinal/lateral tire force distribution evaluation have been discussed. Among all, the one proposed in [15] is effective to decrease total tire workload and keep each tire workload as equal as possible [14]. The objective function is expressed as:

$$\min J_1 = \text{Var}(\gamma_i) + w \cdot E(\gamma_i) = \left(\frac{1}{4} \sum_{i=1}^4 (\gamma_i - \frac{1}{4} \sum_{i=1}^4 \gamma_i)^2 \right)^{\frac{1}{2}} + w \cdot \frac{1}{4} \sum_{i=1}^4 \gamma_i \quad (15)$$

where, $\min$ denotes minimization, Var denotes the variance, E denotes the average value, w is a weight coefficient, and γ_i is the workload of each tire defined by equation (16).

$$\gamma_i = \frac{\sqrt{F_{xi}^2 + F_{yi}^2}}{\mu_i F_{zi}} \quad (16)$$

where, μ_i is the frictional coefficient between each tire and ground, and the subscript set $i = \{fl, fr, rl, rr\}$ refers to front-left, front-right, rear-left, and rear-right tire.

Regretfully, none of these objective functions is sufficient for longitudinal/lateral/vertical tire force distribution. In fact, vertical force values are given in longitudinal/lateral force distribution problem; while for longitudinal/lateral/vertical force distribution problem, vertical forces become variables which need to be distributed as well. Regarding this, applying equation (15) as the objective function can only optimize the ratio between resultant horizontal force and vertical force of each tire, while the distributed longitudinal/lateral/vertical forces may differ a lot even though each tire workload is equal. As seen from the example given in equation (17), longitudinal/lateral/vertical tire forces of rear-right tire are even 1000 times as those of front-left tire although each tire workload is almost equal.

$$\left\{ \begin{array}{l} \gamma_{fl} = \frac{\sqrt{F_{xfl}^2 + F_{yfl}^2}}{\mu_{fl} F_{zfl}} = \frac{\sqrt{0.5^2 + 0.5^2}}{0.85 \times 1} \\ \gamma_{fr} = \frac{\sqrt{F_{xfr}^2 + F_{yfr}^2}}{\mu_{fr} F_{zfr}} = \frac{\sqrt{5^2 + 5^2}}{0.85 \times 10} \\ \gamma_{rl} = \frac{\sqrt{F_{xrl}^2 + F_{yrl}^2}}{\mu_{rl} F_{zrl}} = \frac{\sqrt{50^2 + 50^2}}{0.85 \times 100} \\ \gamma_{rr} = \frac{\sqrt{F_{xrr}^2 + F_{yrr}^2}}{\mu_{rr} F_{zrr}} = \frac{\sqrt{500^2 + 500^2}}{0.85 \times 1000} \end{array} \right. \quad (\gamma_{fl} = \gamma_{fr} = \gamma_{rl} = \gamma_{rr}) \quad (17)$$

As extreme difference between the vertical forces of each tire may result in too large a dynamic suspension displacement, which is unfavorable for vehicle structural stability, and the difference between the distributed vertical force and the respective static vertical load of each tire must be limited to be as small as possible. To discuss this problem, the dynamic ratio of the vertical forces is defined in [equation \(18\)](#) and a new objective function is designed combined with the tire workload in [equation \(19\)](#).

$$\varepsilon_i = F_{zi,0} / F_{zi} \quad (18)$$

$$\min J = Var(\gamma_i) + w_1 \cdot E(\gamma_i) + w_2 \cdot Var(\varepsilon_i) \quad (19)$$

where, ε_i is the dynamic ratio of each vertical tire force, $F_{zi,0}$ is the static load of each tire calculated from [equation \(20\)](#), and w_1 and w_2 are weighting coefficients.

$$F_{zfl,0} = F_{zfr,0} = \frac{l_r}{2l} mg, \quad F_{zrl,0} = F_{zrr,0} = \frac{l_f}{2l} mg \quad (20)$$

As seen from [equation \(19\)](#), the new objective function keeps each distributed vertical force close to the respective static vertical load and also guarantees optimal tire workloads. In other words, vehicle handling stability, dynamic coupling effect of longitudinal/lateral/vertical tire forces and safety concern of dynamic displacement of suspensions are all involved in this objective function. Therefore, it can be used to evaluate the optimization of longitudinal/lateral/vertical force distribution.

Tire Force Constraints

To determine the feasible region of each tire force, the following constraints must be considered: (1) vehicular driving demands; (2) actuator characteristics; (3) frictional limitations between tires and ground.

First, desired forces and moments determined from vehicular driving demands must be provided by the tire forces together. These constraints are given in [equations \(21\), \(22\), \(23\), \(24\), \(25\), \(26\)](#).

$$F_{x,des} = F_{xfl} + F_{xfr} + F_{xrl} + F_{xrr} \quad (21)$$

$$F_{y,des} = F_{yfl} + F_{yfr} + F_{yrl} + F_{yrr} \quad (22)$$

$$F_{z,des} = F_{zfl} + F_{zfr} + F_{zrl} + F_{zrr} \quad (23)$$

$$M_{x,des} = 0.5t_f(F_{zfl} - F_{zfr}) + 0.5t_r(F_{zrl} - F_{zrr}) \quad (24)$$

$$M_{y,des} = -l_f(F_{zfl} + F_{zfr}) + l_r(F_{zrl} + F_{zrr}) \quad (25)$$

$$M_{z,des} = l_f(F_{yfl} + F_{yfr}) - l_r(F_{yrl} + F_{yrr}) + 0.5t_f(-F_{xfl} + F_{xfr}) + 0.5t_r(-F_{xrl} + F_{xrr}) \quad (26)$$

where, F_{xi} , F_{yi} and F_{zi} are the longitudinal, lateral, and vertical force of each tire, t_f and t_r are front and rear tread respectively, $F_{x,des}$, $F_{y,des}$ and $F_{z,des}$ are the desired external longitudinal, lateral and vertical force, and $M_{x,des}$, $M_{y,des}$ and $M_{z,des}$ are the desired external roll, pitch and yaw moment.

Second, the constraint which describes the frictional limitation between each tire and the driving road must be satisfied:

$$\sqrt{F_{xi}^2 + F_{yi}^2} \leq \mu_i F_{zi} \quad (27)$$

Furthermore, the characteristics of the motor-driving, steering and suspension systems must be considered in order to guarantee that the distributed forces are executable by actuators. Hence, the maximal static and dynamic capabilities of driving/steering and suspension systems should be limited. Considering maximal value and maximal changing rate of each actuator, the constraints are given in [equations \(28\), \(29\), \(30\), \(31\), \(32\), \(33\)](#).

$$-T_{i,max} \leq F_{xi} \cdot r \leq T_{i,max} \quad (28)$$

$$-k_{xi,max} / r \leq \Delta F_{xi} / \Delta t \leq k_{xi,max} / r \quad (29)$$

$$-F_{yi,max} \leq F_{yi} \leq F_{yi,max} \quad (30)$$

$$-k_{yi,max} \leq \Delta F_{yi} / \Delta t \leq k_{yi,max} \quad (31)$$

$$F_{zi} \geq 0 \quad (32)$$

$$-k_{zi,max} \leq \Delta F_{zi} / \Delta t \leq k_{zi,max} \quad (33)$$