Abstract—A lane following system is proposed that employs four constrained wheel torques to regulate a vehicle on a reference trajectory. The proposed control algorithm was developed by combining several techniques such as: DYC method, hierarchical control architecture, sliding mode controls, control distributions, and etc. Different from existing approaches, the proposed method has the following advantages: (1) it can be implemented on two-wheel drive vehicles; (2) the error resulting from the hierarchical architecture are minimized and compensated; (3) the controlled wheel torque calculated by the control distribution method is an analytical solution instead of from numerical search. The proposed controller is evaluated by simulations on two more complex vehicle models: a full-state vehicle model and a sedan model from commercial software Carsim. Simulation results indicate that, in both cases, the proposed method can regulate the vehicle to finish a single lane change when the vehicle is moving at an initial speed of 90 km/hr.

I. INTRODUCTION

In recent years, many research employed the direct yaw moment control (DYC) to regulate vehicle trajectories for lane following [1], [2]. This is done by generating a controlled yaw moment on a vehicle. Many researchers employed a steering control system to generate this yaw moment [3], [4], and this approach has been proven to be effective. Some researcher proposed differential brake system to generate this yaw moment. The advantages are that no additional mechanical or hydraulic components are needed and the steering system can remain intact for other control applications [3], [4]. Most differential brake systems are to determine braking torques on tires either at two sides (left and right) or at one side (left or right). This greatly reduces the complexity of control algorithms as compared to determining optimal wheel torques for four tires. However, from the control viewpoint, the less degree-of-freedom of controls could possibly limit the effectiveness and efficiency of the vehicle control.

Many researchers developed their DYC control systems from a hierarchical architecture to reduce the complexity of the controller design [5], [6], [7]. When using this hierarchical architecture, a virtual tire force and/or moment for the vehicle system were firstly determined by control methods. Then, this virtual entity was distributed to four tire forces using optimization techniques. This is so called “control distribution” methods [5], [6], [7]. Lastly, these controlled tire forces need to be transformed into driver’s command such as: driving/braking wheel torques, wheel steering angles, and etc. for practical use. There are two concerns in this hierarchical approach. First, the optimization in “control distribution” methods were often done by numerical search which requires significant computation burden and thus less preferred for practical implementation in ground vehicles [5], [6], [7]. Second, there exists dynamics between tire force and brake/drive torques or between tire forces and steering wheel angles. These dynamics were often neglected in the transformation and controller design, which may result in stability problems.

Most differential torque control systems were proposed for the electric vehicles, which equips in-wheel motors that can generate driving/braking torque for each tire independently. Thus, the designated tire forces, positive or negative, can be easily implemented by controlled wheel torques on each tire [1], [7]. Even though some control algorithms did impose constraints on tire forces, they are for limiting the maximum and/or minimum values of tire forces [5], [6]. Obviously, those approaches can not be applied to two-wheel drive vehicles wherein either two front tires or two rear tires can only have braking torques.

This paper aims to construct a lane following control system for four-wheel, front-steer, and front-drive vehicles. The proposed system use differential torques to regulate vehicle trajectories wherein four wheel torques can be totally different from each other for better performance. The hierarchical architecture and control distribution methods are employed for deriving the controlled wheel torques. Different from existing approaches, the proposed method has the following advantages: (1) it can be implemented on the two-wheel drive vehicles; (2) the transformation error between designated tire forces and tire forces resulting from controlled wheel torques were compensated; (3) the controlled torque were obtained analytically. The first one is achieved by imposing additional constraints on the control distribution methods. The second one is achieved by considering vertical loads in the force distribution and adapting a robust feedback controller design. The third one is achieved by formulate this problem into a format which can be solved by the Karush-Kuhn-Tucker method [10], [11]. The design procedures and the stability analysis of the lane following system are both discussed in details in this paper.
II. VEHICLE MODEL

A vehicle model with three degree-of-freedom (DOF) and two coordinate systems are introduced for the control algorithm derivations. Two coordinate systems, global frame \( \{g\} \) and body frame \( \{b\} \) (see Fig. 1), are used to describe the vehicle dynamics for the ease of equation derivation. By neglecting the vehicle roll and pitch motions, the relations between the global frame and the body frame can be described by vehicle yaw angles \( \psi \).

A. Vehicle Modeling

A 3-DOF vehicle model is shown in (1). The first two equations describe the vehicle translational motions in \( x \)-axis and \( y \)-axis. The third equation describes the vehicle yaw motions, which is used to indicate the heading direction of the vehicle. The last equation describes the relations between tire forces and applying brake/drive torques. The definitions of each state variable and geometric parameter are listed in Tab. I.

\[
\begin{align*}
\dot{x}_b - \psi \dot{y}_b &= F_{x,1} + F_{x,2} + F_{x,3} + F_{x,4} \\
\dot{y}_b + \psi \dot{x}_b &= F_{y,1} + F_{y,2} + F_{y,3} + F_{y,4} \\
I_e \dot{\psi} &= l_t (F_{y,1} + F_{y,2}) - l_t (F_{y,3} + F_{y,4}) \\
&\quad - t_r (F_{x,1} - F_{x,2}) + t_r (F_{x,3} - F_{x,4}) \\
I_e \dot{\omega}_b &= -F_{a,1} r_t + T_t
\end{align*}
\]

where

\[
\begin{align*}
F_{x,i} &= F_{a,i} \cos \delta_i - F_{b,i} \sin \delta_i \\
F_{y,i} &= F_{a,i} \sin \delta_i + F_{b,i} \cos \delta_i.
\end{align*}
\]

For a front-steer and front-drive vehicle discussed in this paper, two front wheel angles \( \delta_1, \delta_2 \) are determined by Ackerman steering principle [8], while two rear wheel angles \( \delta_3, \delta_4 \) are zeros.

The longitudinal tire force \( F_{a,i} \) is not an independent variable but a function of slip ratio, vertical load, road friction, and etc. The lateral tire force \( F_{b,i} \) is a function of the slip angles \( \alpha_i \), road friction, and etc [9]. Under the assumptions of small slip angles and constant friction coefficients, the magnitude of the lateral tire force can be described by a linear tire model [5].

\[
F_{b,i} = C_i \alpha_i
\]

where

\[
\begin{align*}
\alpha_1 &= \delta_1 - \tan^{-1} \left( \frac{y_b}{x_b} \right) \\
\alpha_2 &= \delta_2 - \tan^{-1} \left( \frac{y_b + l_t \psi}{x_b + l_t \psi} \right) \\
\alpha_3 &= -\tan^{-1} \left( \frac{y_b - l_t \psi}{x_b + l_t \psi} \right) \\
\alpha_4 &= -\tan^{-1} \left( \frac{y_b - l_t \psi}{x_b - l_t \psi} \right).
\end{align*}
\]

The velocities \( \dot{x}_b, \dot{y}_b \) represent the velocity observed in a body frame, while the velocities \( \dot{x}_g, \dot{y}_g \) represent the velocity observed in a global frame. Through approximations, the relations between these four velocities can be greatly simplified to the following:

\[
\begin{align*}
\dot{x}_g &= \dot{x}_b \cos \psi - \dot{y}_b \sin \psi \\
\dot{y}_g &= \dot{x}_b \sin \psi + \dot{y}_b \cos \psi.
\end{align*}
\]

III. LANE FOLLOWING CONTROL SYSTEM

A. Control Concept

Similar to the hierarchical architecture employed in many vehicle control systems, the control input is obtained for the tire drive/brake forces first and then is converted into the applying wheel drive/brake torques. In this paper, the longitudinal tire forces \( F_{a,i} \) are used as the intermediate control input to obtain the controlled wheel torques. This

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**TABLE I**

**STATE VARIABLES AND GEOMETRIC PARAMETERS**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>( x_g, \dot{x}_g, \ddot{x}_g )</td>
<td>the longitudinal position/velocity/acceleration of the vehicle CG observed in the global frame</td>
</tr>
<tr>
<td>( y_g, \dot{y}_g, \ddot{y}_g )</td>
<td>the lateral position/velocity/acceleration of the vehicle CG observed in the global frame</td>
</tr>
<tr>
<td>( x_0, \dot{x}_0, \ddot{x}_0 )</td>
<td>the longitudinal velocity/acceleration of the vehicle CG observed in the body frame</td>
</tr>
<tr>
<td>( y_0, \dot{y}_0, \ddot{y}_0 )</td>
<td>the lateral velocity/acceleration of the vehicle CG observed in the body frame</td>
</tr>
<tr>
<td>( \psi, \dot{\psi}, \ddot{\psi} )</td>
<td>vehicle yaw/angle/angular rate/angular acceleration</td>
</tr>
<tr>
<td>( F_{a,i}, F_{b,i} )</td>
<td>longitudinal/lateral tire force, and the subscript ( i ) refers to the four corners in a way: 1—front-left, 2 to 4 in a clockwise motion</td>
</tr>
<tr>
<td>( F_{x,i}, F_{y,i} )</td>
<td>longitudinal/lateral tire force with respect to the body frame</td>
</tr>
<tr>
<td>( F_{a,i} )</td>
<td>vertical loads of the tire ( i )</td>
</tr>
<tr>
<td>( \alpha_i )</td>
<td>slip angle of the tire ( i )</td>
</tr>
<tr>
<td>( m )</td>
<td>vehicle mass</td>
</tr>
<tr>
<td>( r_i )</td>
<td>effective radius of the tire ( i )</td>
</tr>
<tr>
<td>( l_i )</td>
<td>the moments of inertia of the sprung mass system along the vertical axis</td>
</tr>
<tr>
<td>( I_{ao} )</td>
<td>the moments of inertia of four tires</td>
</tr>
<tr>
<td>( h_i, h_t )</td>
<td>the distances from the CG to the front/rear axis</td>
</tr>
<tr>
<td>( r_t )</td>
<td>one-half of the distances of the front/rear axis</td>
</tr>
</tbody>
</table>

---

Fig. 1. Free body diagram of the vehicle and two coordinate systems: global frame and body frame. The dashed-dotted-blue line represents a reference trajectory.
control strategy is briefly described in the following. First, a Lyapunov function is chosen for deriving the control input and ensuring the system stability. Second, through the implementation of the sliding mode controls, a reference yaw rate and constraints on tire forces are deduced to minimize the lateral displacement error. Third, the control distribution method is utilized to determine the longitudinal force for each tire to satisfy the previous force constraints and additional constraint from the requirement of front-drive vehicles. Lastly, by assuming small slip ratios of tire motions, these tire forces are converted into controlled wheel torques.

B. Lyapunov Function

If a reference trajectory (the dashed-dotted-blue line in Fig. 1) is known beforehand, this trajectory can be approximated by a close-form equation using curve-fitting. By entering the current longitudinal position $x_g$ into the equation, the reference lateral position $y_{ref}$ can be determined. And, the reference trajectory following can be achieved by the reference later position control. Since the DYC system is applied to eliminate the lateral displacement error, a Lyapunov function $V$ is chosen to consist of lateral displacement error and rotational motions.

$$V = \frac{1}{2} \dot{y}_g^2 + \frac{1}{2} \dot{\lambda}(y_g - y_{ref})^2$$

where $\lambda$ is a design parameter and must be positive; $\dot{\psi}_{ref}$ represents the reference yaw rate and its value will be determined later on. To analyze the system stability with this Lyapunov function, the time derivative of the Lyapunov function is re-grouped in three terms:

$$\dot{V} = \dot{V}_{p1} + \dot{V}_{p2} + \dot{V}_{p3}$$

$$\dot{V}_{p1} = -\tau_1 e^2 + \dot{e}$$

$$\dot{V}_{p2} = (\ddot{x}_g \dot{y}_g \sin \psi + \dot{x}_g \dot{y}_g \cos \psi) e + \dot{\lambda} (y_g - y_{ref}) e$$

$$\dot{V}_{p3} = e \dot{\psi} - e \dot{\psi}_{ref} + \tau_2 e^2$$

where $\tau_1, \tau_2$ are design parameters and must be positive. By selecting proper values for $\tau_1, \tau_2$, $\dot{V}_{p1}$ can be guaranteed to be negative semi-definite. Then, the system stability is determined by the values of $\dot{V}_{p2}$ and $\dot{V}_{p3}$.

C. Sliding Mode Control Method

By applying the sliding mode control method, the reference yaw rate can be chosen as follows:

$$\dot{\psi}_{ref} = -x_g^{-1} \left[ -\dot{y}_{ref} - \dot{\lambda} (y_g - y_{ref}) + \tau_1 s + \kappa s / \Phi \right]$$

where $\kappa$ is a design parameter which value is small and positive; $\Phi$ represents the implicit boundary layer and must be positive. By choosing the reference yaw rate shown in (6), $\dot{V}_{p2}$ is negative semi-definite outside the implicit boundary layer $\Phi$.

As discussed before, the proposed control algorithms need to be robust to compensate unmodeled dynamics and discrepancy resulting from transforming designated tire force into applying wheel torques. For this reason, prior to the derivation of control inputs, the dynamics of the vehicle yaw motions is rewritten to accommodate system uncertainties and external disturbances.

$$\ddot{\psi} = A_0 + \Delta A + (B_0 + \Delta B)(F_{a0} + \Delta F_a)$$

where $A_0, B_0$ represent the vehicle dynamics obtained from the vehicle model shown in (1); $F_{a0}$ are the designated control inputs; $\Delta A$ represents the uncertain values of $A_0$, which mainly comes from lateral tire forces; $\Delta B$ represents the uncertain values of $B$, which mainly comes from the neglected vehicle attitude (pitch and roll); the uncertain values of the longitudinal tire forces $\Delta F_a$ result from two factors: (1) neglected slip ratios and varying road friction in calculating tire forces; (2) neglected wheel dynamics in calculating the wheel torques. The second one will be discussed later.

Substituting (7) into (5), $\dot{V}_{p3}$ can be rewritten as follows:

$$\dot{V}_{p3} = e [A_0 + \Delta A + (B_0 + \Delta B)(F_{a0} + \Delta F_a) - \ddot{\psi}_{ref} + \tau_2 e]$$

$$\leq \bar{A} + e B_0 F_{a0} + |e| \delta_{i1}|F_{a1}| + |e| \delta_{i2}|F_{a2}| + |e| \delta_{i3}|F_{a3}| + |e| \delta_{i4}|F_{a4}|$$

where

$$\bar{A} = e A_0 + |e| \delta_A + |e| \|B_0\| \|\dot{\delta}_t\| + |e| \delta_B \delta_r - e \psi_{ref} + \tau_2 e^2$$

$$\delta_r = \sup_{t \in [0, \infty]} |\Delta F_a(t)|$$

$$\delta_A = \sup_{t \in [0, \infty]} |\Delta A(t)|$$

$$\delta_{B,i} = \sup_{t \in [0, \infty]} |\Delta B_i(t)|$$

$$\delta_B = \sqrt{\delta_{B,1}^2 + \delta_{B,2}^2 + \delta_{B,3}^2 + \delta_{B,4}^2}$$

D. Control Distribution

Obviously, many sets of longitudinal tire forces $F_{a0}$ can satisfy the constrained equation (8). Besides, the longitudinal force on two rear tires must be negative for a front-drive vehicle. Hence, the determination of the longitudinal tire forces are formulated into a constrained nonlinear optimization problem for an optimal solution.

1) Nonlinear Constrained Optimization: According to the optimization theory [10], the above equations can be
formulated into the following:

\[
\begin{align*}
\min & \quad \frac{1}{2} F_{a_0}^{T} F_{a_0} \\
\text{s. t.} & \quad \bar{A} + eB_0 F_{a_0} + |e| \delta_1 |F_{a_1}| + |e| \delta_2 |F_{a_2}| \\
& \quad + |e| \delta_3 |F_{a_3}| + |e| \delta_4 |F_{a_4}| = 0 \\
& \quad F_{a_3}, F_{a_4} \leq 0
\end{align*}
\]  

(9)

The cost function in (9) is meant to minimize the power consumption. The first constraint in (9) ensures the negative definite of the $V_p$3, while the second constraint ensures no driving force on two rear tires. It is noted that the first constraint is designed as an equality constraint instead of an inequality constraint. This is to shrink the size of solution pools to prevent the solution selected by the optimization process from jumping around, which is impractical and could possibly damage the vehicle.

2) Analytical Solutions: Because of the absolute values presented in (9), the above problem is a nonlinear optimization problem whereas its solution is mostly obtained by numerical search. However, the numerical search is less preferred in vehicle real time applications. Luckily, this nonlinear constrained equation can be skillfully modified so that its analytical solution can be obtained by the Karush-Kuhn-Tucker theorem. This is done by the following steps. Two rear tire forces $F_{a_3}, F_{a_4}$ must be non-positive. Therefore, their absolute values can be represented as $-F_{a_3}, -F_{a_4}$, respectively. Two front tire forces $F_{a_1}, F_{a_2}$ are free in sign, they can be converted by introducing new variables and new constraint equations [11].

\[
F_{a,n} = F_{a,n}^+ - F_{a,n}^-
\]

<table>
<thead>
<tr>
<th>$F_{a,n}$</th>
<th>$F_{a,n}^+$</th>
<th>$F_{a,n}^-$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$n = 1, 2$</td>
<td>$F_{a,n}$</td>
<td>0</td>
</tr>
</tbody>
</table>

where

\[
F_{a,n}^+ = \begin{cases} F_{a,n} & \text{if } F_{a,n} > 0 \\ 0 & \text{if } F_{a,n} \leq 0 \end{cases}
\]

\[
F_{a,n}^- = \begin{cases} 0 & \text{if } F_{a,n} > 0 \\ -F_{a,n} & \text{if } F_{a,n} \leq 0 \end{cases}
\]

Substituting (10) into (9), a new format for nonlinear constrained optimization problem can be written as below:

\[
\begin{align*}
\min & \quad \frac{1}{2} \left[(F_{a,1}^+ - F_{a,1}^-)^2 + (F_{a,2}^+ - F_{a,2}^-)^2 + F_{a,3}^2 + F_{a,4}^2 \right] \\
\text{s. t.} & \quad \bar{A} + BF_a = 0 \\
& \quad -F_{a,1}^+, -F_{a,1}^-, -F_{a,2}^+, -F_{a,2}^-, -F_{a,3}, -F_{a,4} \leq 0
\end{align*}
\]

(11)

where

\[
\begin{align*}
B = & \quad [eB_1 + |e| \delta_1 B_{B_1}, -eB_1 + |e| \delta_2 B_{B_2}, eB_2 + |e| \delta_3 B_{B_2}, -eB_2 + |e| \delta_4 B_{B_2}] \\
F_a = & \quad [F_{a,1}^+, F_{a,1}^-, F_{a,2}^+, F_{a,2}^-, F_{a,3}, F_{a,4}]^T.
\end{align*}
\]

3) Tire Force Distribution Subjected to Vertical Loads: A longitudinal tire force that could vary with its vertical loads is expected to enhance the effectiveness of the control system. For simplicity, we assume the information of the vertical loads is available for the controller design. Thus, the longitudinal tire force, with the consideration of vertical loads, can be done by adding four weighting factors $q_1, q_2, q_3, q_4$ into the cost function.

\[
\begin{align*}
\min & \quad \frac{1}{2} \left[q_1^2 (F_{a,1}^+ - F_{a,1}^-)^2 + q_2^2 (F_{a,2}^+ - F_{a,2}^-)^2 + q_3^2 F_{a,3}^2 + q_4^2 F_{a,4}^2 \right] \\
\text{s. t.} & \quad \bar{A} + BF_a = 0 \\
& \quad -F_{a,1}^+, -F_{a,1}^-, -F_{a,2}^+, -F_{a,2}^-, -F_{a,3}, -F_{a,4} \leq 0
\end{align*}
\]

(12)

where

\[
q_i = F_{a,i}^{-1}.
\]

E. Wheel Torque Calculation

According to the wheel dynamic shown in (1), it is a non-causal system to determine applying torques from designated longitudinal tire forces. However, by assuming small slip ratios for tire forces, the above equation can be approximated by the equation shown in (13). Thus, it becomes possible to do so.

\[
T_i = F_{a,i} r_i
\]

(13)

The longitudinal force is chosen intentionally to be proportional to the vertical loads. This justifies the assumption of small slip ratio to some extent. However, the existence of the slip ratio would still result in deviations in the specified tire force. This discrepancy is compensated by $\Delta F_a$ as shown in (7).

From the above derivation, all three components of $V$ in (5) are guaranteed to be negative semi-definite outside the implicit boundary layer. Therefore, the stability of this control system is guaranteed.

IV. SIMULATION RESULT

A. Reference Trajectory

A reference trajectory is formulated as a single lane change, which describes two parallel lanes are at a distance of 3 m, and then, a vehicle is moving from one lane to the other lane. In the following simulations, the lane following system is active at the initial time and regulate the vehicle along the reference trajectory. The reference trajectory is modeled as below:

\[
y_{ref}(x) = \frac{3}{1 + \exp^{-0.08(x-145)}}
\]

TABLE II

<table>
<thead>
<tr>
<th>Design Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\lambda$</td>
<td>1</td>
</tr>
<tr>
<td>$\eta$</td>
<td>0.1</td>
</tr>
<tr>
<td>$\tau_1$</td>
<td>7.5</td>
</tr>
<tr>
<td>$\tau_2$</td>
<td>22</td>
</tr>
<tr>
<td>$\delta_\lambda$</td>
<td>0.01</td>
</tr>
<tr>
<td>$\delta_\eta$</td>
<td>80</td>
</tr>
<tr>
<td>$\kappa$</td>
<td>5</td>
</tr>
<tr>
<td>$\Phi$</td>
<td>1</td>
</tr>
<tr>
<td>$\delta_{B1-4}$</td>
<td>1e-6</td>
</tr>
</tbody>
</table>
Two simulation cases are used to demonstrate the effectiveness of the proposed lane following system. The first one uses a full-state vehicle model to mimic the real vehicle dynamics on a road, which is a nonlinear 6-DOF vehicle model and consists of 20 states [12]. The second one uses the CS7 E-class sedan model from the commercial software Carsim 7.1 to mimic the real vehicle dynamics. In these simulation cases, a vehicle moves at the initial speed of 25 m/s (= 90 km/hr) and has no steering (δ1 = δ2 = 0). The design parameters of this controller design are listed in Tab. II. In Figures 2, 3, 6, and 7, the vehicle dynamics from the reference trajectory are shown in dashed-blue line. The vehicle dynamics controlled by the proposed lane following system are shown in solid-red line.

B. Simulation Results with a Full-State Vehicle Model

Figures 2 and 3 show that the proposed control system successfully leads the vehicle to finish a single lane change, while the vehicle yaw rate can closely follow its reference yaw rate as discussed in (6).

Figure 4 shows the controlled wheel torques on each tire. The top two plots are for two front tires, while the bottom two plots are for two rear tires. As shown in the plot, two front tires can have driving and braking torques, while two rear tires can only have braking torques. Furthermore, the controlled torques have a high-frequency chattering. This behavior comes from the equality constraint of the optimization problem shown in (12) and it makes the vehicle yaw rate closely follow the reference yaw rate when the system uncertainties are present. Some improvement work to minimize this chattering is underway. Additionally, as shown in Fig. 5, all absolute values of the system uncertainties are smaller than the value of the design parameters, which justifies the assumptions made in deriving the control algorithms (see (7)).

C. Simulation Results with a Vehicle Model from Carsim

The CS7 E-class sedan model from Carsim 7.1 is used to work with the proposed controller design to demonstrate the feasibility of this controller design. As compared to the previous vehicle model (full-state vehicle model), this model is more close to the real vehicle dynamics but we have less knowledge of its system dynamics.
Figure 6 shows that the proposed control system successfully leads the vehicle to finish a single lane change, which is the same as the previous one. However, the vehicle yaw rate does not follow its reference trajectory as shown in Fig. 7. Figure 8 shows the controlled wheel torques on each tire. The calculated wheel torques are similar to those shown in previous case (see, Fig. 4), except the high-frequency chattering is dismissed from the 3rd to 9th second. This may be because a large uncertainties present when vehicle is turning, and it exceeds the uncertainty bound of design parameters. Therefore, the vehicle yaw rate cannot closely follow its reference trajectory and thus the chattering does not shown. Unfortunately, we don’t have the access to all state values to confirm this speculation.

V. CONCLUSIONS

In this paper, a lane following system for a front-drive, front-steer vehicle is presented and verified by simulation results. The proposed method uses differential drive/brake torque to regulate vehicle trajectories. And, this differential torques are obtained analytically by combining several techniques such as: sliding mode controls, control distributions, and Karush-Kuhn-Tucker theorems. In simulations, both a full-state vehicle model and CS7 sedan model from Carsim are used to mimic the real vehicle dynamics on a road, and the vehicle is moving at the initial speed of 90 km/hr. The proposed method demonstrates its feasibility to regulate the vehicle to finish a single lane change. Currently, the controlled wheel torques have a high-frequency chattering. The improvement work is underway.

VI. ACKNOWLEDGMENTS

The authors gratefully acknowledge Dr. Te-Sheng Hsiao for providing the commercial software Carsim 7.1.

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