Ecological cooperative adaptive cruise control of over-actuated electric vehicles with in-wheel motor in traffic flow

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1 INTRODUCTION

Due to the combined challenge of air pollution and limited fossil fuel sources, many studies have focused on the energy-efficient control strategy for the passenger vehicles recently [1, 2]. A substantial proportion of energy is lost for the vehicle, which includes heat lost, heat dissipation, mechanical loss, air drag, rolling friction and brake losses [3]. Fortunately, full electric vehicles with clean energy sources can avoid heat loss and heat dissipation fundamentally. And, for electric vehicles with in-wheel motor, the mechanical losses can be reduced obviously due to the elimination of the transmission, differential and driving axle. In addition, four-wheel-steering (4WS) and four-wheel-drive (4WD) over-actuated electric vehicle, that brings much flexibility to the vehicle design, have been provided with redundant actuators which can be used to achieve more control targets such as stability control and energy-efficient control. In recent years, the applications of intelligent transportation systems (ITS), especially advanced driver assistance systems (ADAS), have been developed rapidly and vehicles can receive a variety of information from surrounding environment [4, 5]. Thus, the energy efficiency control of electric vehicles is extensively focused.

To extend the driving distance, different types of energy-efficient control methods have been studied, such as wireless power transfer, vehicle-to-grid (V2G) technology, energy-storing ultracapacitors system and regenerative braking [6]. In order to improve the energy efficiency, many researchers have paid extensive attention to minimize the tyre friction loss and improve the motor efficiency by optimizing driving/braking force distributions [7]. Chen and Wang [8, 9] proposed an energy-efficient control allocation (EECA) to optimize the longitudinal driving-braking force distribution between the front and rear motors during the whole trip. In their study, the proposed controller can explicitly treat the actuators’ efficiency for energy optimization in over-actuated systems. Fujimoto et al. [10] proposed a model-based travel range extension control system to optimize the front and rear driving-braking force...
distributions by considering the slip ratio of the wheels and the motor loss. Li et al. [11] presented the research on the real-time traction allocation method for the compound electric propulsion system, which combines an induction motor (IM) and two permanent-magnet synchronous motors (PMSM). Chen et al. [12] proposed a feedback hierarchical controller to track the desired total torque demand and distribute the braking torques to the four wheels to improve energy recovery.

The above studies only dealt with the case of front and rear wheel distribution during longitudinal motion, but the controller design will be more complex when considering horizontal and vertical motion. Chen et al. [13] presented a two-layer control structure, which considered the longitudinal motion, lateral motion, and yaw motion of vehicles, to track the planner motions of over-actuated vehicles while improving the operational energy efficiency. Lenzo et al. [14, 15] discussed several wheel torque control allocation methods, which can reduce the energy consumption while achieving the optimal total wheel torque and yaw moment demands. In their study, effects of load transfers due to driving/braking and cornering are studied and discussed in detail. De Filippis et al. [16] analysed the effects of drivetrain power losses and tire slip power losses for the case of identical drivetrains at the four vehicle corners.

Besides the optimization of torques and steering angles, the topic of energy efficient control method based on traffic information has been widely focused by many researchers. Considering that vehicles can obtain rich information about the traffic, they can understand the surrounding traffic environment excellently, meanwhile the advanced control strategy can be developed to improve the energy efficiency in the traffic [17]. Zhang et al. [17] designed an explicit model predictive control (EMPC) method to reduce the energy consumption of an electric vehicle based on the movement prediction of preceding vehicles. Chen et al. [18] proposed an energy management and driving strategy (EMDS) design for minimizing the energy consumption of a trip based on information of global positioning system (GPS) and geographical information system (GIS). Alam et al. [19] presented a method to describe the motion of a group of heavy-duty vehicles (HDVs) at close inter-vehicular distances, known as a platoon, and to increase the fuel efficiency of the group by reducing the overall air drag. Turri et al. [20] discussed a novel control architecture based on look-ahead control (LAC) for fuel-efficient and safety platooning for heavy-duty vehicle. Yu et al. [21] presented an MPC for a hybrid electric vehicle platoon by considering the fuel economy, the aerodynamic drag varied by vehicle spacing, and the road shape information. Sakhdari et al. [22] presented an ecological adaptive cruise control (EACC) system to improve the platoon’s economy and enable the lead vehicle to follow an unconnected preceding vehicle that is not part of the platoon. In [23], an ACC strategy based on MPC method by including the lane-change assistance function is proposed to solve the vehicle over-taking in the platoon.

Although the ACC for the whole vehicle platoon based on platoon model and energy efficient control allocation (CA) for individual electric vehicle based on detailed vehicle dynamics model have been extensively focused separately in the literature, few of studies have integrated the high-level EACC and low-level energy-efficient CA together to improve the energy efficiency of the platoon by considering the detailed low-level vehicle dynamics model.

The main contribution of our paper is the design of a two-level energy efficient control strategy for electric driving motors. The energy efficiency of electric driving motors for whole vehicle is optimised in high-level cooperative adaptive cruise control (CACC), while the energy efficient optimisation for the individual motor in each wheel is proposed in lower-level CA method. In high-level CACC, the change rate of the desired acceleration is minimised to avoid the sharp fluctuation of acceleration and reduce the overall energy consumption of the electric motor. In lower-level CA, the individual motor power consumption is optimised by considering both the power consumption mode and power regenerating mode. Furthermore, the optimisation of tyre friction dissipation is also considered in the lower-level CA. The whole control strategy mainly includes three layers: the finite state machine (FSM), path-planner and CACC MPC in higher level; the yaw motion and longitudinal motion controller in the middle layer; ecological control allocation method in the lower level. Based on the surrounding environment, the FSM determines the lane-keeping mode or the lane-changing mode. For the lane changing mode, the path-planner determines the desired path to be followed by the yaw motion controller in the middle layer. For both of the lane-changing mode and lane-keeping mode, CACC calculates the desired longitudinal speed and acceleration profile which can be achieved by the longitudinal motion controller in the middle layer. The lower level ecological CA (Eco-CA) method is proposed to achieve the target state values from the middle layer controller by allocating steering angle and individual motor driving torque.

The remainder of this paper is organized as follows: the overall structure of the proposed energy-efficient control system of over-actuated electric vehicle in traffic flow is illustrated in Section 2. Section 3 presents the vehicle dynamics model, platoon model and energy consumption model used in the control design and evaluation. Section 4 shows the controller design for the upper layer ecological CACC system, middle layer controller and lower layer Eco-CA system. The simulation examples of proposed control strategies are presented in Section 5. Concluding remarks are given in Section 6.

## 2 | FRAMEWORK

The overall structure of the proposed energy-efficient control system of over-actuated electric vehicle in traffic flow is illustrated in Figure 1, where the upper layer, middle layer and lower layer controllers are responsible for ecological CACC, vehicle motion control and Eco-CA respectively. The subject vehicle, its leading vehicle and following vehicle are regarded as ‘SV’, ‘LV’, ‘FV’ respectively. The putative leading vehicle and putative following vehicle of SV in target lane during lane change are regarded as ‘PLV’, ‘PFV’ respectively.

According to Figure 1, the FSM firstly determines the vehicle lane-keeping or lane-changing intention based on current vehicle states. Then if the vehicle is working in the lane-changing mode, the path planner will determine the desired path and
the middle level yaw motion controller is enabled for the path-tracking. In the lane keeping mode, when the current lane is curve lane, the middle layer yaw motion MPC controller is enabled to achieve the desired lateral position and yaw angle of LV. If the current lane is straight line, the middle layer yaw motion controller is disabled. In addition, for both the lane-keeping and lane-changing mode, the upper-level CACC MPC [23] is proposed to determine the desired velocity and acceleration profile which can be achieved by the middle layer longitudinal motion controller. In this way, the path-planning, path-tracking controller and CACC are combined together for the future intelligent electric vehicles.

In the lower layer, when receiving the desired steering angle, total driving torque and controlled yaw moment values from middle layer controller, the Eco-CA algorithm allocates individual wheel steering angle and driving/brake torque and minimizes the energy consumption of individual actuator, including energy-efficiency of the driving motor and tyre slip dissipation.

3 MODELLING

The vehicle models used for controller evaluation include a vehicle dynamics model, the platoon model and the energy consumption model, which is presented in this section.

3.1 Vehicle dynamics model

In this paper, the dynamics of the over-actuated electric vehicle can be describe as following equation [24, 25] as shown in Figure 2:

$$\begin{align*}
    m \ddot{v}_x &= m v_y r + \sum_{i = pl, fr, rl, rr} F_{x_i} \\
    m \ddot{v}_y &= -m v_x r + \sum_{i = pl, fr, rl, rr} F_{y_i} \\
    I_z \dot{\psi} &= l_f (F_{y_f} + F_{y_rr}) - l_r (F_{y_pl} + F_{y_rr}) \\
    &+ \frac{b_f}{2} (-F_{x_f} + F_{x_rr}) + \frac{b_r}{2} (-F_{x_pll} + F_{x_rrr}) \\
\end{align*}$$

where $v_x$ is the longitudinal velocity; $v_y$ is the lateral velocity; $r$ is the yaw rate; $F_{x_i}$ is the vehicle longitudinal tyre forces; $F_{y_i}$ is the vehicle lateral tyre forces; $i = pl, fr, rl, rr$ are the front left, front right, rear left, rear right wheel respectively; $l_i$ is the moment of vehicle inertia in terms of yaw axis; $m$ is the vehicle mass; $l_f$ is the front wheel position from the central of gravity (C.G.); $l_r$ is the rear wheel position from the C.G.; $b_f$ is the front track width; $b_r$ is the rear track width.
The tyre traction or brake force and side force are defined as $F_{ti}$ and $F_{si}$, respectively, which can be related to the longitudinal and lateral tyre forces by the steering angle $\delta_j$ as follows:

$$F_{ti} = F_i \cos \delta_j - F_i \sin \delta_j$$
$$F_{si} = F_i \sin \delta_j + F_i \cos \delta_j$$

(2)

According to the feature of over-actuated EVs, traction or brake torque of individual wheel can be independently controlled and the wheel dynamics are obtained as follows:

$$I_\omega \dot{\omega}_i = T_{\text{di}} - T_{\text{bi}} - R_\omega F_{ti}$$
$$T_{\text{bi}} = T_{\text{bmi}} + T_{\text{bhi}}$$

(3)

where $I_\omega$ is the wheel moment of inertia; $\omega_i$ is the angular velocity of each wheel; $T_{\text{di}}$ is the traction torque of each wheel; $T_{\text{bi}}$ is the brake torque of each wheel; $T_{\text{bmi}}$ is the motor brake torque; $T_{\text{bhi}}$ is the mechanical brake torque; $R_\omega$ is the wheel radius.

In this paper, it is assumed that the available motor brake torque can meet the braking needs. The maximum traction or braking torque can be calculated as following equation:

$$T_{\text{ti, max}} = F_{\text{ti, max}} R_\omega = \mu F \omega R_\omega$$
$$|T_{\text{di, min}}| \leq \min(T_{\text{di, max}}, T_{\text{ti, max}})$$
$$|T_{\text{bi, min}}| \leq \min(T_{\text{bi, max}}, T_{\text{ti, max}})$$

(4)

where $\mu$ is the tyre-road friction coefficient; $F_{\text{ti}}$ is the vertical load of each wheel.

$$F_{\text{ti}} = \frac{m}{l_f + l_r} \left( \frac{1}{2}g l_f - \frac{1}{2}v_x b + \frac{l_f}{b_y} (v_y b) \right)$$
$$F_{\text{si}} = \frac{m}{l_f + l_r} \left( \frac{1}{2}g l_f - \frac{1}{2}v_x b - \frac{l_f}{b_y} (v_y b) \right)$$
$$F_{\text{vi}} = \frac{m}{l_f + l_r} \left( \frac{1}{2}g l_f + \frac{1}{2}v_x b + \frac{l_f}{b_y} (v_y b) \right)$$
$$F_{\text{vi}} = \frac{m}{l_f + l_r} \left( \frac{1}{2}g l_f + \frac{1}{2}v_x b - \frac{l_f}{b_y} (v_y b) \right)$$

(5)

where $b$ is the height of C.G., $g$ is the acceleration of gravity.

### 3.2 Platoon model

Different architectures of the communication network between vehicles can be described by different platoon models. In this paper, only the communications between the SV and LV/PLV is considered. The diagram of vehicle platoon is shown in Figure 3.

![Energy-efficient vehicle platoon](image)

Figure 3 Energy-efficient vehicle platoon

Note that the vehicle platoon in this paper is a refactored platoon instead of a stable platoon. Similar to traditional vehicle platoon, the most important goal of platoon is to achieve the desired distance and ensure safety which will be achieved by constraining an error $e$. The error dynamics are then described by the following equations by applying a constant time headway rule [22]:

$$e_{s, LV/SV} = s_{SV} - s_{LV} - b_{SV} v_{SV}$$
$$\dot{e}_{s, LV/SV} = a_{LV} - a_{SV}$$
$$\ddot{e}_{s, LV/SV} = -a_{LV} + a_{SV}$$

(6)

where $s_{SV}$, $s_{LV}$ are the position, velocity and acceleration of SV, $a_{LV}$ and $a_{SV}$ are the velocity and acceleration of LV, $b_{SV}$ is the constant headway-time. $e_{s, LV/SV}$ is the error of inter-vehicular distance between LV and SV, $e_{s, LV/SV}$ is the relative velocity between LV and SV.

Even though the lower level actuator controller has small response time, the delay of the actuators and communication cannot be ignored. Thus, the longitudinal acceleration dynamics of SV can be described as follows [22]:

$$a_{SV} = \frac{K_{SV}}{\tau_{SV}} + a_{dSV}$$

(7)

where $a_{dSV}$ is the desired input of SV, $K_{SV}$ is the steady-state gain, $s$ is the complex number of Laplace transform, $\tau_{SV}$ is the lower controller's delay. According to Equations (6) and (7), the platoon model can be described as follows:

$$\dot{x}_{SV} = A x_{SV} + B_a a_{dSV} x_{SV} + B_d a_{SLV}$$

$$A = \begin{bmatrix}
0 & 1 & -b_{SV} & 0 \\
0 & 0 & -1 & 0 \\
0 & 0 & -1 & \frac{1}{\tau_{SV}} - 1 \\
0 & 0 & 1 & 0
\end{bmatrix}, B_a = \begin{bmatrix}
0 \\
0 \\
0 & \frac{K_{SV}}{\tau_{SV}} \\
0 & 0
\end{bmatrix}, B_d = \begin{bmatrix}
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0
\end{bmatrix}$$

(8)

$$x_{SV} = [s_{s, LV/SV} e_{s, LV/SV} a_{SV} v_{SV}]^T$$
It is noted that the string stability of the vehicle platoon is not considered in this study and further analysis on string stability is required in the future work.

3.3 Energy consumption model

Considering the purpose of this paper is to improve the energy-efficiency of vehicles in traffic flow, it is necessary to propose the energy consumption model. The total energy consumption model in this paper can be described as follows [26, 27]:

\[ P_{total} = P_{motor} + P_{wind} + P_{tyre} \]  

where \( P_{motor} \), \( P_{wind} \) and \( P_{tyre} \) are the power of driving motor, air drag dissipation and tyre dissipation respectively, and all of them can be reduced by dynamics control. The tyre slip power dissipation can be obtained as [28]:

\[ P_{tyre} = u_{slide.xi}(t - \Delta t) \frac{X_{yi}(t - \Delta t)}{X_{yi}(t - \Delta t)} F_{yi}(t) \]

\[ + u_{slide.yi}(t - \Delta t) \frac{Y_{yi}(t - \Delta t)}{Y_{yi}(t - \Delta t)} F_{yi}(t) \]  

where \( X_{yi}(t - \Delta t) \) and \( Y_{yi}(t - \Delta t) \) are the longitudinal and lateral slip forces at the slip region in the contact patch in previous time step, respectively; \( u_{slide.xi}(t - \Delta t) \) and \( u_{slide.yi}(t - \Delta t) \) are the longitudinal and lateral slip speed of the tyre in previous time step, respectively; \( X_{yi}(t - \Delta t) \) and \( Y_{yi}(t - \Delta t) \) are the longitudinal and lateral forces of the tyre calculated by the tyre model in previous time step, respectively; \( F_{yi} \) and \( F_{yi} \) are allocated tyre force in current time step.

4 CONTROL DESIGN

In this section, a hierarchical control system is designed to improve the energy-efficiency of over-actuated electric vehicle in traffic flow, which includes the upper-level controller, middle-level controller and low-level controller.

4.1 Control strategy in the upper layer controller

Control strategy in the upper layer controller aims to solve the complex problem of energy-efficient and safe control for the vehicles in platoon, which includes three submodules as follows.

4.1.1 Control strategy for driving mode selection

The driving mode selection is accomplished by ruled-based FSM controller, which will select the lane-changing mode when the vehicle ahead is too slow, or keep the lane-keeping mode. To realize the successful lane changes, it is necessary to identify the relative positions between SV and the surrounding vehicles and ensure a safe inter-vehicle distance. It is assumed in this study that the SV and the surrounding vehicles can maintain the steady following action in the traffic flow.

The minimum safety distance plays an important role in driving mode selection in traffic flow. The minimum safety distance in lane changing mode is much larger than the minimum safety distance in lane keeping mode, and hence a synthesis minimum safety distance is proposed according to [23], which can be expressed as:

\[ d_{LV\_min} = \max (\tau_{THWV\_LV} v_{SV}, \tau_{TTC\_LV} (v_{SV} - v_{LV})) \]

\[ d_{FV\_min} = \max (\tau_{THWV\_FV} v_{FV}, \tau_{TTC\_FV} (v_{FV} - v_{SV})) \]

\[ d_{PLV\_min} = \max (\tau_{THWV\_PLV} v_{PLV}, \tau_{TTC\_PLV} (v_{PLV} - v_{SV})) \]

\[ d_{PFV\_min} = \max (\tau_{THWV\_PFV} v_{PFV}, \tau_{TTC\_PFV} (v_{PFV} - v_{PLV})) \]  

where \( d_{LV\_min} \) is the minimum safety distance between SV and LV, \( d_{FV\_min} \) is the minimum safety distance between SV and FV, \( d_{PLV\_min} \) is the minimum safety distance between SV and PLV in target changing lane, \( d_{PFV\_min} \) is the minimum safety distance between SV and PFV in target changing lane. \( \tau_{THWV\_LV} \) and \( \tau_{TTC\_LV} \) are the safe time headway (THW) and safe time to collision (TTC) between SV and LV; \( \tau_{THWV\_FV} \) and \( \tau_{TTC\_FV} \) are the THW and TTC between SV and FV; \( \tau_{THWV\_PLV} \) and \( \tau_{TTC\_PLV} \) are the THW and TTC between SV and PLV in target changing lane; \( \tau_{THWV\_PFV} \) and \( \tau_{TTC\_PFV} \) are the THW and TTC between SV and PFV in target changing lane. The values of THW and TTC have been proposed in [23].

According to the minimum safety distance in Equation (11), the FSM controller determines the driving mode of SV. The flow chart is presented in Figure 4 to present the selection logic, where \( D_{SV} \) presents the inter-vehicular distances between SV and surrounding vehicles, and \( \tau_{safe} \) is the minimum safety inter-vehicular distances between SV and surrounding vehicles on changing lane. When the vehicle ahead (LV) is too slow and...
inter-vehicular distances between SV and surrounding vehicles on changing lane are larger than the safety distances, the FSM generates the initial lane change intention, otherwise the vehicle is in the lane keeping mode. If SV is in the lane keeping mode, the car-following weight of leading vehicle in the current lane LV $\lambda_{LKA} = 1$, and the car-following weights of the leading vehicle in the target changing lane PLV $\lambda_{LCA} = 0$, which means SV will keep the current lane and follow LV in the current lane. If FSM generates the initial lane change intention, $\lambda_{LCA}$ changes from 1 to 0 and PLV $\lambda_{LKA}$ changes from 0 to 1, which means the SV will take the lane change and follow the PLV in the target changing lane.

4.1.2 Control strategy for ecological CACC

After the driving mode is selected, the ecological CACC strategy is proposed to achieve the desired control performance when following the preceding vehicle or when taking the lane change by considering the safety distance between the SV and the vehicles nearby.

The control objectives and the detailed description of controller design of ecological CACC is presented in this section. The CACC performance index is a quantitative description of the optimization cost function of the MPC algorithm, which plays an important role in the ecological CACC algorithm. Four performance indicators including vehicle following, safety, air drug reduction and motor energy efficiency are selected as the optimization cost functions of CACC algorithm.

(1) Vehicle following performance index

During the lane keeping mode, only the following performance index related to following LV is enabled. During the lane changing mode, switching the leading vehicle from LV (before lane change) to PLV (after lane change) smoothly is the main objective of vehicle following performance index. The cost function of ecological CACC includes the weighted cost of the position and velocity differences between SV and LV/PLV. When LPV exists in the adjacent lane, there are two target vehicles (the LV in current lane and PLV in the target changing lane) for following, and the following target gradually transforms from LV to PLV. At the beginning of lane change, SV is located in the centre of the original lane, so the following target is solely LV. When crossing the lane mark, SV tries to follow PLV while avoiding a collision with LV, thus at this time SV is following the PLV and avoiding PV simultaneously. After successfully crossing the lane mark, SV will neglect LV and just follow PLV instead. Since the tracking performance should be coordinated between the two target vehicles, the cost function of CACC MPC is separately designed as the linear weighted results of the tracking performances between SV and the two leading vehicles. When there is no PLV in the adjacent lane, the SV makes the lane change then works in the cruise control mode.

The cost function of ecological CACC related to the vehicle-following is shown as follows:

$$J_{fu} = \sum_{k=1}^{N_f} j_{fu}(k)$$

$$= \sum_{k=1}^{N_f} \lambda_{LKA}(w_i e_{x_i \text{LV}}(k))^2 + w_i e_{x_i \text{LV}}(k)^2)$$

$$+ \lambda_{LCA}(w_i e_{x_i \text{PLV}}(k))^2 + w_i e_{x_i \text{PLV}}(k)^2)$$

where $N_f$ is the prediction horizon of MPG, $J_{fu}$ is the vehicle following performance index, $\lambda_{LKA}$ and $\lambda_{LCA}$ are the vehicle-following weights of lane keeping and lane change, $w_i$ and $w_v$ are the weights of distance and relative velocity, $e_{x_i \text{LV}}$ and $e_{x_i \text{PLV}}$ are the distance between SV and LV/PLV, $e_{x_i \text{LV}}$ and $e_{x_i \text{PLV}}$ are the relative velocity between SV and LV/PLV.

(2) Safety performance index

Inter-vehicular distance is a critical value to evaluate the safety performance. A small inter-vehicular distance will lead to the risk of rear collision and bring unsafe feeling to the passengers, while a large inter-vehicular distance will lead to frequent insertion of surrounding vehicles which can cause the reduction of vehicle velocity. The main objective of safety control of CACC is to keep a minimum safety distance between SV and surrounding vehicles, which can be achieved by the following constraints:

$$e_{x_{min}} \leq e_{x_{LV}} \leq e_{x_{max}}$$

$$e_{x_{min}} \leq e_{x_{PLV}} \leq e_{x_{max}}$$

where $e_{x_{LV}}$ is the distance between SV and surrounding vehicles, and $e_{x_{PLV}}$ is the velocity difference between SV and surrounding vehicles. $e_{x_{min}}$ and $e_{x_{max}}$ are lower and upper limits of relative distance, $e_{x_{min}}$ and $e_{x_{max}}$ are lower and upper limits of relative velocity.

(3) Air drug reduction performance index

In this section, not only the energy-efficiency of a single vehicle but also the energy-efficiency of vehicles in platoon are focused.

In upper layer ecological CACC, the benefits of platoon can be achieved by close inter-vehicular distances, which reduces the overall aerodynamic drag and thus improves energy-efficiency. An illustration of this benefit is given in Figure 5, which shows that the smaller the inter-vehicular distance is, the more reduction of aerodynamic drag can be achieved. In addition, with the increase of the number of vehicles in platoon, the energy consumption of air drag of all the vehicles can be further reduced. It is noted that when it comes to the coordination between safety and energy saving issues, the scaling factors of each optimisation term should be adjusted to guarantee the minimum safety
distance of the vehicle as the priority and then achieve the best energy saving performance.

The power consumption reduction of air drags can be presented as follows:

\[ P_{\text{wind\_reduce}} = -\frac{1}{2} \rho A_f \Delta C_D v^3 \]  
(14)

where \( A_f \) is the frontal area of the vehicle and \( \rho \) is the air density, \( \Delta C_D \) is the air drag reduction coefficient [30].

\[ \Delta C_D = \left(1 - C_D, \text{plateau/\infty} \right) \left( \epsilon_{r, LV/ SV}(k) \right) C_D \]  
(15)

where \( C_D, \text{plateau/\infty} \) is the drag coefficient ratio, \( C_D \) is the drag coefficient.

The cost function related to ecology performance index can be described as follows:

\[ J_{\text{air}} = \frac{N_p}{\sum_{k=1}^{N_p} P_{\text{wind\_reduce}}(k)} \]  
(16)

After substituting Equation (15) into Equation (16), it can be found that the cost function of ecology performance index can be optimized only by the inter-vehicular distance between vehicles. In this paper, the inter-vehicular distance between the preceding vehicle and following vehicle should achieve \( \Delta s^{*}_{LV/ SV} \):

\[ \Delta s^{*}_{LV/ SV} = d_0 - b_{LV/ SV} \]  
(17)

Thus, the cost function related to air drug performance index should be rewritten as:

\[ J_{\text{air}} = \sum_{k=1}^{N_p} \left[ \epsilon_{r, LV/ SV}(k) - \Delta s^{*}_{LV/ SV}(k) \right]^2 \]  
(18a)

Subject to:

\[ \epsilon_{r, LV/ SV}(k) \geq d_{sa \_fr} \]  
(18b)

where \( d_{sa \_fr} \) is the safety distance between the platoon vehicles.

(4) Energy efficient performance index

The fluctuation of MPC is inevitable when switching between lane changing mode and lane keeping mode. The influence of the fluctuation has already been taken into consideration during the controller design by adding a penalized cost function of acceleration to reduce the sharp change of desired acceleration. Furthermore, the sharp fluctuation of acceleration will also cause the increase of the energy consumption of the electric motor and tyre energy dissipation. Secondly, the acceleration of the vehicle should be constrained within the maximum acceleration that the road surface adhesion coefficient can provide. Therefore, the comfort cost function is designed as the square of the acceleration:

\[ J_{\text{eng}} = \frac{N_p}{\sum_{k=1}^{N_p} \Delta a_{\text{des, SV}}(k)} \]  
(19)

where \( J_{\text{eng}} \) is the energy-efficient performance index. \( \Delta a_{\text{des, SV}} \) is the increment of acceleration control input. The constraint of the acceleration can be expressed as:

\[ \max (a_{\min}, -\mu g) \leq a_{\text{des, SV}}(k) \leq \min (a_{\max}, \mu g) \]  
(20)

where \( a_{\min} \) and \( a_{\max} \) are the upper and lower acceleration limits, and \( \mu g \) is the constraint of acceleration due to the friction circle, which is used to prevent the wheel from locking or slipping.

(5) Optimization problem

The performance indexes of vehicle following, safety, ecology and comfort can be summed as a comprehensive cost function. Based on Equations (12), (13), (18) and (19), the comprehensive cost function \( J_{\text{total}} \) for optimization problem can be formulated.
In this section, the middle layer controller is proposed to coordinate the longitudinal and lateral vehicle position in the upper layer. This desired path and longitudinal acceleration will be sent and achieved by the middle and lower level controllers. In this paper, the MPC is applied to solve optimization problem, Equation (21). The sampling time of MPC is assumed as 0.001 s and the length of predictive horizon is 10.

According to the system states in the current sample time, the cost function and constraint in Equation (21) can be reformulated as follows:

\[
\min_{X_{SV}, Y_{SV}, a_{des, SV}} J_{total} = J_{fuel} + w_{es}J_{es} + w_{air}J_{air} \\
= \sum_{k=1}^{N_p} \lambda_{LCA} \left( w_e (e_{x,SV}^*(k))^2 + w_v e_{y,SV}^*(k)^2 \right) \\
+ \lambda_{LCA} \left( w_e (e_{x,SV}^*(k))^2 + w_v e_{y,SV}^*(k)^2 \right) \\
+ w_{es} \sum_{k=0}^{N_{es}-1} \Delta a_{des,SV}^*(k)^2 \\
+ w_{air} \sum_{k=0}^{N_{air}-1} \left[ \epsilon_{x,SV}^*(k) - \Delta a_{des,SV}^*(k) \right]^2
\]  

(21a)

subject to:

\[
e_{x,SV}^{\text{min}} \leq e_{x,SV} \leq e_{x,SV}^{\text{max}} \\
e_{v,SV}^{\text{min}} \leq e_{v,SV} \leq e_{v,SV}^{\text{max}} \\
\max (a_{\text{min},SV}, \mu g) \leq a_{SV} \leq \min (a_{\text{max},SV}, \mu g) \\
x_{\text{min}} \leq x_{SV} \leq x_{\text{max}} \\
y_{\text{min}} \leq y_{SV} \leq y_{\text{max}}
\]

(21b–d)

where \(w_{es}\) and \(w_{air}\) are weighting factors of energy efficient cost and air drug cost, respectively. \(x_{SV} \) and \(y_{SV}\) are optimized longitudinal and lateral vehicle position in the upper layer. \(a_{SV}\) is the increment of lateral acceleration, which can be obtained by:

\[
\Delta a_{\text{des},SV} = \Delta a_{\text{des},SV}^*(k) + \Delta a_{\text{des},SV}^*(k)
\]

(23a)

where \(H\) and \(f\) are obtained by:

\[
H = 2 (\Theta^T \tilde{Q}_{CACC} \Theta + \tilde{R}_{CACC}) \\
f = 2\Theta^T \tilde{Q}_{CACC} E (k)
\]

(24)

\[
\Theta = C\bar{B}_v. 
\]

The error between actual output and desired output \(E (k)\) can be defined as:

\[
E (k) = \Gamma x_{SV} (k-1) - \Psi a_{\text{des}} (k-1) - \Lambda a_{\text{des},0} - Y_{\text{ref}}
\]

(25)

where \(\Gamma = CA, \Psi = CB_v, \Lambda = C\bar{B}_v.\) The constraints Equations (23b) and (23c) are reformulated based on Equations (21b–d).

### 4.1.3 Path planner

When the SV is working in the lane change mode, the path planner is required to determine the desired path which can be tracked by the middle level path-tracking controller. The desired longitudinal position and lateral position can be described by following equation when given the initial and final positions:

\[
Y_{\text{des}} = c_0 + c_1 X_{\text{des}} + c_2 X_{\text{des}}^2 + c_3 X_{\text{des}}^3 + c_4 X_{\text{des}}^4 + c_5 X_{\text{des}}^5
\]

(26)

\(c_0, c_1, \ldots, c_5 \in R\) are the parameters to determine the planned path. \(X_{\text{des}}\) and \(Y_{\text{des}}\) are desired longitudinal and lateral position, respectively. The desired yaw angle can be determined by following equation:

\[
\psi_{\text{des}} = \frac{Y_{\text{des}}}{X_{\text{des}}}
\]

(27)

It is also noted that when the SV is working in the lane keeping mode but the current lane is curve lane, the upper layer controller takes the measured lateral position and yaw angle from LV and sent to the middle layer yaw motion controller to follow the desired lateral position and yaw angle of LV.

### 4.2 Control strategy in the middle layer controller

In this section, the middle layer controller is proposed to coordinate the longitudinal and lateral motion according to the control
targets from upper layer. This middle layer controller is divided into two sub-modules: longitudinal motion controller and yaw stability controller.

4.2.1 Longitudinal motion controller

Longitudinal motion controller in the middle layer receives the desired longitudinal acceleration from the upper layer. The proportional-integral (PI) feedback control is applied to achieve the desired longitudinal acceleration due to the complexity of powertrain, which can be described as follows:

\[ T_{total} = K_P e_a (t) + K_I \int_0^t e_a (t) \quad (28) \]

where \( K_P, K_I \) are the proportional and integral feedback control gains, \( T_{total} \) is the required total driving torque of wheels and \( e_a \) is the acceleration error between the desired longitudinal acceleration and actual acceleration. The distribution of the total torque into individual wheel will be shown in the lower level CA in Section 4.3.

4.2.2 Yaw motion controller

In order to track the target lateral position \( Y_{ref} \) and yaw angle \( \psi_{ref} \), the vehicle states in vehicle body coordinate system need to be changed into vehicle states in the global coordinate system, the mapping relation can be described as follows:

\[
\begin{align*}
X &= v_x \cos \varphi - v_y \sin \varphi \\
Y &= v_x \sin \varphi + v_y \cos \varphi
\end{align*}
\quad (29)
\]

where \( X, Y \) and \( \varphi \) are the longitudinal position, the lateral position and the heading angle of vehicle in global coordinate system. Combining Equations (1) and (29), the dynamics equation of the over-actuated electric vehicle for path tracking can be derived as:

\[
\begin{align*}
\dot{x} &= f_1 (v_x, v_y, r, \delta_f, \delta_r) \\
\dot{y} &= f_2 (v_x, v_y, r, \delta_f, \delta_r) \\
\dot{r} &= f_3 (v_x, v_y, r, \delta_f, \delta_r, \Delta M_z) \\
\dot{\psi} &= r \\
\dot{X} &= v_x \cos \varphi - v_y \sin \varphi \\
\dot{Y} &= v_x \sin \varphi + v_y \cos \varphi
\end{align*}
\quad (30)
\]

where \( \Delta M_z \) is the additional control yaw moment. The control input and output are:

\[
\begin{align*}
U &= [\delta_f \quad \delta_r \quad \Delta M_z]^T \\
Y &= [\varphi \quad Y]^T
\end{align*}
\quad (31)
\]

To design the MPC controller for path-tracking, the cost function of MPC can be defined based on the discretization form of Equations (31)–(33):

\[
\begin{align*}
\min_{\delta_f, \delta_r, \Delta M_z} J_{lat} &= \sum_{t=1}^{N_f} Q_{lat} \| Y_r (\tau) - Y (\tau) \|^2 \\
&+ \sum_{t=0}^{N_c-1} R_{lat} \| U (\tau) \|^2
\end{align*}
\quad (32)
\]

subject to:

\[
\begin{align*}
Y_{min} &\leq Y (k) \leq Y_{max} \\
U_{min} &\leq U (k) \leq U_{max} \\
\Delta U_{min} &\leq \Delta U (k) \leq \Delta U_{max}
\end{align*}
\quad (33)
\]

where \( N_f = 20 \) and \( N_c = 5 \) are the predictive and control horizons, \( Q_{lat} \) and \( R_{lat} \) are the diagonal weighting matrices. The sampling time of MPC is 0.001 s. The formula derivation process and solution method are like the controller which proposed in ecological CACC.

4.2.3 Control strategy in the lower layer controller

In order to minimize the energy consumption of driving motor and tyre, the CA method is adopted in this paper to allocate the individual torque and steering angle of each in-wheel motor. CA method is a widely applied method to allocate individual control actuator for over-actuated control system. In this study 4WS-4WD electric vehicle has the advantage of redundant actuators that can be utilized not only to achieve the control goals which comes from the upper layer, but also can realize the important goal of energy efficiency optimization in the lower level actuator.

4.2.4 In-wheel electric motor system

The electric vehicle is equipped with four brushless direct current motors and the operational efficiency in driving mode \((\eta_{drive,i})\) and regenerative braking mode \((\eta_{regen,i})\) can be looked up from the efficiency map. Based on the assumption that the energy can be partially re-gained through the regenerative braking function, the total energy conversion between the batteries and the in-wheel motors can be described by the following equation:

\[
P_{motor} = \sum_{i=1}^{4} P_{drive,i} / \eta_{drive,i} + \sum_{i=1}^{4} P_{regen,i} / \eta_{regen,i}
\quad (34)
\]

where \( P_{drive,i} \) is the output power in the energy consuming mode of each motor, and \( P_{regen,i} \) is the input power in the energy gaining mode of each motor, which can be expressed as follows:

\[
\begin{align*}
P_{drive,i} &= T_{di} \omega_i \\
P_{regen,i} &= T_{di} \omega_i
\end{align*}
\quad (35)\]
where $T_{dl}$ and $T_{dr}$ are the torques of each motor in driving mode and regenerative braking mode and $\omega_i$ is the rotational speed of each motor.

### 4.2.5 Ecological CA optimization

The objective of CA is to meet the traction force and yaw moment demands, improve the energy-efficiency of four in-wheel motors and minimize the tyre slip power. The total longitudinal force and the direct yaw moment generated by the driving forces can be described as follows:

\[
F_{\text{total}} = F_{f\beta} + F_{fr} + F_{r\beta} + F_{rr}
\]

\[
\Delta M_{\zeta} = \frac{b_{fr}}{2} (-F_{f\beta} + F_{fr} - F_{r\beta} + F_{rr})
\] (36)

where $F_{\text{total}}$ is the total longitudinal force and $\Delta M_{\zeta}$ is the direct yaw moment. Combining Equation (36) with $F_{l} = T_j/R_\omega$, we can restructure Equation (36) as follows:

\[
K_{opt} U_{opt} = \begin{bmatrix} 1 & 1 & 1 & 1 \\ -b_f & b_f & -b_r & b_r \\ 2R_\omega & 2R_\omega & 2R_\omega & 2R_\omega \\ T_{fr} & T_{fr} & T_{rr} & T_{rr} \end{bmatrix} \begin{bmatrix} T_{fr} \\ T_{fr} \\ T_{rr} \\ T_{rr} \end{bmatrix}
\] (37)

Then, the cost function of CA can be derived as:

\[
\begin{align*}
\min_{U_{opt}} J_{CA} &= \beta \left( K_{opt} U_{opt} - U_{des} \right)^T \left( K_{opt} U_{opt} - U_{des} \right) \\
&+ \alpha P_{motor} + \gamma \sum_{i=1}^{4} \left( \alpha_i^2 F_{fr}^2 + \beta_i^2 F_{r\beta}^2 \right)
\end{align*}
\] (38)

where $U_{des} = [T_{\text{total}} \Delta M_{\zeta}]^T$ is the control target, $\alpha$, $\beta$ and $\gamma$ are the scaling factors for the three optimization terms related to achieving the desired control target in middle layer, the minimize of motor power consumption and tyre slip power dissipation. $a_i$ and $b_i$ can be obtained according to Equation (10):

\[
\begin{align*}
a_i &= u_{\text{slide,si}}^2 (t - \Delta t) \frac{X_{si}^2}{X_{si}^2} (t - \Delta t) \\
b_i &= u_{\text{slide,si}}^2 (t - \Delta t) \frac{Y_{si}^2}{Y_{si}^2} (t - \Delta t)
\end{align*}
\] (39)

It is noted that scaling factors $\alpha$, $\beta$ and $\gamma$ are carefully tuned by normalizing individual optimisation term in cost function Equation (38). The normalized cost function can be beneficial for the tuning of scaling factors by making the individual terms in cost function quantitatively comparable.

It is also noted that the MPC algorithm is applied on upper layer CACC, and the model applied this MPC is linear platoon model based on Equations (6)–(8). The detailed vehicle dynamics model is not considered and computational effort can be reduced. In the middle layer, the PI controller is proposed to achieve the desired longitudinal acceleration and MPC algorithm is applied to achieve the yaw stability control. The PI controller for longitudinal acceleration can be applied in real time, but the yaw stability MPC controller is based on nonlinear vehicle model and requires more computational efforts. Although currently the ‘Fmincon’ solver in MATLAB is used to solve the MPC in our study, the fast MPC solver (such as primal dual interior point (PDIP) solver developed by Embotech) can be applied in high-level CACC and yaw stability MPC in middle layer to achieve the real-time application. In the lower level, the optimization algorithm for CA is applied to allocate individual motor torque in an energy-efficient manner. Unlike MPC with optimization of several forward steps, this control allocation problem in low-level only has one-step optimization and the real-time application can be achieved.

### 5 SIMULATION RESULTS

In this section, the simulation evaluation of the proposed controller is presented to reduce the energy consumption of air drag dissipation and tyre dissipation, and improve the energy efficiency of driving motor. A platoon of vehicles, which can switch among car following mode and lane change mode, is simulated in MATLAB/Simulink environment by integrating dynamics model, platoon model and energy consumption model of over-actuated electric vehicle. Each vehicle communicates with its surrounding vehicles through wireless vehicle-to-vehicle (V2V) communications in traffic flow. The parameter values in the simulation are listed in Table 1.

To verify the performance of the proposed approach, several simulation tests for different scenarios have been carried out in this section. Moreover, the proposed Eo-CA in lower layer controller is evaluated and compared to a traditional allocation method, which does not consider energy efficiency optimization. For the traditional allocation, the control target is only the desired control targets in middle layer and the scaling factors of each term in Equation (40) can be tuned as $\alpha = 0$, $\gamma = 0$ to disable the energy efficient control. The tyre-road friction coefficient is assumed as 0.9.

### 5.1 Scenario A: Pure longitudinal motion

The controller performances of ecological CACC and the lower level controller are demonstrated in scenario A. LV, SV and PV constitute a platoon in this scenario. The US06 is a typical driving cycle to evaluate the electric vehicle drive range, which is conducted in this section to verify the proposed ecological control method. To realize the goal of simulation, we can track an LV which run as the US06 driving cycle. Based on the literature review of CACC, the frequency analysis shows result for different values of headway time [31]. The constant headway time is chosen as 0.6 s.
The first step of scenario A is to validate the following performance of the upper level CACC. Figures 6 and 7 show the velocity profile and acceleration profile of a platoon respectively, which is a US06 driving cycle. It can be seen that the vehicles follow each other with a reasonable velocity and acceleration. In Figures 6 and 7, the following vehicles including SV and FV have a much smoother velocity profile and acceleration profile compared to LV. This smoother trajectory will lead to less unnecessary accelerations and a better energy efficiency for the vehicles in platoon. The acceleration profile in Figure 7 also presents the acceleration limit which was chosen as $a \in [-3.5, 4]$. In Figure 7, the acceleration profiles of SV and FV ($SV_0$ and $FV_0$ which represent the SV and FV when the motor energy efficient term is not considered in the cost function of CACC MPC) have more abrupt acceleration changes compared with SV and FV (when the energy efficient optimisation is considered), and this causes more motor energy consumptions as shown in Figure 12. The constraint on the distance error was chosen as $e_s \in [-0.25, 0.125]$ to ensure the safety of platoon. Figure 8 presents the inter-vehicular distance errors between the three vehicles which are both well behaved and inside the defined constraint.

Aerodynamic drag coefficient of LV, SV and FV have been illustrated in Figure 9. FV has the smallest aerodynamic drag coefficient.
coefficient, compared to the other vehicles, when driving in a platoon. Our proposed CACC strategy is mainly focused on the energy efficiency of individual vehicle, and energy saving of the whole platooning is the additional benefits of the proposed CACC strategy. Figure 10 shows that all the vehicles in platoon operate at close inter-vehicular distances with each other, which reduces the overall aerodynamic drag and thus results in better energy-efficiency. It is reasonably foreseeable that the more vehicles joining in the platoon, the more energy-efficient improvement of air drag dissipation will be achieved.

After simulating the upper layer controller, the next step is to verify the effectiveness of proposed middle layer controller and lower layer controller. Figures 12 and 13 illustrate the energy-efficient improvement of the proposed lower layer Eco-CA. For the traditional allocation method in the pure longitudinal motion scenario, the control target is only the desired longitudinal velocity. For the proposed Eco-CA, the control targets are the desired longitudinal velocity and the energy consumption, and the scaling factors of each term in Equation (38) can be tuned as $\alpha = 12, \beta = 2, \gamma = 5$. The desired vehicle longitudinal velocity of lower level CA is determined by the upper level CACC presented in Figure 6. Figure 11 shows the longitudinal velocity errors between the desired value in upper layer and actual value controller by lower layer Eco-CA. As it can be seen, the proposed lower layer Eco-CA tracks the desired longitudinal velocity excellently.

In Figure 12, the comparison of motor power energy consumption between proposed high-level Eco-CACC and low-level Eco-CA (Eco-CACC and Eco-CA), proposed Eco-CACC and traditional CA (Eco-CACC and CA), and traditional CACC and Eco-CA (CACC and Eco-CA) are shown. Compared with the traditional CA method, the proposed low-level Eco-CA has much better motor power energy saving performance. Similarly, compared with traditional CACC, the proposed Eco-CACC has better motor power energy saving performance. It is noted that in 200–450 s for Figure 12, the energy saving performance of proposed Eco-CA is more obvious, at this time the longitudinal acceleration of the vehicle is relatively small as shown in Figure 7. On the other hand, before 200 s or after 450 s when the longitudinal acceleration is relatively big, the energy saving performance of Eco-CA is less obvious. The reason why acceleration greatly affects the energy saving performance of Eco-CA can be explained as follows: in high acceleration condition, motor torques are usually distributed equally to achieve the high acceleration demand, so the energy saving effect of Eco-CA is not obvious. On the other hand, in the low acceleration condition, the distributed individual motor torques can be different and the energy saving performance of Eco-CA is more obvious. Figure 13 also suggests that Eco-CA has smaller tyre energy dissipation rate compared with tradition CA method.

Figure 14(a) suggests that the individual torque applied on front right wheel for Eco-CA is larger than traditional CA method. However it can be seen form Figure 14(b) that the individual torque applied on rear left wheel for Eco-CA is smaller than traditional CA. The front left and rear right wheels have the same trend as the front right wheel and rear left wheel, so they are not presented in the manuscript. Table 2 can further explain that for the Eco-CA method, the absolute mean value of front wheel torques are larger than traditional CA, while the torques
FIGURE 12  Motor energy consumption in pure longitudinal motion

TABLE 2  Absolute mean value of motor torque of individual wheel (unit: N·m)

|                | Eco-CA         | Without Eco-CA |
|----------------|----------------|----------------|
| Front left wheel| 74.2598        | 69.5948        |
| Front right wheel| 108.3211      | 69.5948        |
| Rear left wheel  | 46.0690        | 69.5948        |
| Rear right wheel | 52.1389        | 69.5948        |

on rear wheels are smaller than traditional CA. On the other hand, the absolute mean values of individual motor torques are same for the traditional CA method, which is reasonable during the pure longitudinal motion for traditional CA.

It is noted that in order to clearly show the advantage of proposed motor energy efficient control in high-level CACC (Eco-CACC) and low-level CA (Eco-CA), Table 3 is added to show the total motor energy consumption of ‘Eco-CACC+Eco-CA’, ‘Eco-CACC+no-Eco-CA’, ‘no-Eco-CACC+Eco-CA’ and ‘no-Eco-CACC+no-Eco-CA’. It can be seen that ‘Eco-

FIGURE 13  Tyre friction dissipation of vehicles in pure longitudinal motion

FIGURE 14  The distributed driving torque of the wheels in pure longitudinal motion: (a) front right wheel (b) rear left wheel
|                  | Eco-CACC       | Without Eco-CACC |
|------------------|----------------|-----------------|
| Eco-CA           | 1.0528         | 1.0663          |
| Without Eco-CA   | 1.5075         | 1.5217          |

CACC+Eco-CA’ has the lowest motor energy consumption while ‘non-Eco-CACC+non-Eco-CA’ shows highest motor energy consumption.

### 5.2 Scenario B: Lane-change manoeuvre

In scenario B, whether the proposed approach can obviously improve the energy-efficiency during lane-change is examined. There is no FV behind the SV in this scenario. In addition, a successful overtaking should not only ensure safety, but also take care of passenger’s comfort during the lane-change. In this scenario, in order to maintain the safety distance between the LV and the SV, it is necessary to decelerate the SV before lane-change.

The vehicle paths with different applied decelerations and the same initial speed of 25 m/s are shown in Figure 15(a). In Figure 15(b), the path tracking performance of four different middle and lower layer MPC controllers are compared, including active front steering (AFS), active front steering and Eco-CA (AFS+CA), four-wheel steering (4WS) and four-wheel steering and Eco-CA (4WS+CA). It can be founded that all the lower controllers can track the desired trajectory perfectly by choosing the appropriate weighting matrix of MPC controller in the middle layer.

During lane-change, the energy consumption of driving motor and the energy dissipation of tyre constitute most of the total losses, and the air drag dissipation is negligible. The energy consumption of motor and tyre are presented in Figure 16, which are obviously affected by different middle and lower layer control method. Figure 16(a) shows that the regenerative braking torque at high deceleration can recycle more energy than at low deceleration. The reason is that the motors are working in their high regenerative braking power region at high deceleration. Compared with the control method without Eco-CA, the proposed Eco-CA control method in lower layer can improve braking energy recovery efficiency significantly. However, the power losses of tyre slip are higher due to tyre sliding at high deceleration, which has been reflected in Figure 16(b). Figure 16(b) also suggests that with the increase of the absolute value of vehicle acceleration, the tyre energy dissipation of controller without Eco-CA increases gradually, and in the meanwhile, the tyre energy dissipation of controller with Eco-CA decreases gradually. In general, as seen in Figure 16(c), the proposed algorithm (4WS with MPC and Eco-CA) can improve the energy efficiency with more regeneration power while lane changing and braking.

The yaw rates and slip angle responses of different methods are depicted in Figure 17(a,b), from which it is obviously found that 4WS controllers have smaller yaw rate and slip angle than those of AFS controller. It also can be found that the proposed control method can achieve similar good yaw rate and side slip angle performance as other methods.

### 6 CONCLUSION

An energy-efficient control system of over-actuated electric vehicle is proposed to improve not only the individual vehicle dynamics control, but also improve the energy efficiency of electric vehicle in traffic flow for extended driving range. The overall energy-efficient ecological CACC controller of the vehicle platoon is designed based on a platoon model. The dynamics control and energy efficiency optimization for the individual vehicle in platoon is designed including middle layer controller and lower layer Eco-CA. To test the performance of the designed energy-efficient control system, two simulation manoeuvres are carried out in MATLAB. In the first simulation case of pure straight lane motion, the energy consumption of driving motor has been obviously reduced by the proposed energy-efficient high-level CACC and low-level CA method. The second simulation case aims to evaluate the energy-efficient control sys-
FIGURE 16 Energy consumption during single lane changing: (a) driving motor regeneration (b) tyre slip (c) total power regeneration

FIGURE 17 Vehicle dynamics performance during single lane changing: (a) yaw rate (b) slip angle

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