Study on Control Strategy of Handling Stability for an Eight In-Wheel Motor Drive AWS Electric Vehicle

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Abstract

This paper took an eight in-wheel motor independent driven and all-wheel independent active steering vehicle as research object, and proposed a hierarchical optimized tire force distribution control strategy (8WD8WS+) based on sliding mode control and control allocation to coordinate the output torques and steering angle of eight wheels. The hierarchical control strategy included an upper vehicle motion controller, a lower tire force distribution controller, and a tire force tracking controller. The upper controller adopted the sliding mode control algorithm focusing on the vehicle motion state control. The lower controller used the optimal control allocation method to realize the tire force distribution control under the constraint condition, and proposed the error approximation objective function and performance objective function. And the tire force distribution was solved by the active set method. The desired tire force tracking controller used the sliding mode control algorithm to realize the tracking of the desired slip ratio and sideslip angle calculated by the inverse tire model, and calculates the actuator output. Finally, the control strategy was verified by simulation based on the MATLAB/Simulink model, and the closed-loop slalom test at high speed verified the effectiveness of the control strategy.

Keywords: handling stability; in-wheel motor; AWS; sliding mode control; optimal control distribution

Nomenclature

Abbreviation

AWS All Wheels Steering
8WD8WS+ Eight Wheels Driving and Eight Wheels Steering Plus
SMC Sliding Mode Control
CA Control Allocation
DOF Degree of Freedom
8WD4WS Eight Wheels Driving and Eight Wheels Steering without The Desired Tire Force Tracking Controller

8WD8WS Eight Wheels Driving and First Four Wheels Steering

Subscript
i The Number of Axle of The Wheel
j The Corresponding Axle Left/Right Wheel

1. Introduction

The research of in-wheel motor driven electric vehicle is an important direction to solve the problems of heavy vehicles in dynamics and maneuverability, and handling stability control of heavy vehicles is a problem that need to be solved urgently.

Vehicle handling stability control methods mainly include direct yaw moment control and active steering control [1]. The direct yaw moment control method was firstly proposed by Bosch in Germany. Active steering was developed with the development of the steer-by-wire technology, mainly controls the lateral force output of the wheel by controlling the active steering angle of the wheel.

Many researches have been carried out on this subject at home and abroad [2]. Wongun Kim proposed a hierarchical coordinated control strategy for an eight in-wheel motor driven vehicle, which considered the tire friction circle constraint in tire force distribution control, and added wheel slip ratio control [3]. Zhao Haiyan conducted a more in-depth discussion on the upper vehicle motion controller, and proposed a three-step nonlinear control method, which realized the decoupling control of the yaw rate and the sideslip angle [4]. Zhai Li added a phase plane stability judgment to the vehicle stability control strategy [5]. Zhang et al. achieved stability improvements with full tire force control for eight-wheel AWS vehicle [6]. But in the limiting condition, vehicle handling stability control methods need to be improved to keep vehicle running steadily.

This paper took the 8×8 in-wheel motor driven AWS electric vehicle as the research object, focusing on the improvement of handling stability in limiting condition based on SMC, CA and tire force tracking control.
2. Control Strategy of Handling Stability

2.1 Vehicle Dynamics Model

The eight in-wheel motor driven AWS electric vehicle is a typical redundant and overdrive system, to establish an accurate vehicle dynamics model is the foundation of dynamics research, like Fig. 1. In this paper, the vehicle 22 DOF model was established, including 6 DOF of the body motion (including longitudinal, lateral, vertical, yaw, pitch, and roll motion) and 8 DOF of wheel rotation and 8 DOF of wheel vertical motion. The vehicle parameters were given in Tab. 1.

Table 1 Vehicle parameters

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle mass m</td>
<td>23000 kg</td>
</tr>
<tr>
<td>Tread B</td>
<td>2.6 m</td>
</tr>
<tr>
<td>Vehicle moment of inertia I_z</td>
<td>36000 kg·m²</td>
</tr>
<tr>
<td>Coefficient of rolling resistance f_r</td>
<td>0.02</td>
</tr>
<tr>
<td>Coefficient of air resistance C_D</td>
<td>0.5</td>
</tr>
<tr>
<td>Vehicle windward area A_f</td>
<td>4.2 m²</td>
</tr>
<tr>
<td>Air density ρ_a</td>
<td>1.2258 kg/m³</td>
</tr>
<tr>
<td>Wheel moment of inertia I_w</td>
<td>150 kg·m²</td>
</tr>
<tr>
<td>Wheel rolling radius R</td>
<td>0.59 m</td>
</tr>
<tr>
<td>In-wheel motor reduction ratio i</td>
<td>11.07</td>
</tr>
</tbody>
</table>

Figure 1 Vehicle oxyz plane motion.

2.1.1 Vehicle Body Kinematic Equation

The 3 DOF kinematic equation for the vehicle body longitudinal, lateral and yaw motion is following:

\[
\begin{align*}
\dot{m}(\text{v}_x - \text{v}_y) &= F_{wx} - mgf - 0.5C_D A_f \rho_a v^2 \\
\dot{m}(\text{v}_x + \text{v}_y) &= F_w \\
I_{\omega_y} &= M_w
\end{align*}
\]

(1)

Where \( F_{x,y,z} \) represents the resultant force/torque formed by the tire longitudinal/lateral force in the vehicle coordinate system, respectively.

2.1.2 Wheel and Tire Model

The force analysis of the wheel rotation motion is shown in Fig. 2. Where \( T_{\text{aij}} \) is the wheel drive torque, \( F_{\text{x,ij}} \) is the wheel longitudinal force, \( F_{\text{z,ij}} \) is the wheel vertical force, \( P_{\text{x,ij}} \) and \( P_{\text{z,ij}} \) represent the body forces applied to the wheel.

The wheel kinematic equation is:

\[
I_{\omega_i} = T_{\text{aij}} - T_{\text{ij}} - F_{\text{wij}}R
\]

(2)

Where \( T_{\text{ij}} \) is the rolling resistance moment of wheel:

\[
T_{\text{ij}} = F_{\text{wij}}R
\]

(3)

Figure 2 Rotation motion of the wheel

The tire force formed by its interaction with the ground directly affects the vehicle's motion, so the tire model is especially critical in vehicle dynamics modeling. The “Magic Formula Tire Model” was widely used in the calculation of tire force [?]. The tire force under the combined condition of the slip and sideslip could be obtained from the basic magic formula expression:

\[
\begin{align*}
F_x &= \mu F_z \sin(C, \arctan(B, \kappa_z - E_x(B, \alpha_z - \arctan(B, \alpha_z))) \\
&+ S_z \cdot \cos(\arctan(r_{in}, \cos(\arctan(r_{in}, \alpha_z)))) \\
&= \kappa + S_z \cdot \cos(\arctan(r_{in}, \cos(\arctan(r_{in}, \alpha_z)))) \\
\kappa_z &= \kappa + S_z \\
F_y &= \mu F_z \sin(C, \arctan(B, \kappa_y - E_y(B, \alpha_y - \arctan(B, \alpha_y))) \\
&+ S_y \cdot \cos(\arctan(r_{in}, \cos(\arctan(r_{in}, \alpha_y)))) \\
&= \kappa_y + S_y \\
M_z &= \mu F_z \sin(C, \arctan(B, \alpha_z - E_z(B, \alpha_z - \arctan(B, \alpha_z))) \\
&+ S_z \cdot \cos(\arctan(r_{in}, \cos(\arctan(r_{in}, \alpha_z)))) \\
&= \alpha_z + S_z
\end{align*}
\]

Where \( \mu \) represents the friction coefficient of the ground. \( \kappa_{ij}, \alpha_{ij} \) represent the slip ratio and sideslip angle of the wheel, respectively.

2.2 The Hierarchical Control Strategy

The hierarchical optimized tire force distribution control architecture was shown in Fig. 3. Firstly, the driver's command, including the steering wheel angle \( \delta \), and the accelerator pedal opening (equivalent to the desired vehicle longitudinal speed, yaw rate, and centroid sideslip angle).
The upper controller calculated the desired resultant force/torque, including the desired longitudinal resultant force \( F_{xu} \), the lateral resultant force \( F_{yu} \), and the yaw moment \( M_{zu} \) based on the deviation between the expected vehicle state value and the actual value feedback from the vehicle model to ensure that the vehicle can follow the driver’s driving. The lower tire force distribution controller was mainly responsible for the realization of the resultant force on the tire. The output of the lower controller was the desired longitudinal forces \( F_{xdi} \) and lateral forces \( F_{ydi} \) of all wheels. A desired tire force tracking controller was added after the lower controller in order to better achieve the desired tire force output, which included an analytical inverse tire model and a slip ratio/sideslip angle tracking controller. The analytic inverse tire model used the inverse "Magic Formula" method to obtain the desired slip ratio \( k_{di} \) and sideslip angle \( \alpha_{di} \) of the wheel corresponding to the desired tire force. A closed loop control system was established based on the vehicle motion state that be returned to the upper controller in real time.

Figure 3 The hierarchical control architecture

2.3 The Upper Vehicle Motion Controller

In the field of vehicle dynamics control, the commonly used control algorithms include linear control algorithms (mainly PID control \([5]\), pole position control, \( H_\infty \) control, etc.) and nonlinear control algorithms (mainly sliding mode control \([2]\), fuzzy control, adaptive control \([6]\), neural network control, etc.). SMC has the advantages of being insensitive to system parameter variation and external disturbance, and has strong robustness. It can better deal with the uncertainty in nonlinear systems. So, the SMC algorithm was used for the upper vehicle motion controller. Ignoring vehicle roll and pitch motion, (1) could be simplified to:

\[
\begin{align*}
\dot{v}_r &= F_w - F_b + \xi_r \\
\dot{v}_y &= (\beta + \omega) F_b + \xi_y \\
I \ddot{\theta}_y &= M_u + \xi \dot{\theta}_y
\end{align*}
\]

(5)

Where \( \xi_r, \xi_y, \xi \) are the uncertainty caused by the simplified model and external disturbance.

The sliding surface was selected as the deviation between the actual state value of the vehicle and the driver's desired value to maximize to the desired vehicle speed, yaw rate and centroid side slip angle tracking:

\[
\begin{align*}
x_r &= v_r - v_d \\
x_y &= v_y - v_d \\
x &= \omega - \omega_d
\end{align*}
\]

(6)

The SMC of the three vehicle reference motion states selected the exponential approach law with saturation function to minimize the chattering problem and ensure the approach speed of the system to the switching surface:

\[
\begin{align*}
s &= -\varepsilon \text{sat}(s/\Phi) - k \cdot s & \varepsilon > 0, k > 0, \Phi > 0 \\
\text{sat}(s/\Phi) &= \begin{cases} 1, & s/\Phi > 1 \\
-1, & s/\Phi < -1 \\
1, & 0 \leq s/\Phi \leq 0 \\
-1, & 0 \leq s/\Phi \leq 0
\end{cases}
\end{align*}
\]

(7)

(8)

Where \( \varepsilon, k, \Phi \) are the SMC parameters. Combining (6), (7) and (8) into (5) can calculate the vehicle desired resultant force/torque. Thus:

\[
\begin{align*}
F_w &= m\left[ \ddot{v}_r + \xi_r - \varepsilon \text{sat}(s_r) - k \cdot s_r \right] + F_b \\
F_y &= m\left[ \dot{\omega}_r + \xi - \varepsilon \text{sat}(s) - k \cdot s \\
M_u &= I\left[ \ddot{\omega}_r + \xi - \varepsilon \text{sat}(s) - k \cdot s \right]
\end{align*}
\]

(9)

2.4 The Lower Tire Force Distribution Controller

The resultant force/torque calculated by the upper controller should be achieved with the actual tire force of each wheels. This paper chose the optimal control distribution method to solve the tire force distribution control problem. The expression of the CA problem with constraints is:

\[
v = Bu
\]

(10)

Where \( v \in \mathbb{R}^n \) is the target control vector, \( u \in \mathbb{R}^m \) is the actual control vector and \( B \in \mathbb{R}^{m \times n} \) is the control validity matrix. For the tire force distribution control:

\[
v = [F_{xu}, F_{yu}, M_u]^T
\]

(11)

\[
u = \begin{bmatrix} F_{x11} & F_{x12} & F_{x21} & F_{x22} & F_{x31} & F_{x32} & F_{x33} & F_{x41} & F_{x42} & F_{x43} \end{bmatrix}^T
\]

(12)

\[
B = \begin{bmatrix} B_1 & B_2 & B_3 \end{bmatrix}
\]

(13)

\[
B_1 = [\cos \delta_{ij}, -\sin \delta_{ij}] \quad (i = 1,2,3,4; \ j = 1,2,3,4)
\]

\[
B_2 = [\sin \delta_{ij}, \cos \delta_{ij}] \quad (i = 1,2,3,4; \ j = 1,2,3,4)
\]
B_3 = [-d_4 \cos \delta_2 + l_1 \sin \delta_1, d_4 \cos \delta_2 + l_1 \sin \delta_2, 
-d_5 \cos \delta_3 - l_2 \sin \delta_2, d_5 \cos \delta_3 + l_2 \sin \delta_2, 
-d_6 \cos \delta_4 - l_3 \sin \delta_3, d_6 \cos \delta_4 + l_3 \sin \delta_3, 
-d_7 \cos \delta_5 - l_4 \sin \delta_4, d_7 \cos \delta_5 + l_4 \sin \delta_4, 
-d_8 \cos \delta_6 - d_9 \sin \delta_5, d_8 \cos \delta_6 + d_9 \sin \delta_5, 
-d_10 \cos \delta_7 - d_11 \sin \delta_6, d_10 \cos \delta_7 + d_11 \sin \delta_6, 
-d_12 \cos \delta_8 - d_13 \sin \delta_7, d_12 \cos \delta_8 + d_13 \sin \delta_7, 
-d_14 \cos \delta_9 - d_15 \sin \delta_8, d_14 \cos \delta_9 + d_15 \sin \delta_8, 
-d_16 \cos \delta_10 - d_17 \sin \delta_9, d_16 \cos \delta_10 + d_17 \sin \delta_9, 
-d_18 \cos \delta_11 - d_19 \sin \delta_10, d_18 \cos \delta_11 + d_19 \sin \delta_10, 
-d_20 \cos \delta_12 - d_21 \sin \delta_11, d_20 \cos \delta_12 + d_21 \sin \delta_11, 
-d_22 \cos \delta_13 - d_23 \sin \delta_12, d_22 \cos \delta_13 + d_23 \sin \delta_12, 
-d_24 \cos \delta_14 - d_25 \sin \delta_13, d_24 \cos \delta_14 + d_25 \sin \delta_13, 
-d_26 \cos \delta_15 - d_27 \sin \delta_14, d_26 \cos \delta_15 + d_27 \sin \delta_14, 
-d_28 \cos \delta_16 - d_29 \sin \delta_15, d_28 \cos \delta_16 + d_29 \sin \delta_15].

The error approximation objective function and the performance objective function was defined to improve tracking accuracy, respectively:

1. The error approximation objective function:

\[ J_1 = \| W_u (Bu - v) \|^2 \quad (14) \]

Where \( W_u \) is the diagonal weight matrix, which represents the importance of the corresponding desired resultant force/torque in an error approximation objective function.

2. The performance objective function

The selection of the performance objective function focused on the optimization of the drive system efficiency and vehicle stability. So the performance objective function can be defined based on the concept of the tire load rate, so that the sum of the load rates of all the tires is minimized, and improved the adhesion performance of each tire. Defined the tire load rate:

\[ \rho_y = \sqrt{\frac{F_{x,y}^2 + F_{y,y}^2}{\mu \cdot F_{x,y}}} , \quad 0 \leq \rho_y \leq 1 \quad (15) \]

Thus the performance objective function:

\[ J_2 = \| W_y u \|^2 \quad (16) \]

\[ W_y = \text{diag} \left( \frac{\lambda_1}{\mu_1 \cdot F_{x,1} \cdot F_{y,1}}, \frac{\lambda_2}{\mu_2 \cdot F_{x,2} \cdot F_{y,2}}, \ldots, \frac{\lambda_n}{\mu_n \cdot F_{x,n} \cdot F_{y,n}} \right) \]

Where \( W_y \) is a diagonal weighting matrix, \( \lambda_i \) is a weighting factor representing the weight value of the corresponding wheel load factor.

The integrated objective function and the actual control output of the lower controller could be obtained by the weighted least squares method:

\[ J = J_1 + J_2 = \eta \| W_y (Bu - v) \|^2 + \| W_u u \|^2 \]

\[ u = \arg \min_{\theta} (J) \quad (17) \]

Where \( \eta \) is a coordination factor, generally taking a small value, so that in the process of solving the optimal solution, it is preferred to ensure that the error approximation target function value is smaller, and the driver's driving intention is better achieved. Equation (17) can be solved by the active set method, and the desired longitudinal/lateral force of each wheel is finally calculated.

2.5 The Tire Force Tracking Controller

2.5.1 The analytical inverse tire model

The current common inverse tire model include both table lookup and analytical methods. The analytical method is widely used because of its small amount of stored data and fast calculation speed. The control idea is based on the tire force, slip rate, sideslip angle return value and the partial derivative of tire force which respects to slip ratio and side angle of the previous moment to solve the desired slip ratio and sideslip angle corresponding to the desired tire force at the current time.

The first order Taylor expansion of magic formula tire model (4) is following:

\[ F_{x,y}(t_i) \approx F_{x,y}(t_{i-1}) + \frac{\partial F_{x,y}}{\partial \kappa} \left[ \kappa(t_i) - \kappa(t_{i-1}) \right] + \frac{\partial F_{x,y}}{\partial \alpha} \left[ \alpha(t_i) - \alpha(t_{i-1}) \right] \quad (18) \]

Solve \( \kappa_{y}(t_i) \) and \( \alpha_{y}(t_i) \):

\[ \begin{bmatrix} \kappa_{y}(t_i) \\ \alpha_{y}(t_i) \end{bmatrix} \approx \begin{bmatrix} \kappa(t_{i-1}) \\ \alpha(t_{i-1}) \end{bmatrix} + M^{-1} \begin{bmatrix} F_{x,y}(t_i) - F_{x,y}(t_{i-1}) \\ F_{y,y}(t_i) - F_{y,y}(t_{i-1}) \end{bmatrix} \quad (19) \]

Where \( M \) is the partial derivative matrix, \( \partial F / \partial \kappa \) and \( \partial F / \partial \alpha \) represent the partial derivatives of the tire force to the wheel slip ratio and the sideslip angle, respectively:

\[ M = \begin{bmatrix} \frac{\partial F_{x,y}}{\partial \kappa} & \frac{\partial F_{x,y}}{\partial \alpha} \\ \frac{\partial F_{y,y}}{\partial \kappa} & \frac{\partial F_{y,y}}{\partial \alpha} \end{bmatrix} \quad (20) \]

2.5.2 The slip rate tracking control

It was also necessary to design the slip ratio and the sideslip angle tracking control to ensure the tire force was realized on the tire after obtaining the desired slip ratio and sideslip angle of the wheel in the reverse tire model. The wheel slip rate tracking was designed using a SMC algorithm like the upper vehicle motion controller.

(1) Slip rate tracking control under driving condition
Slip rate expression under driving condition:

\[
\kappa_s = \frac{\omega_y R - v_{wxy}}{\omega_y R} \tag{21}
\]

Combining the time differential of the above formula into the wheel rotational motion differential equation:

\[
\dot{\kappa}_{ij} = (1 - \kappa_{ij}) \frac{T_{wxy} - (F_{xij} + F_{zij}f_R)}{I_{wj}/\omega_{ij} - \omega_{ij}R} - \frac{\dot{v}_{wxy}}{\omega_{ij} R} \tag{22}
\]

In the slip rate tracking control, the constant velocity approach law with saturation function was used to suppress the chattering problem, which can be expressed as:

\[
\ddot{s} = -\eta_{\text{sat}}(s / \Phi) \tag{23}
\]

The definition of \( sat(s / \Phi) \) was given in (8). The desired output torque of the wheel under driving condition can be obtained:

\[
T_{ij} = \left[ \frac{I_{\omega ij}}{1 - \kappa_{ij}} \kappa_{ij} + \frac{I_{\omega ij}}{R(1 - \kappa_{ij})} \dot{\kappa}_{ij} + F_{xij} R \right] + \frac{F_{zij} f_R - I_{\omega ij}}{1 - \kappa_{ij}} \eta_{\text{sat}}(\kappa_{ij}) \tag{24}
\]

Thus, the desired output torque of the in-wheel motor under driving condition is:

\[
T_{ij} = T_{wij} / i = \left[ \frac{I_{\omega ij}}{1 - \kappa_{ij}} \kappa_{ij} + \frac{I_{\omega ij}}{R(1 - \kappa_{ij})} \dot{\kappa}_{ij} + F_{xij} R \right] / i \tag{25}
\]

Where \( v_c \) is the speed of wheel center in the longitudinal direction of the wheel coordinate system.

(2) Slip rate tracking control under braking condition

The desired output torque of the wheel and in-wheel motor under the braking condition can be obtained similarly to the slip rate tracking control under the driving condition as follows:

\[
T_{wij} = \frac{I_{\omega ij}}{1 - \kappa_{ij}} \kappa_{ij} + \frac{I_{\omega ij}}{R(1 - \kappa_{ij})} \dot{\kappa}_{ij} + F_{xij} R + F_{zij} f_R - \frac{I_{\omega ij}}{1 - \kappa_{ij}} \eta_{\text{sat}}(\kappa_{ij}) \tag{26}
\]

\[
T_{ij} = T_{wij} / i = \left[ \frac{I_{\omega ij}}{1 - \kappa_{ij}} \kappa_{ij} + \frac{I_{\omega ij}}{R(1 - \kappa_{ij})} \dot{\kappa}_{ij} + F_{xij} R + F_{zij} f_R - \frac{I_{\omega ij}}{1 - \kappa_{ij}} \eta_{\text{sat}}(\kappa_{ij}) \right] / i \tag{27}
\]

2.5.3 The sideslip angle tracking control

The steering angles of all wheels are independently controllable in the all-wheel active steering vehicles. The desired steering angle of each wheel \( \delta_{ij} \) can be calculated by the following formula:

\[
\alpha_{ij} = \frac{v_c + L \omega_{ij}}{v_c + B / 2 \cdot \omega_{ij} + \epsilon} - \delta_{ij} \tag{28}
\]

Where \( L \) is the distance between the axle of wheel and the center of mass.

2.6 Simulation Analysis

A closed-loop slalom test simulation was carried out at ground adhesion coefficient 1 and vehicle speed 60 km/h to verify the effectiveness of the hierarchical optimized tire force distribution control strategy. The eight in-wheel motor drive AWS electric vehicle slalom test refers to the setting of the vehicle with total mass greater than 15t in the national standard “GBT6323-014 Vehicle Steering Stability Test Method”. This paper used 8WD8WS and 8WD4WS as comparative analysis object to comparative analysis.

It can be seen from the simulation results that the three control strategies can substantially complete the slalom test condition from Fig. 4 to Fig. 10. But compared with 8WD8WS and 8WD8WS+, 8WD4WS had a poor performance, especially in the longitudinal speed tracking and centroid side angle control. And from Fig. 8, the 8WD4WS control strategy had appeared the situation that the first four wheels of longitudinal force reached the ground attachment limit during the turning.

In comparison, the 8WD8WS+ had the most outstanding control effect. On one hand, 8WD8WS+ can keep the vehicle longitudinal velocity closer to 60km/h than 8WD8WS from Fig. 4. On the other hand, the centroid side angle of 8WD8WS+ was near to 0, but 8WD8WS was near to 0.1 from Fig. 6. That is mainly due to the 8WD8WS+ can track the desired tire force by tracking the slip rate and the sideslip angle directly after adding the tire force tracking controller. It comes to achieve a better desired resultant force/torque tracking of the upper controller.

3. Conclusion

In this paper, a hierarchical optimized distribution control strategy was proposed, which was based on SMC. The lower controller achieves the tracking of full tire longitudinal/lateral force better through the slip rate sliding mode control. The simulation results show that the hierarchical optimized control allocation strategy can achieve vehicle stability control by coordinating the output torque and steering angle of each wheel compared with the vehicle without rear two-axle steering and no lower controller for all-wheel steering vehicle.

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Reference


