

Design of driving control strategy of torque distribution for two - wheel independent drive electric vehicle

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Abstract. In order to coordinately control the torque distribution of existing two-wheel independent drive electric vehicle, and improve the energy efficiency and control stability of the whole vehicle, the control strategies based on fuzzy control were designed which adopt the direct yaw moment control as the main line. For realizing the torque coordination simulation of the two-wheel independent drive vehicle, the vehicle model, motor model and tire model were built, including the vehicle 7 - DOF dynamics model, motion equation, torque equation. Finally, in the Carsim - Simulink joint simulation platform, the feasibility of the drive control strategy was verified.

1. Introduction

The automobile is an important part of social development, and promotes the development of the global economy. Under the influence of the global energy crisis, traditional fuel vehicles are gradually being replaced by clean and pollution-free electric vehicles^[1]. In order to improve the control stability of independent drive electric vehicle, the direct yaw moment control based on fuzzy control^[2] and independent drive torque distribution based on stability are proposed^[3-5], and the direct yaw moment control becomes one of the main research directions in this field. At the same time, the electronic differential control and steering coordination control of independent drive electric vehicle are the key technologies to guarantee the stability of the operation^[6]. Up to now, the independent drive control technology of electric vehicle develops rapidly. In this paper, the direct yaw moment control is taken as the main line, which has achieved good results in the test simulation, and provides a reference for the exploration of independent driving control technology of electric vehicle.

2. Theoretical analysis

2.1. Whole vehicle model

Ignoring the roll, pitch and vertical motions of the vehicle, the seven - degree - of - freedom vehicle model is established based on the motion of vehicle longitudinal, lateral, yaw and four wheel rolling, then we can get^[7]:

$$\text{longitudinal motion equation: } ma_x = (F_{x1} + F_{x2}) \cos \delta - (F_{y1} + F_{y2}) \sin \delta + F_{x3} + F_{x4} - F_w \quad (2.1)$$

$$\text{lateral motion equation: } ma_y = (F_{x1} + F_{x2}) \sin \delta + (F_{y1} + F_{y2}) \cos \delta + F_{y3} + F_{y4} \quad (2.2)$$

yaw motion equation:

$$I_z \dot{\gamma} = [(F_{y1} + F_{y2}) \cos \delta + (F_{x1} + F_{x2}) \sin \delta] \cdot a - (F_{y3} + F_{y4}) \cdot b + (F_{x2} - F_{x1}) \cos \delta \cdot \frac{d_f}{2} + (F_{y1} - F_{y2}) \sin \delta \cdot \frac{d_f}{2} + (F_{x4} - F_{x3}) \cdot \frac{d_r}{2} \quad (2.3)$$



From(2.1-2.3), m -quality of automobiles, kg; a_x -longitudinal acceleration, m/s²; a_y -lateral acceleration, m/s²; δ -front steering angle, rad; $F_{xi} = (i = 1 \sim 4)$ -longitudinal force of each wheel, N; $F_{yi} = (i = 1 \sim 4)$ -lateral force of each wheel, N; V -speed of vehicle, km/h; u -longitudinal speed, km/h; v -lateral speed, km/h; γ -yaw rate, rad/s; I_z -the moment of inertia about the Z axis; a 、 b -the distance of the center of mass to the front and rear axles, m; d_f 、 d_r -font and rear wheel track, m.

2.2 Ideal yaw angular velocity calculation

The ideal value of the control variable reflects the ideal running state, and the expected reference value can be obtained by 2 - DOF model: in order to verify the feasibility of the control strategy, the ideal centroid side-slip angle is set to 0, $\beta_d = 0$.

calculation formula of ideal yaw angular velocity:

$$\gamma_d = \min \left\{ \left| \frac{u}{L(1 + Ku^2)} \cdot \delta \right|, \left| \frac{\mu \cdot g}{u} \right| \right\} \text{sign}(\delta) \quad (2.4)$$

if the vehicle driving is stable, the steady state yaw angular velocity should meet the following conditions:

$$|\gamma_d| \leq \left| \frac{\mu \cdot g}{u} \right| \quad (2.5)$$

K -stability factor; L - axle - spacing of two shafts, m; μ -coefficient of road surface attachment; g -acceleration of gravity, m/s².

2.3 Tire model calculation

According to the a steering model under ideal conditions, speed equations of two front wheels can be obtained. The parameters such as longitudinal force and lateral force of the wheel can be obtained by tire model. In this paper, the Dugoff tire model is selected, and the calculation formulas are as follows:

$$\text{longitudinal force: } F_{xi} = C_x \frac{s}{1 + s} f(\lambda_i) \quad (2.6)$$

$$\text{lateral force: } F_{yi} = C_y \frac{\tan \alpha_i}{1 + s} f(\lambda_i) \quad (2.7)$$

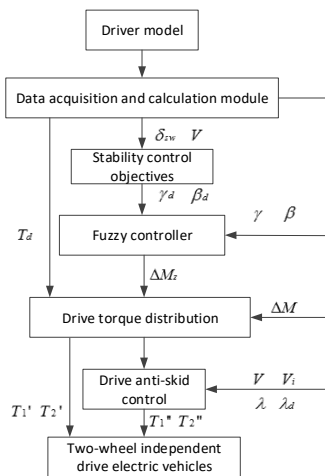
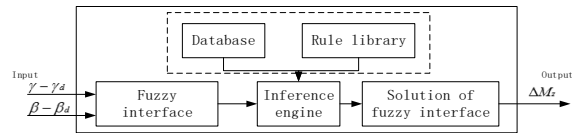
From(2.4-2.7), δ -steering angle, deg; L -wheelbase, m; C_x -longitudinal stiffness of tire, KN·m/rad; C_y -lateral stiffness of tire, KN·m/rad; α_i -tire camber angle, rad;

As we know, the speed of each wheel is related to the actual speed of the vehicle V , the steering angle δ , the relevant structural parameters of the body, the wheelbase L and the tread.

3. Design of vehicle control strategy

3.1 Design of torque coordination control strategy

The control logic of torque coordination is to change the adhesion of the driving wheels by controlling the driving torque of each motor to achieve the desired operational state for obtaining the beneficial yaw moment to the vehicle steering. Fig.3.1 the structure diagram of the drive torque coordination control system.

**Figure 3.1** Coordination control system**Figure 3.2** Fuzzy controller structure

3.2 Determination of yaw moment based on fuzzy control theory

Reducing the side-slip angle of the center of mass can ensure the vehicle in the range of ideal yaw angular velocity, and make the system more stable. In this paper, the deviation of yaw angular velocity $\Delta\gamma$ and the side-slip angle of the center of mas $\Delta\beta$ are input fuzzy controller, the true value of the yaw moment ΔM_z can be obtained after the ambiguity treatment, the structure of the controller is shown in Fig.3.2 above.

(1)Fuzzy processing:The triangular membership function is selected for fuzzy processing, and the analytical expression is as shown in formula (3.1).

$$f(x) = \begin{cases} 0 & x \leq a \\ \frac{x-a}{b-a} & a \leq x \leq b \\ \frac{c-x}{c-b} & b \leq x \leq c \\ 0 & x \geq c \end{cases} \quad (3.1)$$

Yaw moment of controller real - time output ΔM_z , the basic universe is $[-1,1]$, in order to control the motion state of the vehicle more accurately, ΔM_z it is divided into 7 grades: $\{ (NB), (NM), (NS), (ZO), (PS), (PM), (PB) \}$.

(2) Set up fuzzy rules: When adjusting the control parameters, the following rules shall be met:

①When the vehicle is turning out, a positive yaw moment is applied to the outer wheel of the vehicle to allow it to move smoothly;

②When the vehicle is turning too much, a negative yaw moment is applied to the inner wheel of the vehicle, so that it can turn smoothly.

Using the Mamdani reasoning algorithm for fuzzy reasoning, and combined with the fuzzy membership function and the control rules designed in the front, the design of fuzzy inference rules table is shown in Table 3.1.

Table 3.1 Fuzzy rules table of fuzzy controller

$\Delta\beta \backslash \Delta\gamma$	NB	NS	ZO	PS	PB
NB	PB	PB	NS	NB	NB
NS	PB	PM	NS	NM	NB
ZO	PM	PS	ZO	NS	NM

PS	PB	PM	PS	NM	NB
PB	PB	PS	PS	NS	NB

3.3 Design of torque distribution

Since the yaw moment and the ideal driving torque have been determined, we need to distribute the torque of each wheel. The single value of the accelerator pedal is linearly proportional to the driver's expected driving torque, and the driving torque T_d is determined according to the pedal opening degree, then we can design the torque distribution based on the real-time yaw moment value.

Because of the tire friction circle limit, the drive torque distribution is constrained. On this basis, the distribution algorithm based on stability is designed.

Here has two constraints: the value of yaw moment in the vehicle 7 - DOF dynamics equation and the torque of each wheel after distribution each wheel should meet the expected output driving torque T_d and yaw moment value ΔM . We can get the following formula from 3.2-3.4.

$$T_d = K S_{ac} \quad (3.2)$$

$$2T_d = T_1'' + T_2'' \quad (3.3)$$

$$\Delta M = (F_{x2} - F_{x1}) \cos \delta \cdot \frac{d_f}{2} + (F_{x4} - F_{x3}) \cdot \frac{d_r}{2} \quad (3.4)$$

On this basis, the limit of the yaw moment shall be less than the maximum yaw moment value that can be provided on the ground. When the driving wheel does not slip, the anti-slip control strategy shall not be implemented, at this time, $T_1'' = T_1'$, $T_2'' = T_2'$.

S_{ac} - Pedal signal, m; K - coefficient; T_1 , T_2 - driving torque of two front wheels, N m.

The driving force of the vehicle can be realized by controlling the torque of the wheel hub motor:

$$F_{xi} = T_i / R \quad (3.5)$$

At the same time, there is a constraint of the tire friction ellipse, that is, the sum vector of the longitudinal force and the lateral force of the tire should be less than or equal to the maximum adhesion provided by the road surface.

4. Experimental simulation and verification

4.1 Establishment of simulation model

After analyzing the vehicle system model, and establishing the driving control strategy, this chapter carries on the simulation model building and the simulation verification of the control strategy. The vehicle model, the motor model in Fig. 4.1, the tire model in Fig. 4.2 based on the Dugoff, and some driving control strategies models are established in Carsim and Simulink^[8], finally, the joint simulation platform is built as shown in Fig. 4.3.

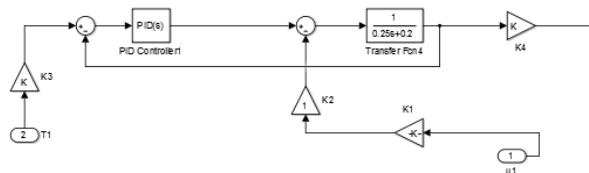


Figure 4.1 Motor model

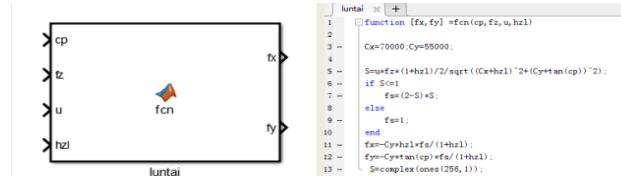


Figure 4.2 Tire model

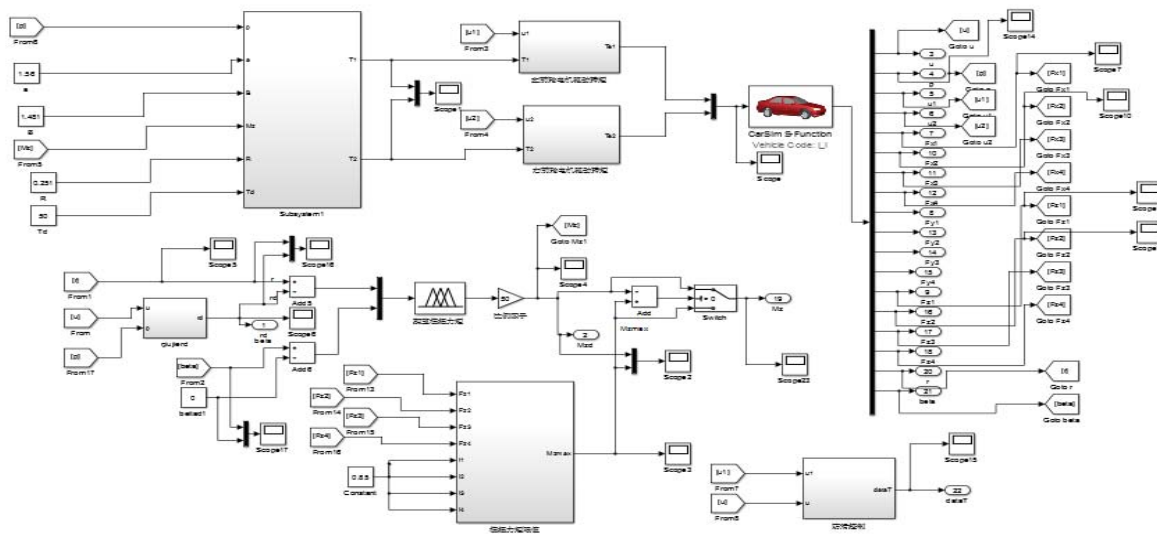


Figure 4.3 Simulink - Carsim joint simulation model

4.2 Simulation analysis

In the simulation of torque coordination, the given steering wheel angle is shown in Fig. 4.4. When the steering wheel angle is positive, the vehicle carries on the left turn, when the turning angle is negative, the vehicle carries on the right turn. At this time, the coefficient of road adhesion is 0.85, no slip and no anti-slip control.

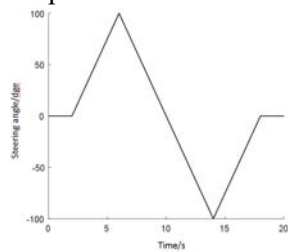


Figure 4.4 Steering angle

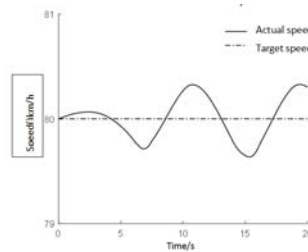


Figure 4.5 Vehicle speed

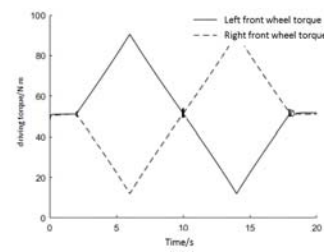


Figure 4.6 Motor output torque

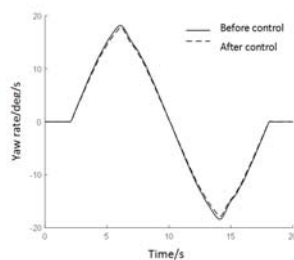


Figure 4.7 Yaw rate chart

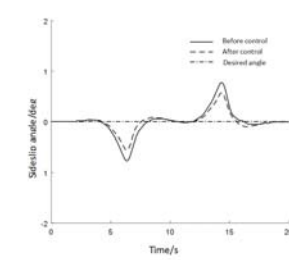


Figure 4.8 Side-slip angle

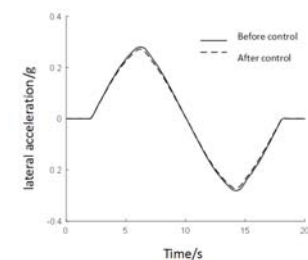


Figure 4.9 Lateral acceleration

The simulation results are shown in Figure 4.5-4.9. Fig. 4.5 is a comparison chart between actual vehicle speed and target speed. Fig. 4.6 is the output torque of the rear left and right drive wheel motor. The change of torque corresponds to the change of wheel speed of driving wheel. Fig. 4.7 is a graph of the yaw angular velocity varying with time, the simulation results are very close to the yaw angular velocity of the Carsim own model, and have good tracking effects. Fig. 4.8 is the curve of the side-slip angle of the center of mass with time, and the side-slip angle of the control is closer to 0. Fig. 4.9 is the curve of lateral acceleration versus time, and the vehicle is in accordance with the trend of the Carsim

model, which has less deviation and good tracking effect.

5. Conclusion

In this paper, the torque distribution control strategy was designed based on the fuzzy control, and the results of the simulation made in the Carsim-Simulink joint platform testified the driving coordination control strategy which can make the two-front-wheel independent driving vehicle drive more smoothly under steering process.

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