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Coordinated control of four-wheel drive electric vehicle EPS/4WD system

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Abstract. Each wheel torque of a four-wheel drive electric vehicle can be independently controlled, which play an important role in ensuring the vehicle good handling stability and energy conservation & environmental protection. A coordinated control strategy of EPS and 4WD based on vehicle driving safety margin is proposed. The vehicle dynamics model and tire model are established. The driving safety margin of the vehicle is determined according to the dynamic characteristics of the vehicle, the dynamic coordination control factors of EPS/4WD controller in the upper controller are determined by driving safety margin. The simulation results validate the EPS and 4WD coordinated control strategy based on driving safety margin can improve the vehicle's driving stability, thus the effectiveness of the control strategy is verified.

1. Introduction

Along with the people to the vehicle driving safety, the energy conservation environmental protection and so on performance request unceasing enhancement,there will be more and more four-wheel drive electric vehicles.Hub motor refers to the motor installed at the hub of the wheel,so that the motor and the wheels form a whole,drive the wheels directly with a motor,thus saves the traditionalfuel vehicles' complex mechanical transmission system. The motor output torque's control precision of four-wheel independent drive electric vehicle is high, and the response speedis fast.Because its four wheels are independent controllable,and vehicle dynamics is more complex than traditional vehicles in different driving conditions.Many scholars at home and abroad have studied its dynamic controlproblems.

Meng et al[1] built the physical model of hub motor driving electric vehicle,designed the control strategy of electric vehicle in starting, advancing, reversing, braking, neutral and stopping operating conditionsetc.Wang et al[2] studied the control strategy aiming at reducing steering wheel's hand force and assisting steering wheels return left and right front wheel's torque distribution.The vehicle yaw correction is realized by rear wheel differential,and the application ability of differential power steering technology is improved.Lu et al[3] proposed the efficiency model of multi-permanent magnet synchronous motor system at the same speed and torque.It is proved that the average distribution of torque at the same speed can optimize the efficiency of multi-permanent magnet synchronous motor system.Literature [4] identified the optimal slip rate of different pavements,designed a fuzzy-PID joint controller with the optimal slip rate as the control objective,ensure that the vehicle still can obtain the better driving anti-skid effect when driving in bad road conditions.With the goal that the front and rear axle wheels reached the adhesion limit at the same time, Li et al[5]got the ideal distribution relationship between the front and rear axle driving force and braking force.In addition, the handling stability was simulated and analyzedunder different allocation modes. Literature [6] combined the simulation and experiment, referred to a single axle drive electric vehicle prototype,the power system of electric vehicle with four-wheel hub motor was calculated,built the prototype and hub motor electric vehicle model with Crusie. Yang et al[7] studied the yaw stability control system of direct yaw moment control for four-wheel drive electric vehicles,the lateral stability can be effectively controlled,prevented the vehicle from lateral instability. In order to achieve the drive anti-skid function controlled by pure motor on the dual-motor four-wheel drive electric vehicles, a single neuron adaptive control algorithm based on radial basis function system identification on the different adhesion coefficients roads was proposed,adjusted the PID control parameters online adaptively,and the response speed and robustness of the control algorithm were improved[8]. Literature [9] designed a velocity estimation algorithm based on extended kalman filter, proposedthe direct yaw moment control strategy and the calculation method of the expected torque and the expected yaw moment, based on the wheel slip state and vehicle state parameters.The direct yaw moment control strategy can select the intervention wheel correctly and control the wheel torque in timebased on the deviation of yaw rate,so as to maintain the lateral stability of the vehicle.

Hayato et al[10]considered the influence factors such as lateral force and yaw moment of the vehicle,calculated the lateral deflection angle of the front/rear wheels and the driving forces of the left/right hub motors based on the least square method,which improved the driving range of electric vehicles and the stability of vehicles on curves. Etsuo[11]of Toyota proposed the decoupling control strategy of hub motor driving vehicle’s three-dimensional torque, considered the vertical reaction of suspension to motor during driving. Theoretical calculation was carried out and the driving force of each wheel was distributed and controlled.Kamachi etal[12] formulated the wheel system/driving force proportional distribution coefficient between the four wheelsseparately. Literature [13] estimated the distribution ratio between the roll control system and the yaw moment control system based on the rollangle index,the integration between RSC and YSC system was realized.Ono et al[14] allocated the target driving force of each wheel with nonlinear optimal allocation methodin order to minimize the adhesion utilization rate of tire-pavement.The stable running of the vehicle had more adhesion margin.

In order to overcome the above disadvantages, this paper adopts the method of combining modeling and simulation. Firstly, the nonlinear vehicle dynamics model is established. Then design the EPS and 4WD system coordinated controller, determine the safety margin of the vehicle, design the lower controller. Finally, the simulation analysis is carried out. Simulation results demonstrate the effectiveness of the control strategy.

2. Vehicle dynamics modeling

2.1. Four-wheel vehicle model

The four-wheel vehicle model is shown in Fig. 1.

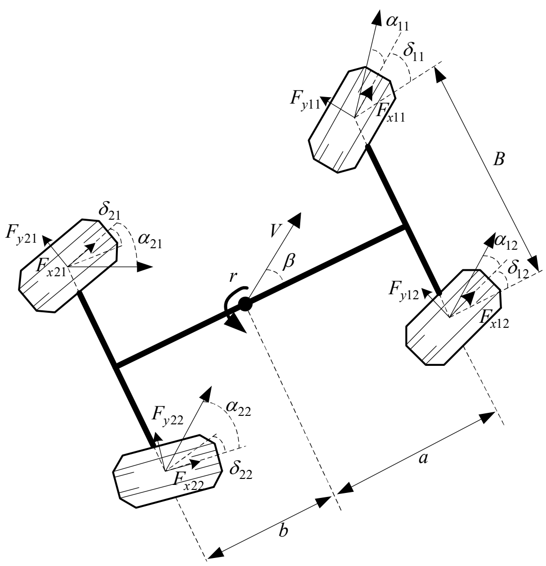


Figure 1. Four-wheel vehicle model

The following dynamics equations of four-wheel vehicle can be obtained

$$m(\dot{u}-vr)=F_{x11}\cos\delta_{11}+F_{x12}\cos\delta_{12}+(F_{x21}+F_{x22})-F_{y11}\sin\delta_{11}-F_{y12}\sin\delta_{12} \tag{1}$$

$$\begin{aligned}I_z\dot{r}=&a(F_{y11}\cos\delta_{11}+F_{y12}\cos\delta_{12}+F_{x11}\sin\delta_{11}+F_{x12}\sin\delta_{12})-b(F_{y21}+F_{y22})+\frac{B}{2}(F_{x11}\cos\delta_{11}-F_{y11}\sin\delta_{11}\\&-F_{x12}\cos\delta_{12}+F_{y12}\cos\delta_{12}+F_{x21}-F_{x22})\end{aligned} \tag{2}$$

$$m(\dot{v}+ur)=F_{x11}\sin\delta_{11}+F_{x12}\sin\delta_{12}+F_{y11}\cos\delta_{11}+F_{y12}\cos\delta_{12}+F_{y21}+F_{y22} \tag{3}$$

$$F_{z11,12}=mg\frac{b}{2(a+b)}-ma_x\frac{h}{a+b}\pm ma_y\frac{hb}{B(a+b)} \tag{4}$$

$$F_{z21,22}=mg\frac{a}{2(a+b)}+ma_x\frac{h}{a+b}\pm ma_y\frac{ha}{B(a+b)} \tag{5}$$

Where, m is the vehicle mass; u is the vehicle speed; r is the yaw rate; β is the sideslip angle; δ_{11},δ_{12} are the front steering wheel angles; I_z is the moment of inertia; a,b are the distance from the center of mass to the front and rear axes; B is the wheel base; a_x,a_y are the longitudinal acceleration and lateral acceleration of the vehicle respectively; h is the height of the center of mass; $F_{x11},F_{x12},F_{x21},F_{x22}$ are the longitudinal forces of four wheels ; $F_{y11},F_{y12},F_{y21},F_{y22}$ are the lateral forces of the four wheels; $F_{z11},F_{z12},F_{z21},F_{z22}$ are the vertical forces of the four wheels.

2.2. Tire model

The tire model adopts the Dugoff model, and the longitudinal force of the tire is expressed as

$$F_x = C_\sigma \frac{\sigma_x}{1+\sigma_x} f(\zeta) \tag{6}$$

The lateral force of the tire is

$$F_y = C_\alpha \frac{\tan(\alpha)}{1+\sigma_x} f(\zeta) \tag{7}$$

$$f(\zeta) = \begin{cases} (2-\zeta)\zeta & \zeta < 1 \\ 1 & \zeta \geq 1 \end{cases} \tag{8}$$

$f(\zeta)$ is the nonlinearity of tire slip angle and the vertical force

$$\zeta = \frac{\mu F_z (1+\sigma_x)}{2\sqrt{(C_\sigma \sigma_x)^2 + (C_\alpha \tan(\alpha))^2}} \tag{9}$$

Where, F_x , F_y , F_z are the longitudinal force, lateral force and vertical force acting on the wheels; C_σ is the longitudinal stiffness of the tire; σ_x is the longitudinal slip rate; C_α is the lateral stiffness of the tire; α is the slip angle of the tire; μ is the road adhesion coefficient.

3. Design of EPS and 4WD system coordinated controller

EPS and 4WD coordinated control system consists upper controller and lower controller.Analyze and calculate the safety margin of the vehicle by the output value calculated by the vehicle nonlinear dynamics system and road adhesion coefficient etc other information. The dynamic coordination control factors λ_1, λ_2 of each subsystem under the current working condition are obtained based on the driving condition of the vehicle and the functional potential of the subsystem,input EPS controller and 4WD controller respectively, and the two controllers output the motor assistant torque $T_m^{'}$ and the yaw moment $M_z^{'}$ respectively and act on the nonlinear dynamics system of the vehicle. So that the vehicle has good handling stability and driving safety.The schematic diagram of the vehicle control system is shown in Fig. 2.

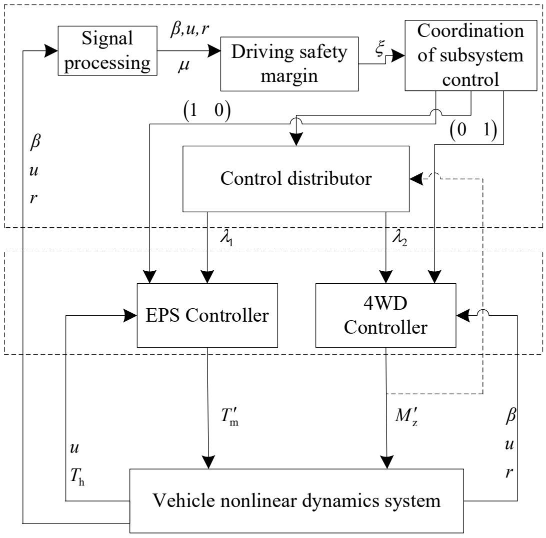


Figure 2. Schematic diagram of the vehicle control system

3.1. Determination of the vehicle driving safety margin

The driving safety margin of the vehicle is determined in this paper. Definition of S is shown in Fig. 3.

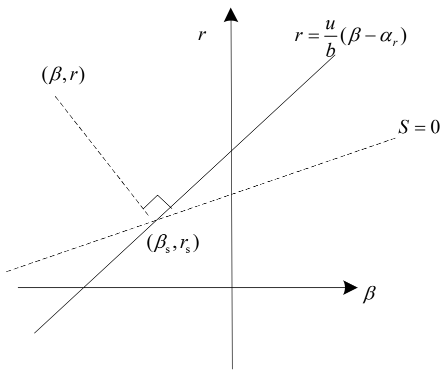


Figure 3. Schematic diagram of definition S

(β, r) represents the current state of the vehicle, (β_s, r_s) is on the safety boundary, the distance to (β, r) is the shortest. The calculation formula is as follows

$$\begin{cases} \beta_s = \dfrac{b^2\beta + bur + u^2 \tan\alpha_r}{u^2 + b^2} \\ r_s = \dfrac{bu\beta + u^2 r - bu\tan\alpha_r}{u^2 + b^2} \end{cases} \tag{10}$$

The vectordistance between (β, r) and (β_s, r_s) ,that is, the vectordistance between the vehicle state and the safety boundary is defined as the vehicle driving safety margin

$$\xi = \frac{1}{1+b_0^2}\sqrt{(\beta - b_0 r - b_1)^2 + (b_0^2 r - b_0\beta + b_0 b_1)^2} \tag{11}$$

A dynamic coordinated control factor λ_1, λ_2 based on the driving safety margin is introduced,guide each controller to change the control signals to control the current working state by the dynamic coordinated control factor.When the current state point of vehicle movement is within the safe boundary, The shorter the distance (β, r) to the safe boundary, λ_1 is the smaller, λ_2 is larger.When the safeboundary is exceeded, λ_2 is 1.When the motor starts to provide power, the steering wheel assistance threshold is T_0 .

The divided effective area of each subsystem for good vehicle handling and lateral stability and the specific value principle of λ_1, λ_2 is as follows

1) When $\xi > 0, T_{\mathrm{h}} > T_0$ and $\beta < \beta_{\mathrm{d}}$,the vehicle can turn steadily at high speed, EPS control is required at the time, so that the vehicle has better handling stability in the process of steering, and $\lambda_1 = 1, \lambda_2 = 0$;

2)When $\xi = 0, T_{\mathrm{h}} > T_0$ and $\beta \ge \beta_{\mathrm{d}}$, the vehicle will lose its stability when turning at high speed.Therefore, EPS and 4WD should be used to control the vehicle at the same time. To maintain good handling stability and lateral stability.

3) When $\xi < 0$, the vehicle loses its ability to turn, thereforeit is necessary to restore the vehicle to a stable state through 4WD,and $\lambda_1 = 0, \lambda_2 = 1$.

$$\begin{pmatrix} T'_{\mathrm{m}} \\ M'_{\mathrm{z}} \end{pmatrix} = \begin{pmatrix} \lambda_1 & 0 \\ 0 & \lambda_2 \end{pmatrix} \begin{pmatrix} T_{\mathrm{m}} \\ M_{\mathrm{z}} \end{pmatrix} \tag{12}$$

Where, $T_{\mathrm{m}}, M_{\mathrm{z}}$ are the outputs of each sub-controller under separate control, $T'_{\mathrm{m}}, M'_{\mathrm{z}}$ are the outputs with coordinated control. Table 1 shows the vehicle EPS/4WD system coordination control rules.

Table 1. Vehicle EPS/4WD system coordination control rules

ξ	T_{h}	β	Control mode
$\xi > 0$	$T_{\mathrm{h}} > T_0$	$\beta < \beta_{\mathrm{d}}$	EPS control
$\xi = 0$	$T_{\mathrm{h}} > T_0$	$\beta \ge \beta_{\mathrm{d}}$	coordinated control
$\xi < 0$	–	–	4WD control

3.2. Design of the lower controller

3.2.1. EPS controller design. Firstly, the dynamics equation of steering column is established.

$$J_s \ddot{\theta}_s + C_s \dot{\theta}_s + k_s \left(\theta_s - \frac{x_r}{r_p} \right) = T_s \tag{13}$$

Where, J_s is the moment of inertia on the steering column, θ_s is the steering axis rotation angle, C_s is the damping coefficient of the steering axis, k_s is the torsional stiffness of the steering axis, T_s is the torque exerted by the driver on the steering wheel.

For the rack

$$m_r\ddot{x}_r + C_r\dot{x}_r + k_t x_r = \frac{k_s}{r_p}\left(\theta_s - \frac{x_r}{r_p}\right) \tag{14}$$

Where, m_r is the mass of the rack, x_r is the displacement of the rack, C_r is the damping coefficient of the rack, k_t is the stiffness coefficient of the rack, and r_p is the radius of steering pinion.

For assistant motor

$$J_m\ddot{\theta}_m + C_m\dot{\theta}_m + k_m\left(\theta_m - i_g\frac{x_r}{r_p}\right) = T_m \tag{15}$$

Where, J_m is the rotary inertia of the motor, θ_m is the rotation angle of the motor shaft, C_m is the damping coefficient of the motor, k_m is the stiffness of the motor shaft, and T_m is the electromagnetic torque of the motor.

3.2.2. 4WD controller design. The 4WD control distributor optimizes the yaw moment calculated by the upper controller to four drive motors. At the same time, the tire adhesion utilization rate, actuator and other constraints were considered.

The constraint of the actuator is

$$\begin{cases} \dfrac{T_{b11\_\mathrm{m}}}{r} \le F_{11} \le \dfrac{T_{d11\_\mathrm{m}}}{r} \\ \dfrac{T_{b12\_\mathrm{m}}}{r} \le F_{12} \le \dfrac{T_{d12\_\mathrm{m}}}{r} \\ \dfrac{T_{b21\_\mathrm{m}}}{r} \le F_{21} \le \dfrac{T_{d21\_\mathrm{m}}}{r} \\ \dfrac{T_{b22\_\mathrm{m}}}{r} \le F_{22} \le \dfrac{T_{d22\_\mathrm{m}}}{r} \end{cases} \tag{16}$$

T_{b11_m}, T_{b12_m}, T_{b21_m}, T_{b22_m} are the maximum braking torque of the four wheelsrespectively; T_{d11_m}, T_{d12_m}, T_{d21_m}, T_{d22_m} are the maximum driving torque of the four wheelsrespectively; r is the radius of the wheel.

The yaw moment generated by the four-wheel drive is

$$M_z = \left(F_{x12} + F_{x22} - F_{x11} - F_{x21}\right)\frac{B}{2} \tag{17}$$

The optimization objective is to reduce the tire utilization rate of the vehicle

$$\min \quad J = \sum_i \frac{F_{xi}^2 + F_{yi}^2}{\left(\mu_i F_{zi}\right)^2} \tag{18}$$

4. Simulation analysis

According to the research needs,establish a sevenDOF nonlinear vehicle dynamics model regardless of the vertical motion of the vehicle,including three DOF of the body,longitudinal, lateral movement and vertical yaw motion,and four DOF for wheel rotation. When the current front wheel rotation angle is 5^o, the relationship between each index with road coefficients and vehicle velocity is shown in table 2.

Table 2. The front wheel angle =5o

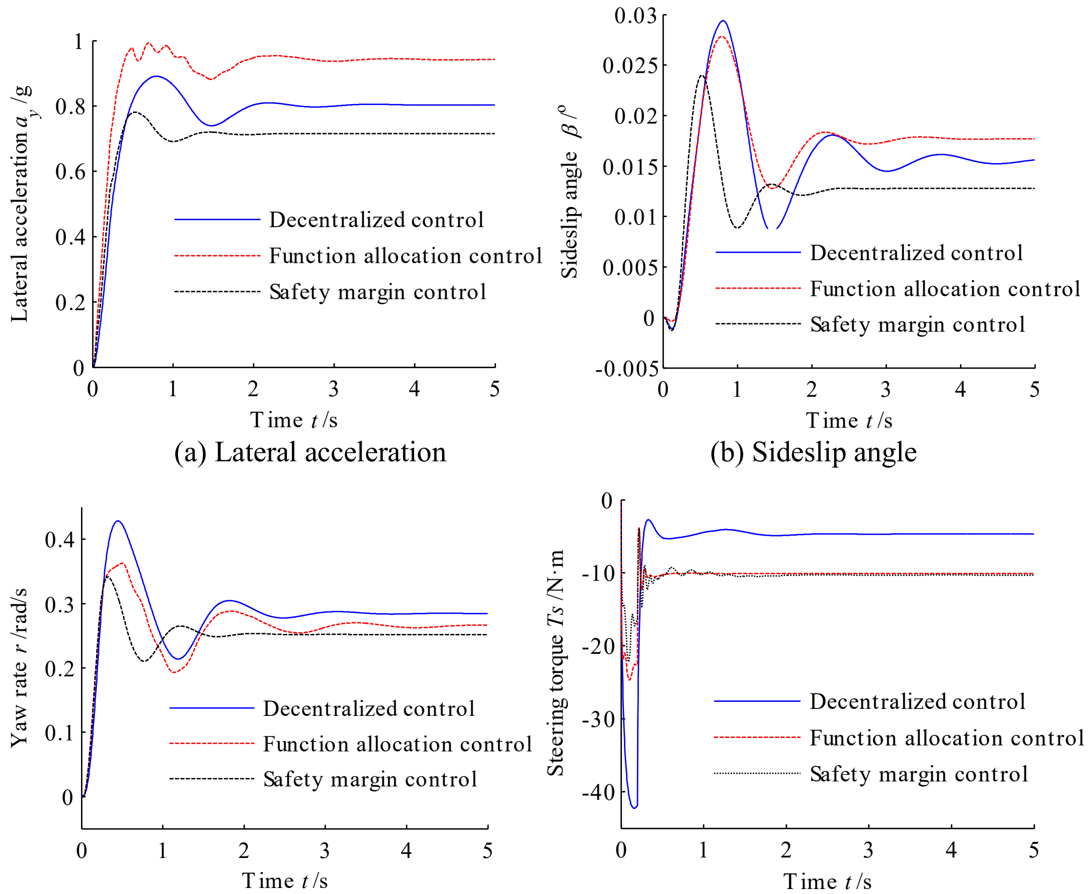
μ	u	$\lvert\beta - b_0 r\rvert \le b_1$		$\lvert r - b_3\beta\rvert \le b_4$	
		b_0	b_1	b_3	b_4
$0.8 \le \mu \le 1$	$5 \le u \le 15$	0.115	0.1863	-0.3181	0.8737
$0.6 \le \mu < 0.8$	$15 < u \le 25$	0.0575	0.1449	2.9207	0.7144
$0.4 \le \mu < 0.6$	$25 < u \le 35$	0.0383	0.1035	5.9258	0.7421
$0.2 \le \mu < 0.4$	$35 < u \le 45$	0.0287	0.0621	10.5442	0.7071
$\mu < 0.2$	$u > 45$	0.0256	0.0207	20.3604	0.4321

EPS/4WD decentralized control refers to that both EPS and 4WD work independently at the same time. In order to verify the effectiveness of the proposed coordinated control strategy, this study compares with the results of EPS/4WD decentralized control and function allocation coordinated control under the same working conditions. The main parameters of the vehicle are shown in table 3.

Table 3. Main parameters of the vehicle

Parameter	Numerical value
Mass m/kg	1280
The vehicle's moment of inertia I_z/(kg · m^2)	1850
Distance from the mass center to the front axis a/m	1.12
Distance from the mass center to the rear axis b/m	1.43
Wheel base d/m	1.52
Height of the mass center h/(m)	0.5
Wheel radius r/(m)	0.3
Wheel's moment of inertia I_w/(kg · m^2)	0.6
Motor rated power $P_N/(kW)$	20

The simulation condition is as follows, the vehicle velocity is set as 80km/h in the step condition, when the time is 1second the steering wheel rotates 90^o.Matlab software is used for programming, and calls the established coordinated control Simulink model for simulation. The simulation results are shown in Fig.4.



(a) Lateral acceleration (b) Sideslip angle

(c) Yaw rate response (d) Steering wheel torque

Figure 4. Time-domain response diagram under step simulation condition

In conclusion, compared with decentralized control and function allocation coordinated control, EPS/4WD coordinated control method based on the vehicle safety margin can improve the maneuverability and stability of the vehicle.

5. Conclusion

According to the internal relations between kinematics variables and vehicle parameters, the driving safety margin of the vehicle is determined by the driving state, vehicle parameters and road adhesion coefficient, etc. Since the hub motor drives the electric vehicle each wheel torque can be directly controlled, An EPS/4WD coordinated control strategy based on vehicle safety margin is proposed. Through the comparison and analysis of simulation under different working conditions, EPS/4WD coordinated control strategy based on vehicle driving safety margin gives full play to the advantages of hub motor driving electric vehicles independently, can guarantee the driving stability of the vehicle effectively.

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