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Integrated DYC and ASR Mixed Control of Distributed Drive Electric Vehicle Using Co-Simulation Methodology

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ABSTRACT

The improvement of handling and stability performance of distributed drive electric vehicle (DDEV) is analyzed, visualized and designed by proposing and deploying the mixed control strategies in this paper including Direct Yaw Control (DYC), Anti-slip Regulation (ASR) and dual-mode switching control. The practicability and real time visualization of driving efficiency and timeliness of DDEV is achieved to reduce the margin of error for the desired torque value by employing the DYC strategy which uses fuzzy PID algorithm. Furthermore the ASR strategy which adopts the optimal slip rate algorithm to determine the requirement of desired torque value based on the different road conditions is used to reduce slip phenomenon effectively and to maintain handling control of DDEV. In response to different scenes especially conflict and coexistence between DYC and ASR, the dual-mode switching control strategy is applied to find more suitable slip rate range by using the root mean square error method (REME). Finally, co-simulation platform of ADAMS/Car and MATLAB/Simulink is built to simulate the mixed control strategies by integrating dual-mode switching control, DYC and ASR. The simulation results show that this strategy has a more significant control effect needed to meet the requirements of normal vehicle handling and stability. The mixed control strategy is adopted and downloaded into the electronic control unit of our student type formula vehicle called Flash V6 which was designed and developed by a team of students, the ZUST ATTACKER Team.

Keywords: Distributed drive electric vehicle(DDEV), mixed control, co-simulation, Direct Yaw Control (DYC), Anti-slip Regulation(ASR).

Introduction

Distributed drive electric vehicle (DDEV) is an important development and design direction of future electric vehicles. Their main features are that, the drive motor is embedded in the wheel, the transmission chain is shorter, speed response is faster and the torque control accuracy is high¹. The wheels equipped with the drive motors can not only be independently controlled but also the wheel torque can be arbitrarily distributed to achieve optimum and better control effect. Based on these characteristics, a torque optimization control algorithm was designed² and combined with the quadratic programming method to design torque average distribution algorithm³. Zhu et al.⁴ proposed a particle swarm optimization and nonlinear predictive control algorithm through torque optimization and distribution to improve the vehicle stability while Liu et al.⁵ used the expected additional torque obtained by a linear quadratic regulator (LQR).

The more common schemes in torque control include uses the yaw rate as the feedback object which is obtained by the optimal allocation strategy⁶ and combined with the slip rate normally used to construct ABS which prevent vehicle instability⁷. The yaw rate and the side slip angle are not only used as parameters to observe the vehicle response⁸⁻⁹ but also combined with sliding mode controller. These strategies based on multiple sliding mode controls¹⁰⁻¹² were used to design a torque distribution system needed for torque redistribution while ensuring the stability of vehicle braking.

Meanwhile, the slip rate can also be used as a reference value to study the ASR strategy. Based on the independent driving electric vehicles models, the conventional method of slip rate calculations were used to improve the vehicle longitudinal driving stability¹³. Through fuzzy PID controller, the motor torque is adjusted to obtain the best slip rate control¹⁴⁻¹⁶ which shortens the driving acceleration distance and improves the steering stability¹⁷. Based on fuzzy PID control theory¹⁸, the integrated control strategy of ASR and DYC with a feed forward-feedback method were designed as the active front wheel steering controller (AFS) and DYC which are used as new controllers to obtain a target drive torque that could effectively improve vehicle stability¹⁹. A dynamic DYC combined with feed forward and feedback control system was proposed to improve the stability of the vehicle²⁰. The path tracking and MPC controller were introduced in DYC to achieve the purpose of correcting the driving state of the vehicle²¹. Multi-mode control strategies were designed to improve the longitudinal stability of the vehicle²². A multi-mode switching control strategy was designed based on the longitudinal acceleration

characteristics of the vehicle. The effectiveness and practicability of the control strategies were finally verified using a real vehicle.

Based on the above excellent research works, a mixed control strategy can be integrated into the dual-mode switching control of DYC algorithm with fuzzy PID controller and ASR algorithm with optimal slip rate PID controller. The strategy can be established based on the DDEV under normal motion conditions.

A co-simulation platform can be set up to simulate the vehicle model with single lane change and step steer tests using ADAMS®/Car and MATLAB®/Simulink.

The results from the mixed control strategy integration consisting of DYC and ASR can effectively prevent DDEV from entering the dangerous limit driving state when turning and driving. The outcome can greatly improves the driving safety of the vehicle.

Vehicle modeling

The electric vehicle model with distributed driven wheels was constructed using ADAMS/Car software while the motor model and control strategy with the MATLAB/Simulink software. Setting the relevant parameters at the communication interface between ADAMS/Car and MATLAB/Simulink, the real-time data interaction channel of ADAMS/Car and MATLAB/Simulink was established, and the two parts of the model were jointly simulated. The overall architecture is illustrated as shown in in Fig.1.

Figure 1. Schematic diagram of vehicle and control models architecture.

Establishment of ADAMS/Car model

The vehicle model established in ADAMS/Car software according to the parameters in Table 1. And the vehicle model built by ADMAS/Car is shown in Fig. 2.

Symbol	Parameters	Value and units
m	Vehicle mass	1500 kg
a	Distance from centroid to front axle	1.185 m
b	Distance from centroid to rear axle	1.283 m
T_f	Front tread	1.52 m
T_r	Rear tread	1.59 m
h	Height of center of gravity	0.28 m
I_z	Torque about Z-axis	2280 kg·m ²
K_1	Front calpha	-60533 N/rad
K_2	Rear calpha	-70052 N/rad

Table 1. Related parameters of vehicle model

Figure 2. ADAMS/Car vehicle model. 1-Front and rear tire system 2-Rear suspension system 3-Body subsystem 4-Steering subsystem 5-Brake subsystem 6-Front suspension system

Establishment of two degree of freedom (2-DOF) model

In order to design the control model, the 2-DOF linear vehicle model⁹ is used as the reference model:

$$\begin{cases} K_1 \left(\beta_d + \frac{a\gamma_d}{v} - \delta_f \right) + K_2 \left(\beta_d - \frac{b\gamma_d}{v} \right) = m(\dot{v}_y + u\gamma_d) \\ aK_1 \left(\beta_d + \frac{a\gamma_d}{v} - \delta_f \right) + bK_2 \left(\beta_d - \frac{b\gamma_d}{v} \right) = I_z \dot{\gamma}_d \end{cases} \quad (1)$$

According to the transformation formula:

$$\begin{cases} \dot{\gamma}_d = \frac{a^2 K_1 + b^2 K_2}{I_z v} \gamma_d + \frac{aK_1 - bK_2}{I_z} - \frac{aK_1}{I_z} \delta_f \\ \dot{\beta}_d = \left(\frac{aK_1 - bK_2}{mv^2} - 1 \right) \gamma_d + \frac{K_1 + K_2}{mv} \gamma_d - \frac{K_1}{mv} \delta_f \end{cases} \quad (2)$$

Where v is the longitudinal speed; v_y is the velocity of the center of mass; δ_f is the front wheel angle.

According to formula (2), δ_f with v are used as input variables; the desired yaw rate γ_d and side slip angle β_d are obtained by establishing 2-DOF linear vehicle model in Simulink.

Modeling of simulink motor

Emrax 188 motor is used in the DDEV, because of the reliability of permanent magnet synchronous motors. Their mechanical characteristics include simple structure, small volume, less cumbersome, rapid response, high power density and efficiency. According to the relevant parameters in Table 2, a simplified motor model is established as the torque demand command actuator in the vehicle control strategy. By adjusting the current in the motor model, the motor can output smooth torque to the driving wheel.

Symbol	Parameters	Value and units
P_t	Peak power	65 kw
P_c	Continuous power	35 kw
T_t	Peak torque	90 N·m
T_c	Continuous torque	50 N·m
n_e	Rated speed	7000 rpm
n_t	Max. speed	8500 rpm

Table 2. Related parameters of EMRAX 188 motor

In order to make the motor torque command response as close to the real motor characteristics as possible, the motor driver is regarded as the first-order inertial link when building the simplified model of the motor as shown in the formula:

$$T_w = \frac{1}{1 + \tau s} \times T_m \quad (3)$$

Where T_w is the actual torque of the motor, τ is the time constant obtained from the motor characteristic parameters, T_m is the external characteristic torque of the motor.

In order to describe the characteristics of the motor, the transfer function equation of the motor is established and combined with the motor parameters. The model diagram is shown in Fig. 3.

Figure 3. Motor model diagram.

Where T_{dl} is target torque of left wheel, T_{dr} represents right wheel target torque, e_T is the torque difference, U_x is the motor terminal voltage value, e_U is the voltage difference, I is the armature current, K_e is the electromagnetic torque constant, K_w is the induced electromotive force constant, U_w is the induced electromotive force value, T_{ml} is the left wheel motor output torque, T_{mr} is the right wheel motor output torque, w_{rr} actual angular velocity of right wheel, w_{rl} actual angular velocity of left wheel, T_{rl} is the actual torque of left wheel, T_{rr} is the actual torque of right wheel.

Vehicle model

As shown in Fig. 4, the actual yaw rate β and side slip angle γ parameters were obtained from ADAMS/Car model, β_d and γ_d were obtained from 2-DOF linear model, which is transferred to the fuzzy PID controller to obtain the correction torque ΔM_1 . On the other hand, the actual linear speed of left and right wheels v_{rl} , v_{rr} and w_{rl} , w_{rr} are obtained by ADAMS/Car model, which are transmitted to the slip-rate PID control module to obtain the correction torque ΔM_2 . The DYC or ASR torque value is obtained by the torque distribution module and transmitted to the mode switching module. T_{dl} and T_{dr} are obtained by ADAMS/Car model while T_{rl} and T_{rr} are input into the motor model established in the previous section. The outputs T_{ml} and T_{mr} act on the driving wheel to realize the control of vehicle handling and stability.

Figure 4. Vehicle control structure diagram.

Where K is the pedal opening valve, θ_c is the steering wheel angle, and μ is the adhesion coefficient.

Control strategy design

In this section, the DYC and ASR are explained, and the effectiveness of the control strategies is verified by simulation.

DYC module

The DYC module is mainly used to calculate the additional yaw torque produced by the vehicle around the center of mass when turning, adjust the yaw motion state of the vehicle, and then participate in the driving torque distribution module of the vehicle.

Based on the 2-DOF linear vehicle model, the ideal state of the vehicle is obtained as γ_d and β_d . By tracking these two variables which reflect the actual state of the vehicle and how it can quickly become stable thus the actual value can

continuously tend to the expected value. By making a difference, we can get $e_1 = \gamma - \gamma_d$ and $e_2 = \beta - \beta_d$ input fuzzy PID controller to get ΔM_1 as shown in Fig.5.

Figure 5. Structure diagram of DYC.

Defined variable:

$$S = \xi_1 \times (\gamma - \gamma_d) + \xi_2 \times (\beta - \beta_d) \quad (4)$$

Where ξ_1 and ξ_2 are the weight coefficients, which are obtained by the established fuzzy rule controller in Table 3.

$\Delta k_p / \Delta k_i / \Delta k_d$		e_c						
		NB	NM	NS	ZO	PS	PM	PB
e	NB	PB/NB/PS	PB/NB/NS	PM/NM/NB	PM/NM/NB	PS/NS/NB	ZO/ZO/NM	ZO/ZO/PS
	NM	PB/NB/PS	PB/NB/NS	PM/NM/NB	PS/NS/NM	PS/NS/NM	ZO/ZO/NS	NS/ZO/ZO
	NS	PM/NB/ZO	PM/NM/NS	PM/NS/NM	PS/NS/NM	ZO/ZO/NS	NS/PS/NS	NS/PS/ZO
	ZO	PM/NM/ZO	PM/NM/NS	PS/NS/NS	ZO/ZO/NS	NS/PS/NS	NM/PM/NS	NM/PB/ZO
	PS	PS/NM/ZO	PS/NS/ZO	ZO/ZO/ZO	NS/PS/ZO	NS/PS/ZO	NM/PM/ZO	NM/PB/ZO
	PM	PS/ZO/PB	ZO/ZO/NS	NS/PS/PS	NM/PS/PS	NS/PS/PS	NM/PB/PS	NM/PB/PB
	PB	ZO/ZO/PB	ZO/ZO/PM	NM/PS/PM	NM/PM/PM	NM/PM/PM	NB/PB/PS	NB/PB/PB

Table 3. Fuzzy controller rules

According to formula (4) and Table 3, the ΔM_1 is obtained:

$$\Delta M_1 = k_1 \times \text{fuzzy}(S) \quad (5)$$

Where k_1 is the fuzzy function known as Mamdani fuzzy rule controller.

Neglecting the road interference in the process of vehicle driving, the ΔM_1 obtained needs to be converted to the torque value acting on the driving wheel and the following formula is used :

$$\Delta M_1 = \Delta F \times T_r = (\Delta F_1 - \Delta F_r) \times T_r / 2 \quad (6)$$

Where ΔF is the corrected driving force.

It is noted that, the relationship between driving torque and driving force of wheel is given as follows:

$$\Delta T = \Delta F \times r = \frac{|\Delta M_1| \times r}{T_r} \quad (7)$$

Where r is the wheel rolling radius, and ΔT is the torque the motor need to adjust when maintaining the body posture under the current driving conditions.

The redistributed torque values are obtained by combining equations (6) and (7):

$$\begin{cases} T_{dl} = T_d, & T_{dr} = T_d & \Delta M_1 = 0 \\ T_{dl} = T_d - \Delta T / 2, & T_{dr} = T_d + \Delta T / 2 & \Delta M_1 > 0 \\ T_{dl} = T_d + \Delta T / 2, & T_{dr} = T_d - \Delta T / 2 & \Delta M_1 < 0 \end{cases} \quad (8)$$

Where T_d is the product of the switch opening and the theoretical maximum output torque of the motor.

Based on the DYC, the joint simulation model of ADAMS/Car and MATLAB/Simulink is established as shown in Fig.6.

Figure 8. Curve of slip rate and adhesion coefficient.

The actual slip rate λ , which reflects the actual state of the vehicle is obtained by ADAMS/Car model. w_{rl} 、 w_{rr} and v_{rl} 、 v_{rr} according to the slip rate formula relate and expressed as:

$$\begin{cases} \lambda_{rl} = \frac{w_{rl} \times r - v_{rl}}{w_{rr} \times r} \times 100\% \\ \lambda_{rr} = \frac{w_{rr} \times r - v_{rr}}{w_{rr} \times r} \times 100\% \end{cases} \quad (11)$$

Where λ_{rl} 、 λ_{rr} are the actual slip rate of the left and right rear wheels respectively.

The difference $e_3 = \lambda - \lambda_{opt}$ is transmitted to the slip rate PID controller output ΔM_2 as shown in Fig.9.

Figure 9. Structure diagram of ASR.

From this we can see that,

$$\Delta M_2 = k_2 \times fpid(e_3) \quad (12)$$

Where k_2 is the experience value, $fpid(e_3)$ is the PID controller.

Considering the longitudinal and lateral acceleration of vehicle during acceleration and steering, the vertical load of left and right wheels are obtained as follows:

$$\begin{cases} F_{zrl} = \frac{mga}{2(a+b)} + \frac{mha_x}{2(a+b)} - \frac{mha_y}{2T_r} \\ F_{zrr} = \frac{mga}{2(a+b)} + \frac{mha_x}{2(a+b)} + \frac{mha_y}{2T_r} \end{cases} \quad (13)$$

Where F_{zrl} is the vertical load of the left rear wheel; F_{zrr} is the vertical load of the right rear wheel; g is the acceleration of gravity; a_x is the longitudinal acceleration; a_y is the lateral acceleration.

When the vehicle is steering, the load transfer will occur and the limit adhesion of the inner wheel will decrease while that of the outer wheel will increase. Therefore, in order to prevent the inner wheel from slipping, the driving torque of the inner wheel should be limited.

$$\frac{T}{r} + F_y \leq F_{zx} \times \mu \quad (14)$$

If the above formula is satisfied, then:

$$\begin{cases} T_{dl} = T_d, & T_{dr} = T_d & \Delta M_2 = 0 \\ T_{dl} = T_d - \Delta T/2, & T_{dr} = T_d + \Delta T/2 & \Delta M_2 > 0 \\ T_{dl} = T_d + \Delta T/2, & T_{dr} = T_d - \Delta T/2 & \Delta M_2 < 0 \end{cases} \quad (15)$$

If not, then:

$$T = (F_{zx} \times \mu - F_y) \times r \quad (16)$$

The co-simulation model of ADAMS/Car and MATLAB/Simulink is built based on the ASR as shown in Fig.10.

Test schemes	Parameters	Value and units
Step steer	v	60,80,100km/h
	θ_c	0.5π rad
	μ	0.8
Single lane change	v	80,90,100km/h
	θ_c	0.5π rad
	μ	0.8
Constant radius cornering	R	20,30,40m
	a_y	$0.1 \text{ m/s}^2/\text{s}$

Table 4. Simulation test parameters of DYC

Two curves are obtained by simulation taking $REME(\gamma)$ as ordinate, w_r as the sum of curves in abscissa and λ as ordinate while w_r is the abscissa curve.

In Fig.13 the curve under the step steer test condition is analyzed when λ is above 0.15, $REME(\gamma)$ is small and the DYC effect at this stage is better. The curve analysis under single lane changes the test condition when λ is above 0.15 and the corresponding $REME(\gamma)$ curve is smaller which shows that the DYC effect is significant in this stage. Under the constant radius cornering test condition, the $REME(\gamma)$ curve is obviously small with w_r and when λ is between 0.15 and 0.4, the $REME(\gamma)$ curve suddenly changes from a larger value to a smaller value. The analysis shows that the DYC has more advantages when λ is between 0.15 and 0.4.

Figure 13. Simulation curve of DYC.

The simulation parameters in Table 5 are set for ASR.

Test schemes	Parameters	Value and units
Step steer	v	30,40,50km/h
	θ_c	0.5π rad
	μ	0.3
Single lane change	v	40,50,60km/h
	θ_c	0.5π rad
	μ	0.3

Constant radius cornering	R	20,30,40m
	a_y	0.2 m/s ² /s

Table 5. Simulation test parameters of ASR

Two curves are obtained by simulation and by taking $REME(\gamma)$ as ordinate, w_r as the sum of curves in abscissa, λ as ordinate and w_r as the abscissa curve.

Figure 14. Simulation curve of ASR.

In Fig.14, it can be seen from the curve under the step steer test condition that λ increases with w_r when λ is higher than 0.4. The control effect is better when λ is above 0.4. Analyzing the curve of single lane change test, it can be concluded that the corresponding $REME(\gamma)$ curve is smaller than other stages when λ is above 0.4 which indicates that the ASR effect is more significant in this stage. Under the constant radius cornering test condition, the corresponding $REME(\gamma)$ curve is obviously smaller than other stages with w_r . When λ is above 0.4 the $REME(\gamma)$ curve changes greatly at this stage. It can be seen that the effect of ASR has been shown. In conclusion, when λ is between 0.4 and 1 the $REME(\gamma)$ curve tends to decline or is in a low range which indicates that the ASR effect is superior.

Combining with the simulation curve analysis in Fig.13 and Fig.14 .When λ is between [0.15 0.4], the effect of DYC strategy is better. When λ is between [0.4 1], the effect of ASR strategy is better. Therefore, the threshold values of mode switching are 0, 0.15, 0.4 and 1.

Mode switching judgement

In order to prevent the frequent change of λ which is caused by complex road surface that eventually lead to misjudgment of vehicle ASR and DYC system. λ is used as the evaluation reference when $0.15 \leq \lambda \leq 0.4$, DYC is switched to apply torque to reduce λ . When $0.4 \leq \lambda \leq 1$, ASR is switched to control λ_{opt} . Once the corresponding control mode is switched in the whole process, it will keep running until it is judged that λ is less than 0.15.

Mode switching means that after switching the control system, the torque command of the drive motor is output by the corresponding control module. When the control mode is exited, the torque command of the drive motor is not directly output through the control module. The switching process is designed as $y = 0, y = 1$ and $y = 2$, which is shown in Fig.15.

Figure 15. Flow chart of switching mode.

Mode switching simulation test

The purpose of simulation test is to verify the effectiveness of the two control modes. The single lane change test simulation means that the driving vehicle completes the lane changing action during the driving process which is of great significance to the vehicle stability. According to the relevant standard¹⁰, v is designed to be 50km/h, μ is 0.4, and θ_c within 1s was set to $\pi/2$ rad to the left and returns to normal at 7s for 8s run. The curve is shown in Fig.16.

Figure 16. Simulation test of single lane change condition.(a)yaw rate,(b)side slip angle,and (c)longitudinal acceleration.

Through the analysis of the yaw rate curve in Fig.16, it can be seen that the vehicle can perform lane change action well at the beginning of simulation. With the change of vehicle steering driving time, the uncontrolled vehicle β in Fig.16 has a large offset and the controlled vehicle β is reduced by 0.04rad while the longitudinal acceleration a_x in Fig.16 is reduced to 2/3 of the previous value and the values of β and γ have been greatly improved. The control model established has a significant improvement on the driving state of vehicles.

The transient response characteristics of the vehicle can be obtained through the step steer test simulation. In the process of simulation, the vehicle will change θ_c in a certain period of time with an input in the form of step from the initial value to the final value. According to the relevant standard¹⁰, v is set as 50km/h, μ is 0.7, and θ_c at 1s to the left is 0.5π rad and the step action lasted for 3s. The simulation result curve is shown in Fig.17.

Figure 17. Simulation test of step steer condition.(a)yaw rate,(b)side slip angle,and (c)longitudinal acceleration.

According to the yaw rate response curve in Fig.17, it can be seen that when the vehicle is in 1s due to θ_c input, the vehicle may be turning or overtaking and its γ changes greatly. After about 1s adjustment, the controlled γ quickly reaches stability and the steady state value is 0.35rad/s making the control effect very good. After the steering action of the uncontrolled vehicle is finished in a second, the γ curve is still in the growth trend and the whole vehicle has the tendency to side slip which is not conducive for stable driving of the vehicle. It can be seen from the curve in Fig.17 that the uncontrolled and controlled changes have taken place near 1.5s and the controlled β curve change is controlled within 0.005rad making the stability of the vehicle better. The change of β curve without control to -0.025rad is 5s of the maximum value of β with control. The a_x curve also shows that the controlled vehicle value is 1/2 of the uncontrolled value, indicating that the vehicle has better handling stability after control.

Driver in the loop simulation test

In order to further verify the real effect of the DYC strategy, it is necessary to observe the control effect via real vehicle test. However, due to the difficulty of real vehicle verification, the existing driver simulator Logitech G27 equipment is used to verify the control strategy to continuously optimize the strategy.

In Fig. 18, the serial communication port between G27 and MATLAB/Simulink is established first and the joystick output module in MATLAB/Simulink is used to adjust the θ_c of Logitech G27 Pedal (switch) opening K and other signals are collected. For the vehicle model in this paper, we mainly collected the θ_c of G27 and K signals. In particular, due to the θ_c of Logitech G27, the total travel range is 5π rad in the vehicle's mathematical model in MATLAB/Simulink. The travel range of θ_c is 6π rad making it is necessary to collect the θ_c of joystick output signal when processed by gain to satisfy the θ_c of vehicle mathematical model.

Figure 18. Drivers in the loop simulation.

Based on G27, the θ_c is designed to test conditions of turning π rad to the left and then turning π rad to the right as shown in the Fig.19.

Figure 19. G27 simulation test conditions.(a)steering wheel angle,(b)velocity,(c)torque of right rear wheel,and (d)torque of left rear wheel.

It can be seen from the curves of torque variation of the rear wheels in Fig.19 that the torque values on the uncontrolled wheels rapidly decrease and then increase at about 4s, 8s and 11s. The change in trend of θ_c curve shows that when the vehicle is steering, the driving torque of the wheel fluctuates violently. After controlling the torque, fluctuations on the rear wheels decrease and are made relatively gentle which indicates that the DYC strategy and the design redistributes the torque to the wheels to improve the handling stability of the vehicle under the premise of meeting the normal driving demand of the vehicle.

Conclusion

The control effects of DYC and ASR on handling and stability performance are presented based on the DDEV model in this study. Firstly, the subsystem model of each part of the vehicle is established using the ADAMS/Car software while the motor model is established based on MATLAB/Simulink software. The DYC model and the ASR model are designed, built and used in the established co-simulation platform.

The analysis from the simulation is carried out by referencing the two models which led to the design of dual-mode switching control strategy. The results of simulations from the curves guaranteed a satisfactory control effect. Finally the results from the driver in the loop simulation of the DYC strategy, real vehicle test simulation and the curve analysis show that the DYC strategy has superior control effect on vehicle handling stability which provides a robust methodology for the development of vehicle control strategy in future.

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Figures

Figure 1

Schematic diagram of vehicle and control models architecture.

Figure 2

ADAMS/Car vehicle model. 1-Front and rear tire system 2-Rear suspension system 3-Body subsystem 4-Steering subsystem 5-Brake subsystem 6-Front suspension system

Figure 3

Motor model diagram.

Figure 4

Vehicle control structure diagram.

Figure 5

Structure diagram of DYC.

Figure 6

Joint simulation model of DYC.

Figure 7

Single lane change test condition with DYC.(a) yaw rate,(b) slip rate of left rear wheel, (c) slip rate of right rear wheel.

Figure 8

Curve of slip rate and adhesion coefficient.

Figure 9

Structure diagram of ASR.

Figure 10

Joint simulation model of ASR

Figure 11

Single lane change test condition with ASR.(a) yaw rate,(b) slip rate of left rear wheel,(c) slip rate of right rear wheel.

Figure 12

Classifications of test conditions.

Step steer

Single lane change

Constant radius cornering

Figure 13

Simulation curve of DYC.

Step steer

Single lane change

Constant radius cornering

Figure 14

Simulation curve of ASR.

Figure 15

Flow chart of switching mode.

Figure 16

Simulation test of single lane change condition.(a) yaw rate,(b) side slip angle,and (c) longitudinal acceleration.

Figure 17

Simulation test of step steer condition.(a) yaw rate,(b) side slip angle,and (c) longitudinal acceleration.

Figure 18

Drivers in the loop simulation.

(a)

(b)

(c)

(d)

Figure 19

G27 simulation test conditions.(a)steering wheel angle,(b)velocity,(c)torque of right rear wheel,and (d)torque of left rear wheel.