Study on Maneuverability Control of Four In-Wheel Motor Electric Vehicle

Zeyang Zhang, Jianpeng Shi, Chunlai Zhao, Qiulai Wang, and Hongtao Li

Abstract This paper mainly studies the improvement of the maneuverability of four In-wheel motor Drive (4IWD) electric vehicle by torque vector control (TVC) under normal working conditions. A TVC algorithm is proposed based on the theory of feed-forward control and feedback control. It includes four control modules: the calculation of the vehicle state parameters and variables, the design of vehicle steering characteristic, the control of additional yaw moment and the allocation method of drive torque. The control method of additional yaw moment and the control method of allocation method of 4IWD are emphatically studied. The TVC algorithm is verified by using simulation experiments where the 4IWD electric vehicle model is established based on CarSim and Simulink joint simulation method. Results show that maximum value of the yaw rate, lateral acceleration and sideslip angle with TVC are 35%, 23.79%, 54.87% lower than that without TVC, respectively. It means that the maneuverability of the 4IWD under normal working conditions can be enhanced effectively by the proposed TVC algorithm.

Keywords In-Wheel motor · Electric vehicle · Ideal yaw rate · Maneuverability · PID control

1 Introduction

Torque vector control (TVC) is one of the important directions of four In-wheel motor Drive (4IWD) electric vehicle [1]. Currently, most of the research on vehicle stability control focus on the under extreme working conditions. [2], while that on the maneuverability research to improve steering performance under the non-extreme conditions is relatively less. In most cases, the car is driven under non-extreme conditions, thus, improvements under this condition can significantly improve the driving experience. The TVC usually includes two aspects: firstly, the ideal distribution of driving torque on the front and rear axles [3], secondly, the dynamic adjustment of

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driving torque on the left and right wheel [4]. The former is called indirect yaw moment control, which mainly affects the steady-state yaw rate gain, also known as the steady-state steering performance. The latter can directly add yaw moment control to the vehicle through the difference of driving torque among each wheel [5], thus improve the transient response quality and improve the stability of the vehicle.

There are many research methods on vehicle yaw moment control, such as feed-forward control, feedback control, feed-forward and feedback control, fuzzy control, sliding mode variable structure control and neural network control, etc [6]. The feed-forward and feedback control method can takes into account the advantages of both the feed-forward control method and the feedback control method, and it also has more practical application value than other methods. Therefore, this paper realizes the maneuverability control of the 4IWD electric vehicle based on the theory of feed-forward and feedback control. Firstly, a hierarchical control method of the TVC is proposed, and then the realization method is introduced in detail: the ideal yaw rate is calculated based on the linear 2-DOF reference model of the vehicle, and the deviation between the ideal yaw rate and the actual value is used as the feedback value to determine the additional yaw moment. Torque vector control is realized by distributing the additional yaw moment on each driving wheel. Finally, 4IWD electric vehicle model is built based on CarSim and Simulink software, and the developed control algorithm is verified by simulation results.

2 TVC Logic

A hierarchical control method is proposed to study the TVC algorithm of 4IWD electric vehicle, as shown in Fig. 1. The whole vehicle is taken as the controlled object, and the four-wheel driving torque determined by the TVC algorithm of the receiving controller is responded. The vehicle state parameters such as vehicle speed, acceleration and yaw velocity are feed back to the control algorithm. Firstly, the lateral stiffness and vertical load distribution of the front and rear axles are calculated via the parameter identification method. The ideal yaw rate is set as the tracking control target of the TVC through the design of steering characteristics. Finally, the ideal yaw rate and the actual value are used for PID control to determine the additional yaw moment needed, and the drive torque of four driving wheels are allocated by the control module of allocation method of the drive torque.

3 Design of Control Module

A TVC algorithm is proposed based on the theory of feed-forward control and feed-back control in this paper. It includes four control modules: the calculation of the vehicle state parameters and variables, the design of vehicle steering characteristic, the control of additional yaw moment and the allocation method of drive torque.

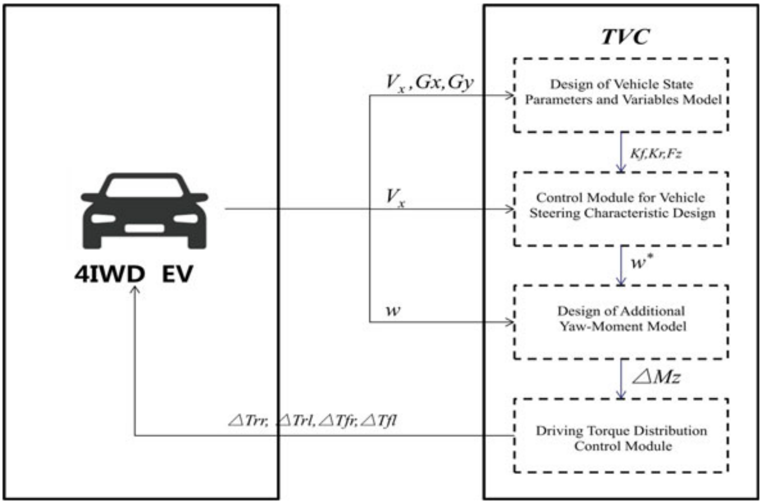

Fig. 1 TVC logic

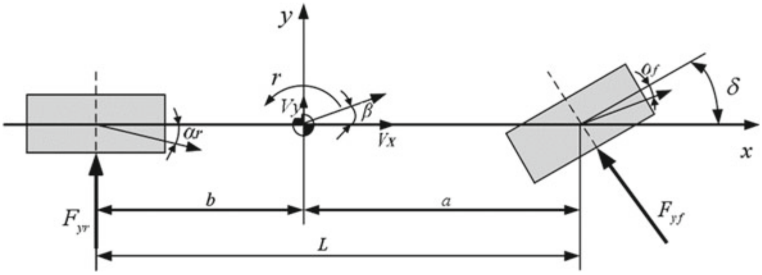

Fig. 2 2-DOF vehicle dynamics model

3.1 Vehicle Model for Controller Design

The linear 2-DOF model is used as a reference model to analyze corresponding vehicle characteristics in this study because it can well reflect the stable running state of vehicle [7] (Fig. 2).

The governing equations of the lateral and yaw motionscan be expressed as follows:

$$m \cdot V_x \cdot (\dot{\beta} + \omega_r) = (K_f + K_r) \cdot \beta + \frac{1}{V_x} \cdot (a \cdot K_f - b \cdot K_r) \cdot \omega_r - K_f \cdot \delta \quad (1)$$

$$I_Z \cdot \dot{\omega}_r = (a \cdot K_f + b \cdot K_r) \cdot \beta + \frac{1}{V_x}(a^2 \cdot K_f + b^2 \cdot K_r) \cdot \omega_r - a \cdot K_f \cdot \delta + M_Z \quad (2)$$

Where m, V_x, β, ω_r, K_f, K_r, a, b, δ, I_Z and Mz represent the vehicle mass, the vehicle longitudinal speed, the vehicle body side slip angle, the vehicle yaw rate, the lateral stiffness of front axles, the lateral stiffness of rear axles, the distances of the mass center from the front axles, the distances of the mass center from the rear axles, the vehicle steering angle, the vehicle mass moment of inertia about Z-axis, and the yaw moment, respectively.

The dynamic response process of yaw rate input with the steering wheel angle can be regarded as a second-order system [8]. The response process of the front wheel steering angle and additional yaw moment to the yaw rate are illustrated as following equations:

$$\omega_r(s) = G_r \cdot \frac{1 + \tau_r \cdot s}{\frac{1}{\omega_n{}^2} \cdot s^2 + \frac{2\zeta}{\omega_n} \cdot s + 1} \cdot \delta \tag{3}$$

$$\omega_r(s) = G_{mz} \cdot \frac{1 + \tau_{mz} \cdot s}{\frac{1}{\omega_n{}^2} \cdot s^2 + \frac{2\zeta}{\omega_n} \cdot s + 1} \cdot M_Z \tag{4}$$

Where G_r, G_{mz}, ω_n, ζ and τ_r represent the steady-state gain of the yaw rate to the input of front wheel steering angle, the steady-state gain of yaw rate to the input of additional yaw moment, the natural circular frequency of the system, the damping ratio and the response time constant, respectively.

3.2 Design of Vehicle State Parameters and Variables Model

The control module identifies the lateral stiffness (K_f, K_r) of the front and rear axle and calculates the vertical load F_Z according to the vehicle parameters.

The lateral stiffness under corresponding speed can be obtained based on the relationship curve between the lateral stiffness and the vehicle velocity which is identified by the real vehicle test. It can be shown as follows (Fig. 3):

The curves are derived from the analysis of the angular pulse test results of the vehicle, and the lateral stiffness is related to the vehicle speed.

Static loads of the front and rear axle are as follows:

$$F_{zf} = M \cdot g \cdot \frac{b}{L} \tag{5}$$

$$F_{zr} = M \cdot g \cdot \frac{a}{L} \tag{6}$$

Where F_{zf} and F_{zr} are the static load of front and rear axle.

Load transfer during vehicle acceleration can be shown as follows:

$$\Delta F_Z = \left(F_{xf} + F_{xr}\right) \cdot \frac{H}{L} \tag{7}$$

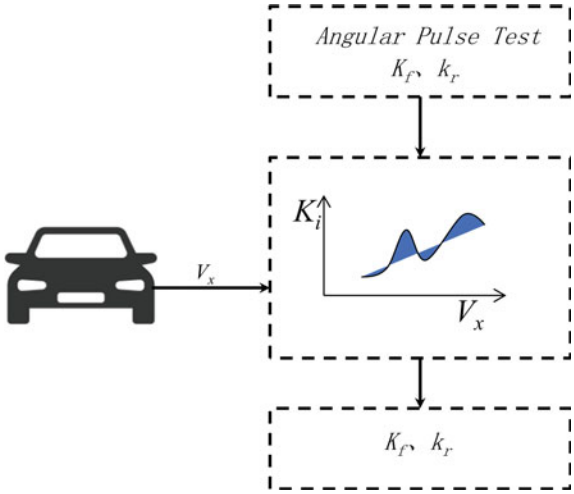

Fig. 3 Design of vehicle state parameters and variables model

Where F_{xf}, F_{xr} and H represent the longitudinal force of front axle, the longitudinal force of rear axle and centroid height, respectively.

Thus, the vertical load with acceleration can be obtained. Considering the influence of the change of the vertical load on the lateral stiffness, the actual lateral stiffness is corrected.

3.3 Design of Vehicle Steering Characteristic Model

According to the 2-DOF model for vehicle, the core of the control algorithm is the calculation of the ideal yaw rate. In view of the character that the driving torque of each driving wheel of the 4IWD electric vehicle can be controlled independently, the yaw motion of the vehicle can not only be affected by the front wheel angle, but also by the additional yaw moment which is obtained by differential drive/brake torque.

Thus, the steering characteristics of vehicle can be designed to break through the limitations of the original mechanical system. It means that the stability factor K, the natural circular frequency of the system ω_n, the damping ratio ζ, the response time constants τ_r and τ_{Mz} can be designed to improve the steering characteristics of vehicle in a certain range, that is improves the maneuverability of the vehicle.

3.4 Design of Additional Yaw-Moment Model

In this control module, the key parameters of the feed-forward model are calculated according to vehicle state parameters and control requirements, and the desired yaw

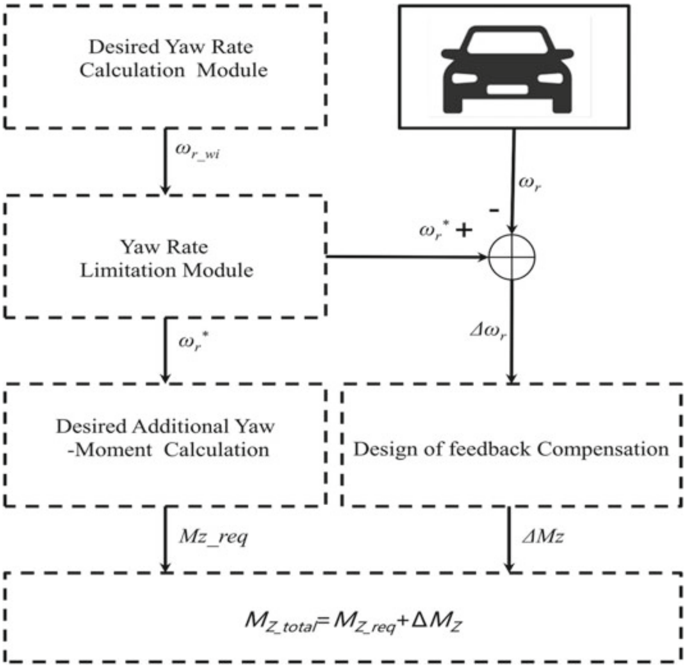

Fig. 4 Additional yaw moment control

rate which is obtained by the reference model is regarded as the control target. The final ideal yaw rate is obtained via the Yaw Rate Restriction Module. Then, the corrected additional yaw moment ΔMz is obtained by the PID control. The ΔMz, is added to the theoretical desired yaw moment M_{Z_req} to obtain the final additional yaw moment M_{Z_total}. The control logic is shown in Fig. 4.

3.4.1 Calculation of Desired Yaw Rate

The desired yaw rate ω_{r_wi} is calculated by using the formula (3) in the Sect. 3.1.

3.4.2 Limitation of Yaw Rate

The vehicle yaw rate is limited by the road adhesion, so as to prevent the vehicle from losing stability under the low-attached road. It is necessary to ensure that the desired yaw rate does not exceed the maximum yaw rate that the vehicle can achieve on the current road surface. Thus, the lateral acceleration of the vehicle satisfies the following formula:

$$a_y \leq \mu \cdot g \tag{8}$$

The maximum allowable yaw rate of pavement as follows:

$$\omega_{r\_max} \approx \frac{a_y}{V_x} \tag{9}$$

Considering the road surface adhesion, the ultimate ideal yaw rate can be obtained as follows:

$$\omega_r^* = min\{|\omega_{r\_wi}|, |\omega_{r\_max}|\} \cdot sgn(\omega_{r\_max}) \tag{10}$$

3.4.3 Calculation of Desired Additional Yaw-Moment

The desired additional yaw moment M_{Z_req} is calculated by using the formula (4) in the Sect. 3.1.

3.4.4 PID Feedback Control

The PID control is mainly composed of three correction links: proportional (P), integral (I) and differential (D). It is simple and practical because the whole process is based on differential feedback and regulation and there is no need for precise mathematical model in the control process. The expression is

$$\mathrm{U} = K_p e(t)dt + K_i \int e(t)dt + K_d de(t)/dt \tag{11}$$

Take the difference between the actual yaw rate of the 4IWD pure electric car model built by CarSim and the ideal yaw rate calculated by Simulink model, as the input of the PID controller. The yaw moment $\triangle M_Z$ was corrected through the design of Kp and Ki parameters, and is output by the vehicle controller. The desired yaw moment M_{Z_req} which is deduced from theory is added to get the final additional yaw moment M_{Z_total}.

3.5 Driving Torque Distribution Control Module

In this control module, the additional yaw moment M_{Z_total} is added according to the demand of the Additional Yaw Moment Control Module. It is distributed to four driving wheels through the drive moment distribution control module.

Firstly, the final additional yaw moment M_{Z_total} determined by the decision is compared with the allowable additional yaw moment M_{Z_max} actually provided by the system.

If the output of the system exceeds the allowable range, the maximum yaw moment M_{Z_max} of the system is used as the output value. The expression is as follows:

$$M_{Z\_real} = M_{Z\_max} \tag{12}$$

If the output of the system is in the allowable range, the final additional yaw moment M_{Z_total} is added according to the demand, then the actual output of the system is as follows:

$$M_{Z\_real} = M_{Z\_total} \tag{13}$$

When the TVC function of the system is turned on, the actual additional yaw moment M_{Z_real} of the aforementioned system is distributed equally to each driving wheel in this study, Thus four wheels have the same effect on the additional yaw moment. For a single wheel i, the additional torque is

$$T_{x\_i} = \frac{\frac{1}{2} M_{Z\_real}}{\frac{1}{2} W_i} \cdot R_i \tag{14}$$

Where T_{x_i}, W_i and R_i represent the additional torque, the wheelbase, and the rolling radius of a wheel i.

Finally, additional torque is applied to each wheel to complete the TVC of the whole vehicle.

4 TVC Simulation

In order to verify the developed method, the TVC model is built by using the MATLAB/Simulink software, which includes vehicle state parameters and variable calculation control module, vehicle steering characteristic design control module, additional yaw moment control module and drive moment distribution control module. The CarSim software was used to build a model of 4IWD electric vehicle. The simulation of TVC is shown in Fig. 5.

Simulated working conditions: double lane change, vehicle speed 80 km/h, road adhesion coefficient 0.8, some parameters of vehicle model as shown in Table 1.

The simulation results are shown in the following figures.

In Fig. 6, the peak value of the yaw rate is 0.13 (rad/s) at 2.8 s and −0.17 (rad/s) at 5 s without TVC. And those under TVC are 0.09 (rad/s) and −0.11 (rad/s) respectively. The maximum amplitude value decreases by 35%. In Fig. 7, without TVC, the peak value of the lateral acceleration is 2.85 (m/s2) at 2.8 s and −3.91 (m/s2) at 5 s,

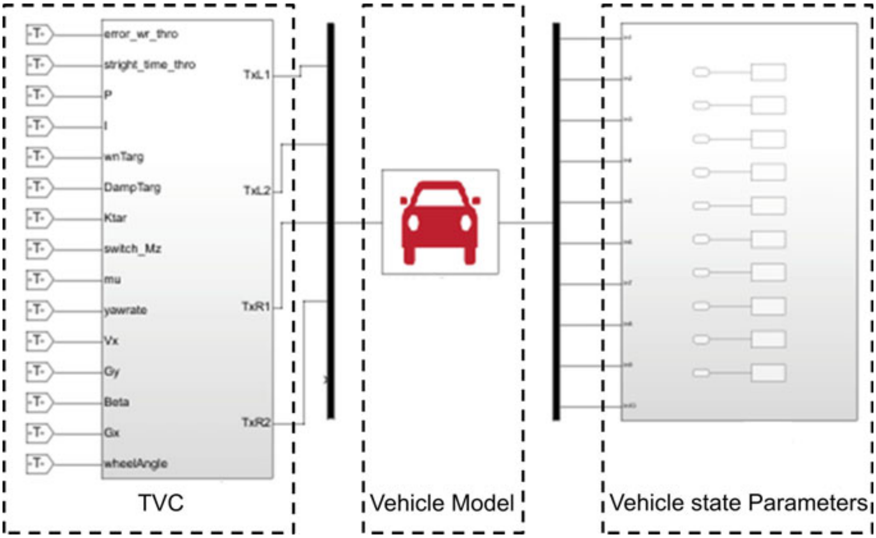

Fig. 5 TVC simulation

Table 1 Some parameters of the vehicle model

Parameter	Value
m/kg	1110
L/m	2.60
w_f/m	1.695
w_r/m	1.695
a/m	1.04
b/m	1.56
h/m	0.54
I_x/(kg.m^2)	440.6
I_y/(kg.m^2)	1343.1
I_z/(kg.m^2)	1343.1
R/m	0.31

and those under TVC is 2.45 (m/s2) and −2.98 (m/s2) respectively. The maximum amplitude value decreases by 23.79%. In Fig. 8, the peak value of the sideslip angle is 10.87E^{-3} (deg) at 5.1 s without TVC, while the peak value of sideslip angle is 4.91 E^{-3} (deg) at 4.2 s with TVC. The maximum amplitude value decreases by 54.83%. From Figs. 6, 7 and 8, we can see that the results under TVC are obviously reduced, which shows that the algorithm can significantly improve the vehicle maneuverability.

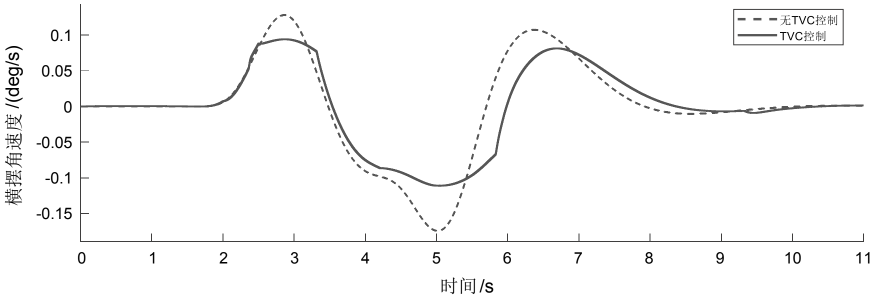

Fig. 6 Yaw rate

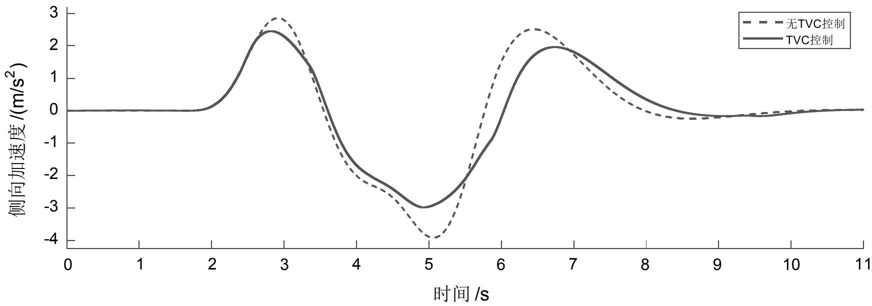

Fig. 7 Lateral acceleration

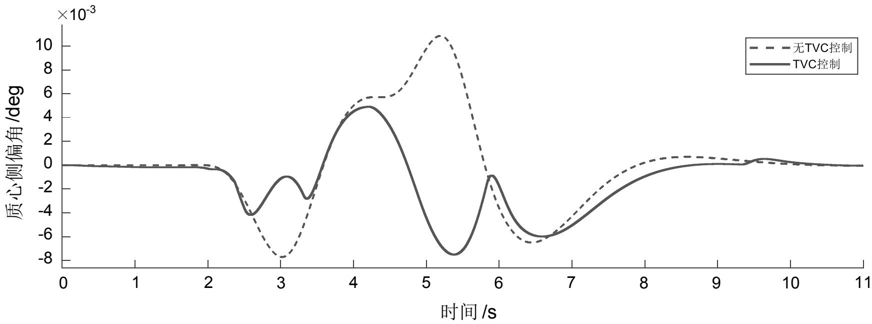

Fig. 8 Sideslip angle of the mass center

5 Conclusions

(1) A TVC algorithm is proposed based on the theory of feed-forward and feedback control. It takes into account the advantages of both the feed-forward control method and the feedback control method, and also has more practical application values.

(2) The desired yaw rate following PID control method and drive torque allocation control method are studied. The algorithm is verified by the CarSim and Simulink joint simulation method. Results show that maximum value of the yaw rate, lateral acceleration and sideslip angle with TVC are 35%, 23.79%, 54.87% lower than that without TVC. It means that the maneuverability of the 4IWD under normal working conditions can be enhanced effectively by the proposed TVC algorithm.

(3) The control algorithm only verified by using the simulation results of yaw rate transient response under double lane change condition, and lack of relevant experimental verification. Further research should be carried out on corresponding test results to make up for this deficiency, and the control algorithm will be further validated on the prototype vehicle.

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