Engineering Advances, 2024, 4(2), 93-101
https://www.hillpublisher.com/journals/ea/
ISSN Online: 2768-7961

Handling and Stability Control Strategy for Distributed Drive Electric Vehicles

Yuan Yuan*, Hang Wang, Lidong Zhou

Taiyuan University of Science and Technology, Taiyuan, Shanxi, China.

How to cite this paper: Yuan Yuan, Hang Wang, Lidong Zhou. (2024) Handling and Stability Control Strategy for Distributed Drive Electric Vehicles. *Engineering Advances*, 4(2), 93-101.
DOI: 10.26855/ea.2024.04.004

Received: March 20, 2024
Accepted: April 17, 2024
Published: May 14, 2024

***Corresponding author:** Yuan Yuan, Taiyuan University of Science and Technology, Taiyuan, Shanxi, China.

Abstract

In order to fully utilize the advantages of distributed drive electric vehicles, such as high control precision, fast response speed, and easy parameter adjustment, a hierarchical vehicle stability control strategy is employed to enhance the handling and stability of the vehicle. On the basis of Model Predictive Control (MPC), the upper layer introduces an analysis of the vehicle stability mechanism based on the phase plane of the mass center sideslip angle and mass center sideslip velocity. The double-line method is utilized to delineate the stability region, enhance the MPC is improved, and calculate the additional yaw moment needed by the vehicle by integrating it with the 2-DOF vehicle dynamic reference model. Subsequently, direct yaw moment control (DYC) is implemented across the entire vehicle. The lower layer considers the minimum tire load rate as the objective function and establishes multiple output torque constraints to optimize the motor torque, which is determined by the additional yaw torque calculated by the upper layer. Finally, the simulation verification was carried out under the condition of double line shifting. The results indicate that the enhanced MPC control strategy is superior to the unimproved MPC control strategy in ensuring the stability of the vehicle.

Keywords

Model predictive control algorithm, Phase plane, Direct yaw moment control, Torque optimization, Stability control strategy

1. Research background

Due to the lack of energy resources in recent years, new energy electric vehicles have attracted more and more attention, and distributed drive electric vehicles (DDEV) are one of the current research hotspots. Because of its higher control accuracy, faster response speed, and high transmission efficiency advantages, it is easy to cause accidents in turning and bad driving conditions when the driver's operation is not appropriate, so the stability control strategy of the vehicle is particularly important for the stable driving of the vehicle.

Liu Hongshuo combined model predictive control and direct yaw moment control, designed cooperative control rules, realized multi-objective control, solved the problem of conflict between accuracy and stability, and improved trajectory tracking accuracy and lateral stability [1]. Dong J proposed a hierarchical control strategy composed of a linear time-varying model predictive controller (LTV-MPC) and an optimal torque distributor, which decoupled the motor torque and wheel Angle linearization technology to form an optimal allocation framework for the trade-off between driving safety and energy saving, realized multi-objective optimization and verified the effectiveness of the strategy [2]. Yaakoubi H conducted system modeling within the framework of piecewise Affine (PWA). Based on the integration of the active front wheel steering control method and the direct yaw moment control method, the hybrid model predictive control (MPC) law was established to ensure the lateral stability of the vehicle [3]. Wu J proposed an Adaptive Model Predictive

Control (AMPC) scheme based on Autoregressive exogenous input (ARX) for direct yaw torque control, and then multi-objective optimization of torque distribution to improve the lateral stability of the vehicle [4].

In summary, the advantages of the MPC algorithm include high-performance control, multivariable control, constraint handling, robustness, scalability, etc., which provides strong support for achieving accurate, efficient, and reliable system control. Therefore, the vehicle stability control strategy based on the MPC algorithm is used in this paper.

2. Vehicle dynamics modeling

2.1 Modeling of 2-DOF vehicle dynamics

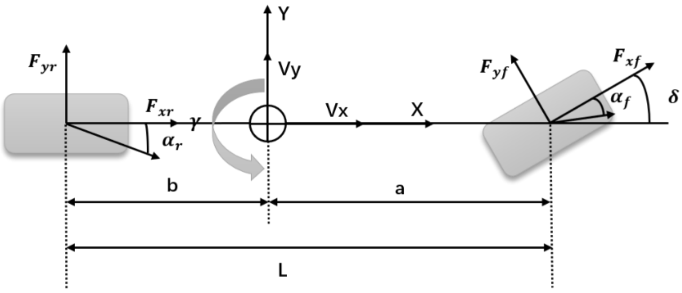

Figure 1. 2-DOF vehicle dynamics model.

In order to facilitate analysis and calculation, this paper establishes a simplified two-degree-of-freedom vehicle dynamics model [5] as shown in Fig. 1.

The dynamic equations are derived from the model:

$$F_{yf}\cos\delta + F_{yr} = m(\gamma v_x + \dot{v}_y) \tag{1}$$

$$aF_{yf}\cos\delta - bF_{yr} = I_z\dot{\gamma} \tag{2}$$

Where, v_x, v_y is the longitudinal and lateral velocity; δ is the front wheel Angle; $\dot{\gamma}$ is yaw Angle acceleration; F_{yf}, F_{yr} is the lateral force of front and rear wheels; And a, b is the distance from the center of mass to the front and rear axes; m is the vehicle mass; I_z is the moment of inertia around the z-axis.

$$\alpha_f = \beta + \frac{a\gamma}{v_x} - \delta \tag{3}$$

$$\alpha_r = \beta - \frac{b\gamma}{v_x} \tag{4}$$

Where, α_f, α_r is the lateral Angle of the front and rear wheels; β is the lateral deflection Angle of the center of mass.

Since the front wheel Angle is small, the formula is transformed as follows:

$$m(\dot{v}_x + v_x\gamma) = C_f(\beta + \frac{a\gamma}{v_x} - \delta) + C_r(\beta - \frac{b\gamma}{v_x}) \tag{5}$$

$$I_z\dot{\gamma} = aC_f(\beta + \frac{a\gamma}{v_x} - \delta) - bC_r(\beta - \frac{b\gamma}{v_x}) \tag{6}$$

According to reference [6], the desired yaw rate and the desired centroid sideslip Angle are as follows:

$$\begin{cases} \gamma_d = \min\left\{ \dfrac{v_x\delta}{(a+b)\left(1+Kv_x^2\right)}, \dfrac{\mu g}{v_x} \right\} \\ \beta_d = 0 \end{cases} \tag{7}$$

Where K is the stability coefficient, $K=\frac{m}{(a+b)^2}\left(\frac{a}{C_r}-\frac{b}{C_f}\right)$; When the center of mass sideslip Angle is 0, the vehicle will better respond to the driver's line diagram, so the expected center of mass sideslip Angle is set to 0.

3. The vehicle stability control strategy is predicted based on the improved model

Design vehicle stability control strategy based on improved model prediction as shown in Fig. 2. By introducing the stability evaluation of the center of mass sideslip Angle and the center of mass sideslip velocity phase plane, an improved model predictive control direct yaw moment controller is designed. Combined with the optimization of tire utilization, the controller can respond to the dynamic changes of the vehicle more accurately, and apply the appropriate yaw moment to improve the handling stability of the vehicle. This control strategy can better adapt to different road conditions and driving situations, reduce the risk of sideslip and loss of control, reduce the accident rate, and provide safer, smoother and reliable vehicle handling.

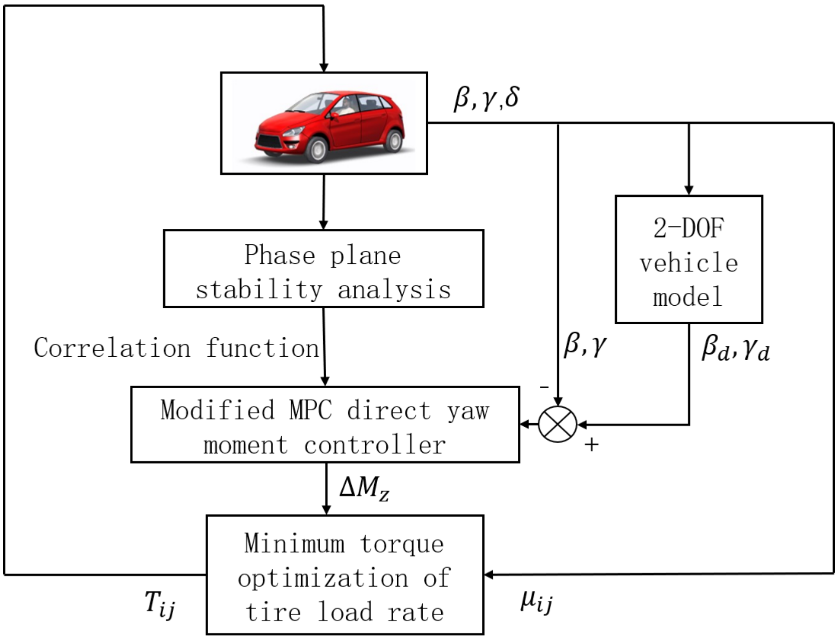

Figure 2. Principle of stability control strategy.

3.1 Vehicle handling and stability analysis

The stability characteristics and potentially unstable behavior of the vehicle can be visually observed by plotting the phase plots between the centroid sidecar Angle and the velocity of the centroid sidecar Angle in Fig. 3.

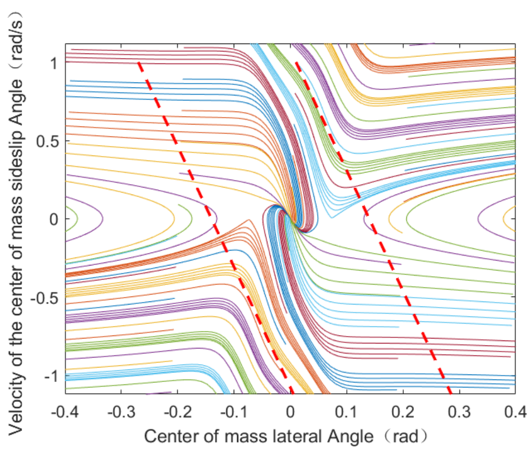

Figure 3. Center of mass sideslip Angle -center of mass sideslip velocity phase plane.

The double line method is a common method used to determine the stability boundary of the vehicle in the phase plane of the center of mass sideslip Angle-center of mass sideslip velocity. According to reference [7], the method for determining the stability region of the phase plane of the centroid lateral declination Angle is as follows:

$$|\dot{\beta}+e_1\beta|=e_2 \tag{8}$$

$$\begin{cases} e_1=-25.01\mu^2+30.5\mu+2.518 \\ e_2=-7.266\mu^2+10.81\mu-1.768 \\ \beta_{up}=e_2/e_1 \end{cases} \tag{9}$$

Where, e_1 and e_2 are the boundary coefficients, e_1 represents the slope of the boundary line, and e_2 represents the boundary intercept.

3.2 Principle of Model predictive control algorithm

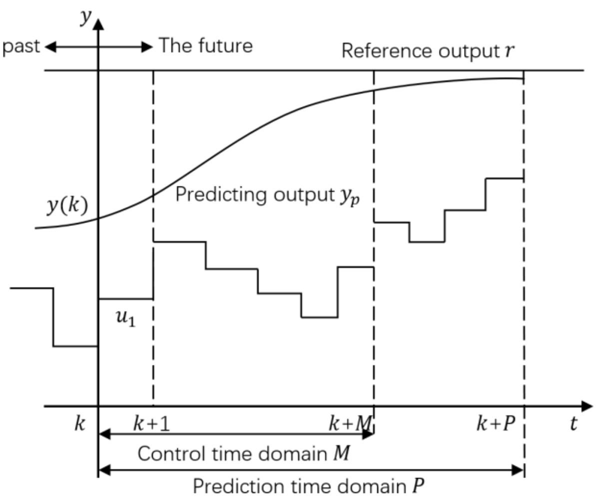

Figure 4. Principles of Model Predictive Control.

Model predictive control generates the optimal control policy based on the prediction of the dynamic model of the system, and its principle is divided into the following steps: First, a mathematical model of the system is established to describe the dynamic behavior of the system. Then, a control time domain range is chosen to transform the control problem into an optimization problem. In each control cycle, the system model is predicted based on the current system state, and the control objectives and constraints are considered in the optimization problem. By solving the optimization problem, the optimal control policy from the current time to the future time is obtained. In practice, only the first time step of the optimal control strategy is used as the current control command, and the control commands of other time steps are updated in the next control cycle, as shown in Fig. 4. The MPC recomputes and implements the optimal control policy in each control cycle to adapt to changes in the system state. Its advantage is that it can deal with multivariable, nonlinear and constrained control problems, and the control objective can be adjusted flexibly. By predicting the system behavior and considering the constraints, MPC can optimize the system performance and achieve better control effects within a given control time domain.

3.3 Improved model predictive control algorithm flow

The 2-DOF vehicle model is used as the prediction model, as follows:

$$\begin{bmatrix} \dot{\beta} \\ \dot{\gamma} \end{bmatrix}=\begin{bmatrix} \dfrac{C_f+C_r}{mv_x} & \dfrac{aC_f-bC_r}{mv_x^2}-1 \\ \dfrac{aC_f-bC_r}{I_z} & \dfrac{a^2C_f+b^2C_r}{I_zv_x} \end{bmatrix}\begin{bmatrix} \beta \\ \gamma \end{bmatrix}+\begin{bmatrix} -\dfrac{C_f}{mv_x} & 0 \\ -\dfrac{aC_f}{I_z} & \dfrac{1}{I_z} \end{bmatrix}\begin{bmatrix} \delta \\ \Delta M_z \end{bmatrix} \tag{10}$$

With the center of mass sideslip Angle and yaw rate as the state variables, the additional yaw moment as the input input variable, and the front wheel Angle input by the driver as the interference term, the formula is simplified as follows:

$$\dot{x} = A_c x + B_c u \tag{11}$$

The Euler method is used to discretize the equation as follows:

$$x(k+1) = Ax(k) + Bu(k) \tag{12}$$

Where, $A = I + A_c \Delta T$, $B = B_c \Delta T$, ΔT is the simulation step size.

Consider the following equation

$$u(k) = u(k-1) + \Delta u(k) \tag{13}$$

Simultaneous equations (12) and (13) are obtained :

$$\begin{aligned} \tilde{x}(k+1) &= \tilde{A}\tilde{x}(k) + \tilde{B}\Delta u(k) \\ y(k) &= \tilde{C}\tilde{x}(k) \end{aligned} \tag{14}$$

Where, $\tilde{x}(k) = \begin{bmatrix} x(k) \\ u(k-1) \end{bmatrix}, y(k) = \begin{bmatrix} \beta \\ \gamma \end{bmatrix}, \tilde{C} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \end{bmatrix}.$

The MPC controller predicts the future output at time k as follows:

$$\begin{aligned} & y(k+1) = C\tilde{A}\tilde{x}(k) + C\tilde{B}\Delta u(k) \\ & y(k+2) = C\tilde{A}^2\tilde{x}(k) + C\tilde{A}\tilde{B}\Delta u(k+1) + C\tilde{B}\Delta u(k) \\ & y(k+3) = C\tilde{A}^3\tilde{x}(k) + C\tilde{A}^2\tilde{B}\Delta u(k+2) + C\tilde{A}\tilde{B}\Delta u(k+1) + C\tilde{B}\Delta u(k) \\ & \vdots \\ & \vdots \\ & y(k+N) = C\tilde{A}^N\tilde{x}(k) + C\tilde{A}^{N-1}\tilde{B}\Delta u(k+N-1) + \cdots\cdots + C\tilde{A}\tilde{B}\Delta u(k+1) + C\tilde{B}\Delta u(k) \end{aligned} \tag{15}$$

Can be simplified as:

$$Y(k) = \bar{A}\tilde{x}(k) + \bar{B}\Delta U(k) \tag{15}$$

The objective function of the controller is designed to better follow the expected yaw rate and the yaw Angle of the center of mass. Due to the intervention of the additional yaw moment, the unilateral wheel torque will rise/fall, and there is a risk of vehicle instability. The pure pursuit of following accuracy will ignore the handling stability of the vehicle. Therefore, when designing the objective function, it is necessary to consider the output of the additional yaw moment as small as possible. At the same time, the stability degree of the phase plane of the mass center of sideslip Angle is also taken into account in the objective function, and the weight of each part is calculated in real time according to the driving state of the vehicle. The objective function is as follows:

$$\begin{aligned} J = & \sum_{i=1}^{N} \left[y_d(k+i) - y(k+i) \right]^T Q \left[y_d(k+i) - y(k+i) \right] \\ & + \sum_{i=0}^{N-1} \Delta u(k+i)^T R \, \Delta u(k+i) + \sum_{i=0}^{N-1} R_c(k+i)^T P R_c(k+i) + \rho \varepsilon^2 \end{aligned} \tag{16}$$

For the handling and stability of the vehicle, the control variables, control increments and output variables are constrained:

(1) Control variable constraints:

$$\begin{bmatrix} u_{\min} \\ u_{\min} \\ \vdots \\ u_{\min} \end{bmatrix} \le \left(\begin{bmatrix} 1 & 0 & \cdots & 0 \\ 1 & 1 & \cdots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 1 & 1 & \cdots & 1 \end{bmatrix} \otimes I_{N_u} \right) \begin{bmatrix} \Delta u(k) \\ \Delta u(k+1) \\ \vdots \\ \Delta u(k+N-1) \end{bmatrix} + \begin{bmatrix} u(k-1) \\ u(k-1) \\ \vdots \\ u(k-1) \end{bmatrix} \le \begin{bmatrix} u_{\max} \\ u_{\max} \\ \vdots \\ u_{\max} \end{bmatrix} \tag{18}$$

Can be expressed as :

$$U_{\min} - U(k-1) \le E\Delta U(k) \le U_{\max} - U(k-1) \tag{17}$$

(2) Control incremental constraints:

$$\begin{bmatrix} \Delta u_{\min} \\ \Delta u_{\min} \\ \vdots \\ \Delta u_{\min} \end{bmatrix} \le \begin{bmatrix} \Delta u(k) \\ \Delta u(k+1) \\ \vdots \\ \Delta u(k+N-1) \end{bmatrix} \le \begin{bmatrix} \Delta u_{\max} \\ \Delta u_{\max} \\ \vdots \\ \Delta u_{\max} \end{bmatrix} \tag{18}$$

Can be expressed as:

$$\Delta U_{\min} \le \Delta U(k) \le \Delta U_{\max} \tag{21}$$

(3)Output constraints:

$$\begin{bmatrix} y_{\min} \\ y_{\min} \\ \vdots \\ y_{\min} \end{bmatrix} \le \begin{bmatrix} y(k) \\ y(k+1) \\ \vdots \\ y(k+N-1) \end{bmatrix} \le \begin{bmatrix} y_{\max} \\ y_{\max} \\ \vdots \\ y_{\max} \end{bmatrix} \tag{19}$$

Can be expressed as:

$$Y_{\min} \le Y(k) \le Y_{\max} \tag{20}$$

According to the objective function and prediction equation, considering each constraint and relaxation factor simultaneously, the following optimization problem can be obtained.

$$\begin{gathered} \| Y_d(k) - Y(k) \|^2_{Q_{MPC}} + \| \Delta U(k) \|^2_{R_{MPC}} + \| R_c(k) \|^2_{P_{MPC}} + \rho\varepsilon^2 \\ Y(k) = \bar{A}\tilde{x}(k) + \bar{B}U(k) \\ s.t. \qquad \begin{bmatrix} \Delta U_{\min} \\ 0 \end{bmatrix} \le \begin{bmatrix} \Delta U(k) \\ \varepsilon \end{bmatrix} \le \begin{bmatrix} \Delta U_{\max} \\ \varepsilon_{\max} \end{bmatrix} \\ \begin{bmatrix} -E & 0 \\ E & 0 \\ -\bar{B} & 0 \\ \bar{B} & 0 \end{bmatrix} \begin{bmatrix} \Delta U(k) \\ \varepsilon \end{bmatrix} \le \begin{bmatrix} -(U_{\min} - U(k-1)) \\ U_{\max} - U(k-1) \\ -(Y_{\min} - \bar{A}\tilde{x}(k)) \\ Y_{\min} - \bar{A}\tilde{x}(k) \end{bmatrix} \end{gathered} \tag{24}$$

MATLAB software was used to solve the above optimization problem, and when the optimal solution sequence was obtained, only the first component was selected as the output variable according to the principle of model predictive control algorithm.

3.4 Torque optimization distribution

The objective function of the minimum tire load rate is as follows:

$$\min J = \frac{1}{4}\sum(\rho_{ij} - E(\rho))^2 + \eta E(\rho) \tag{25}$$

Where, $E(\rho) = \frac{\rho_{fl} + \rho_{fr} + \rho_{rl} + \rho_{rr}}{4}$ is the mean tire load rate; $\frac{1}{4}\sum(\rho_{ij} - E(\rho))^2$ is the variance of tire load rate; η is the weighting coefficient.

In order to avoid complex calculations and ignore the lateral forces that are difficult to obtain, the tire load rate can be expressed as follows :

$$\rho_{ij}=\sqrt{\frac{F_{xij}^2}{\mu_{ij}^2 F_{zij}^2}}=\frac{F_{xij}}{\mu_{ij}F_{zij}} \tag{21}$$

(1) Constraint of road adhesion coefficient:

$$-\sqrt{\mu_{ij}^2 F_{zij}^2 - F_{yij}^2} \le F_{xij} \le \sqrt{\mu_{ij}^2 F_{zij}^2 - F_{yij}^2} \tag{22}$$

(2) Driving force demand constraint:

$$F_{xfl}\cos\delta + F_{xfr}\cos\delta + F_{xrl} + F_{xrr} = F_{xd} \tag{23}$$

(3) Additional yaw moment constraints:

$$\frac{d}{2}F_{xfr} - \frac{d}{2}F_{xfl} + \frac{d}{2}F_{xrr} - \frac{d}{2}F_{xrl} = \Delta M_z \tag{24}$$

(4) In-wheel motor torque constraints:

$$\frac{T_1}{R} \le F_{xij} \le \frac{T_2}{R} \tag{25}$$

Where, T_1 is the maximum braking torque and T_2 is the maximum driving torque.

The objective function and constraints are transformed into a standard quadratic programming problem, which is solved by the quadratic programming method.

4. Simulation verification and analysis

The Carsim/Simulink co-simulation platform is built. According to the current national standard GB/T 30677-2014, the double line shift condition of 80km/h is set, and the sampling interval is 0.001s. Some vehicle parameters are shown in Table 1.

Table 1. Some vehicle parameters

Name and Organization	Symbol	Numerical value
Complete vehicle mass /(kg)	m	1400
Mass on spring /(kg)	m_s	1270
Distance from center of mass to front axis /(m)	a	1.015
Distance from center of mass to rear axis /(m)	b	1.895
Front wheel pitch /(m)	d1	1.675
Rear wheel pitch /(m)	d2	1.675
Moment of inertia /(kg·m^2)	Iz	1536.7
Distance from the center of mass to the ground /(m)	h	0.5
Radius of wheel /(m)	R_e	0.325

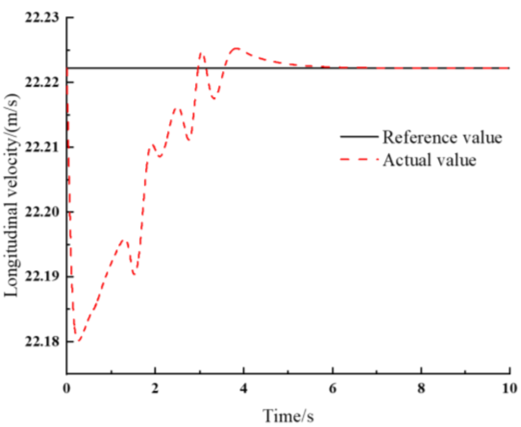

Figure 5. Longitudinal velocity.

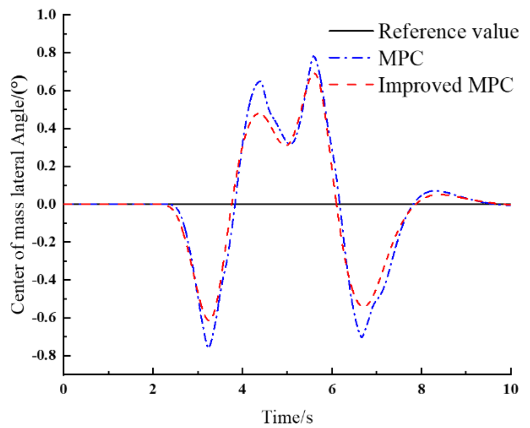

Figure 6. Center of mass lateral Angle.

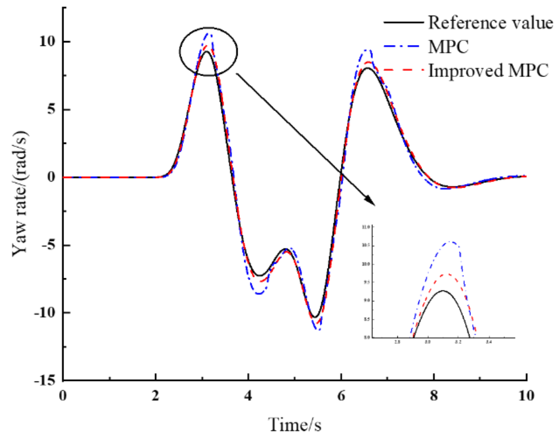

Figure 7. Yaw rate.

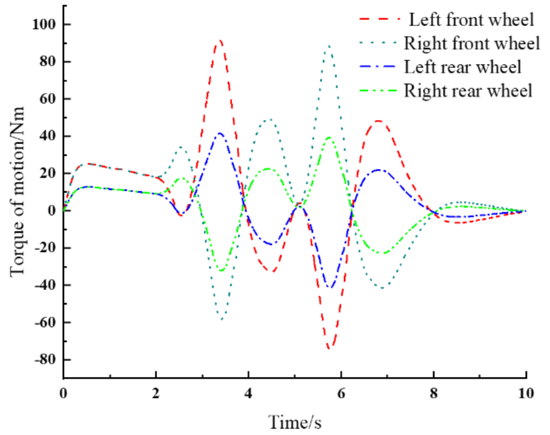

Figure 8. Optimal torque distribution.

Table 2. The evaluation index of the deviation Angle of the center of mass

Control strategy	MAE	RMSE
MPC	2.2646	1.0700
Improved MPC	1.9836	0.9306
Percentage error reduction	12.41%	13.03%

Table 3. Evaluation index of yaw rate error

Control strategy	MAE	RMSE
MPC	3.9181	1.8370
Improved MPC	2.0594	1.0150
Percentage error reduction	47.44%	44.75%

According to the simulation results in Figs. 5-8, the longitudinal velocity fluctuation is less than 0.5km/h under the double line shift condition, which can well follow the desired longitudinal velocity. The stability control strategy based on improved MPC is better than the simple MPC stability control strategy in controlling the centroid sideslip Angle and yaw rate. The mean absolute error and root mean square error of the centroid sidecar Angle of the unimproved MPC stability control strategy are 2.2646 and 1.0700, and the mean absolute error and root mean square error of the centroid sidecar Angle of the improved MPC stability control strategy are 1.9836 and 0.9306, respectively. Compared with the unimproved MPC stability control strategy, the mean absolute error and root mean square error of the centroid sideclination Angle are reduced by 12.41% and 13.03%, respectively. The mean absolute error of yaw rate is reduced by 47.44%, and the root mean square error is reduced by 44.75%.

5. Conclusion

In order to improve the driving stability and safety of DDEV, the model predictive control algorithm is improved, and the stability mechanism of the vehicle is analyzed. The phase plane method of center of mass sidesangle Angle and center of mass sidesangle velocity is introduced into the model predictive control algorithm. Combined with the two-degree of freedom dynamics model, the additional yaw moment required by the vehicle is calculated. Multiple constraints are added to optimize the torque distribution, which is solved by quadratic programming. The simulation results show that the improved model predictive control is better than the unimproved model predictive control in the control of the center of mass Angle and yaw moment, which are increased by 12.41% and 47.44% respectively. According to the analysis, the improved model predictive control improves the handling stability of the vehicle.

Funding

This paper is supported by the Shanxi Province Scientific and Technological Achievements Transformation Guidance Project (No. 202204021301060).

References

[1] Liu Hongshuo, Tao Gang, Li Derun, et al. Cooperative Control of Trajectory Tracking and Yaw Stability for Intelligent Vehicle [J]. Journal of Automotive Engineering, 2022, 12(06):773-781.

[2] Dong J, Li J, Gao Q, et al. Optimal coordinated control of active steering and direct yaw moment for distributed-driven electric vehicles [J]. Control Engineering Practice, 2023, 134: 105486.

[3] Yaakoubi H, Haggège J, Rezk H, et al. Explicit hybrid MPC for the lateral stabilization of electric vehicle system [J]. Energy Reports, 2024, 11: 1100-1111.

[4] Wu J, Wang Z, Zhang L. Unbiased-estimation-based and computation-efficient adaptive MPC for four-wheel-independently-actuated electric vehicles [J]. Mechanism and Machine Theory, 2020, 154: 104100.

[5] Fan Ying, Xu Yuan. Research on Control Strategy of Four-wheel Steering Vehicle [J]. Journal of Taiyuan University of Science and Technology, 2022, 43(04):335-339+345.

[6] Wen Shucheng. Distributed drive electric vehicle state estimation with steering stability control [D]. Jiangxi university of science and technology, 2023. DOI: 10.27176 /, dc nki. Gnfyc. 2023.000603.

[7] Zhao Yongqiang. Four wheel drive electric motor torque optimization allocation method research [D]. Jilin university, 2021. DOI: 10.27162 /, dc nki. Gjlin. 2021.007448.