Research on Stability Control Method of Vehicle Based on Steering Torque Superposition

ZHENYANG LI\textsuperscript{1}, YING ZHAO\textsuperscript{1}, KAI XU\textsuperscript{2}, JIAN WU\textsuperscript{3}, QINGFENG KONG\textsuperscript{3}, AND XIAOPING LIU\textsuperscript{3}

\textsuperscript{1}School of Mechanical and Automotive Engineering, Liaocheng University, Liaocheng 252000, China
\textsuperscript{2}Niuquan Township People’s Government of Laiwu District, Jinan 271100, China
\textsuperscript{3}School of Mechanical and Automotive Engineering, Liaocheng University, Liaocheng 252000, China

Corresponding author: Ying Zhao (zhaoying@lcu.edu.cn)

This work was supported in part by The Key Research and Development Plan of Shandong Province, China, under Grant2019GGX104047, and in part by the Natural Science Foundation of Shandong Province, China, under Grant ZR2019PEE006.

ABSTRACT At present, most stability control systems based on active steering are superposition control of steering Angle. For the steering system configuration commonly used in automatic driving vehicles, the active steering control system of steering angle superposition cannot be realized. In order to achieve man-machine collaborative driving, the co-driving intelligent vehicles usually realize automatic steering control based on the torque superposition, which solves the drawback that the driver of the Angle superposition cannot participate in the steering. In this paper, an active steering control method of superposition of steering torque based on Receding horizon control is proposed for man-machine collaborative driving of co-drive intelligent vehicles. This control method combines steering system with vehicle dynamics system, and takes steering motor torque as the sole control input, which ensures that the driver has the ultimate control right and makes the control of the system more stable. In order to solve the system uncertainties caused by model mismatch and external disturbances, the optimal control rate in predictive time domain can be calculated according to real-time vehicle state parameters by receding horizon control. In the design of the controller, the yaw rate deviation is the input of the controller, and then the calculated superposition torque is input into the vehicle model in real time, and the steering wheel angle is observed as a state variable. The control strategy can effectively compensate for the uncertainty of the system such as model mismatch and external disturbance, and enhance the robustness of the system.

INDEX TERMS Stability control, torque superposition, receding horizon control, system robustness.

I. INTRODUCTION

Before the realization of automatic driving, man-machine cooperative driving control is the necessary stage [1]. Man-machine co-driving system is a human-machine co-driving mode through the interaction of steering system, it’s based on the steering system to realize the torque interaction between driver and automatic driving control system [2], [3]. At the same time, intelligent driving also pays more attention to the use and conservation of energy [4], [5]. Based on torque control, the dynamic energy utilization efficiency can be improved. Intelligent driving will also have dangerous working status such as instability in complex driving conditions. The active steering is the key technology to achieve stability control of vehicles, and the stability control is mostly achieved by superposition control of steering angle. For the steering system configuration commonly used in automatic driving vehicles, the active steering control system with steering angle superposition cannot be realized [6]. For the disadvantage of steering conflict between driver and auxiliary steering system caused by steering angle control, this paper adopts active steering control method with steering torque superposition.

Active steering is the key technology to achieve vehicle stability control. Therefore many scholars have studied the active steering control strategy [7]–[11]. In recent years, fuzzy control or hybrid fuzzy control and optimal control theory have been widely used in the research of active steering control. Reference [12] designed yaw rate tracking control method based on PID and combined control method of PID and fuzzy logic, and compared the control effects. Reference [13] considering the constraints of ground
adhesion, steering and braking, a fuzzy PID chassis integrated controller is designed, which has a good effect on vehicle stability control. Reference [14] studied the linear quadratic regulator (LQR) based on variable-ratio steering gear to seek the optimal feedback input of the controller, realized the ideal steering characteristics of low speed with high steering sensitivity and high speed with high steering stability. Reference [15] designed a new LQG control strategy for active steering system by considering all kinds of disturbances and noises that may exist in the system.

In order to enhance the anti-interference ability of the system and the uncertainty of the model, Reference [16] studied the active rear wheel steering control based on $H_\infty$ robust control method, realized the setting of response characteristic of yaw rate by designing a feedback controller based on $H_\infty$ control. Robust control $H_\infty$ has a strong robustness to disturbances and uncertainties of the system, but when the robustness of the system is strong, the control performance of the controller will be sacrificed. Especially when the actual disturbances and uncertainties of the system are small, the control system will become very conservative and the control effects will become worse. Reference [17] designed a general internal model robust controller for active steering aiming at the contradiction between the performance and robustness of the controller. Reference [18] designed active steering robust controllers for electric vehicles, which effectively resisted external disturbances and guaranteed the robustness of the system. Reference [19] designed an active steering controller based on $H_\infty$ control for two-degree-of-freedom model (2-DOF) linear vehicle with yaw and lateral, they improved the performance and robustness of the controller by compensating the uncertainties of vehicle parameters.

The above active steering control method has made a good progress. In previous studies, the steering angle was used as control input to achieve stability control, while in intelligent vehicle steering system, the superposition control of active steering Angle could not be realized. In order to consider the driver’s driving automation in-the-loop, Reference [20] put forward the man-machine co-driving mode, which usually realizes automatic steering control based on the torque loop, they solved the disadvantage that the man could not participate in the steering of the steering angle loop. In this paper, a receding horizon control algorithm based on steering torque superposition is proposed for man-machine cooperative driving mode of co-driving intelligent vehicles. Compared with the superposition control of steering angle, the torque superposition control system has faster response and ensures more stable control while guaranteeing the driver’s final control right. The control input is yaw rate, the steering angle is liberated as a state variable, and the steering system model is combined with the vehicle dynamics system model. Receding horizon control can effectively utilize the external disturbances in the measured values or the error information of the model, and better promote the stability of the system performance. Experiments show that the control method effectively improves vehicle driving stability and system robustness, and has a good effect on active steering stability control.

The rest of this paper is arranged as follows: Section II introduces the control strategy and steering for active steering and the modeling of vehicle system dynamics, and analyses the whole model. Section III carries on the corresponding simulation experiment, and analyses the result of the experiment. Then Section IV builds the hardware experimental platform of active steering for co-driving vehicle, and carries out the hardware in-the-loop experiment of active steering in the steering experimental simulator, and processes and analyses the experiment results accordingly. Finally, a brief conclusion of this paper is given in Section V.

II. RECEDING HORIZON CONTROL STRATEGY AND MODELING
The principle of active steering control strategy designed in this study is shown in Fig. 1.

The reference model obtains the ideal yaw rate $\dot{\psi}$ according to the driver’s driving behavior and the vehicle speed, then the receding horizon controller according to the deviation between ideal yaw rate and actual yaw rate and the variation rate of deviation to obtain the value of corrected torque, which is superimposed with the actual steering torque to obtain the ideal steering torque. Finally, the steering of the vehicle is controlled and the driving stability is improved.

A. STEERING SYSTEM MODEL
The steering system model is shown in Figure 2. The steering torque is the only control input of the steering system.

In order to integrate steering dynamics into vehicle dynamics system, the same force is applied to the kingpin. According to the characteristics of steering system:

$$J_{eq} \ddot{\delta} + B_{eq} \dot{\delta} + \tau_{dis} + \tau_{r} \text{sign}(\dot{\delta}) = N_s N_m \tau_m$$

(1)

where $N_s N_m$ is the motion ratio of motor reducer and steering system, $\tau_m$ is the steering motor torque, $\tau_r$ is equivalent to the friction torque of the steering system, $\tau_{dis}$ is the steering resistance torque, which is generated by the front tire force. The equivalent inertia and damping of the steering system of
the kingpins $J_{eq}$ and $B_{eq}$ are as follows:

$$J_{eq} = J_{fw} + (N_sN_m)^2J_m + N_z^2(j_c + m_r r_p^2)$$

$$B_{eq} = B_{fw} + (N_sN_m)^2B_m + N_z^2(B_c + B_r r_p^2)$$

where $J_m$, $B_m$ are the inertia and damping of steering column, $J_{fw}$, $B_{fw}$ are the inertia and damping of the steering tie rod, $B_r$, $m_r$ are the damping and quality of the gear rack.

According to equation (1), the state space equation can be obtained by taking $x_1 = \begin{bmatrix} \delta \end{bmatrix}$ as a state variable.

$$\dot{x}_1 = A_1 x_1 + B_1 \tau_m + N_1 w_1$$

$$y_1 = C_1 x_1$$

(2)

where,

$$A_1 = \begin{bmatrix} 0 & 1 \\ 0 & \frac{B_{eq}}{J_{eq}} \end{bmatrix}$$

$$B_1 = \begin{bmatrix} 0 \\ \frac{N_mN_s}{J_{eq}} \end{bmatrix}$$

$$N_1 = \begin{bmatrix} 0 \\ -\frac{1}{J_{eq}} \end{bmatrix}$$

$$C_1 = \begin{bmatrix} 1 & 0 \end{bmatrix}$$

$$w_1 = \tau_{dis} + \tau_{sign}(\delta)$$

The state equation (2) shows the relationship between $\tau_m$ and $\delta$.

B. VEHICLE DYNAMICS MODEL AND TIRE MODEL

This section describes tire and vehicle models for controller design. The non-linearity of tire can be well described by the Fiala tire model [21];

$$F_y = \begin{cases} 
-\mu F_z \left(1 - \left(\frac{C_a |\tan \alpha|}{3 \mu F_z}\right)^3\right) \text{sgn}(\alpha), \\
-\mu F_z \text{sgn}(\alpha), 
\end{cases}$$

(3)

$\alpha$ and $C_a$ represent cornering angle and over-bending stiffness, $F_z$ and $\mu$ represent vertical force and road friction coefficient, respectively.

As shown in Figure 3, a non-linear 2-DOF vehicle model is used to describe the lateral dynamics of the vehicle during the operation of active steering [22]. Without directly affecting the vehicle speed, the controller only responds to the speed change specified by the driver. For simplicity, it is assumed that the current vehicle speed will remain unchanged throughout the predicted range. Therefore, it is assumed that the longitudinal velocity of the vehicle model used by the controller is constant:

$$\dot{\beta} = \frac{F_{sf}(\alpha_f)}{mv_x} + \frac{F_{yr}(\alpha_r)}{mv_x} - \psi$$

$$\ddot{\psi} = -\frac{aF_{ry}(\alpha_f)}{I_z} - bF_{yr}(\alpha_r)$$

(4)

where, $m$ is vehicle mass; $v_x$ is vehicle longitudinal speed; $\beta$ is the sideslip angle of vehicle; $\psi$ is vehicle yaw angle; $I_z$ is vehicle rotational inertia; $a$ is the distance from the front wheel to the vehicle centroid; $b$ is the distance from the rear wheel to the vehicle centroid; $F_{sf}$ and $F_{yr}$ are the lateral forces of front and rear tires; $\alpha_f$ and $\alpha_r$ are the sideslip angles of front and rear wheels, respectively, so that we can obtain:

$$\alpha_f = \frac{v_y + a \dot{\psi}}{v_x} - \delta \alpha_r = \frac{v_y - b \dot{\psi}}{v_x}$$

(5)

To simplify the calculation, the lateral force of a non-linear tire is approximated as an affine function near the working point.

$$F_{sf}(\alpha_f) = F_{sf0} - C_{f0}(\alpha_f - \alpha_{f0})$$

$$F_{yr}(\alpha_r) = F_{yr0} + C_{r0}(\alpha_r - \alpha_{r0})$$

(6)

In which, $C_{f0}$ and $C_{r0}$ are the slopes of the Fiala tire model at points $\alpha_{f0}$ and $\alpha_{r0}$.

In order to observe the vehicle state and control, the sideslip angle and yaw angle of vehicle are taken as state variable $x_2 = \begin{bmatrix} \beta & \dot{\psi} \end{bmatrix}$. By introducing equation (5), (6) into (4), the vehicle dynamic state equation can be obtained as follows:

$$\dot{x}_2 = A_2 x_2 + B_2 \delta + N_2 w_2^2$$

$$y_2 = C_2 x_2$$

(7)

where,

$$A_2 = \begin{bmatrix} \frac{C_{f0} + C_{r0}}{mv_x} & \frac{aC_{f0} + bC_{r0}}{mv_x} - v_x \\
\frac{aC_{f0} - bC_{r0}}{I_z v_x} & \frac{a^2C_{f0} + b^2C_{r0}}{I_z v_x} \end{bmatrix}$$

$$B_2 = \begin{bmatrix} \frac{C_f}{mv_x} \\
-\frac{aC_f}{I_z} \end{bmatrix}$$

$$C_2 = \begin{bmatrix} 1 & 0 \\
0 & 1 \end{bmatrix}$$

$$N_2 = \begin{bmatrix} \frac{F_{sf0} + F_{yr0} - C_{r0}\alpha_{f0} - C_{f0}\alpha_{r0}}{mv_x} \\
\frac{aC_{f0} - bC_{r0} - aC_{r0}\alpha_{f0} + bC_{f0}\alpha_{r0}}{I_z} \end{bmatrix}$$
Equation (7) gives the relationship between steering angle input and vehicle dynamic output. For the purpose of this study, it is necessary to combine vehicle dynamic model and steering system model to form a combined dynamic model.

C. COMBINATORIAL DYNAMICS MODEL
According to the foregoing, a fourth-order combinatorial dynamic model can be obtained by combining formula (2) with formula (7),

$$\dot{x} = Ax + Br_m + Nw$$

$$y = Cx$$

where,

$$A = \begin{bmatrix} A_1 & 0 \\ B_2C_1 & A_2 \end{bmatrix}, \quad B = \begin{bmatrix} B_1 \\ 0 \end{bmatrix}$$

$$N = \begin{bmatrix} N_1 & 0 \\ 0 & N_2 \end{bmatrix}, \quad C = \begin{bmatrix} 0 & C_2 \end{bmatrix}$$

The above combined model releases $\delta$ as a state variable and disturbance $w_2$ is calculated by dynamic equation [23].

D. ESTABLISHMENT OF REFERENCE MODEL
In this paper, linear 2-DOF, which can well characterize the characteristics of the system, is taken as a reference model [24] [25]. The ideal yaw rate can be obtained,

$$\dot{\psi}_r = \frac{v_x/L}{1 + Kv_x}\delta$$

the corresponding ideal sideslip angle can also be obtained,

$$\beta_r = \frac{v_x^2}{L(1 + Kv_x)^2} \left( \frac{b}{v_x} + \frac{ma}{k_2L} \right)$$

where, $k_2$ is the lateral tilt stiffness of rear wheel, $L$ is the wheel base, and the stability factor $K$ can be obtained by the following equation:

$$K = \frac{m}{L^2} \left( \frac{a}{k_2} - \frac{b}{k_1} \right)$$

E. DESIGN OF RECEDING HORIZON CONTROLLER
The receding horizon control strategy solves the control input which minimizes the control target according to the system state within a limited period of time after the current sampling time. $k$ is the current moment, $j$ is the time interval, $N$ is the predictive time domain, $m$ is the control time domain, and $i_f = k + N$ is the terminal time of receding horizon prediction. The following assumptions are needed to solve the control time domain problem:

The system control amount outside the control time domain remains unchanged, that is:

$$\Delta u(k + i) = 0, \quad i = m, m + 1, \cdots, N - 1$$

The detectable interference remains unchanged after the current control time, that is:

$$\Delta \omega_1(k + i) = \Delta \omega_2(k + i) = 0, \quad i = 1, 2, \cdots, N - 1$$

Discrete equation can be obtained by discretizing equation (7):

$$x_{k+j+1|k} = A_dx_{k+j|k} + B_du_{k+j|k} + N_d\omega_{k+j|k}$$

$$y_{k+j|k} = C_dx_{k+j|k}$$

The performance indicators are as follows:

$$J(x_{k|k}, x^{r}, u_{k+|k})$$

$$= \sum_{j=0}^{i_f-k-1} \left[ (x_{k+j|k} - x^{r}_{k+j|k})^T Q (x_{k+j|k} - x^{r}_{k+j|k}) + u_{k+j|k}^T R u_{k+j|k} \right] + \left( x_{i_f|k} - x^{r}_{i_f|k} \right)^T R_f (x_{i_f|k} - x^{r}_{i_f|k})$$

where,

$$Q = C_d^T \overline{Q} C_d$$

$$R_f = C_f^T \overline{Q} C_d$$

Construct Hamilton function:

$$H_{k+j|k} = \left[ (x_{k+j|k} - x^{r}_{k+j|k})^T Q (x_{k+j|k} - x^{r}_{k+j|k}) + u_{k+j|k}^T R u_{k+j|k} \right]$$

$$+ \left( x_{i_f|k} - x^{r}_{i_f|k} \right)^T R_f (x_{i_f|k} - x^{r}_{i_f|k})$$

The partial derivation of equation (15) is calculated,

$$p_{k+j} = \frac{\partial H_{k+j|k}}{\partial x_{k+j|k}} = 2Q(x_{k+j|k} - x^{r}_{k+j|k}) + A^T p_{k+j+1|k}$$

$$p_{i_f|k} = \frac{\partial h(x_{i_f|k}, i_f)}{\partial x_{i_f|k}} = 2Q_f(x_{i_f|k} - x^{r}_{i_f|k})$$

where:

$$h(x_{i_f}, i_f) = (x_{i_f} - x^{r}_{i_f})^T Q_f (x_{i_f} - x^{r}_{i_f})$$

Let $\frac{\partial H_{k+j|k}}{\partial u_{k+j|k}} = 0$,

$$\frac{\partial H_{k+j|k}}{\partial u_{k+j|k}} = 2R u_{k+j|k} + B^T p_{k+j+1|k}$$

The optimal control output is:

$$u^*_{k+j|k} = -\frac{1}{2} R^{-1} B^T p_{k+j+1|k}$$

Assuming that:

$$p_{k+j+1|k} = 2K_{k+j, i_f} x_{k+j|k} + g_{k+j, i_f}$$

$$v_{k+j+1|k}$$
The boundary conditions are:

\[
\begin{align*}
K_{j,j_y|k} &= Q_j \\
G_{j,j_y|k} &= -Q_j x_{j_y|k}
\end{align*}
\]  

(20)

Equation (11) is brought into equation (19) and equation (18) is substituted for \( u_{k+j} \),

\[
p_{k+j+1|k} = 2K_{k+j+1,j_y|k}(A_d x_{k+j+1|k}) - \frac{1}{2}B_d R^{-1}B^T p_{k+j+1|k}) + 2g_{k+j+1,j_y|k}
\]  

(21)

To solve equation (21),

\[
p_{k+j+1|k} = [I + K_{k+j+1,j_y|k}B_d R^{-1}B^T]^{-1} \\
x [2K_{k+j+1,j_y|k}A_d x_{k+j+1|k} + 2g_{k+j+1,j_y|k}]
\]  

(22)

Bringing equation (22) into equation (18), the receding horizon optimal solution of the future N steps can be obtained,

\[
u_{k+j|k}^* = -R^{-1}B^T[I + K_{k+j+1,j_y|k}B_d R^{-1}B^T]^{-1} K_{k+j+1,j_y|k}A_d x_{k+j+1|k} + g_{k+j+1,j_y|k}
\]  

(23)

Bringing equation (22) into equation (16), to get:

\[
p_{k+j+1|k} = 2Q(x_{k+j} - x^r_{k+j}) + A^T[I + K_{k+j+1,j_y|k}B_d R^{-1}B^T]^{-1} [2K_{k+j+1,j_y|k}A_d x_{k+j+1|k} + 2g_{k+j+1,j_y|k}]
\]  

(24)

To get:

\[
\begin{align*}
K_{k+j,j_y|k} &= A^T[I + K_{k+j+1,j_y|k}B_d R^{-1}B^T]^{-1} K_{k+j+1,j_y|k}A + Q \\
A^T K_{k+j+1,j_y|k}A &= A^T K_{k+j+1,j_y|k}A + Q \\
B(R + B^T K_{k+j+1,j_y|k}B) &= B(R + B^T K_{k+j+1,j_y|k}B) - B^T K_{k+j+1,j_y|k}A + Q \\
g_{k+j,j_y|k} &= A^T[I + K_{k+j+1,j_y|k}B_d R^{-1}B^T]^{-1} x_{k+j+1,j_y|k} - Q_{x_{k+j}}
\end{align*}
\]  

(25)

Performance indicators in receding horizon based on current time \( k \) and prediction interval \([k, k+N]\):

\[
J(x_{k|k}, x^r, u_{k+|k}) = \sum_{j=0}^{N-1} (x_{k+j} - x^r_{k+j})^T Q (x_{k+j} - x^r_{k+j}) + u_{k+j|k}^T R u_{k+j|k}
\]  

(26)

The optimal solution of receding horizon control with predictive interval \([k, k+N]\):

\[
u_{k+j|k}^* = -R^{-1}B^T[I + K_{k+j+1,k+N}B_d R^{-1}B^T]^{-1} K_{k+j+1,k+N}A x_{k+j} + g_{k+j+1,k+N}
\]  

(27)

where,

\[
\begin{align*}
K_{k+j,k+N|k} &= A^T[I + K_{k+j+1,k+N}B_d R^{-1}B^T]^{-1} K_{k+j+1,k+N}A + Q \\
g_{k+j,k+N|k} &= A^T[I + K_{k+j+1,k+N}B_d R^{-1}B^T]^{-1} g_{k+j+1,k+N}
\end{align*}
\]  

(28)

\[
\begin{align*}
K_{k+N,k+N|k} &= Q_f \\
g_{k+N,k+N|k} &= -Q_f x_{k+N}^r
\end{align*}
\]  

(29)

The optimal control quantity at the current time \( k \) of receding horizon optimal control is \( u_k \). The optimal control quantity can be expressed as:

\[
u_k^* = -R^{-1}B^T[I + K_{k+1,k+N}B_d R^{-1}B^T]^{-1} K_{k+1,k+N}A x_{k+1} + g_{k+1,k+N}
\]  

(30)

\[
K_{k+1,k+N} \quad \text{and} \quad g_{k+1,k+N}
\]  

can be obtained by equation (28).

Receding horizon control can simplify the expression of optimal control quantity by deleting reference values:

\[
u_{k+j}^* = -R^{-1}B^T[I + K_{k+j+1,j_y}B_d R^{-1}B^T]^{-1} K_{k+j+1,j_y}A x_{k+j} + g_{k+j+1,j_y}
\]  

(31)

where,

\[
\begin{align*}
K_{k+j,j_y} &= A^T[I + K_{k+j+1,j_y}B_d R^{-1}B^T]^{-1} K_{k+j+1,j_y}A + Q \\
g_{k+j,j_y} &= A^T[I + K_{k+j+1,j_y}B_d R^{-1}B^T]^{-1} g_{k+j+1,j_y}
\end{align*}
\]  

(32)

\[
\begin{align*}
K_{k+N,j_y} &= Q_f \\
g_{k+N,j_y} &= -Q_f x_{k+N}^r
\end{align*}
\]  

(33)

Substituting \( u_k \) and \( K_{k+1,k+N} \) for \( u_k \) and \( K_{k+1,k+N} \),

\[
u_k^* = -R^{-1}B^T[I + K_{k+1,k+N}B_d R^{-1}B^T]^{-1} K_{k+1,k+N}A x_{k+1} + g_{k+1,k+N}
\]  

(34)

Similarly,

\[
\begin{align*}
K_{k+N,k+N} &= Q_f \\
g_{k+N,k+N} &= -Q_f x_{k+N}^r
\end{align*}
\]  

(35)

By solving the above process, the optimal control output is obtained.

\[
u_k^* = -R^{-1}B^T[I + K_{k+1,k+N}B_d R^{-1}B^T]^{-1} K_{k+1,k+N}A x_{k+1}
\]  

(36)
III. SIMULATION ANALYSIS

In order to verify the control effect of the controller, the simulation results are compared with the proportional control PID scheme and the uncontrolled scheme. The three groups of simulation experiments are carried out at a constant speed of 100 km/h under the classical double lane change conditions. Figure 4 shows the lateral displacement in the three schemes. It can be clearly seen in the figure that there is a large lag in the uncontrolled scheme when crossing the double-lane curve and the maximum overshoot reaches about 40%, which forms a steeper route with larger errors, and the vehicle is in a state of instability. Compared with PID control, receding horizon control has smaller lag and is more stable when the vehicle completes double lane change, and can make the system tend to be stable state more quickly.

In Figure 5a, it can be seen that the torque input fluctuation of receding horizon steering control is greater than that of PID method, which can achieve more effective and fast control in vehicle steering and further improve steering stability. As can be seen from Figure 5b, the amplitude of the uncontrolled steering angle is much larger than that of the vehicle adopting the control method. From the third seconds, the uncontrolled steering angle has an obvious lag compared with the steering angle using receding horizon control method. The peak overshoot is about 70%. Compared with receding horizon control, the PID control has an obvious lag and a certain amount of overshoot.

The yaw rate of the three schemes are shown in Figure 6. The simulation results show that the receding horizon control method has faster response, higher control accuracy, stronger anti-interference ability and stronger system robustness.

IV. HARDWARE IN THE LOOP EXPERIMENT

In order to further verify the performance of the designed controller, the hardware-in-the-loop experiment is carried out on the simulated driving experiment table. In the established hardware in-loop system, the CarSim embedded with PXI real-time control system is used to simulate the vehicle and send out the road condition. Through PXI’s CAN card, the vehicle state parameters such as torque, steering angle, yaw rate, sideslip angle and speed are transferred to controller based on dSPACE. In dSPACE, the information of vehicle state parameters is used to calculate the required steering moment in real time according to controller designed. Target torque is transmitted to the bottom steering controller, which drives the torque motor to perform the torque calculation and transmits the actual steering angle to CarSim. The steering to HIL is realized through the above steps. The stability of active steering under receding horizon control and PID control strategy is analyzed under the condition of simulating the original working conditions. In experiment, the speed is 100 km/h, and the double lane change condition is adopted. The experimental scheme is shown in Figure 7.

The data collected from the experiment table and the related curves are as follows. Figure 8 is the lateral displacement collected in the experiment. It can be seen that when the uncontrolled vehicle passes through the double lane change path, the formed route is steep, and the vehicle seriously deviates from the ideal path, and the vehicle is in a state of instability. Vehicles imposed with receding horizon
FIGURE 7. Experimental scheme diagram of active steering.

FIGURE 8. Lateral displacement.

FIGURE 9. a. Steering wheel angle. b. Stability control torque.

FIGURE 10. Sideslip angle.

FIGURE 11. RMS value of controlled motor current.

control and PID control can smoothly complete the double lane change condition, vehicles with receding horizon control are closer to the ideal track and can be in a stable state faster.

Figure 9a is the steering wheel angle collected from the test. It can be seen that the uncontrolled steering angle increases sharply at about 3 seconds and has a large lag, the maximum peak overshoot is about 60%, and the vehicle is in an unstable state. The receding horizon control and the PID control can complete the working condition smoothly, but the PID control has a certain lag, while the receding horizon control has a faster response, and better guarantees the stability of the vehicle. Figure 9b is the control torque. From the figure, it can be seen that the relative error of the PID control torque is larger, the control amount in receding horizon is smaller, and it is closer to the ideal trajectory and reduces the driver’s burden.

Figure 10 is the contrast figure of the sideslip angles collected by the experiment. The sideslip angle of the uncontrolled vehicle is large when it crosses the bend, and the vehicle is in a state of instability. The sideslip angles of the vehicle with PID control and receding horizon control can be maintained in a stable range. The sideslip angle of vehicle with receding horizon control is smaller, which provides better control for vehicle stability. When the vehicle runs for 8 seconds, the vehicle with receding horizon control completes the emergency obstacle avoidance and is in a stable state, while the vehicle controlled by PID still has a certain degree of jitter.

Figure 11 shows the RMS of motor current under different control modes. It can be seen from the figure that the RMS value of motor current in receding horizon control is the smallest, so the energy loss of receding horizon control vehicle is the smallest. The comparison shows that the receding horizon control algorithm can effectively suppress noise and has high responsibility.

V. CONCLUSION

In this paper, the main factors affecting the stability of active steering system are studied. The change of system stiffness, disturbance of movement and system parameters...
are not considered. The next step will be to consider more indicators to establish a more eligible control system. The research emphasis of this paper mainly includes the following aspects:

1. The proposed active steering control method with torque superposition solves the disadvantage that the driver of the corner ring cannot interfere with the steering. It alleviates the steering conflict between the driver and the auxiliary steering system to a certain extent.

2. The simulation results and experimental results show that the designed controller can effectively realize the stability control of the vehicle in the actual system, and improve the performance and stability of the system.

3. The receding horizon controller can effectively solve the problems of system model adaptation and external interferences, it has a good robustness and anti-interference ability.

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JIAN WU received the Ph.D. degree in vehicle from the Nanjing University of Aeronautics and Astronautics, Nanjing, China, in 2015. He did two years research at Tsinghua University, as a Post-doctoral Researcher, from 2016 to 2018. He has been an Associate Professor with Liaocheng University, since 2018. His research interests include key technologies of intelligent driving and intelligent driving assistance system hardware in the loop technology.

XIAOPING LIU received the B.S. degree in vehicle engineering from the Liaoning University of Technology, Liaoning, China, in 2019. He is currently pursuing the master's degree in intelligent driving with Liaocheng University. His research interests include intelligent driving control technology and other technologies.

QINGFENG KONG received the B.S. degree in vehicle engineering from the Shandong University of Technology, Shandong, China, in 2019. He is currently pursuing the M.S. degree with Liaocheng University. His research interests include intelligent driving control technology and other technologies.