Yaw stability control of automated guided vehicle under the condition of centroid variation

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Title page

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Biographical notes

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Abstract

The centroid of an automated guided vehicle changes due to the irregular position and unbalanced weight of the merchandises on the load platform, which affects the completion of the handling task between stations in intelligent factories. This paper presents a hierarchical control strategy to improve yaw stability considering centroid variation. Firstly, the vehicle body and hub motor models are established based on dynamics. Secondly a hierarchical controller is designed by using the method of

extension theory, model predictive control (MPC) and sliding mode control. Then based on CarSim and Simulink, the step co-simulation of the low-speed condition of the automated guided vehicle is carried out. Compared with the uncontrolled condition, the maximum deviation of the yaw rate is reduced from 0.58 rad/s to 0.52 rad/s, and the error with the theoretical value is reduced from 16% to 4%; the maximum deviation of the centroid sideslip angle is reduced from -0.84 rad to -0.77 rad, and the error with the theoretical value is reduced from 12% to 3%. Finally, a four-wheel drive and four-wheel steering automated guided vehicle is manufactured to carry out inter station steering experiments in simulated factory environment. The error between simulation and experiment is less than 5%. The results show that the designed controller is effective, and the research can provide theoretical and experimental basis for the low-speed steering control stability of automated guided vehicle.

Keywords Four-wheel steering • extension theory • model predictive control (MPC) • sliding mode control • centroid variation

Declarations

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Consent to publish All authors have read and agreed to the published version of the manuscript.

Yaw stability control of automated guided vehicle under the condition of centroid variation

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Abstract

The centroid of an automated guided vehicle changes due to the irregular position and unbalanced weight of the merchandises on the load platform, which affects the completion of the handling task between stations in intelligent factories. This paper presents a hierarchical control strategy to improve yaw stability considering centroid variation. Firstly, the vehicle body and hub motor models are established based on dynamics. Secondly a hierarchical controller is designed by using the method of extension theory, model predictive control (MPC) and sliding mode control. Then based on CarSim and Simulink, the step co-simulation of the low-speed condition of the automated guided vehicle is carried out. Compared with the uncontrolled condition, the maximum deviation of the yaw rate is reduced from 0.58 rad/s to 0.52 rad/s, and the error with the theoretical value is reduced from 16% to 4%; the maximum deviation of the centroid sideslip angle is reduced from -0.84 rad to -0.77 rad, and the error with the theoretical value is reduced from 12% to 3%. Finally, a four-wheel drive and four-wheel steering automated guided vehicle is manufactured to carry out inter station steering experiments in simulated factory environment. The error between simulation and experiment is less than 5%. The results show that the designed controller is effective, and the research can provide theoretical and experimental basis for the low-speed steering control stability of automated guided vehicle.

Keywords Four-wheel steering • extension theory • model predictive control (MPC) • sliding mode control • centroid variation

1 Introduction

Distributed drive electric vehicles can effectively alleviate the environmental protection and safety problems in the development of automobile industry, and become a popular carrier for many scholars at home and abroad to research vehicle stability control [1,2]. It has the advantages of short transmission chain, high transmission efficiency and compact structure, which provides more possibilities for the overall intelligent by wire control and dynamics control of the chassis.

Four-wheel independent drive and four-wheel independent steering is one of the main forms of distributed drive. Compared with traditional vehicles, it optimizes the torque distribution of each driving wheel, makes full

use of road adhesion, changes the steering direction of front and rear wheels at different speeds, and improves vehicle safety and handling stability [3]. The stability of vehicle steering driving has a certain impact on the safety of the vehicle. The control system of four-wheel drive and four-wheel steering electric vehicle is more integrated and more complex than the traditional vehicle. Therefore, scholars at home and abroad have done a lot of research on the yaw stability of four-wheel steering vehicle and obtained some achievements. The evaluation indexes of yaw stability mainly include yaw rate and sideslip angle of centroid. In reference [4,5], the extension theory, sliding mode control integrated control algorithm and LQR control algorithm are used to control the yaw rate and sideslip angle of centroid at the same time, and the weights of the two kinds of

control are changed in real time according to the actual road conditions. The solved compensation torque is constrained and then distributed to four wheels motors. The test results show that the response speed of vehicle parameters and driving stability are improved. Real time tracking of parameters in ideal state can effectively improve vehicle handling stability and yaw stability. According to the problems to be solved in reference [6-12], the corresponding reference model is established. Different control algorithms are used to track the theoretical value, and the controller parameters are dynamically adjusted according to different steering states to reduce the error between the actual value and the theoretical value.

The above yaw stability research is mainly through optimizing the control algorithm to strengthen the control of yaw rate and sideslip angle of centroid, so as to improve the yaw stability. Another research hot topic of yaw stability control system is control strategy. In many conditions, a single control strategy can't achieve the desired goal, so the application of integrated control can further improve the vehicle yaw stability. In reference [13-16], a control strategy integrating direct yaw control (DYC) with different controls is adopted to make up for the deficiency of tire linear control and improve the handling performance in nonlinear range, so as to reduce the demand of yaw moment and improve vehicle yaw stability. Reference [17], the integrated control strategy of direct yaw control (DYC) and steering angle adaptive compensation is adopted. The control objective is to zero the sideslip angle of centroid. Through the distribution of braking torque and the correction of steering angle, the error between the sideslip angle of centroid and zero value is reduced, and the yaw stability is improved. Reference [18] innovatively combines grey predictive control theory, extension theory and fuzzy control theory to realize "leading control", and preprocesses the yaw rate and

sideslip angle of centroid. The results show that it can effectively guarantee the yaw stability of the vehicle and reduce energy consumption. Reference [19-21] proposed to build a four-wheel steering model based on Ackerman's angle theorem. When the four-wheel steering torque is the same, the steering stability can be improved by changing the wheel angle distribution.

The main object of the above research is the traditional car with constant centroid, and the application scenarios are obstacle free and wide road. It does not take into account the limited movement space, strict speed requirements of the factory environment, the goods placed in an irregular position and the weight imbalance lead to the variate of the centroid of the automated guided vehicle, so it is easy to produce the problem of insufficient steering or excessive steering, which affects the yaw stability, so it is necessary to study this working condition. In order to solve this problem, a control method is proposed to improve the stability of low-speed yaw, which combines extension theory, model predictive control theory, fuzzy control theory and sliding mode control to form a hierarchical control based on model predictive control. According to the stability of the vehicle, the yaw moment of the four driving wheels is dynamically compensated, so as to improve the yaw stability of the vehicle.

The rest of this paper is arranged as follows. In the second chapter, the dynamic model of the automated guided vehicle is established. In the third chapter, the yaw stability controller is designed. In the fourth chapter, the controller is simulated and analyzed. In the fifth chapter, the algorithm verification experiment is carried out. Finally, the conclusion is given in the sixth chapter.

2 Dynamic model

2.1 3-DOF model of vehicle

The stability of the vehicle is mainly determined by the lateral motion and yaw motion, so the vehicle model only needs to consider the longitudinal motion, lateral motion and yaw motion. The established dynamic model is used for model predictive control, and the specification is performed on the basis of describing the vehicle dynamics process to reduce the calculation amount of the control algorithm. In vehicle dynamics modeling, the following idealized assumptions should be made. Firstly, it is assumed that the automated guided vehicle runs on a flat road and ignores the

vertical motion; secondly, it only considers the pure cornering characteristics of the tire and ignores the coupling of transverse and longitudinal tire forces; then, it does not consider the lateral load transfer of the tire; finally, it ignores the influence of transverse and longitudinal aerodynamics on the yaw characteristics of the automated guided vehicle.

Based on the above assumptions, the yaw dynamic model can be obtained, as shown in Fig. 1. O is the instantaneous center; CG is the centroid; L is the distance from the front axle to the rear axle; M_Z is the yaw moment; β is the sideslip angle of centroid.

Fig.1 Top view of vehicle

According to the force balance and moment balance, the motion equation of three degrees of freedom is obtained as follows:

Equation of longitudinal motion

$$m(\dot{v}_x - v_y \gamma) = F_{xfl} \cos \delta_{fl} + F_{xfr} \cos \delta_{fr} + F_{xrl} \cos \delta_{rl} + F_{xrr} \cos \delta_{rr} + F_{yrl} \sin \delta_{rl} + F_{yrr} \sin \delta_{rr} - F_{yfl} \sin \delta_{fl} - F_{yfr} \sin \delta_{fr} \quad (1)$$

Where, m is the mass of the vehicle; $\dot{v}_x$ is the longitudinal acceleration; v_y is the longitudinal velocity of the centroid in the car body coordinate system; γ is the yaw rate; F_{xi} and F_{yi} ($i=fl, fr, rl, rr$) are the longitudinal force and lateral force of the tire respectively; δ_{fl} and δ_{fr} are the left and right steering angles of the front

wheels respectively; δ_{rl} and δ_{rr} are the left and right steering angles of the rear wheels respectively.

Equation of lateral motion

$$m(\dot{v}_y + v_x \gamma) = F_{yfl} \sin \delta_{fl} + F_{yfr} \cos \delta_{fl} + F_{yfr} \sin \delta_{fr} + F_{yfr} \cos \delta_{fr} + F_{yrl} \cos \delta_{rl} + F_{yrr} \cos \delta_{rr} - F_{xrl} \sin \delta_{rl} - F_{xrr} \sin \delta_{rr} \quad (2)$$

Where, $\dot{v}_y$ is the lateral acceleration and v_x is the longitudinal velocity of the centroid in the car body coordinate system.

Equation of yaw motion

$$\begin{aligned}
I_z \dot{\gamma} = & l_f (F_{yfl} \sin \delta_{fl} + F_{yfl} \cos \delta_{fl} + F_{yfr} \sin \delta_{fr} + F_{yfr} \cos \delta_{fr}) \\
& + l_r (F_{yrl} \sin \delta_{rl} - F_{yrl} \cos \delta_{rl} + F_{yrr} \sin \delta_{rr} + F_{yrr} \cos \delta_{rr}) \quad (3) \\
& + d_1 (F_{yfl} \sin \delta_{fl} - F_{yfl} \cos \delta_{fl} - F_{yrl} \sin \delta_{rl} - F_{yrl} \cos \delta_{rl}) \\
& + d_2 (F_{yfr} \cos \delta_{fr} - F_{yfr} \sin \delta_{fr} + F_{yrr} \cos \delta_{rr} + F_{yrr} \sin \delta_{rr})
\end{aligned}$$

Where, I_z is the moment of inertia around the z-axis; $\dot{\gamma}$ is the yaw angular acceleration; L_f and L_r are the distances from the centroid to the front and rear axles respectively; d_1 and d_2 are the distances from the left wheel and the right wheel to the equivalent wheel respectively.

2.2 Hub motor model

In this paper, the hub motor adopts brushless DC motor, and the research focuses on the yaw stability control strategy of four-wheel steering vehicle. The model of hub motor needs the following specifications. Firstly, the eddy current loss and hysteresis loss of the motor are ignored, and the saturation of the motor core is ignored; secondly, the air gap magnetic field distribution is approximately a trapezoidal wave with a flat top angle of 120 degrees; then, the cogging effect is ignored, and the armature conductor is evenly distributed on the armature surface; finally, the freewheeling diode and power transistor are regarded as ideal components.

The voltage equation of hub motor is as follows:

$$\begin{aligned}
\begin{bmatrix} u_A \\ u_B \\ u_C \end{bmatrix} &= \begin{bmatrix} R & 0 & 0 \\ 0 & R & 0 \\ 0 & 0 & R \end{bmatrix} \begin{bmatrix} i_A \\ i_B \\ i_C \end{bmatrix} \\
&+ \begin{bmatrix} L-M & 0 & 0 \\ 0 & L-M & 0 \\ 0 & 0 & L-M \end{bmatrix} \frac{d}{dt} \begin{bmatrix} i_A \\ i_B \\ i_C \end{bmatrix} + \begin{bmatrix} e_A \\ e_B \\ e_C \end{bmatrix} \quad (4)
\end{aligned}$$

$$P_e = e_A i_A + e_B i_B + e_C i_C \quad (5)$$

$$P_e = T_e * \Omega \quad (6)$$

It can be obtained from equations (5) and (6):

$$T_e = \frac{e_A i_A + e_B i_B + e_C i_C}{\Omega} \quad (7)$$

The motion equation of the motor is:

$$T_e - T_l = J \frac{d\Omega}{dt} + B_v \Omega \quad (8)$$

Where, T_e is the electromagnetic torque; Ω is the mechanical angular velocity of the motor; T_l is the load torque; J is the rotor moment of inertia; B_v is the viscous friction coefficient; u_a , u_b and u_c are the ABC three-phase winding voltage; i_a , i_b and i_c are the ABC three-phase current; e_A , e_B and e_C are the ABC three-phase back electromotive force; L is the phase winding self-inductance; M is the phase winding mutual inductance.

Fig.2 Simulink model of hub motor

3 Design of yaw stability controller

The vehicle control strategy is shown in Fig.

3. This paper adopts hierarchical control, which is divided into two layers. The upper controller is used to solve the additional yaw moment required by the yaw stability control, which is

composed of yaw rate controller, sideslip angle of centroid controller, extension joint controller and sliding mode controller; the lower controller is the driving torque distribution controller,

which distributes the additional yaw moment calculated by the upper controller to four driving wheels after constraint.

Fig.3 Vehicle control strategy

By inputting the target speed and steering angle into the ideal reference model and dynamic model, the theoretical values γ^* , β^* and the actual values γ , β of yaw rate and centroid sideslip angle are obtained. The upper controller inputs γ^* , β^* , γ , β , the yaw rate controller, the centroid sideslip angle controller and the extension joint controller outputs the yaw moment ΔM_γ , ΔM_β , $M_{\gamma+\zeta_2\Delta M_\beta}$, where ζ_1 and ζ_2 are the weights of the yaw rate controller and the centroid sideslip angle controller respectively. Then the final additional yaw moment ΔM_z is obtained by sliding mode control. The control domain module in the lower controller receives the inputs of ΔM_z and γ^* , β^* , γ , β from the upper controller, judges the control domain according to the vehicle state, and allocates $T_{di}(i=fl, fr, rl, rr)$ to the four hub motors

through constraint conditions.

3.1 Ideal reference model

When automated guided vehicle turning, we hope that the sideslip angle of centroid can be as small as possible, so the ideal sideslip angle of centroid is zero. The linear two degree of freedom model can effectively reflect the linear relationship between yaw rate, centroid sideslip angle and steering angle, and can be used as an ideal reference model. The ideal reference model of automated guided vehicle can be expressed as:

$$\dot{x}_d = A_d * x_d + B_d * u_d \quad (9)$$

Where,

$$x_d = [\beta_d \quad \omega_r]^T \quad (10)$$

$$A_d = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix} = \begin{bmatrix} \frac{(2K_f + 2K_r)}{mv_x} & \frac{(2aK_f + 2bK_r)}{mv_x^2} - 1 \\ \frac{1}{I_z}(2aK_f + 2bK_r) & \frac{1}{I_z v_x}(2a^2K_f + 2b^2K_r) \end{bmatrix} \quad (11)$$

$$B_d = \begin{bmatrix} b_{11} & b_{12} & b_{13} & b_{14} \\ b_{21} & b_{22} & b_{23} & b_{24} \end{bmatrix} = \begin{bmatrix} -\frac{K_f}{mv_x} & -\frac{K_f}{mv_x} & -\frac{K_r}{mv_x} & -\frac{K_r}{mv_x} \\ -\frac{aK_f}{I_z} & -\frac{aK_f}{I_z} & -\frac{bK_r}{I_z} & -\frac{bK_r}{I_z} \end{bmatrix} \quad (12)$$

$$u_d = [\delta_{fl} \quad \delta_{fr} \quad \delta_{rl} \quad \delta_{rr}]^T \quad (13)$$

Where, K_f is the front wheel cornering stiffness; K_r is the rear wheel cornering stiffness; I_z is the yaw moment of inertia; a and b are the distances from the center of mass of the whole vehicle to the front and rear axles respectively.

Namely

$$\begin{bmatrix} \dot{\beta}_d \\ \dot{\omega}_{rd} \end{bmatrix} = \begin{bmatrix} 0 & 0 \\ 0 & -1/\tau_\omega \end{bmatrix} \begin{bmatrix} \beta_d \\ \omega_{rd} \end{bmatrix} + \begin{bmatrix} 0 \\ G\omega/\tau_\omega \end{bmatrix} \delta_f^* \quad (14)$$

Where, τ_ω is the time constant of inertia link; $G\omega$ is the steady-state gain of yaw rate to wheel angle of ideal vehicle model.

$$G_\omega = \frac{1}{1 + Kv_x^2} \frac{v_x}{a + b} \quad (15)$$

Where, K is the steering characteristic stability factor of the vehicle.

Considering that the different adhesion coefficient of the road will limit the tire force, the yaw rate of the vehicle needs to meet the requirements

$$|\omega_{rd}| = \left| \frac{0.85\mu g}{v_x} \right| \quad (16)$$

Where, μ is the road adhesion coefficient.

3.2 Model predictive control

Model predictive control (MPC) is a feedback control strategy, according to the established model to predict the future state of the system. The error between the predicted output and the actual output can be reduced by the rolling optimization method, and the optimization problem can be continuously updated to achieve the optimal control. This paper combines model predictive control, fuzzy control theory and extension theory to form fuzzy extension control based on model predictive control. The control flow chart is shown in Fig. 4.

Fig. 4 model predictive fuzzy extension controller

In Fig. 4. Firstly, the difference $\Delta\gamma$ and $\Delta\beta$ between the MPC predicted values γ, β and the theoretical values γ^*, β^* are taken as the inputs

of the yaw rate fuzzy controller and the centroid sideslip angle fuzzy controller respectively, $\Delta\gamma$ and β as inputs of extension joint controller.

Secondly, output to the lower controller through sliding mode control. Then, through the lower controller, the additional yaw moment ΔM_z is converted into driving force and distributed to the vehicle dynamics model. Finally, MPC refreshes and optimizes the predicted value of the model state until the optimal control is realized.

3.3 Design of upper controller

According to the extension theory, the automated guided vehicle can be divided into three states when turning at low speed: stable region, single control region and joint control region. In the stable region, the automated guided vehicle runs smoothly without control; In the single control domain, the automated guided vehicle gradually loses stability or tends to lose stability, and the yaw rate controller starts to work; In the joint control domain, the automated guided vehicle has become unstable, and the yaw rate controller and the centroid sideslip angle controller work at the same time, in which the control weight is determined by the extension theory. The division of control domain is shown in Fig. 5.

Fig.5 Control domain division

Firstly, the fuzzy controller of yaw rate is designed. When the automated guided vehicle transports the goods in the workshop, the randomness of the goods' position and weight causes the vehicle's yaw rate to be too large or too small, which leads to the instability of the body when steering. The controller uses the combination of model predictive control and

fuzzy control to modify the yaw rate. The membership functions of input and output fuzzy subsets are shown in Fig. 6 and Fig. 7. The fuzzy control rules of yaw rate controller are shown in Tab 1.

Fig.6 Membership function of input fuzzy subsets of yaw rate controller

Fig.7 Membership function of output fuzzy subsets of yaw rate controller

In Fig. 6 and Fig. 7, the membership functions of fuzzy subsets of input and output of yaw rate controller adopt triangle function. The input is the difference $\Delta\gamma$ between the ideal yaw rate and the predicted yaw rate and the change rate of the difference $\Delta\dot{\gamma}$, the output is the additional yaw moment ΔM_y . The abscissa is the domain, and the ordinate is the degree of membership. According to the system model, the input domain is $[-6, 6]$, the output domain is $[-3, 3]$, and the fuzzy sets are all [NB, NM, NS, ZO, PS, PM, PB].

Tab1 Fuzzy control rules of yaw rate controller

ΔM_y	$\Delta\gamma$	NB	NM	NS	ZO	PS	PM	PB
NB		PB	PB	PM	PM	PS	ZO	ZO
NM		PB	PB	PM	PS	PS	ZO	NS
NS		PM	PM	PM	PS	ZO	NS	NS
ZO		PM	PM	PS	ZO	NS	NM	NM
PS		PS	PS	ZO	NS	NS	NM	NM
PM		PS	ZO	NS	NM	NM	NM	NB
PB		ZO	ZO	NM	NM	NM	NB	NB

Taking the yaw rate as the control parameter, the 49 fuzzy inference rules in Tab 1 are designed by repeatedly adjusting the simulation experiment. Input the data in Table 1 into Matlab, input the data in Table 1 into Matlab,

use "Mamdani" type reasoning, area center of gravity method to de-fuzzify, and get the additional yaw moment response diagram of yaw rate controller, as shown in Fig. 8.

Fig.8 Additional yaw moment response diagram of yaw rate controller

In Fig. 8, the x-axis and y-axis are the error and error change rate between the actual value and the theoretical value of the yaw rate, and the z-axis is the output additional yaw moment. It can be seen that the output surface of the additional yaw moment in the domain can be seen directly as the yaw rate controller changes with the working conditions.

Secondly, the fuzzy controller of centroid sideslip angle is designed. When the automated guided vehicle transports goods between two stations in the factory, the randomness of the goods position makes the automated guided vehicle produce understeer and oversteer in the process of low-speed steering, and deviate from the expected path. The controller uses the combination of MPC and fuzzy control to modify the sideslip angle of centroid. The membership function of input and output fuzzy subsets is shown in Fig. 9 and Fig. 10. The fuzzy control rules of centroid sideslip angle controller are shown in Tab 2.

Fig.9 Membership function of input fuzzy

subsets of centroid sideslip angle controller

Fig.10 Membership function of output fuzzy subsets of centroid sideslip angle controller

In the Fig.9 and Fig.10, the membership functions of fuzzy subsets of input and output of centroid sideslip angle controller adopt double S-type function. The input is the difference $\Delta\beta$ between the ideal and predicted centroid sideslip angles and the change rate of the difference $\Delta\dot{\beta}$, and the output is the additional yaw moment ΔM_β . The abscissa is the domain, and the ordinate is the degree of membership. According to the system model, the input domain is $[-6, 6]$, the output domain is $[-3, 3]$, and the fuzzy sets are all [NB, NM, NS, ZO, PS, PM, PB].

Tab2 Fuzzy control rules of centroid sideslip angle controller

$\Delta\beta$	NB	NM	NS	ZO	PS	PM	PB
$\Delta\dot{\beta}$							
NB	NB	NB	NM	NM	NS	ZO	ZO
NM	NB	NB	NM	NS	NS	ZO	ZO
NS	NB	NM	NS	NS	ZO	PS	PS
ZO	NM	NM	NS	ZO	PS	PM	PM
PS	NM	NS	ZO	PS	PS	PM	PB
PM	ZO	ZO	PS	PS	PM	PB	PB
PB	ZO	ZO	PS	PM	PM	PB	PB

Taking the sideslip angle of centroid as the control parameter, 49 fuzzy inference rules in Tab 2 are designed through repeated adjustment of simulation experiment. Input the data in Tab 2 into Matlab, use "min-max" type reasoning and center of gravity method to get the additional yaw moment response diagram of yaw rate controller, as shown in Fig.11.

Fig11 Additional yaw moment response diagram of centroid sideslip angle controller

In Fig. 11, the x-axis and y-axis are the error and error change rate between the actual value and the theoretical value of the centroid sideslip angle, and the z-axis is the output additional yaw moment. It can be seen that the output surface of the additional yaw moment in the domain can be seen directly as the centroid sideslip angle controller changes with the working conditions.

Thirdly, the extension joint controller is established. In order to realize extension joint control, it is necessary to determine the size of extension set, that is, the stability boundary of automated guided vehicle. Then, the weight distribution of yaw rate and centroid sideslip angle is realized in the joint control domain. According to the following steps, the extension joint controller is established.

1) the control quantity is selected. Yaw rate and sideslip angle of centroid are the main performance indexes to evaluate vehicle driving stability. The yaw rate represents the stability of the vehicle when turning, and the sideslip angle of the centroid shows the deviation of the actual running track from the expected path. In this paper, the difference $\Delta\gamma$, $\Delta\beta$ between the predicted γ , β of MPC model and the theoretical values of γ^* , β^* of ideal reference model are selected as the control variables, and the vehicle driving state is divided into stability region, single control region and joint control region.

2) the extension set is divided. Select the value β^* predicted by the MPC model based on the actual centroid slip angle as the abscissa, the difference $\Delta\gamma$ between the value γ predicted by the actual yaw rate and the ideal yaw rate γ^* calculated by the linear 2-DOF model is used as the ordinate. Among them, the tolerance zone division method is adopted for the yaw rate deviation of the ordinate: Stable region $|\Delta\gamma| < |\xi_1\gamma^*|$; Single control domain $|\xi_1\gamma^*| \leq |\Delta\gamma| \leq |\xi_2\gamma^*|$; Joint control domain $|\Delta\gamma| > |\xi_2\gamma^*|$.

Where, ξ_1 and ξ_2 are constants, taking 0.05 and 0.15 respectively. The boundary β_1 of the stability region is calculated by substituting the limit value δ_{max} of the wheel angle in the linear region of the vehicle into the 2-DOF reference model of the double track. The boundary of single control region $\beta_1 = \arctan(0.02\mu g)$ changes with the adhesion coefficient of pavement, where μ is the adhesion coefficient of pavement.

3) the correlation function is constructed. The origin O (0, 0) is the best point in the extension set. Take a point Q in the single control region, connect and extend the two points Q and O , and intersect with the boundary of the stable region and the joint control region at points Q_1 , Q_2 , Q_3 and Q_4 , as shown in fig. 5. It can be seen that the line segment OQ is the shortest distance that Q is close to the best point. The two-dimensional extension set is transformed into a one-dimensional extension set to calculate the extension distance. The one-dimensional extension set is shown in Fig. 12.

Fig.12 One dimensional extension set

Let the set of stable regions be X_w , the set of single control regions be X_d , the extension distance from point Q to the stable region is $\rho(Q, X_w)$, the extension distance from point Q to the single control region is $\rho(Q, X_d)$, and Depending on the position of the Q point, the extension distance from the point to the control domain is different. Taking the extension distance from point Q to single control field as an example.

$$\rho(Q, X_d) = \begin{cases} |QQ_2|, Q \in \langle -\infty, Q_2 \rangle \\ -|QQ_2|, Q \in \langle Q_2, 0 \rangle \\ |QQ_2|, Q \in \langle 0, Q_3 \rangle \\ |QQ_2|, Q \in \langle Q_3, +\infty \rangle \end{cases} \quad (17)$$

The correlation function is

$$K(S) = \frac{\rho(Q, X_d)}{D(Q, X_d, X_w)} \quad (18)$$

Where, $D(Q, X_d, X_w) = \rho(Q, X_d) - \rho(Q, X_w)$

4) the weight of joint control is determined.

When it is in the stable region, the vehicle runs smoothly and does not need extra control, at this time, $\xi_1 = 0$, $\xi_2 = 0$. When it is in the single control area, the vehicle gradually loses stability, and only need to apply the yaw rate control alone to make the car run smoothly, at this time, $\xi_1 = 1$, $\xi_2 = 0$. When it is in the joint control area, the vehicle is unstable, and the yaw rate control and sideslip angle control need to be applied at the same time, at this time, $\xi_1 = |K(s)| / 100$, and $\xi_2 = 1 - |K(s)| / 100$.

Fourthly, the sliding mode controller is designed. Sliding mode control is mainly to select the sliding mode surface. Different switching surfaces have different control effects. This paper studies the simultaneous control of the yaw rate and the center of mass side slip angle, so the following switching surface is constructed:

$$s = \xi_1(\gamma - \gamma_d) + \xi_2(\beta - \beta_d) \quad (19)$$

This control method can not only track the ideal yaw rate quickly, but also keep the centroid sideslip angle not far from the theoretical value, ensuring the steering stability of the vehicle when turning.

Sliding mode control is theoretically a perfect control method, but the resulting chattering phenomenon makes it difficult to apply in practice. In this paper, the sliding mode control based on the exponential reaching law is adopted, which gradually converges to the sliding mode switching surface from the starting point, which effectively reduces the chattering phenomenon of sliding mode control. The constructed reaching law is as follows:

$$\dot{s} = -\varepsilon \operatorname{sgn}(s) - \eta s, \varepsilon > 0, \eta > 0 \quad (20)$$

Where, ε is the approaching rate, $\operatorname{sgn}(s)$ is the sign function, and η is the exponential coefficient.

Finally, the additional yaw moment is

calculated. The derivation of equation (19) leads to the conclusion that:

$$\dot{s} = \xi_1(\dot{\gamma} - \dot{\gamma}_d) + \xi_2(\dot{\beta} - \dot{\beta}_d) \quad (21)$$

The additional moment of body stability is

$$\begin{aligned} \Delta M_z = & l_f (F_{yfl} \sin \delta_{fl} + F_{yfr} \sin \delta_{fr}) \\ & + l_r (F_{yrl} \sin \delta_{rl} + F_{yrr} \sin \delta_{rr}) \\ & - d_1 (F_{xfl} \cos \delta_{fl} + F_{xrl} \cos \delta_{rl}) \\ & + d_2 (F_{xfr} \cos \delta_{fr} + F_{xrr} \cos \delta_{rr}) \end{aligned} \quad (22)$$

Equation (3) can be transformed into

$$\dot{\gamma} = \frac{1}{I_z} \begin{bmatrix} \Delta M_z + l_f (F_{yfl} \cos \delta_{fl} + F_{yfr} \cos \delta_{fr}) \\ + l_r (F_{yrr} \cos \delta_{rr} - F_{yrl} \cos \delta_{rl}) \\ + d_1 (F_{yfl} \sin \delta_{fl} - F_{yrl} \sin \delta_{rl}) \\ + d_2 (F_{yrr} \sin \delta_{rr} - F_{yfr} \sin \delta_{fr}) \end{bmatrix} \quad (23)$$

Substituting equation (25) into equation (23), get

$$\dot{s} = \xi_1 \left\{ \frac{1}{I_z} \begin{bmatrix} \Delta M_z + l_f (F_{yfl} \cos \delta_{fl} + F_{yfr} \cos \delta_{fr}) \\ + l_r (F_{yrr} \cos \delta_{rr} - F_{yrl} \cos \delta_{rl}) \\ + d_1 (F_{yfl} \sin \delta_{fl} - F_{yrl} \sin \delta_{rl}) \\ + d_2 (F_{yrr} \sin \delta_{rr} - F_{yfr} \sin \delta_{fr}) \end{bmatrix} - \dot{\gamma}_d \right\} + \xi_2 (\dot{\beta} - \dot{\beta}_d) \quad (24)$$

Combined with exponential approach rate, additional torque can be obtained

$$\begin{aligned} \Delta M_z = & - \begin{bmatrix} l_f (F_{yfl} \cos \delta_{fl} + F_{yfr} \cos \delta_{fr}) \\ + l_r (F_{yrr} \cos \delta_{rr} - F_{yrl} \cos \delta_{rl}) \\ + d_1 (F_{yfl} \sin \delta_{fl} - F_{yrl} \sin \delta_{rl}) \\ + d_2 (F_{yrr} \sin \delta_{rr} - F_{yfr} \sin \delta_{fr}) \end{bmatrix} \\ & + \frac{I_z}{\xi_1} \begin{bmatrix} -\varepsilon \operatorname{sgn}(s) \\ -\eta s + \xi_1 \dot{\gamma}_d \\ -\xi_2 (\dot{\beta} - \dot{\beta}_d) \end{bmatrix} \end{aligned} \quad (25)$$

3.4 Design of lower controller

The role of the lower controller is to solve the additional yaw moment obtained by the upper controller, and then convert it into driving force and transmit it to the driving wheel.

The balance formula between the driving force of each wheel and the additional yaw moment is as follows

$$\begin{cases} M_f = L_f (F_{xfl} \sin \delta_{fl} + F_{xfr} \sin \delta_{fr}) \\ \quad - d_1 F_{xfl} \cos \delta_{fl} + d_2 F_{xfr} \cos \delta_{fr} \\ M_r = L_r (F_{xrl} \sin \delta_{rl} + F_{xrr} \sin \delta_{rr}) \\ \quad - d_1 F_{xrl} \cos \delta_{rl} + d_2 F_{xrr} \cos \delta_{rr} \end{cases} \quad (26)$$

$$M_f = iM_z; M_r = (1-i)M_z \quad (27)$$

$$\begin{cases} F_{xfl} + F_{xfr} = j \frac{T_d}{r} \\ F_{xrl} + F_{xrr} = (1-j) \frac{T_d}{r} \end{cases} \quad (28)$$

Where, M_f and M_r are the additional torque of the front and rear wheels respectively; i is the additional torque adjustment coefficient; j is the dynamic adjustment coefficient; T_d is the driving torque of the whole vehicle; r is the tire radius.

According to the above constraints and the additional yaw moment, the longitudinal forces of the four wheels can be calculated:

$$\begin{cases} F_{xfl} = \frac{j \frac{T_d}{r} (L_f \sin \delta_{fl} + d_1 \cos \delta_{fl}) - iM_z}{(d_1 + d_2) \cos \delta_{fl}} \\ F_{xfr} = \frac{j \frac{T_d}{r} (d_2 \cos \delta_{fr} - L_f \sin \delta_{fr}) + iM_z}{(d_1 + d_2) \cos \delta_{fr}} \\ F_{xrl} = \frac{(1-j) \frac{T_d}{r} - \frac{(1-i)M_z}{d_1 + d_2}}{2r} \\ F_{xrr} = \frac{(1-j) \frac{T_d}{r} + \frac{(1-i)M_z}{d_1 + d_2}}{2r} \end{cases} \quad (29)$$

The obtained longitudinal force is transformed into driving torque:

$$T_i = F_{xi} r + I \dot{\omega}_r \quad (30)$$

Where, I is the moment of inertia of the wheels; F_{xi} is the four-wheel drive force.

The calculated four-wheel drive torque is used as the input of CarSim vehicle model, and

the co-simulation platform is built with Simulink to realize the yaw stability research of automated guided vehicle under low-speed condition.

4 Simulation results and analysis

In order to verify the effectiveness of the control method, this paper carries out co-simulation based on CarSim and Matlab/Simulink, establishes the vehicle dynamics model and ideal reference model established above in CarSim environment, establishes the hub motor and controller model in Simulink, and tests the control effect of the control strategy on the yaw stability of automated guided vehicle under low-speed condition. The step condition which is prone to instability under low-speed steering condition is selected for simulation analysis. The parameters of the whole vehicle are shown in Tab 3.

In the vehicle dynamics model, the low-speed steering driving condition is established, and the longitudinal speed is 25km/h, the adhesion coefficient of the ground is 0.3. After the automated guided vehicle normally runs at a constant speed for 3 seconds, it is affected by a 20 degrees (about 0.349 rad) step signal, resulting in the instability of the body of the vehicle. The controller corrects the yaw rate and the sideslip angle of the centroid to restore the stability of the body, and the whole working condition lasts for 20 seconds. The simulation results of low-speed step are shown in Fig. 13.

Tab3 Main parameters of CarSim vehicle model

Name	Value
Vehicle quality (m/kg)	128
Vehicle curb quality (m/kg)	148
Distance from centroid to front axle L_f	0.69

Distance from centroid to rear axle L_r	0.57
Track width (d/m)	0.73
Wheel rolling radius (r/m)	0.10
Yaw moment of inertia $I_z/(kg \cdot m^2)$	70
Tire moment of inertia $I/(kg \cdot m^2)$	0.42

(a) Response of γ under low-speed step condition

(b) Response of β under low-speed step condition

Fig.13 Response at low-speed step condition

It can be seen from Fig. 13. In the first 3s of normal driving, the automated guided vehicle is in the stable region, and γ and β can track the changes of theoretical values well. After 3s, the vehicle starts to turn. Affected by the step signal, γ and β begin to deviate from the theoretical value and gradually increase, reaching the maximum value around 5s. The maximum yaw rate is 0.58rad/s, which exceeds the theoretical value by 16%, there is a hysteresis delay of 0.4s at the maximum value. The maximum value of the centroid sideslip angle is -0.84rad, which is 12% beyond the theoretical value. There is a 0.3s lag delay at the maximum value, and fluctuations appear after the maximum value. It

indicates that the automated guided vehicle has a tendency to lose stability or has already lost stability, and its stability performance is not good. After adding hierarchical control, γ and β can closely track the theoretical value. The errors of γ , β and the theoretical value are maintained at 4% and 3% respectively. The hysteresis delay is small and negligible, which improves the low-speed operating conditions of the unmanned vehicle Yaw stability.

5 Algorithm verification experiment

5.1 Development of prototype vehicle

The prototype vehicle in this experiment is a four-wheel drive and four-wheel steering automated guided vehicle designed and manufactured independently. The physical map is shown in Fig. 14. The vehicle parameters and motor parameters are shown in Tab 4. The automated guided vehicle is equipped with a DSP of model F28335 as the lower computer, and the embedded mini-industrial computer is the upper computer. It uses the hub motor as the driving source and the stepping motor as the steering source. The four-wheel driving, braking and Steering; the load platform uses the push rod motor as the power source to realize up and down movement, which is convenient for

loading and unloading goods.

During the experiment, the vehicle's position and attitude information, such as the vehicle's two-dimensional position, vehicle speed, yaw rate, and centroid sideslip angle, are determined by the high-precision INS inertial navigation system. The real-time parameters of the hub motor, such as the motor speed and torque, are measured by the motor encoder and torque sensor respectively. The signal is collected in real time by the DSP single-chip microcomputer and fed back to the upper computer. The upper computer issues control instructions to the single-chip microcomputer through serial communication, and the single-chip directly controls the corresponding equipment according to the command.

Fig.14 Physical map of automated guided vehicle

Tab 4 Main parameters of vehicle and wheel motor

Parameter	Value
Overall dimensions of the vehicle (long/wide/high) /mm	89/69/56
wheelbase /mm	70
Tire diameter /mm	200

Track width /mm	58
Vehicle quality /kg	128
Braking type	Disc electric brake
Rated output power /W	50-300
Rated speed /RPM	400
Rated torque / (N·m)	5
Rated voltage /VDC	24-48

5.2 Experimental results and analysis

The operating environment of the automated guided vehicle research in this paper is the low-speed yaw stability study between two workstations in the factory. For safety reasons, the road with an adhesion coefficient of 0.6 is selected for this test, and the average vehicle speed is maintained at 25Km/h, the station layout and vehicle operating condition route are shown in Fig. 15 and Fig.16. The two chairs in the picture represent the loading area and the unloading area respectively. The automated guided vehicle loads from the loading area, then drives out of the loading area and turns into the unloading area for unloading. The black line is the driving path of the automated guided vehicle.

Fig.15 Station layout

Fig.16 Experimental trajectory

The automated guided vehicle enters the test site at a constant speed and arrives at the loading area to load the goods. After 3 seconds, it starts to turn, and after the 20s, it ends turning and arrives at the loading area for unloading. The experimental results are shown in Fig. 17.

(a) Response of γ under low-speed steering test

(b) Response of β under low-speed steering test

Fig.17 Response at low-speed steering test

It can be seen from Fig. 17 that after the automated guided vehicle is loaded with goods, the irregular placement of the goods causes the centroid of the vehicle to variate. When the steering starts in 3s, the yaw stability begins to fluctuate and the controller starts to work. From 4s to 7s, the automated guided vehicle is in the single control region, and the yaw rate controller works alone; at the 8s, the automated guided vehicle is in the joint control region, and the yaw rate controller and the centroid sideslip angle controller work together; From 9s to 17s, the automated guided vehicle returns to the single control region, and the centroid sideslip angle controller stops working; from 18s to 20s, the automated guided vehicle gradually enters the stable region and the yaw rate controller stops working. Experiments show that the input of the controller, the yaw rate and the centroid sideslip angle can closely track the theoretical value, and the deviation from the theoretical value at the 8s peak value is kept within 5%, which improves the low-speed steering of the automated guided vehicle yaw stability, also proves the effectiveness of the control strategy.

6 Conclusion

(1) This paper takes the four-wheel drive and four-wheel steering automated guided vehicle as the research object, and establishes

the three-degree-of-freedom model of vehicle and the hub motor model to provide a theoretical basis for establishing the model in Simulink.

(2) Aiming at the problem of the degree of completion of the automated guided vehicle's handling tasks between stations, a control strategy combining model predictive control (MPC), sliding mode control and extension theory is designed. The upper layer is the additional yaw moment solution layer, which calculates the compensation moment when the automated guided vehicle is turning. The lower layer is the driving force distribution layer, which reasonably distributes the compensation torque calculated by the upper controller to the four wheels to ensure the yaw stability control of the automated guided vehicle at low speed.

(3) Establish a joint simulation platform based on Carsim and Matlab/Simulink for testing. The vehicle model and simulation conditions are established in the Carsim environment, and the motor and controller models are established in Simulink. In the uncontrolled state, the yaw rate and the centroid sideslip angle exceed the theoretical value of the reference model by 16% and 12% respectively. After adding layered control, the yaw rate and center of mass side slip angle can closely track the theoretical value, and the error remains within 4%. Experimental results show that the established joint control strategy is reasonable under low-speed steering conditions and can

improve the yaw stability of the unmanned guided vehicle.

(4) Carry out real-vehicle experiments on the control strategy, and develop a four-wheel drive and four-wheel steering automated guided prototype vehicle. Simulating the steering and handling task between two stations in the factory, the experiments show that the yaw rate and the centroid sideslip angle fluctuate within 5% of the theoretical value. The designed control strategy can reduce the delay, speed up the response, and improve the yaw stability of the automated guided vehicle. The experimental results are basically consistent with the simulation results, verifying the correctness of the algorithm.

7 Declaration

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Data and materials availability

The datasets supporting the conclusions of this article are included within the article.

Authors' contributions

The author's contributions are as follows: Wei Liu was in charge of the whole trial; Qingjie Zhang wrote the manuscript; Yidong Wan, Yue Yu, Ping Liu, Jun Guo assisted with sampling and laboratory analyses.

Conflict of interests

The authors declare no competing financial interests.

Ethics approval

Not applicable.

Consent to participate

Authors agree to the authorship order.

Consent to publish

All authors have read and agreed to the published version of the manuscript.

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Figures

Figure 1
Top view of vehicle

Figure 2

Figure 3

Vehicle control strategy

Figure 4

model predictive fuzzy extension controller

Figure 5

Control domain division

Figure 6

Membership function of input fuzzy subsets of yaw rate controller

Figure 7

Membership function of output fuzzy subsets of yaw rate controller

Figure 8

Additional yaw moment response diagram of yaw rate controller

Figure 9

Membership function of input fuzzy subsets of centroid sideslip angle controller

Figure 10
 Membership function of output fuzzy subsets of centroid sideslip angle controller

Figure 11
 Additional yaw moment response diagram of centroid sideslip angle controller

Figure 12

One dimensional extension set

(a)Response of γ under low-speed step condition

(b)Response of β under low-speed step condition

Figure 13

Response at low-speed step condition

Figure 14

Physical map of automated guided vehicle

Figure 15

Station layout

Figure 16

Experimental trajectory

(a) Response of γ in low-speed steering test

(b) Response of β in low-speed steering test

Figure 17

Response at low-speed steering test