

# TRACKING CONTROL OF FRONT WHELL ANGLE BASED ON SLIDING MODE ACTIVE DISTURBANCE REJECTION FOR TRACTOR AUTOMATIC STEERING

## 面向自动转向的拖拉机前轮转角滑模自抗扰跟踪控制

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### ABSTRACT

A sliding mode active disturbance rejection tracking control scheme based on the Smith predictive control structure is proposed for the front steering wheel of tractors, aiming to address the challenges of time delay and unknown disturbances in the angle tracking process. In the control system design, the effects of time delay and disturbances on the angle tracking performance are thoroughly considered, based on the established steering system model. A time delay processing structure is designed based on the Smith predictive control principle, and linear extended state observer is designed to realize synchronous estimation for the un-delayed angle output and the controller output. A simple Sliding Mode Controller (SMC) is introduced to replace the linear feedback controller in active disturbance rejection control (ADRC) to improve the accuracy and disturbance rejection performance. Simulation charts and performance evaluating indexes show that the proposed control method has obvious advantages in tracking accuracy, robustness and disturbance rejection in comparison with other methods. The angle tracking error is no more than 0.055rad even under the interference of white noise signal. The proposed method effectively mitigates the impact of time delay and disturbance, it can significantly enhance the dynamic response performance of the steering system.

### 摘要

针对拖拉机前轮转角控制系统存在滞后、易受未知干扰等不确定因素影响问题，提出一种基于史密斯预估控制结构的滑模自抗扰转角跟踪控制方案。根据确定的转向系统模型，考虑延迟环节、内外部干扰对转角跟踪效果的影响，基于史密斯控制预估思想，设计时滞处理结构，通过线性扩展状态观测器实现对无滞后角度输出和控制器输出的同步估计。设计简单的滑模控制器替代自抗扰控制中的线性反馈控制，进一步提高转向角跟踪控制的精度和扰动抑制能力。仿真曲线及性能评价指数显示，与其它控制方法相比，本文提出的控制方法在跟踪精度、鲁棒性和抗扰动性等各方面都具有明显的优越性，即使在白噪声信号干扰下，对设定角度的跟踪误差也不超过 0.055rad。结果表明，提出的方法能够有效应对系统中存在的时滞及扰动对转角跟踪效果造成的影响，可进一步提高转向系统的动态响应性能。

### INTRODUCTION

As is well known, the environment sensing system, path planning system, and automatic steering system constitute the main components of an autonomous navigation tractor (Yang *et al.*, 2024). The automatic steering system is a key component for achieving automatic control. Based on the automatic steering system, the technologies of automatic navigation (Chen *et al.*, 2024; Song *et al.*, 2025; Wang *et al.*, 2023) and stability control (Gao *et al.*, 2023; Song *et al.*, 2022; Wang *et al.*, 2024) of tractors have become research hotspots in the field of agricultural machinery. Usually, in order to achieve precise tracking for the preset steering angle, the steering controller controls the steering actuator to generate the corresponding torque according to the command from the upper-level controller (Sun *et al.*, 2023). The tracking performance of the steering system is of crucial significance for realizing functions of the upper-level controller (Yang *et al.*, 2022).

The field of automatic steering system is well established and the available literature on the topic is numerous (Yin *et al.*, 2019). Generally speaking, the research of automatic steering system mainly focuses on two aspects, namely driving mechanism and control methods. At present, there are mainly three types of ways for generating the driving force of tractor steering actuators, that are electro-hydraulic drive, steering wheel motor drive and column motor drive.

Among these methods, electro-hydraulic drive is known for its high power and quick response, such as Lee et al. developed and implemented an electro-hydraulic steering system, in their study, a proportional feedforward control algorithm was adopted to accurately compensate for the non-linear characteristics existing in the current system (Lee et al., 2022). But the electro-hydraulic drive method is associated with significant challenges in terms of modification complexity and incurs relatively high costs. The steering wheel motor drive method is easy to install and application, for example, in Zhao et al. study, steering wheel motor is adopted to track the desired angle to avoid obstacle. Conversely, the column motor drive method is highly regarded for its straightforward structure and ease of replication (Zhao et al., 2023). Lu et al. designed a front wheel tracking system based on electro-hydraulic drive fore to realize a dual-mode control scheme (Lu et al., 2023). An et al. designed an electric automatic steering system incorporating a brushless motor and reducer assembly (An et al., 2024). Yang et al. designed an auxiliary steering system including auxiliary wheel mechanism, hydraulic drive system and electro-hydraulic servo controller to extend the steering range (Yang et al., 2022). To improve the steering performance, hydraulic characteristics of the electrohydraulic proportional valve and steering trapezoidal mechanism in the automatic steering system were tested and analyzed (Xu et al., 2024). An exemplary steering control system must possess the capability to rapidly and precisely track the preset angle. The method of steering control has a direct impact on the steering performance of tractors. To realize autonomous steering, a substantial number of scholars have conducted extensive research on various control algorithms. The PID controller has been widely used in the field of tractor steering control because of its simple structure and ease of implementation in engineering (Cheng et al., 2024). Ye et al. analyzed the effect of the time delay and external disturbance on the steering performance and built the time delay dynamic models of automatic steering system. They proposed control scheme based on PID method to improve the stability and accuracy for the automatic steering system (Ye et al., 2020). He et al. adopted power steering motor and designed PID controllers to control the inner speed and outer angle respectively using cascade control scheme (He et al., 2023). Bhosinak et al. proposed a control scheme adopting fuzzy logic algorithm to optimize the PID parameters (Bhosinak et al., 2022). Wang et al. developed a front-wheel steering angle compensation controller based on an enhanced second-order sliding mode algorithm (Wang et al., 2023).

The above control methods have all demonstrated good effects and significantly promoted the development and application of tractor automatic control technology. However, due to factors such as mechanical friction and viscous damping inside the steering actuator may cause time delay during the steering control process, they will affect the stability and dynamic performance of steering control. In addition, during the operation of tractors, changes in the working environment and steering loads can also bring unknown disturbances for the steering control, and in severe cases, they may even lead to oscillation or divergence of the control output, which add the complexity in the process of adjusting parameters for PID controller.

ADRC is favored because of simple parameter adjustment and easy engineering implementation. And SMC is widely employed because of its fast dynamic response and insensitivity to perturbations. Inspired by the advantages of the two methods, a steering control scheme is proposed in this study, which provides superior disturbance rejection performance and stronger robustness. The contributions of this article will be summarized as follows: the influence of time delay and disturbance on the steering tracking effect is fully considered, and a sliding mode active disturbance rejection steering tracking control scheme based on Smith predictor structure is designed taking the advantages of active disturbance rejection control and sliding mode control. The integration of the Smith predictor control structure and the linear extended state observer enables the synchronous estimation of the controller output and the non-delayed system output. This combination effectively estimates, compensates and rejects both internal and external disturbances within the system, which enhances the robustness and precision of the control mechanism. The controller design method proposed in this paper is simple, which is convenient for implementing the control algorithm in the hardware system with limited resources. The experimental results demonstrate that the control method proposed in this study exhibits notable characteristics. It can achieve rapid tracking speed and high accuracy while maintaining robust performance, particularly in terms of disturbance rejection.

## MATERIALS AND METHODS

### Structure and modeling of steering system

As the actuating mechanism responsible for adjusting the position or posture of the tractor, the steering tracking control system serves as the foundation for achieving automatic navigation or stability control.

When the upper-level controller determines the desired steering angle based on the control requirements, the steering tracking control system is responsible for accurately tracking this desired steering angle. This study adopts the electro-hydraulic unit to provide the steering driving force and presents the overall block diagram of the system shown in Fig.1, which consists of four main parts: the steering controller, the electromagnetic proportional reversing valve, the steering mechanism and the angle transducer. Upon receiving the desired steering command from the upper-level controller, the steering controller executes calculations via the tracking control algorithm and subsequently outputs a precise voltage signal. This voltage signal modulates both the opening direction and the orifice area of the electromagnetic proportional reversing valve, thereby controlling the velocity of the piston in the steering cylinder. With the movement of the piston, the steering connecting rod mechanism moves accordingly, which realizes the adjustment of the steering wheels. Meanwhile, a closed-loop feedback control system is established with an angle transducer to ensure accurate tracking of the desired angle.

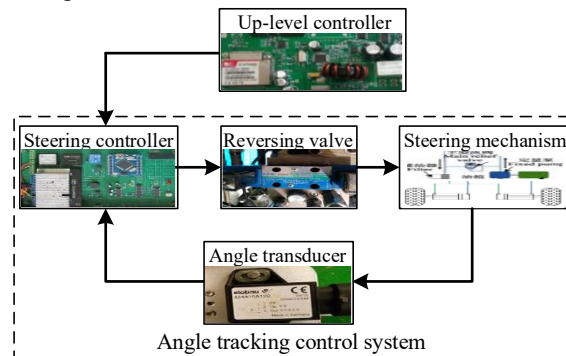


Fig. 1 - Structure diagram of tractor angle tracking system

The mathematical description of the steering system is regarded as a function of the angle of the tractor front wheel  $\delta(s)$  to the control input voltage  $u(s)$ . The input voltage controls the change rate of the front wheel angle through the solenoid valve, the cylinder and the connecting rod mechanism. The electromagnetic reversing valve can be regarded as a second-order element. The cylinder and the connecting rod mechanism can be regarded as a first-order element, and the function from the rotational speed to the rotational position is regarded as a pure integrator element. As a result, the mathematical model of the steering system is obtained as:

$$P_m(s) = \frac{\delta(s)}{u(s)} = \frac{K}{s(Ts + 1)(s^2 + 2\xi\omega_n s + \omega_n^2)} \tag{1}$$

The natural frequency of the electromagnetic directional valve is usually much higher than that of the steering system. Based on this premise, the steering system model, as shown in Eq (1), can be simplified to a second-order model. To improve the accuracy of the model, the influence of friction and steering load are considered, and delay is introduced in the process of the steering. Therefore, with the control voltage as the input and the front wheel as the output, the open-loop transfer function of the steering system can be simplified and expressed as follows:

$$P_m(s) = Ke^{-\theta s} / s(Ts + 1) \tag{2}$$

**Controller design**

For the steering system model established in Eq. (2), a novel sliding-mode ADRC strategy integrated with a Smith predictor structure is proposed, as illustrated in Fig. 2. The control architecture is designed to address the inherent time delay and external disturbances in the steering dynamics, thereby enhancing angle tracking performance and system robustness.

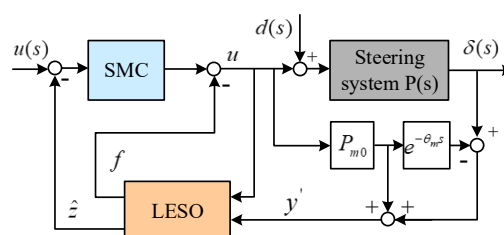


Fig. 2 - Control block diagram of angle tracking system

In the proposed control scheme, the block labeled  $P_{m0}$  denotes the delay-free dynamics of the steering system, while  $\theta$  and  $\theta_m$  represent the delay existing in the steering system and its model respectively. The actual steering system is denoted by  $P(s) = P_0 e^{-\theta s}$ , and its nominal model is represented by  $P_m = P_m e^{-\theta_m s}$ . The reference input  $u(s)$  corresponds to the voltage signal associated with the desired steering angle, and  $\delta(s)$  denotes the actual front wheel steering angle. The term  $d(s)$  encapsulates the lumped disturbances acting on the steering system, including external perturbations and unmodeled dynamics. The control signal generated by the controller is denoted by  $u$ . From the control scheme, it is evident that to address the time delay occurring in the control process, the adoption of an improved Smith predictor control structure ensures that the controller output and the system output angle can be synchronously fed into the extended state observer. This allows for the effective estimation and compensation of the time delay, thereby improving the overall performance of the steering tracking control system. This is conducive to achieving precise estimation of the system state and disturbances, thereby it can enhance the performance of disturbance rejection for the angle tracking system. At the same time, a SMC is introduced to improve the tracking performance of steering angle control and enhance the robust stability of the system.

For the steering system model established in Eq. (2), its second-order state-space representation can be explicitly formulated as follows:

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = bu + f \\ y = x_1 \end{cases} \quad (3)$$

where  $f = -\frac{x_2}{K}$ ,  $b = K/T$ . The state variable  $x_1$  represents the front wheel angle, and  $x_2$  represents the front wheel steering speed. Here,  $f$  is taken as the total disturbance of the system, which includes the unmodeled parts within the system and the external disturbances.

Extended state observer design

In accordance with the design philosophy of the extended state observer within the ADRC framework, the total disturbance is augmented as an additional state variable. This leads to the construction of a third-order extended state-space model:

$$\begin{cases} \dot{X} = AX + Bu + Ef \\ y = CX \end{cases} \quad (4)$$

where  $X = [x_1 \ x_2 \ x_3]^T$ ,

$$A = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ 0 & 0 & 0 \end{bmatrix}, \quad B = \begin{bmatrix} 0 \\ b \\ 0 \end{bmatrix}, \quad C = \begin{bmatrix} 1 \\ 0 \\ 0 \end{bmatrix}^T, \quad E = \begin{bmatrix} 0 \\ 0 \\ 1 \end{bmatrix}.$$

Depending on the treatment of the estimation error, extended state observers can be categorized into linear and nonlinear configurations. A typical nonlinear structure is given in the literature (Fu et al., 2017).

$$\begin{aligned} \dot{Z} &= f(Z, u, y) \\ f_i(Z, u, y) &= \begin{cases} z_{i+1} - l_i \psi_i(z_1 - y) & i = 1, 2, \dots, n \\ -l_{n+1} \psi_{n+1}(z_1 - y) & i = n + 1 \end{cases} \\ f_n(Z, u, y) &= z_{n+1} - l_n \psi_n(z_1 - y) + bu \end{aligned} \quad (5)$$

where  $z_i (i = 1, 2, \dots, n + 1)$  are the estimated variable of the state parameters by the observer,  $l_i (i = 1, 2, \dots, n + 1)$  are the parameters of the observer,  $\psi_i(e) = K(e) \cdot e$  is the tracking value error function, where:

$$K(e) = \begin{cases} 1 / \delta^{1-\alpha} & |e| \leq \delta \\ |e|^{\alpha-1} / \delta^{1-\alpha} & |e| > \delta \end{cases} \quad (6)$$

By introducing the adaptive gain  $K(e)$ , the convergence speed of the extended state observer can be significantly improved, and the estimated error can be reduced.  $\alpha$  is a nonlinear parameter, while  $\delta$  is the boundary parameter. Compared with the nonlinear extended state observer, design method for a linear state observer is simpler in principle and easier to apply in engineering practice. Therefore, in this study, a standard linear observer is selected to estimate the dynamic state and disturbance in the steering system.

$$\begin{cases} \dot{Z} = A_1 Z + B_1 u + L_1 (y' - \hat{y}') \\ \hat{y}' = C_1 Z \end{cases} \quad (7)$$

$$A_1 = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ 0 & 0 & 0 \end{bmatrix}, B_1 = \begin{bmatrix} 0 \\ b \\ 0 \end{bmatrix}, L_1 = \begin{bmatrix} l_1 \\ l_2 \\ l_3 \end{bmatrix}, C_1 = [1 \ 0 \ 0]$$

where  $z_1, z_2$  are the estimated state variables through the extended observer respectively,  $z_3$  is the expanded variable by the observer, it is the estimation of the generalized uncertain function  $f$ . And  $y' = uP_{m0} + u(P_0e^{-\theta s} - P_{m0}e^{-\theta_m s})$ , under perfect model matching, it replicates the un-delayed plant output, whereas under parametric or structural mismatch it instantaneously quantifies the attendant model uncertainty. Through the state estimation, the performance of disturbance rejection of the system to uncertain disturbances as well as the tracking accuracy to the desired front wheel angle can be improved.  $L_1(y - \hat{y}')$  are the closed loop correction factors, they can adaptively adjust the unknown parameters of the system and external disturbances based on the observer parameters.  $L_1 = [l_1 \ l_2 \ l_3]^T$  is the parameter vector of the extended state observer, and the observer parameters can be determined using the standard design method:

$$L_1 = [\beta_1 \omega_0 \ \beta_2 \omega_0^2 \ \beta_3 \omega_0^3]^T \tag{8}$$

where  $\omega_0$  denotes the bandwidth of the linear extended-state observer and constitutes its sole tuning knob. The above design can make  $s^3 + l_1s^2 + l_2s + l_3$  the Hurwitz expression. And by setting the observer gain, the eigenvalues of Hurwitz expression are all  $-\omega_0$ , namely  $s^3 + l_1s^2 + l_2s + l_3 = (s + \omega_0^3)$ . The three gain parameters of the observer can be obtained through the following expression:

$$\beta_i = \prod_{k=0}^{i-1} \frac{n+1-k}{i-k}, 1=1,2,3, n=2 \tag{9}$$

During the design process, the estimated values of the observer variables will converge to the actual state values by choosing proper bandwidth value. Therefore, the extended state observer can not only achieve an accurate estimation of the system state variables, but also estimate the total disturbance  $f$  using the extended state variable  $z_3$ .

Sliding mode controller design

In the traditional design method of active disturbance rejection control, the feedback control is usually composed of the linear combination of the differential differences between the estimated value and the set value. To improve the overall performance of angle tracking, a simple sliding mode control is adopted to replace the feedback control in active disturbance rejection control. As shown in Fig. 2, the output of the linear extended state observer is used as the input of the sliding mode observer. Thus, the estimated system output  $z_1$  is used to replace the measurable value  $y'$ . In addition, by introducing the state observer, the discrepancy between the observer-reconstructed output and its reference set point is directly employed as the angle-tracking error, delivering a delay-free feedback variable for precise closed-loop control.

$$\hat{e} = z_1 - r(s) \tag{10}$$

According to the design method of the sliding mode controller, a linear function is adopted to establish the sliding mode surface:

$$s = \lambda \hat{e} + \dot{\hat{e}} \tag{11}$$

where  $\lambda$  is the gradient of the sliding mode surface. According to the sliding mode control theory, the sliding state and the approaching state are the two states of sliding mode control. The system moves on the sliding surface only when the switching function and its derivative are zero.

$$\dot{s} = \lambda(z_2 - \dot{r}) + bu + z_3 - \ddot{r} - l_2(y' - \hat{y}') \tag{12}$$

By adopting a conventional reaching-law formalism:

$$\dot{s} = -ks - \varepsilon \text{sgn}(s) \tag{13}$$

Combined with Eq (12) and Eq (13), the output expression of the controller can be obtained as:

$$u = [-\lambda(z_2 - \dot{r}) + \ddot{r} - ks - \varepsilon \text{sgn}(s) - z_3 - l_2(y' - \hat{y}')]/b \tag{14}$$

To suppress the chattering phenomenon near the sliding mode surface, the continuous approximation is employed to design the switching control.

$$\text{sgn}(s) = s / (|s| + \delta) \tag{15}$$

Combining Eq (14) and Eq (15), the control law acting on the steering system can be obtained as:

$$u = [-\lambda(z_2 - \dot{r}) + \ddot{r} - ks - \varepsilon s / (|s| + \delta) - z_3 - l_2(y' - \hat{y}')]/b \tag{16}$$

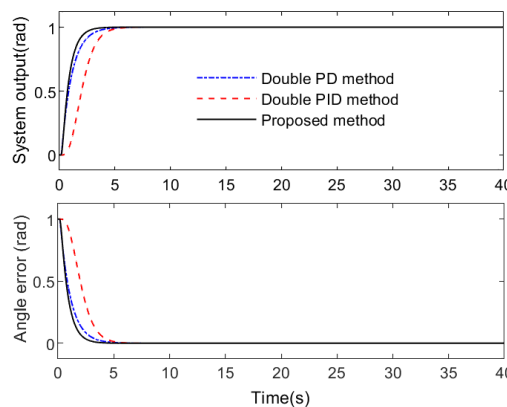
The foregoing expression constitutes the sliding mode controller output, where  $k$  is the parameter of the sliding mode controller, it is a constant greater than zero,  $\varepsilon$  is the switching coefficient, and  $\delta$  is a normal constant with a very small amplitude.

**RESULTS**

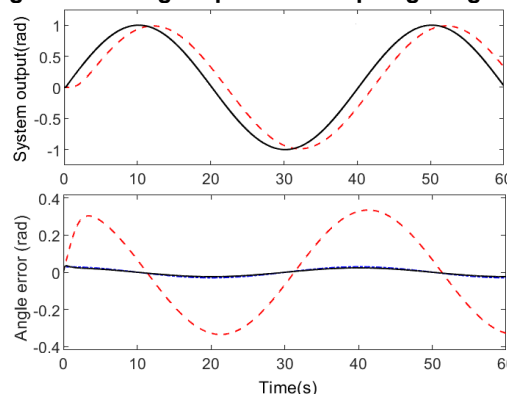
The parameters of the steering system model are  $K = 18.56$ ,  $T = 0.93$ ,  $\theta = 0.15$ . The parameters of the extended state observer and the sliding mode controller are determined as follows:  $w_0=67$ ,  $\lambda=10$ ,  $k=1.5$ ,  $\varepsilon=0.001$ ,  $\delta=0.001$ . To provide a performance comparison, other two methods based on the improved Smith predictor control structure are introduced. Both methods have demonstrated their superiority over many methods through simulation and experiment. They are the control scheme with double PD controllers (*Somak et al., 2021*) and the scheme with double PID controllers (*Yin et al., 2020*). Based on the determined steering system model and controllers, six simulation conditions are selected to demonstrate the performances of accuracy, robustness and disturbance rejection of the control scheme.

**Tracking precision simulation**

A step signal with an amplitude of 1 is set at time  $t=0s$  to the input and the system responses are shown in Fig. 3. It can be seen from the figure that all the three methods can track the desired angle without overshoot. Compared with the other two methods, the method proposed has a faster response speed, and the rise time is 1.453 s. Correspondingly, the angular tracking error performance of the proposed method is also the best. Similarly, a sine signal with amplitude of 1 and period 20 s is set at time  $t=0s$  to the input. The system output and tracking error for the desired angle are shown in Fig. 4. It can be seen that for the varying angle, the angle tracking precision of both the double PD method and the proposed method are superior, and their maximum tracking error is approximately 0.035 rad, while tracking performance of the double PID method is worse.



**Fig. 3 - Tracking response to step angle signal**



**Fig. 4 - Tracking response to sinusoidal angle signal**

**Robustness simulation**

In order to verify the robustness of the system, two groups of parameter perturbation states are designed for demonstration in this section. For model mismatch case I, it is assumed that the amplification factor in the actual steering control system increases by 50%, while other parameters remain unchanged, that is  $P = 28e^{-0.15s} / (0.093s + 1)$ . By adding a unit step change to the set point, the system responses are shown in Fig.5.

It can be observed from the system output responses that the change of model parameters does not have a significant impact on the tracking accuracy for the three control methods, and there are no overshoot or oscillation in system outputs. However, from the controller output responses, it can be found that before the output reaches the steady state, the control laws of the three methods are adjusted rapidly. In addition, the local amplification diagram further illustrates that after the system reached the steady state, there are still relatively large adjustment phenomena in the controller outputs for the other two methods due to the influence of model mismatch, especially there is more serious adjustment for the double PD method.

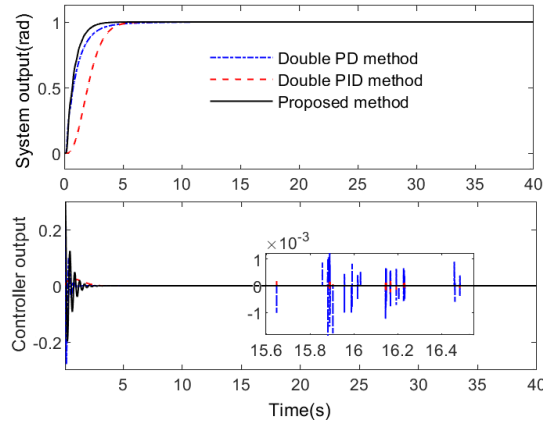


Fig. 5 - Tracking response to step angle under model mismatch I

In order to quantitatively evaluate the smoothness of the controller output, controller output fluctuation evaluation index TV is introduced. According to this index, it can be concluded that the method proposed in this study performs the best in terms of the smoothness of the controller output, and its performance evaluation index is  $TV_{prod} = 1.558$ . In contrast, the fluctuation amplitude of the controller output in the double PD method is the largest with  $TV_{dPD} = 5.927$ , which indicates deficiency in application. Furthermore, with the sinusoidal wave signal as the preset target value, the responses of the perturbed system are shown in Fig. 6. It can be observed from the results that in the case of model mismatch, the three methods all show good tracking performance for the angle signals changing slowly. Meanwhile, the evaluation indexes are obtained as  $TV_{dPD} = 1.276$ ,  $TV_{dPID} = 0.38$ ,  $TV_{prod} = 0.2125$ .

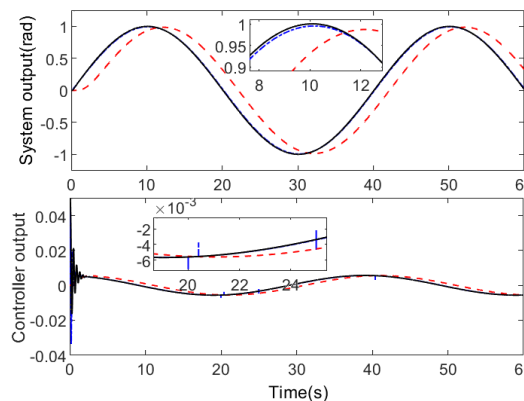


Fig. 6 - Tracking response to sinusoidal angle under model mismatch I

Based on the evaluation indicators, a conclusion can be drawn that when tracks the sinusoidal signal, the output of the proposed controller is relatively smooth. Furthermore, model mismatch condition II is supposed that both the model amplification factor and the time delay in the actual steering system are changed, that is  $P = 25e^{-0.18s} / (0.093s + 1)$ . In this case, taking the step signal as the desired front angle, the responses of the system are shown in Fig. 7. Compared with the model mismatch I condition, it can be seen that the three methods all demonstrate relatively better performance.

**Disturbance rejection simulation**

To demonstrate the performance of disturbance rejection of the proposed method, two disturbance signals are introduced respectively before the steering system.

Firstly, a unit step change to the set point is added at  $t = 0s$  and a step change with an amplitude of  $-0.5$  as a disturbance signal is introduced  $t = 20s$ . The responses of the system to the disturbance are shown in Fig. 8. It can be found that the disturbance has negligible influence on the system output angle for the proposed method in this study. In contrast, the system output of the double PID method exhibits an amplification effect of more than twice. Moreover, the controller output performance also confirms the superiority of the proposed method to the disturbance rejection. In this simulation condition, the calculated performance indexes of the controller output are  $TV_{dPD} = 103.9$ ,  $TV_{dPID} = 3.268$ ,  $TV_{prod} = 1.524$ .

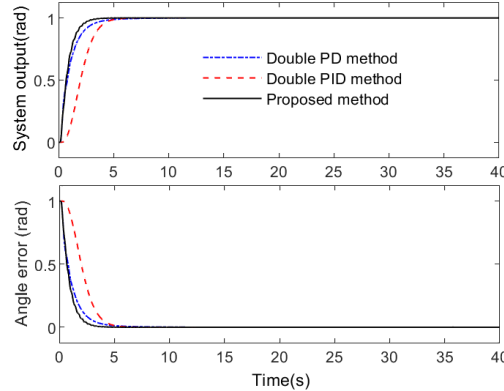


Fig. 7 - Tracking response to step angle under model mismatch II

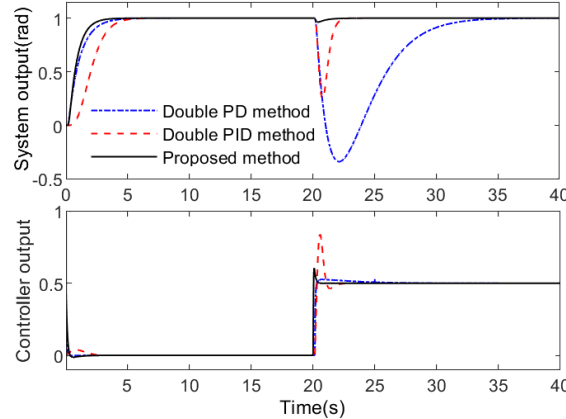


Fig. 8 - Tracking response to step angle with step disturbance

Secondly, white noise signal with power 0.01 and sampling time 0.01 is introduced at  $t = 0s$ . The system outputs are shown in Fig. 9. Obviously, the noise has greatly affected the angle tracking performance for the other two methods, and there are angle errors greater than 0.4 rad after the system reaches the steady state. While the maximum output error of the sliding mode active disturbance rejection angle tracking system designed in this study is only 0.055 rad.

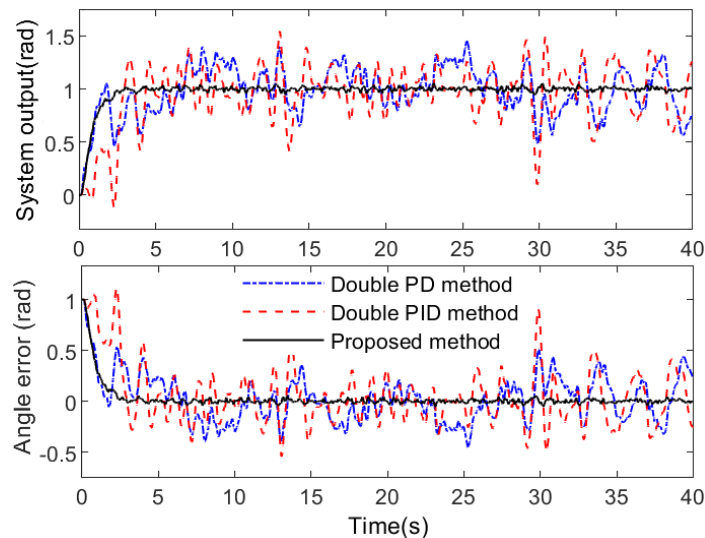


Fig. 9 - Tracking response to step angle with white noise disturbance

For quantitative comparison, performance indexes such as integral of absolute error (IAE), integral square error (ISE), integral time-weighted absolute error (ITAE) are adopted to evaluate the system performance. Mathematically,

$$IAE = \int_0^{\infty} |e(t)|dt, \quad ISE = \int_0^{\infty} e^2 dt, \quad ITAE = \int_0^{\infty} t|e(t)|dt \quad (17)$$

Smaller value denotes the superior performance of the control method. With the step signal as the set point input, the performance indexes for the three methods under three conditions, namely, no disturbance, model mismatch case II, and model mismatch case II with step disturbance are calculated respectively, they are shown in Table 1. It can be clearly seen that the performances of the proposed method are significantly superior to those of the other two methods.

The simulation results indicate that the steering angle tracking control method proposed in this study exhibits satisfactory tracking speed, accuracy, and robustness. Notably, its disturbance rejection performance is particularly outstanding. In the proposed control scheme, the synchronization between the controller output and the system output is achieved through the incorporation of the Smith predictor concept to handle the time delay. Moreover, the designed control structure essentially represents a two-degree-of-freedom control framework. This structure not only enables the estimation and compensation of disturbances but also enhances the overall performance of the front-wheel tracking system.

Table 1

Performance evaluation indexes				
State	Method	IAE	ISE	ITAE
Step signal without disturbance	Double PD	1.119	0.6522	1.062
	Double PID	2.156	1.607	2.832
	Proposed	0.9163	0.5791	0.6421
Step signal with model mismatch II	Double PD	1.12	0.6166	1.248
	Double PID	2.156	1.594	2.851
	Proposed	0.9002	0.5619	0.6305
Step signal with disturbance	Double PD	7.77	6.76	159.8
	Double PID	2.837	1.963	17.1
	Proposed	0.951	0.5799	1.318

## CONCLUSIONS

An enhanced front wheel angle tracking control scheme was developed, integrating the Smith predictor control structure and active disturbance rejection control methodology to address time delays and disturbances in the steering system. Based on comparative analysis with existing control methods, the following conclusions can be drawn. The simulation results under six typical working conditions showed that the sliding mode active disturbance rejection control based on the Smith predictor structure demonstrated superior control performance in the case of parameter perturbations and unknown disturbances.

The combination of the Smith predictor control structure and the linear extended state observer could achieve synchronous estimation of the controller output and the non-delay output of the system, and could effectively estimate, compensate and reject the internal and external disturbances of the system. In the wheel tracking control scheme, the nonlinear sliding mode control was employed to replace the linear feedback control within the active disturbance rejection framework. This substitution further enhanced the performance in terms of disturbance rejection and robustness, as well as the tracking precision for the desired front-wheel angle.

Judging from the simulation curves and performance evaluation indices, the proposed method is theoretically capable of meeting the accuracy and stability requirements for automatic steering control in tractors. Future research will involve conducting real tractor tests to further validate the practical application effectiveness of the proposed control scheme.

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