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Research on Three Kinds of Lane Keeping Method based on Anti-saturation and Adaptive Method

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Abstract

The lane keeping problem of automatic driving for vehicles was studied based on the simplified linear lateral dynamic model. Three kinds of sliding mode methods were proposed by adopting integral sliding mode, anti-saturation and adaptive method. The stability of each method was proved by constructing Lyapunov method. What is worthy pointing out is that the anti-saturation method can not only reduce the oscillations by using soft function and sigmoid function, but also it can solve the saturation problem of front wheels by using the bounded characteristics of above two functions. And this method is also very simple and it has a reasonable physical meaning which makes the chosen of control parameter very easy. At last, detailed numerical simulations were done for proposed three methods and simulation results were compared with other three methods in the past references to testify the rightness of the proposed method.

Keywords: *Lane keeping, traffic engineering, sliding mode control, double loop, robustness*

1. Introduction

In recent years, the scholars at home and abroad have been attracted by the related technology of automatic driving for vehicles, whose control including lateral control and longitudinal control. In these control technologies, the lane keeping control and lane change control were the key technologies of automatic driving for vehicles. In the technology of lane keeping control, by the offset between the actual direction of vehicle moving and the central direction of the lane, it calculated the corner of the steering wheel with corresponding control algorithm, then based on which the vehicle was controlled to drive safely following the lane. At present, there have been many achievements on the research for the lane keeping control, which could be summarized as follows.

The first one mainly focused on the research for motion modeling of lane keeping, such as literature [1], it made the point on the vehicle nearest to the lane as the tracking point, then the lane keeping model based on the strategy of lane preview was established and according to this model, fuzzy self-adaptive PID controller was designed; In literature [2], it established the motion model of autonomous navigation for intelligent vehicle based on vision, and designed the optimal controller; In literature [3, 4], the lateral control model of vehicle was constructed on the basis of the theory of "preview-follow".

The second aspect mainly focused on the control methods that were to select a suitable control strategy, such as adaption, variable structure, or neural network and so on, to keep the robustness of lane keeping pointed at the uncertainty of parameters in the lateral control model for vehicles. As proposed in literature [5], fuzzy PD control was chosen to solve the uncertainty of model mass, the moment of inertia. In literature [6], direct adaptive method was presented to solve problems such as the uncertainty of path

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curvature. In literature [7], terminal sliding mode control approach was applied to uncertain system of lane keeping, in order to enhance the rapidity of lane keeping.

The third aspect mainly conducted thorough research in view of the practical problems of the vehicle lateral motion model, the problems mainly included steering delay of front wheels, time-lag of system, and saturation of steering angle of front wheels. Such as literature [8] considered the dynamic characteristics of the rotation of front wheels, adopted the first-order inertial component to simulate its delay characteristic, and proposed self correction method to deal with its uncertainty of model. In literature [9], considering the delay characteristics of lateral motion model for vehicle, LMI was used to design sigmoid sliding mode controller.

However, limited by physical factors, the maximum steering angle of the front wheels could not exceed the limit value. Furthermore, for the reason of safe driving and reducing tire wear, the maximum steering angle should also be limited to a given value. This problem was a typical saturation problem, and now there was seldom literatures considering anti-saturation method when solving lane keeping. Thus in this paper, for the saturation of steering angle of front wheels, a kind of double sliding mode control law was proposed to realize anti-saturation with using softening function and bipolar sigmoid function.

2. Model Description

On the premise of assuming the rotation angle of wheel as a small angle, the lateral position error and the lateral yaw angle error were chosen as the basic state variables to describe the model of lateral motion for vehicles. The mode was written as the following [9-12] formula:

$$\ddot{y} = -\frac{2(C_f + C_r)}{m v_x}\dot{y} - [v_x + \frac{2(C_f l_f - C_r l_f)}{m v_x}]\dot{\psi} + \frac{2C_f}{m}\delta_z \tag{1}$$

$$\ddot{\psi} = -\frac{2(C_f l_f^2 + C_r l_r^2)}{I_z v_x}\dot{\psi} - \frac{2(C_f l_f - C_r l_r)}{I_z v_x}\dot{y} + \frac{2C_f l_f}{I_z}\delta_z \tag{2}$$

In which, ψ was shown as the lateral yaw angle; y was shown as the lateral position; I_z was described as the moment of inertia for vehicle; v_x was indicated as the longitudinal velocity for vehicle; m was meant for the mass of the vehicle; l_f and l_r were shown as the distance from the centre of mass to the front axle and the back axle respectively in vehicle; C_f and C_r were written as the stiffness of the front wheels and the back wheels; δ was shown as the rotating angle of the front wheels.

Referenced as literature [13], if exploration strategy was assumed to be used, then two lateral sensors were installed at the front and rear bumpers. In this way, the lateral offset of the wheel from the central line and the angle between the moving direction of vehicle and the tangent of the path could be both measured. Let the explored distance be taken as d, the lateral moving model in geometry of vehicles could be shown as follows:

$$\dot{y}_s = \dot{y} + v_x\psi_r + d\dot{\psi}_r \tag{3}$$

$$\ddot{\psi}_r = \ddot{\psi} - \ddot{\psi}_d \tag{4}$$

In which, ψ_r was shown as the lateral yaw angle error, that was the angle between the moving direction of vehicles and the tangent of the path; y_s was shown as the lateral position error off the central line; ψ_d was the desired yaw angle, χ was the curvature of the center line in the lane, that the equation t$\dot{\psi}_d = v_x\chi$ could be obtained.

3. Transformation and Hypothesis of the Model

According to equations (1) to (4), the related variables were defined as follows:

$$a_1 = C_f + C_r\,,\quad a_2 = C_f l_f - C_r l_r\,,\quad a_3 = C_f l_f^2 + C_r l_r^2 \tag{5}$$

$$a_{11} = \frac{2a_2}{I_z}\,,\quad a_{12} = 2\left[\frac{da_2}{I_z v_x} - \frac{a_3}{I_z v_x}\right] \tag{6}$$

$$a_{13} = -\frac{2a_2}{I_z v_x}\,,\quad b_1 = -\frac{2C_f l_f}{I_z}\,,\quad d_1 = -\frac{2a_3}{I_z v_x}\dot{\psi}_d - \ddot{\psi}_d \tag{7}$$

$$a_{21} = 2\left[\frac{a_1}{m} + \frac{da_2}{I_z}\right],\quad a_{22} = 2\left[\frac{da_1}{mv_x} - \frac{a_2}{mv_x} + \frac{d^2 a_2}{I_z v_x} - \frac{da_3}{I_z v_x}\right] \tag{8}$$

$$a_{23} = -2\left[\frac{a_1}{mv_x} + \frac{da_2}{I_z v_x}\right],\quad b_2 = 2\left[\frac{C_f}{m} + \frac{dC_f l_f}{I_z}\right] \tag{9}$$

$$d_2 = -\left[v_x + \frac{2a_2}{mv_x} + \frac{2da_3}{I_z v_x}\right]\dot{\psi}_d - d\ddot{\psi}_d \tag{10}$$

Then the lateral moving model of vehicles could be described as the following form:

$$\begin{bmatrix} \ddot{\psi}_r \\ \ddot{y}_s \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} & 0 & a_{13} \\ a_{21} & a_{22} & 0 & a_{23} \end{bmatrix} \begin{bmatrix} \psi_r \\ \dot{\psi}_r \\ y_s \\ \dot{y}_s \end{bmatrix} + \begin{bmatrix} b_1 \\ b_2 \end{bmatrix} \delta_z + \begin{bmatrix} d_1 \\ d_2 \end{bmatrix} \tag{11}$$

With $\dot{\psi}_r = \omega$ and $\dot{y}_s = v_y$ defined, the above model could be written as standard state-variable mode.

$$\begin{bmatrix} \dot{\psi}_r \\ \dot{\omega} \\ \dot{y}_s \\ \dot{v}_y \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ a_{11} & a_{12} & 0 & a_{13} \\ 0 & 0 & 0 & 1 \\ a_{21} & a_{22} & 0 & a_{23} \end{bmatrix} \begin{bmatrix} \psi_r \\ \omega \\ y_s \\ v_y \end{bmatrix} + \begin{bmatrix} 0 \\ b_1 \\ 0 \\ b_2 \end{bmatrix} \delta_z + \begin{bmatrix} 0 \\ d_1 \\ 0 \\ d_2 \end{bmatrix} \tag{12}$$

Referenced by the literature [10], it was considered further the characteristic of inertial lag from the steering to the front wheels angle δ_z. The first-order inertial component was introduced to describe its dynamic characteristic shown as below:

$$\dot{\delta} = \frac{1}{T_e}u - \frac{1}{T}\delta \tag{13}$$

Considering the practice and complexity in the problem of lane keeping, the hypothesis was made for model with reference to literature [11].

HYPOTHESIS 1: The variables m ,I_z,l_f,l_r,C_f and C_r of vehicles are all unknown.

HYPOTHESIS 1: The curvature χ of the pavement was also the unknown parameter, but its value was bounded.

4. Design of Two Kinds of Anti-saturation Sliding Controller

Without loss of generality, the expected value of the lateral position error y_s was defined as $y_{r\,\mathrm{in}}$ .And without special introductions, $y_{r\,\mathrm{in}} = 0$ .The error variable was defined as $e_y = y_s - y_{r\,\mathrm{in}}$, then the following equation could be obtained.

$$\dot{e}_y = \dot{y}_s - \dot{y}_{r\,\mathrm{in}} \tag{14}$$

The sliding surface was chosen as below

$$s_y = \dot{e}_y + c_1 e_y + c_2 \int e_y dt \tag{15}$$

The derivation of the equation (15) was written as

$$\dot{s}_y = \ddot{e}_y + c_1 \dot{e}_y + c_2 e_y \tag{16}$$

The above equations could be deduced to the following equation.

$$\begin{aligned} \dot{s}_y &= \ddot{y}_s + c_1 \dot{y}_s + c_2 e_y \\ &= \dot{v}_y + c_1 v_y + c_2 e_y \\ &= a_{21}\psi_r + a_{22}\omega + (a_{23} + c_1) v_y + c_2 e_y + b_2 \delta + d_2 \end{aligned} \tag{17}$$

Corresponding to the calculation of practical parameters, $b_2 > 0$, so equation (17) could be written as follows further.

$$\frac{1}{b_2}\dot{s}_y = \frac{a_{21}}{b_2}\psi_r + \frac{a_{22}}{b_2}\omega + \frac{(a_{23}+c_1)}{b_2} v_y + \frac{c_2}{b_2} e_y + \frac{d_2}{b_2} + \delta \tag{18}$$

First considering the certain case, that was parameters of the vehicle model and the curvature of the path were known completely, the first kind of integral sliding mode control law could be designed as below:

$$\delta_1 = -\frac{a_{21}}{b_2}\psi_r - \frac{a_{22}}{b_2}\omega - \frac{(a_{23}+c_1)}{b_2} v_y - \frac{c_2}{b_2} e_y - \frac{d_2}{b_2} + \Omega_1 \tag{19}$$

Among which:

$$\Omega_1 = -k_1 s - \frac{k_2 s}{|s| + \varepsilon} - k_3 \frac{1 - e^{\tau s}}{1 + e^{\tau s}} \tag{20}$$

Taken the unknown parameters into account, some of them were assumed to satisfy the following bounded condition.

$$\left| -\frac{a_{21}}{b_2}\psi_r - \frac{a_{22}}{b_2}\omega - \frac{(a_{23}+c_1)}{b_2} v_y - \frac{c_2}{b_2} e_y - \frac{d_2}{b_2} \right| < k_2 + k_3 \tag{21}$$

Then the control law described above could be simplified as the below form of anti-saturation sliding mode control.

$$\delta_2 = -\frac{k_2 s}{|s| + \varepsilon} - k_3 \frac{1 - e^{\tau s}}{1 + e^{\tau s}} \tag{22}$$

Among which, $k_2 + k_3$ was the physical limitation of maximum steering angle for front wheels. For example, if the maximum value of steering angle was limited as 15°, k_2 and k_3 could be set as $k_2 = 7/57.3$, $k_3 = 8/57.3$.

Chosen Lyapunov function $V = s^2/2$, it was not difficult to prove that the above two control laws could make the system stable under the corresponding assumptions.

It was worth illustrating that assumed equation (21) was easy to be satisfied when k_2 and k_3 were designed as above methods, which meant the front wheels had sufficient ability to complete control mission. At the same time, the above design could make the second control law satisfied the following bounded condition.

$$\left|\delta_2\right| = \left| -\frac{k_2 s}{\left|s\right| + \varepsilon} - k_3 \frac{1 - e^{-\tau s}}{1 + e^{-\tau s}} \right| \leq k_2 + k_3 \tag{23}$$

That meant this control program possessed the feature of anti-saturation.

5. Design of a Kind of Adaptive Integral Sliding Mode Controller

When the parameters of model for vehicle were unknown completely, the below adaptive method could be adopted to design control program. For equation (18), the definitions were written as:

$$g_1 = \frac{a_{21}}{b_2}, g_2 = \frac{a_{22}}{b_2}, g_3 = \frac{(a_{23} + c_1)}{b_2}, g_4 = \frac{c_2}{b_2}, g_5 = \frac{d_2}{b_2} \tag{24}$$

Then we had the following equation:

$$\frac{1}{b_2}\dot{s}_y = g_1\psi_r + g_2\omega + g_3 v_y + g_4 e_y + g_5 + \delta \tag{25}$$

The third kind of adaptive integral sliding mode control law was designed as below:

$$\delta_3 = -\hat{g}_1\psi_r - \hat{g}_2\omega - \hat{g}_3 v_y - \hat{g}_4 e_y - \hat{g}_5 + \Omega_1 \tag{26}$$

In which, Ω_1 was defined as above. The error variables were defined as following:

$$\tilde{g}_1 = g_1 - \hat{g}_1,\ \ \tilde{g}_2 = g_2 - \hat{g}_2,\ \ \tilde{g}_3 = g_3 - \hat{g}_3,\ \ \tilde{g}_4 = g_4 - \hat{g}_4,\ \ \tilde{g}_5 = g_5 - \hat{g}_5 \tag{27}$$

Then we would get the equation:

$$\frac{1}{b_2}\dot{s}_y = \tilde{g}_1\psi_r + \tilde{g}_2\omega + \tilde{g}_3 v_y + \tilde{g}_4 e_y + \tilde{g}_5 + \Omega_1 \tag{28}$$

The regulation rules of adaptive weights for unknown parameters were designed as below equations.

$$\dot{\hat{g}}_1 = -\Gamma_1 s_y \psi_r,\ \ \dot{\hat{g}}_2 = -\Gamma_2 s_y \omega,\ \ \dot{\hat{g}}_3 = -\Gamma_3 s_y v_y,\ \ \dot{\hat{g}}_4 = -\Gamma_4 s_y e_y,\ \ \dot{\hat{g}}_5 = -\Gamma_5 s_y \tag{29}$$

Chosen Lyapunov function as below:

$$V = \frac{b_2}{2} s_y^2 + \sum_{i=1}^{5} \frac{1}{\Gamma_i}\tilde{g}_i^2 \tag{30}$$

Find the derivation of this function, we got:

$$\dot{V} = -s_y\Omega_1 \leq 0 \tag{31}$$

It was visible that system was stable according to the theory of Lyapunov stability, thus $e_y \to 0$, the control of lane keeping was realized.

6. Simulation and Analysis

The parameters were assigned as the following value: the mass of the vehicle $m = 1573\ kg$; the moment of inertia $I_z = 2873\ kg \cdot m^2$; the front wheels base $l_f = 1.1\ m$; the rear wheelbase $l_r = 1.58\ m$; the stiffness of the front wheels $C_f = 80\ kN / rad$ and the stiffness of the rear wheel $C_r = 80\ kN / rad$; the longitudinal velocity of vehicle was $20\ m/s$; the delayed time constant for the steering $T_e = 0.05\ s$; the initial error of the yaw angle was 0 degree; the initial error of the lateral position was 1 m; other initial states were considered as 0.

The following Figure 1 and Figure 2 showed the control effect of traditional sliding mode control and self-tuning sliding mode control in paper [12]. Figure 3 and Figure 4 gave the control effect of the direct adaption elaborated in literature [11].

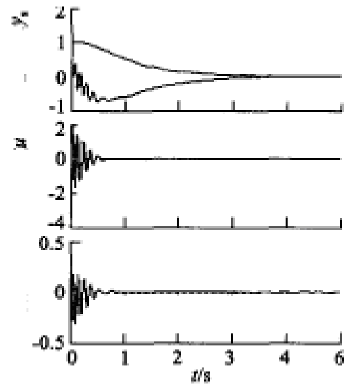

Figure 1. Control Effect of Traditional SMC

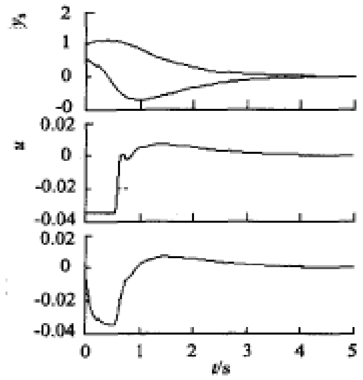

Figure 2. Control Effect of Self-tuning SMC

It could be seen from Figure 1, the convergence time of traditional sliding mode control was about 3s, the maximum steering angle of wheel reached to more than 17°; while from Figure 2, the convergence time of self-turning sliding mode control is about 3s, but the maximum steering angle of wheel was only about 2°. Of course the parameters of vehicles in Figure 1 and Figure 2 are slightly different from the parameters adopted in the following Figure 3 and Figure 4.

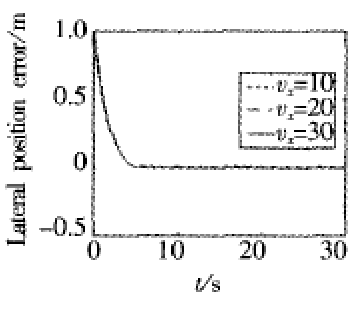

Figure 3. Control Effect of Direct Adaption

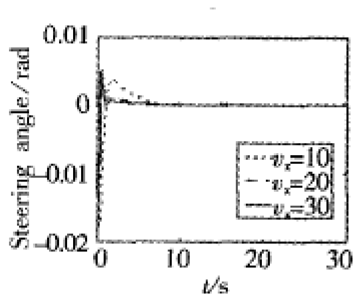

Figure 4. Control of Steering Angle in Front Wheels with Direct Adaption

Figure 3 and Figure 4 were shown that convergence of direct adaptive control was slow, at about 6s, while its maximum steering angle of front wheels was also small, at about 1°.

The following Figure 5 and Figure 6 were shown the control effect for the first kind of integral sliding mode control designed in this paper. In this simulation, nominal value was adopted in control law and random perturbation was executed for parameters in vehicle mode. Simulation were repeated 50 times, with the stiffness of the tires C_f and C_r chosen as the random values in the region $[40,80]$, the curvature radius of the path selected as the random value in the range of $[100,500]$. The results were shown in the following Figures.

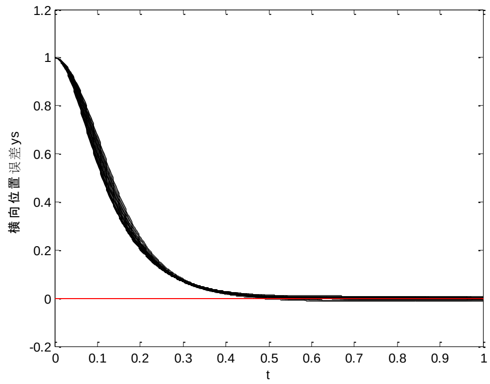

Figure 5. Error of Lateral Position

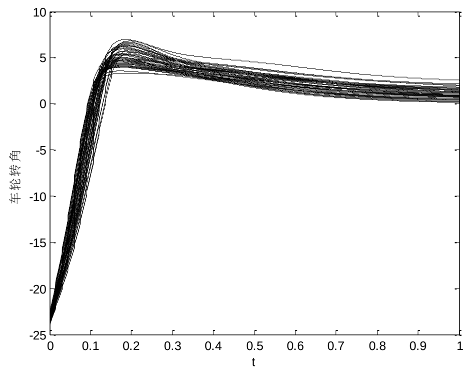

Figure 6. Steering Angle of Front Wheels (°)

Parameter perturbation was selected as the same above, control variable was also chosen the same as the first kind of integral sliding mode control method. Figure 7 and Figure 8 showed the control effect for the second kind of anti-saturation sliding mode control law.

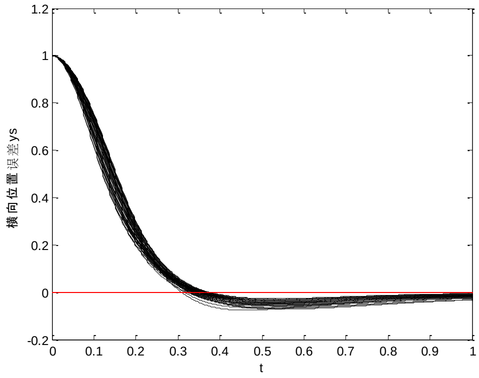

Figure 7. Error of Lateral Position

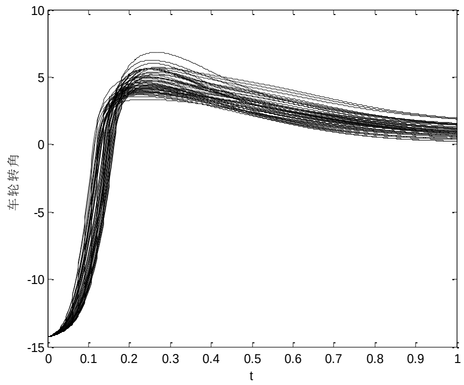

Figure 8. Steering Angle of Front Wheels (°)

From the above simulated curves, the rapidity of both control methods were almost the same. However, the first control method could not assure the steering angle not exceed the limited value of 15°, while the second method could solve the problem of saturation better.

The control effect of the third kind of adaptive integral sliding mode control law designed in this paper was shown in the following Figure 9 and Figure 10.

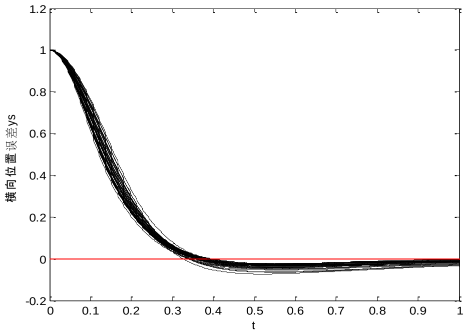

Figure 9. Error of Lateral Position

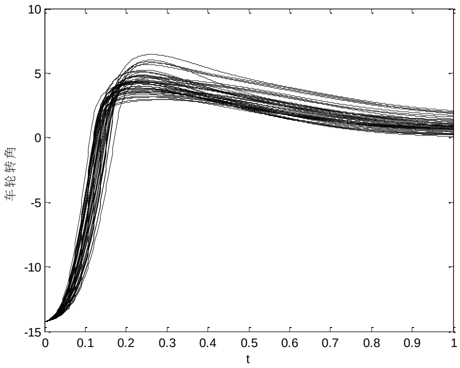

Figure 10. Steering Angle of Front Wheels (°)

From the above Figure 9 and Figure 10, it was shown that the control law of adaptive integral sliding mode had better rapidity and robustness, and steering angle of front wheels was also controlled relatively good under the condition of small adaptive regulated coefficient for unknown parameters. In this paper, several control strategies put forward before were elaborated, such as traditional sliding mode control, self tuning sliding mode control and direct adaptive sliding mode control; also several control methods were proposed in this paper now, such as integral sliding mode control, anti-saturation sliding mode control and adaptive integral sliding mode control. By synthetically comparing these methods, it was found that the later two had better rapidity, and with random perturbation of model parameters, there would not appear the situation that steering angle of front wheels exceeded the available range.

7. Conclusion

For simplified linear model of lateral dynamics for vehicles, three kinds of sliding mode control methods were presented in this paper to eliminate lateral position error. Particularly worth mentioning, the second method of anti saturation sliding mode for lane keeping, possessed advantages of simple algorithm and clear physical meaning. Furthermore, it adopted softening function and sigmoid function, which not only eliminated the chattering phenomenon in traditional sliding mode, but also could ensure the steering angle of front wheels did not exceed the physical limit. This aspect preferably solved the problem of saturation during the control for lane keeping. Finally, by random perturbation of model parameters, multi-simulation were repeated with these three methods. And compared with traditional sliding mode control, self tuning sliding mode control and direct adaptive sliding mode control introduced before, the method proposed in this paper was demonstrated its effectiveness.

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