Nonlinear differential and integral sliding mode control for wave compensation system of ship-borne manipulator

Zhiqiang Xu¹, Zhiyong Wang¹, Zhixin Shen¹ and Yougang Sun²

Abstract
Ship-borne manipulator system is extremely unstable under the complex marine environment, which seriously threatens the safety of operating equipment and operators. In this paper, the dynamics and robust control of wave compensation system for ship-borne manipulator are studied. First, based on the oil circuit variable amplitude control of ship-borne manipulator, the coupling dynamic model of valve-controlled cylinder parallel accumulator is established. Then, since traditional sliding mode needs high-order derivative of feedback angle, it is difficult to implement traditional sliding mode in real hardware system. To solve these problems, a nonlinear differential and integral sliding mode control strategy is proposed. The integral term is introduced to reduce the influence of unmodeled disturbance and parameter perturbation. The stability analysis proves that the system state can track the desired target signal, and the tracking error \( e(t) \) tends to zero. In addition, in order to weaken the phenomenon of system chattering, this paper introduces a nonlinear differential control to increase the damping coefficient of the system. The simulation and experimental results show that the control law has good dynamic performance, high control accuracy, and strong anti-disturbance ability without chattering phenomenon. It is of great significance to improve the efficiency and safety of ship-borne manipulator operation, and this paper also provides useful reference for wave compensation system of other marine equipment.

Keywords
Ship-borne manipulator, wave compensation, dynamic model, sliding mode control

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Introduction
Ship-borne manipulator system is the most important tool for marine cargo trans-shipment and salvage.¹⁻⁴ As shown in Figure 1, it can greatly reduce manual labor, improve operational efficiency, and achieve safe production. The ship-borne manipulator system equipped with a luffing mechanism is used to construct automatic wave compensation equipment.² To eliminate the effect of ship rolling on hoisting and moving cargoes, the luffing mechanism controls the telescopic arm to be suspended at a fixed angle rapidly when a wave makes the ship shake from side to side steeply.

The researchers have proposed a variety of control strategies for the wave compensation systems. Hu et al.⁵ presented a parallel wave compensation system to improve the anti-pendulation capability. Wang et al.⁶ designed a novel offshore crane combined compensation approach named four-post combined compensation based on the three-post direct ship motion compensation. They all had good wave compensation effect, but the mechanisms are complex and not easy to repair. Some studies are presented from the point of control algorithm. Kuchler et al.³ restrained the payload motion due to the wave disturbance efficiently using an active controller together with a prediction algorithm. Sun et al.⁷ proposed a double-layer sliding mode control law to eliminate the influence of the sea condition on the cargo sway. Both the simulations and experiments are included to show the effectiveness of the proposed control law. Takagi and Mishimura⁸ utilized a centralized control method with coupling between the boom luff and rotation directions to

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reduce the swing of a jib-type crane. Woodacre et al.\textsuperscript{9} proposed a marine active-heave compensation (AHC) system with Model Predictive Control (MPC) controller and a set-point prediction algorithm. Liu et al.\textsuperscript{1} developed a modified fuzzy proportional–integral–derivative (PID) controller for the harsh sea condition salvage crane, which can improve the salvage effect. Sun et al.\textsuperscript{10} proposed a novel nonlinear stabilizing control strategy for underactuated ship-mounted crane systems. Niu et al.\textsuperscript{11} designed a heave compensation system for a 200T winch system based on a semi-active method. The experiment results are included to show the effectiveness of the proposed method.

Because of the complexity of the marine environment, the wave motion has a great impact on the maneuverability of the ship-borne manipulator, such as the different levels of wind and sea conditions and the gravity deviation caused by the ship hull cargo at the heading angle. In addition, there are some parameters which are difficult to be described by accurate mathematical models or which will change in the long-term operation of machinery and equipment, such as the mass of the valve core, the current-force gain coefficient of the ratio of viscous damping coefficient, and the gradient of the valve core opening. If the robustness of the control system is not strong, the control performance of ship-borne manipulator system will inevitably deteriorate, resulting in equipment damage and casualties caused by control failure.

For this practical and urgent problem, robust control is a highly feasible solution.\textsuperscript{11,17–20} Robust control refers to the design of a closed-loop feedback control system to stabilize the state of the system without modeling external disturbances or a certain degree of parameter perturbation. At present, there are many methods to achieve robust control strategies, such as $H_\infty$ control theory,\textsuperscript{21,22} sliding mode variable structure control (SMC),\textsuperscript{23,24} structural singular value theory ($\mu$ theory),\textsuperscript{25} and Kharitonov interval theory.\textsuperscript{26} Among them, from the application point of view, a new area of robust control is the SMC.\textsuperscript{27–36} When the system is on the sliding mode surface, the system is insensitive to modeling errors, parameter perturbations, and unmodeled external disturbances. Especially, it has good control effect on the nonlinear system, which makes it possible to apply SMC in ship-borne manipulator system.

Previous studies have not fully solved the problem of the nonlinear control for ship-borne manipulator under persistent disturbances and system uncertainty of the parameters. In this paper, we first establish the dynamic model of ship-borne manipulator based on the variable amplitude control oil circuit. Next, in order to address the problem of non-existence of the feedback variable high-order derivative, a nonlinear differential and integral sliding mode control strategy is proposed to compensate disturbance of waves. Also, a differential switching controller is utilized to reduce the chattering phenomenon. Moreover, the stability analysis proves that the system state can track the desired target signal, and the tracking error $e(t)$ tends to zero. Finally, the validity and robustness of the design method are verified by numerical simulation and experiment. The contribution of this paper can be summarized as follows:

1. To our best knowledge, this paper is the first attempt to use nonlinear differential and integral sliding mode controller (NDISMC) for wave compensation system. This method is more suitable for the ship-borne manipulator system with complex nonlinear characteristics.

2. The proposed method solves the problem that the higher derivative of the feedback variable
does not exist, and the chatter of the sliding mode is significantly reduced by the differential switching control law. The speed of the system approaching to the sliding surface is accelerated.

3. The wave compensation experimental results show that the method has better control performance in tracking accuracy, response speed, and robustness over traditional method.

The rest of this paper is organized as follows. In section “Dynamic model,” the coupling dynamic model of valve-controlled cylinder parallel accumulator is established. Section “Improved nonlinear differential and integral sliding mode control” designs a modified a nonlinear differential and integral sliding mode control strategy to remove the requirement of high-order derivative of the feedback angle and reduce chattering. Simulation results are provided in section “Simulation results.” Experimental results are shown in section “Hardware experiments.” Finally, the conclusions and future work directions are drawn in section “Conclusion.”

Dynamic model

The hydraulic system loop of the ship-borne manipulator’s amplitude control is shown in Figure 2. The angle displacement sensor is used to monitor the displacement signal and angle signal of the amplitude-changing cylinder, and the sliding valve opening is adjusted by the controller and power amplifier.

Figure 3 is a schematic diagram of the connection between the valve-controlled cylinder system and accumulator. When the cylinder is extended, the hydraulic oil from the valve mouth and accumulator jointly flows into the rodless chamber of the cylinder, which satisfies the condition that the rodless chamber urgently needs a large flow of hydraulic oil, to improve the response rate of the cylinder and shorten the response time of the system control.

Based on the flow continuity equation and dynamic balance equation of valve opening, cylinder, and accumulator, the coupling dynamic model of valve-controlled cylinder system paralleled with accumulator is established.

Cylinder dynamic equilibrium equation

Compared with the working pressure, the pressure in the rod chamber of the cylinder can be neglected, and the force balance equation of the piston rod of the cylinder is given as follows

\[ pA = m_i\ddot{x} + B_p\dot{x} + k_2x + F_L \]  

(1)

where \( p \) denotes the working pressure, \( A \) denotes the action area of rodless cavity, \( m_i \) denotes the mass of the piston, \( x \) denotes the displacement of piston, \( B_p \) denotes
the viscosity damping coefficient, \( k_z \) denotes the spring stiffness, and \( F_L \) represents the external load force acting on the piston.

### Cylinder flow continuity equation

The cylinder flow continuity equation can be represented as follows\(^{37–39}\)

\[
QL = A \dot{x} + C_i p
\]

where \( Q_L \) is the cylinder flow rate and \( Q_L = Q_1 + Q_X \), \( Q_1 \) is the orifice flow; \( Q_X \) is the accumulator flow; \( C_i \) is the cylinder leakage coefficient.

### Valve orifice flow equation

The flow rate of spool valve is a function of working pressure and the valve core displacement. The spool valve can be seen as a zero-opening four-way spool valve. The flow rate equation of spool valve can be expressed as

\[
Q_1 = C_d w x_r \sqrt{\frac{2}{\rho} (p_S - p_1)} = k_q x_r
\]

where \( Q_1 \) is the valve orifice flow rate, \( C_d \) is the valve orifice flow coefficient, \( w \) is the valve orifice area gradient, \( x_r \) is the valve core displacement, and then \( x_r = k_r \cdot \mu \) with \( k_r \) the valve core proportional coefficient and \( \mu \) the current signal of valve core, \( \rho \) is the oil density, \( p_S \) is the oil source pressure, \( p_1 \) is the pressure of valve core chamber, \( k_q \) is the flow gain coefficient of valve orifice and \( k_q = C_d \cdot w \cdot \sqrt{\frac{2}{\rho} (p_S - p_1)} \).

### Accumulator dynamic equation

The simplified model of the structure and force of the bladder accumulator is shown in Figure 4.

The accumulator chamber is pre-filled with pressure nitrogen, and the weight of nitrogen can be neglected. The accumulator dynamic equation is given by

\[
p \cdot A_a = k_a \cdot x_a
\]

where \( A_a \) is the action area of the accumulator, \( k_a \) is the gas stiffness coefficient, \( x_a \) is the gas displacement in the accumulator.

### Accumulator flow equation

Because the output flow of accumulator is equal to the volume change rate, equation (5) can be obtained

\[
Q_X = \dot{x}_a \cdot A_a
\]

where \( Q_X \) is the accumulator flow rate.

The state–space expression of the valve-controlled cylinder system obtained by simultaneous equations (1)–(5) is as follows

\[
\begin{align*}
\dot{p} &= \frac{A \cdot \dot{x} + B_p \cdot \dot{x} + k_z \cdot x + F_L}{m_t} \\
A \cdot \dot{x} + C_i \cdot p &= k_q \cdot x_r + \frac{\rho}{k_q} \cdot \sqrt{\frac{2}{\rho} (p_S - p_1)} \\
x_r &= k_r \cdot \mu
\end{align*}
\]

According to organizing equation (6), equation (7) is given by

\[
\begin{align*}
\dot{x} &= \frac{(B_p - C_i \cdot k_q)}{m_t \cdot A_a} \cdot \dot{x} - \frac{(k_z)}{m_t} - \frac{(C_i \cdot k_a \cdot B_p + k_a \cdot A)}{m_t \cdot A_a} + \frac{(k_a \cdot C_i \cdot F_L)}{m_t \cdot A_a} \\
&+ \frac{(k_r \cdot \dot{x}_a \cdot k_a \cdot k_z)}{m_t \cdot A_a} - \frac{(k_r \cdot k_q \cdot k_a \cdot A)}{m_t \cdot A_a} + \frac{(k_a \cdot C_i \cdot F_L)}{m_t \cdot A_a} \\
&+ \frac{(k_r \cdot \dot{x}_a \cdot k_a \cdot k_z)}{m_t \cdot A_a} - \frac{(k_r \cdot k_q \cdot k_a \cdot A)}{m_t \cdot A_a} + \frac{(k_a \cdot C_i \cdot F_L)}{m_t \cdot A_a}
\end{align*}
\]

Assuming the state of the system is \( x_1 = x \), \( x_2 = \dot{x} \), \( x_3 = \dot{x}_a \), the dynamic model of the valve-controlled cylinder system is

\[
\begin{align*}
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= x_3 \\
\dot{x}_3 &= \theta_1 x_1 + \theta_2 x_2 + \theta_3 x_3 + g \mu + d \\
x &= x_1
\end{align*}
\]

where

**Figure 4.** Simplified model of the bladder accumulator.
Improved nonlinear differential and integral sliding mode control

Considering the dynamic model of valve-controlled cylinder system, traditional sliding mode control method is designed to achieve the stable hovering state of the fixed angle of the manipulator under the influence of external disturbance. In other words, the state \( x_m(t) \) of the system can track the sine-cosine wave compensation motion of salvage equipment under complex sea condition and set the target position signal \( r(t) \) of sine-cosine compensation motion, which can make the position tracking error \( |x_m(t) - r(t)| \) converge to the neighborhood of zero.

**Assumption 1.** The desired target position signal \( r(t) \) and its first and second derivatives have a range, which should be \( \{ \max (|r|, |\dot{r}|, |\ddot{r}|) \leq \delta_r \} \), where \( \delta_r \) is an unknown positive number.

**Assumption 2.** The external disturbance \( F_L \) has an upper limit, namely \( |F_L| \leq F_{L_{\text{max}}} \), where \( F_{L_{\text{max}}} \) is an unknown positive number. In fact, different levels of sea state disturbance cannot be infinite, so there will always be an upper limit.

The traditional sliding mode model controller requires calculating high-order derivatives of the angle of the manipulator, and there is a phenomenon that the high-order derivative does not exist, where the existence of singularities increases the difficulty of calculation and which also provides a certain challenge to the hardware of the controller. It is difficult to design the motion law of the amplitude-changing cylinder with the sliding model controller. In order to solve the problem of non-existence of high-order derivatives and better deal with the uncertainties and parameter perturbations during sea lifting, a nonlinear differential and integral SMC strategy is proposed.

Nonlinear parametric uncertainty of SMC

The expressions of uncertain high-order nonlinear parameters of the luffing cylinder system for salvage equipment under complex sea condition are given by

\[
\begin{align*}
\theta_1 &= \frac{c_i b_k k_i}{m_i A_i} \\
\theta_2 &= -\left( \frac{b_i}{m_i} - \frac{c_i b_k k_j}{m_i A_i} \right) \\
\theta_3 &= -\left( \frac{b_i}{m_i} - \frac{c_i b_k k_j}{m_i A_i} \right) \\
g &= -\frac{k_i k_j k_s}{m_s A_s} \\
d &= \frac{k_i k_j k_s}{m_s A_s}
\end{align*}
\]  

where \( i = 1, 2, \ldots, n-1 \), \( x_i \) and \( x_n \) are the state variables of salvage equipment under complex sea condition (i.e. the value and derivatives of the amplitude-changing cylinder of the manipulator), \( d(t) \) is the load of the external manipulator, \( \mu \) is the control system signal input, \( y \) is the angle of the manipulator, \( f(x) \) and \( g(x) \) are known functions of the hydraulic system, and for any \( x \), \( \beta g(x) \neq 0 \). \( \theta \) and \( \beta \) are uncertain parameters for salvage luffing mechanism.

An integral sliding mode control system is designed to make the displacement of the cylinder of the system track the expected trajectory \( y_d \) of sine-cosine compensation

\[
\lim_{t \to 0} (y - y_d) = 0 \tag{11}
\]

In wave compensation systems (equation (10)), the switching function shown below is common in general SMC

\[
z_0 = c_1 e_1 + c_2 e_2 + \cdots + c_n e_n - e_0 \tag{12}
\]

where \( e_i = y^{(i-1)} - y_d^{(i-1)} \), \( i = 1, \ldots, n \).

The limit for the amplitude-changing cylinder system to reach the sliding surface \( z_0 = 0 \) is \( z_0 z_0 < 0 \). In order to realize the function, assuming that the parameter \( \beta \) is unknown, the following expression of sliding mode switching control signal is given by

\[
\mu = \frac{1}{\beta g(x)} \left[ \sum_{j=1}^{n-1} c_j e_j + y_d^{(0)} - M[f(x)|\text{sgn}(z_0)| - D \text{sgn}(z_0)] \right] \tag{13}
\]

where \( M \) and \( D \) are the upper bounds of the system parameter \( \theta \) and external disturbance \( d(t) \), respectively, that is, \( |\theta| < M, |d(t)| < D \).

As seen in equation (12), the feedback acquisition value of the cylinder length is indispensable, and the high-order derivative of the return signal needs to be worked out. Therefore, when the high-order derivative of cylinder tracking displacement does not exist, wave compensation will be affected, and integral calculation will be considered. Similarly, the symbolic switching, in the amplitude-changing cylinder system, would cause the vibration of the cylinder and reduce the compensation effect. So, the differential link is considered to suppress the switching frequency. By synthesizing the two computational methods, a strategy of combining nonlinear differential with integral sliding mode parameter structure (DI-SVSC) is proposed.

Integral sliding mode control

**Non-adaptive sliding mode control of unknown parameters.** For system (10), suppose \( |\theta| < A, b < \beta < B, |d(t)| < D \).

It is difficult to calculate each derivative term in the amplitude-changing cylinder system. So, the integral terms of feedback errors are introduced into the control signal expression, and each error term is replaced with each state variable.
Definition 1. In the improved sliding mode integral control design, the function of the control signal is defined as
\[
z = c_1 x_1 + c_2 x_2 + \cdots + c_{n-1} x_{n-1} + x_n + k \int_0^t (x_1 - y_d) \, d\tau \tag{14}
\]
where \( k \) and \( c_i \) are optional constants.

Theorem 1. When system (10) is taken as the research object, the switching function of equation (14) is always equal to zero; at the same time, the amplitude-changing cylinder system reaches to balance. The expression of the transfer function of the amplitude-changing system is as follows
\[
\frac{X_1(s)}{Y_d(s)} = \frac{k}{s^n + c_{n-1}s^{n-1} + c_2 s^2 + c_1 s + k} \tag{15}
\]
where the Laplace transforms of \( X_1(s) \) and \( Y_d(s) \) are input \( x_1 \) and output \( y_d \), respectively.

It is proved that the derivative of equation (14) is
\[
z = c_1 x_2 + c_2 x_3 + \cdots + c_{n-1} x_n + x_n + k x_1 - k y_d \tag{16}
\]
For \( z = 0 \), \( \dot{z} = 0 \). So \( c_1 x_2 + c_2 x_3 + \cdots + c_{n-1} x_n + x_n + k x_1 - k y_d = 0 \).

By Laplace transformation, the transfer function of input and output in the system is as follows
\[
\frac{X_1(s)}{Y_d(s)} = \frac{k}{s^n + c_{n-1}s^{n-1} + c_2 s^2 + c_1 s + k} \tag{17}
\]
where \( \beta = \frac{b}{b} \).

The proof is completed.

According to the generalized conditions of achieving the sliding mode surface \( \dot{z} < 0 \), the following controller is designed as
\[
\mu = \frac{1}{b}(\mu_0 + \Delta \mu_1 + \Delta \mu_2) \tag{18}
\]
where \( c_1 x_2 + c_2 x_3 + \cdots + c_{n-1} x_n + x_n + k x_1 - k y_d = 0 \).

\( \Delta \mu_1 \) is used to eliminate the influence of the change of \( \theta \) and external load on the system; the values are designed as follows
\[
\Delta \mu_1 = [-A(f(x))] - D) \text{sgn}(z) \tag{19}
\]
\( \Delta \mu_2 \) is used to eliminate the influence of the change of \( \beta \) on the system. The method of collecting cylinder length is used to calculate its expression which can be expressed as follows
\[
\Delta \mu_2 = \phi_1(x_1 - y_d) + \sum_{i=2}^n \phi_i x_i \tag{20}
\]
where \( \phi_1 = \frac{b-B}{k}|\text{sgn}(ez)|, \phi_i = \frac{b-B}{k} |c_{i-1}| |\text{sgn}(x_i z)| 
\]
\( i = 2, 3, \ldots, n \).

The following theorem can be obtained from the equation: \( e = x_1 - y_d \).

Theorem 2. For system (10), when the switching function defined by Theorem 1 and the controller composed of equations (14), (18)–(20) are adopted, the transfer function between the input and the output of the whole system is as follows
\[
\frac{X_1(s)}{Y_d(s)} = \frac{k}{s^n + c_{n-1}s^{n-1} + c_2 s^2 + c_1 s + k} \tag{21}
\]
where the Laplace transforms of \( X_1(s) \) and \( Y_d(s) \) are the control signal \( x_1 \) and the return value of cylinder length \( y_d \), respectively.

It is proved that equation (18) is substituted for equation (14) and system (10) is considered
\[
\dot{z} = c_1 x_2 + c_2 x_3 + \cdots + c_{n-1} x_n + \theta f(x) + d(t) + ke + (\mu_0 + \Delta \mu_1 + \Delta \mu_2) + \frac{\beta - b}{b} (\mu_0 + \Delta \mu_1 + \Delta \mu_2) \tag{22}
\]
where \( \beta = \frac{b}{b} \).

Equations (19) and (20) are substituted into the above formula to obtain
\[
\dot{z} = \theta f(x) - A(f(x)) \text{sgn}(z) + d(t) - D \text{sgn}(z) + \frac{\beta - b}{b} [-A(f(x)) \text{sgn}(z) - D \text{sgn}(z)] + \frac{1}{b} [\beta \phi_1 - k(\beta - b)] e + \frac{1}{b} \sum_{i=2}^n [\phi_i x_i - (\beta - b) c_{i-1}] x_i \tag{23}
\]
For \( \dot{z} = 0 \). So
\[
\dot{z} = \theta f(x) - A(f(x)) |z| + zd(t) - D |z| + \frac{\beta - b}{b} [-A(f(x)) |z| - D |z|] + \frac{1}{b} [\beta \phi_1 - k(\beta - b)] e + \frac{1}{b} \sum_{i=2}^n [\phi_i x_i - (\beta - b) c_{i-1}] x_i \tag{24}
\]
For \( \dot{z} = \xi \), \( |\theta| < A \), \( |d(t)| < D \)
\[
\xi \theta f(x) - A(f(x)) |z| + zd(t) - D |z| < 0 \tag{25}
\]
For \( 0 < b < \beta < B \)
\[
\frac{\beta - b}{b} [-A(f(x)) |z| - D |z|] < 0 \tag{26}
\]
For \( \phi_1 = ((b-B)/b) |k| |\text{sgn}(ez)|, \phi_2 = ((b-B)/b) |c_1| \text{sgn}(x_1 z)| \)
\[
\frac{1}{b} [\beta \phi_1 - k(\beta - b)] e = \frac{1}{b} \left[ \beta \frac{b-B}{b} |k| e - k(\beta - b) e - \frac{b-B}{b} |k| e - k(\beta - b) e \right] \tag{27}
\]
For \( 0 < b < \beta < B \)
For the same reason

$$1 \sum_{i=2}^{n} [\phi_i x_i - (\beta - b) c_{i-1}] x_i z < 0 \quad \text{(31)}$$

According to those equations, $z \dot{z} < 0$.

When the deviation $z$ is greater than zero, the deviation change rate is less than zero, and the system has the tendency of moving to zero; when the deviation $z$ is less than zero, the deviation change rate is greater than zero, and the system has the tendency of moving in the opposite direction to zero. When the amplitude-changing cylinder system reaches the sliding surface, the cylinder will continue to expand and contract alternately along the sliding surface to complete sine and cosine wave compensation, and the angle of the manipulator will remain constant. This state is not disturbed by external system and can be automatically positioned. From Theorem 1, the transfer function between cylinder length and control signal is obtained as follows

$$X_1(s) \over Y_d(s) = \frac{k}{s^n + c_{n-1}s^{n-1} + c_2 s^2 + c_1 s + k} \quad \text{(32)}$$

According to the transfer function (equation (15)) of the valve-controlled cylinder system of salvage equipment, the characteristic equation of the amplitude-changing system is as follows

$$s^n + c_{n-1}s^{n-1} + c_2 s^2 + c_1 s + k = 0 \quad \text{(33)}$$

According to the controllers (equations (18) to (20)) designed above, the integral sliding mode control method is used only with the expected length $y_d$ of the cylinder length, not including the third, fourth, and fifth derivatives, which eliminates the assumption that the feedback signal and multi-derivatives must exist in the previous SMC.

**Adaptive sliding mode control of unknown parameters.** Integral sliding mode adaptive control can control the length of cylinder in real time and keep the angle of manipulator constant. However, in the process of maintaining the angle of interception constant, the output control signal of the controller will update and change in real time under the action of adaptive calculation, so the completely amplitude-changing cylinder system is constantly switching. This will result in non-smooth motion compensation in the system. When the switching is too frequent, it may cause damage to components, such as the piston rod oil seal damage in the luffing system, the high oil temperature, and the clamp position of the electro-hydraulic proportional directional valve. According to the unknown characteristics of system parameters, the parameter adaptive control system is added on the base of integral sliding mode control, as shown in Figure 5.

For the adaptive integral sliding mode control system, the parameters of the controller are constantly changing under the action of the adaptive law. In the initial stage of the adaptive calculation, the trend of parameter change is generally unchanged. The estimated parameters are in a large range. When the estimated parameters are approaching the actual values slowly, the estimated parameters are displayed in the form of up-down oscillation near the actual values. The continuous switching of control values directly manifests this kind of handover in a small range.

$$\phi(e) \text{ is a nonlinear function and should satisfy two conditions of deviation nonlinear function in the previous chapter.}$$

The design here is still as follows

$$\phi(e) = \frac{1}{\eta e^2 + \varepsilon} \quad \text{(34)}$$

where $e = y - y_d = x_1 - y_{ds}$, $\eta$, and $\varepsilon$ are positive constants.

Adaptive integral sliding mode controller is designed as follows

\[ \text{Switching function design} \]

\[ \text{Integral sliding mode controller} \]

\[ \text{System} \]
\[
\mu = \frac{1}{\beta g(x)} \cdot (\mu_0 + \Delta \mu_1 + \Delta \mu_2)
\]  
(35)

Select the adaptive law as follows
\[
\begin{align*}
\dot{\theta} &= -\frac{\theta f(x)}{\theta^2 + \phi(\epsilon)g(x)} \\
\dot{\beta} &= -\frac{\beta g(x)\mu}{\gamma^2 + \phi(\epsilon)\beta^2} \\
\dot{d} &= -\frac{d}{\lambda^2 + \phi(\epsilon)}
\end{align*}
\]  
(36)

Theorem 3. When the controller is composed of formula (35) and the adaptive rate (equation (36)) in system (10), the system is asymptotically stable.

It is proved that system (10) and the controller composed of equations (37) and (38) can obtain
\[
\dot{z} = \frac{1}{1 + \phi(\epsilon)} \left[ \theta f(x) + \beta g(x) + \dot{d} - \dot{h}z \right]
\]  
(37)

Likewise, the Lyapunov function is defined as
\[
V = \frac{1}{2} z^2 + \frac{1}{2} \theta^2 + \frac{1}{2} \beta^2 + \frac{1}{2} \lambda d^2
\]  
(38)

Derivatives from equation (38) are derived as follows
\[
\dot{V} = \frac{\dot{\theta} f(x) + \beta \dot{g}(x)\mu - \dot{h}z^2}{1 + \phi(\epsilon)} + z \ddot{d}(t) + \theta \ddot{\theta} - \gamma \beta \ddot{\beta} - \lambda \ddot{d}
\]  
(39)

The adaptive law (equation (40)) is substituted for the upper equation
\[
\dot{V} = -\frac{h \dot{z}^2}{1 + \phi(\epsilon)} \leq 0
\]  
(40)

Due to the same reason, the system is still asymptotically stable. When \(t \rightarrow \infty, z \rightarrow 0\) and it will always be zero. It can also be seen from Theorem 1 that the system has the transfer function shown in equation (13). The proof is completed.

**Nonlinear differential and integral sliding mode control**

Due to the existence of discontinuous switching function \(\text{sgn}(z)\) in the sliding mode controller, the phenomenon that the vibration of the amplitude-changing cylinder compensates the angle of the manipulator will appear. In the previous formula function \(\text{sgn}(z)\), the function \(z/(|z| + \epsilon)\) is used to replace it. In this section, the nonlinear differential is used to suppress the rate of change.

In SMC, the essential cause of system vibration is the constant change of \(z\) value of switching function. Therefore, nonlinear differential control is implemented to keep the value of switching function stable, thereby weakening the vibration expansion of amplitude-changing cylinder system. When the absolute value of displacement tracking of variable amplitude-changing cylinder is close to zero, proper increase of differential coefficient will slow down the switching speed of amplitude-changing cylinder. In addition, when the absolute value of displacement tracking of variable amplitude cylinder is larger, the differential coefficient will be reduced to compensate the angle deviation quickly; thus, the effect of differential suppression compensation will be reduced. Figure 6 shows the block diagram of the whole system.

The compensation expression of the nonlinear differential controller is as follows
\[
\mu_d = -s(\epsilon) \dot{z}
\]  
(41)

where the differential coefficient \(s(\epsilon)\) in equation (41) is a nonlinear function expression, and its variation law is as follows:

1. \(s(\epsilon) > 0\);
2. When \(\epsilon > 0\), \(s(\epsilon)\) decreases with the increase of deviation; when \(\epsilon < 0\), \(s(\epsilon)\) increases with the increase of deviation.

In this paper, the nonlinear function chosen first is as follows
\[
s(\epsilon) = \frac{1}{\epsilon + \eta \epsilon^2}
\]  
(42)
where \( e = y - y_d = x_1 - y_d \), \( \eta \), and \( \varepsilon \) are positive constants.

It can be seen from equation (42) that \( s(e) > 0 \) and the value of the function \( s(e) \) decreases with the increase of the value of the function \( |e| \). Therefore, the two characteristics (1 and 2) of the nonlinear function \( s(e) \) are satisfied. The differential controller composed of equations (41) and (18) are simultaneously used to be given by

\[
\mu = \frac{\mu_c + \mu_d}{bg(x)}
\]  

(43)

where \( \mu_c = \mu_0 + \Delta \mu_1 + \Delta \mu_2 \).

The following theorems can also be obtained.

**Theorem 4.** For system (10), if controller composed of equation (43) is adopted, the transfer function between the control signal and the displacement of the amplitude-changing cylinder of the whole system is still as follows

\[
\frac{X_1(s)}{Y_d(s)} = \frac{k}{s^n + c_{n-1}s^{n-1} + c_2s^2 + c_1s + k}
\]  

(44)

where the Laplace transforms of \( X_1(s) \) and \( Y_d(s) \) are the control signal \( x_1 \) and the displacement of the amplitude-changing cylinder \( y_d \), respectively.

It is proved that the switching function and its derivatives of rewriting equation (15) are given by

\[
z = c_1x_1 + c_2x_2 + \cdots + c_{n-1}x_{n-1} + x_n
\]  

+ \( k \int_0^t (x_1 - y_d) dt \) \hspace{1cm} (45)

\[
z = c_1x_2 + c_2x_3 + \cdots + c_{n-1}x_{n-1} + x_n + kx_1 - ky_d
\]  

(46)

The controller composed of equation (43) is substituted into the upper formula to obtain the derivative of the switching function

\[
\dot{z} = c_1x_2 + c_2x_3 + \cdots + c_{n-1}x_{n-1} + \theta(x) + d(t) + ke
\]  

+ \( \frac{B}{\beta} \mu_c + \frac{B}{\beta} \mu_e = \dot{\varphi} + \frac{\beta}{\beta} s(e) \dot{z}
\]  

(47)

So

\[
\dot{z} = \frac{\varphi}{1 + \frac{\beta}{\beta} s(e)}
\]  

(48)

\[
z = \frac{z\varphi}{1 + \frac{\beta}{\beta} s(e)} = \frac{z\xi}{1 + \frac{\beta}{\beta} s(e)} \leq 0
\]  

(49)

Therefore, the amplitude-changing cylinder system can still reach the switching surface \( z = 0 \), and there is a stable running state of the sliding surface. Similarly, the transfer function between the input and the output of the amplitude-changing cylinder of the salvage equipment is still as follows

\[
\frac{X_1(s)}{Y_d(s)} = \frac{k}{s^n + c_{n-1}s^{n-1} + c_2s^2 + c_1s + k}
\]  

(50)

The proof is completed.

It can be seen from equation (49) that when the angle deviation of the manipulator becomes larger, it causes larger tracking error of the length of the amplitude-changing cylinder, smaller value of \( 1 + (\beta/b)s(e) \), and larger calculation result of the \( z \) value of the switching function. Therefore, the angle deviation of the manipulator tends to zero as soon as possible and the absolute value of the tracking error \( e \) of the displacement of the amplitude-changing cylinder decreases rapidly. When the angle deviation of the manipulator becomes smaller, the tracking error of the length of the amplitude-changing cylinder decreases correspondingly, and the value of \( 1 + (\beta/b)s(e) \) increases gradually. The calculation result of the \( z \) value of the switching function becomes smaller, which weakens the vibration of the amplitude-changing cylinder. This is also verified in the later simulation calculation.

**NDISMC for electro-hydraulic servo velocity tracking**

According to Theorem 1, the control function is designed as follows

\[
z = c_1x_1 + x_2 + k \int_0^t \xi dt
\]  

(51)

where \( e = y - y_d \).

The derivation of equation (52) is as follows

\[
\dot{z} = c_1x_2 + \dot{x}_2 + kx_1 - ky_d
\]  

(52)

The following controller is designed

\[
\mu_c = \frac{1}{\beta g(x)} (\mu_0 + \Delta \mu_1 + \Delta \mu_2)
\]  

(53)

where \( \mu_0 = -c_1x_2 - k(x_1 - y_d) \).

\( \Delta \mu_1 \) is used to eliminate the influence of the change of \( \theta \) and external load on the system. Its values are designed as follows

\[
\Delta \mu_1 = [-A|f(x)] - D|\text{sgn}(z)
\]  

(54)

\( \Delta \mu_2 \) is used to eliminate the influence of the change of \( \beta \) on the system. The method of using amplitude-changing cylinder length compensation is used to design the control quantity

\[
\Delta \mu_2 = f_1(x_1 - r) + f_2x_2
\]  

(55)

Due to \( |\theta| < A, b < \beta < B, |d(t)| < D \) and Theorem 1, when the controller composed of equations (51)-(54) is adopted, the transfer function between the input and the output of system (10) is as follows

\[
\frac{X_1(s)}{Y_d(s)} = \frac{k}{s^n + c_{n-1}s^{n-1} + c_2s^2 + c_1s + k}
\]  

(56)
where the Laplace transforms of $X(s)$ and $Y_d(s)$ are the control signal $x_1$ and the displacement of the amplitude-changing cylinder $y_d$, respectively. The nonlinear differential controller is designed as follows

$$\mu_d = -g(e)\dot{z}$$

where $g(e) = 1/(e + h\dot{e})$ is the nonlinear function. $\mu_c = \mu_0 + \Delta\mu_1 + \Delta\mu_2$

$$\mu = \frac{1}{\beta g(x)}(\mu_c + \mu_d)$$

Similarly, due to Theorem 2, when the controller composed of equations (51), (57), (58) is adopted, the transfer function between the input and the output of salvage equipment amplitude-changing system (10) is still as follows

$$\frac{X(s)}{Y_d(s)} = \frac{k}{s^3 + c_1s + k}$$

Therefore, the system also has strong anti-disturbance ability.

**Simulation results**

According to the real ship-borne manipulator, the parameters of the system are chosen as follows: $c_1 = 10$, $c_2 = 2$, $k = 8$, $\beta_c = 1.05$, $C_r = 0.1$, $\dot{e} = 1.5$, $\dot{h} = 6.3$, $A = 21$, $b = 3$, $B = 18$, and $D = 30$. Subsequently, the performance of the NDISMC is divided into the following two situations for simulation:

1. Case 1: Step response;
2. Case 2: Tracking control of wave compensation.

In order to verify the effectiveness of the proposed control method, the simulations of NDISMC are conducted in the MATLAB/Simulink environment. The simulation results are provided in Figures 7–10.

As shown in Figures 7–10, the static error under the proposed improved sliding mode control is small. Despite the wave disturbance, the tracking error with NDISMC under the wave compensation condition is rather small and can satisfy the requirement of the wave compensation systems. Although the controller outputs have high-frequency oscillation, which is called chattering, the chattering phenomenon can be reduced significantly with NDISMC. To some extent, these results show that the proposed improved sliding mode controller is robust facing external disturbances and parameter perturbations.

**Hardware experiments**

The model of high sea condition ship-borne manipulator on the rolling motion platform is shown in
Figure 11, which includes a luffing cylinder mechanism to compensate for the swaying of ship hull in Z direction. The rolling motion platform simulates the rolling of ship motion and simulates the motion of different sea conditions by setting the inclination angle and frequency.

The counter of Siemens S7-1200 controller can collect the pulse of the cylinder’s expansion length and the analog function is used to collect the pressure of the cylinder’s cavity and chamber. The 485 serial port is used to read the angle of the manipulator. The analog value of the controller outputs 4–20 mA current. The amplified current after the proportional amplifier controls the electro-hydraulic proportional directional valve, as shown in Figure 12.

The valve-controlled cylinder system is mainly composed of inclination sensor, PLC control circuit and amplifier, and so on. In order to fully study the proposed control scheme, two groups of experiments were conducted to detect the tracking performance, anti-disturbance ability, and dynamic performance under disturbance. One group is PID, the other is NDISMC, as shown in Figure 13.

Figure 11. Experimental platform of ship-borne manipulator.

Figure 12. Hydraulic and acquisition system.

Figure 13. Wave compensation effect of ship-borne manipulator under proposed NDISMC controller: (a) experimental results of wave compensation for obliquity of manipulator, (b) cylinder large cavity pressure, and (c) small chamber pressure of cylinder.
We can learn from Figure 13, after wave compensation, the amplitude of ship-borne manipulator can be kept within 3.0° with the proposed controller. The pressure of the cylinder varies steadily, and the distribution is less than 40 bar. Robust tracking control method of the manipulator can achieve superior control performance on the premise of smooth control quantity and has satisfactory robustness under wave disturbance and parameter perturbation. The disadvantage of this control method is that the actual application programming is more complex and time-consuming than PID controller. The statistics of experimental results are listed in Table 1.

### Table 1. Statistics of experimental results.

| Item                          | PID         | NDISMC       |
|-------------------------------|-------------|--------------|
| Maximum angle deviation       | 3.29°       | 1.27°        |
| Average angle deviation       | 1.47°       | 0.76°        |
| Minimum air pressure of cylinder large cavity pressure | 79.7 bar    | 78.9 bar     |
| Maximum air pressure of cylinder large cavity pressure | 91.9 bar    | 91.1 bar     |
| Minimum air pressure of small chamber pressure of cylinder | 83.7 bar    | 82.9 bar     |
| Maximum air pressure of small chamber pressure of cylinder | 113.2 bar   | 110.1 bar    |

PID: proportional–integral–derivative; NDISMC: nonlinear differential and integral sliding mode controller.

### Conclusion

Aiming at the external disturbance and system parameter perturbation during the operation of the valve-controlled cylinder in the wave compensation system of the ship-borne manipulator, an improved sliding mode controller is proposed by analyzing the dynamic model and constructing the sliding mode manifold, which can realize the manipulator angle fast and accurately and resist the wave disturbance effectively. The simulation and experiment results show that the control algorithm can achieve better control effect than the traditional PID and sliding mode control method. Compared with the existing methods, this method can basically eliminate the chattering phenomenon in sliding mode control with better compensation effect and enhance the robustness of the control system. The inclination angle can be reduced significantly in harsh sea conditions. In conclusion, the proposed control law can improve the stability performance of the ship-borne manipulator effectively under the action of waves and enhance the operational ability and safety of the ship-borne manipulator in the marine environment. Our future works will study the application of the proposed control law to other marine equipment’s wave compensation system.

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