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Cascade Control of Active Heave Compensation Nonlinear System for Marine Crane

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Abstract: During the rough marine environment, heave compensation is used to offset the heave motion of the vessel when a marine crane lifts and lands the load. Thus, load motion and vessel motion are realized decoupled. In previous studies, the interference items such as hydraulic cylinder friction, underwater drag force and nonlinear friction in the active heave compensation system of a marine hydraulic crane are compensated as a concentrated interference force to be estimated. In this paper, we disassembled the interference items; the disturbance observer and adaptive rate are designed to estimate unmodeled disturbance force and system uncertain parameters, respectively; and we designed an active heave compensator with the adaptive nonlinear cascade controller which has the disturbance observer (DOB-ANCC). For the heave compensation of load displacement, this paper derived the control law of the nonlinear system model based on the backstepping method. The outer loop control is displacement control and the inner loop control is pressure control. The simulation verifies the effectiveness of the control strategy proposed in this paper and the availability of heave displacement compensation for a marine crane hoisting load. The compensation efficiency of the designed controller (DOB-ANCC) for the heave motion of the load can reach more than 95%, and the maximum displacement tracking error of the controller can reach $\pm$ 0.035 m.

Keywords: nonlinear cascade control; parameter adaptive law; disturbance force observer; hydraulic servo system; active heave compensator

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1. Introduction

When operating in the formidable marine environment with wind, waves and so on, vessels generally have six degrees of freedom motion, such as roll, pitch, yaw, surge, sway and heave. Particularly, the heave motion has the most adverse impact on the offshore crane operating system, which will lead to the instability of the offshore operating device and seriously affect the work efficiency and operational security. In the future, offshore oil and gas fields will be developed in a large area, and processing and transportation equipment will be carried out at sea [1]. Therefore, in the harsh sea conditions, to ensure that the marine crane can be stable, a lifting load needs to rely on the support of the compensation system which can improve the high sea state adaptability of equipment. When marine crane lifting load operation needs to be completed in severe sea conditions, in order to achieve efficient development efficiency, offshore operations require high operational and high-precision equipment system support. Therefore, the design and application of AHC system controller is particularly important [2].

Southerland [3] first proposed active and passive heave compensation system solutions. The passive heave compensation system is designed to keep the line tension of the lifting load constant [4]. The active heave compensation system uses a simple mechanical feedback system to reduce the impact of vessel motion. At the beginning, a passive heave compensation system was widely used, but with the progress of the control system, a more accurate and advanced control algorithm being verified and applied, and the progress of the hydraulic system as faster and more efficient, active heave compensation or a hybrid active–passive heave compensation system is more respected and has gained the attention of researchers. A complete linear drill string model was proposed by Korde [5]. The control system for the active heave compensation device on the drilling vessel is based on linear control. The theoretical results show that the designed control system can completely decouple the load motion and the vessel motion, but its decoupling premise is based on idealized calculation. Based on the linear drill string model established by Korde, Hattleskog and Dunnigan, [6] built a linear transfer function model applied to the active–passive hybrid system and adopted feedback displacement control and a PD feedback loop for actuator position feedback. However, nonlinear systems are converted into ideal linear system models, and the friction effects are mostly considered as linear models or ignored in practical applications. In the heave compensation system of the hydraulic system, friction will have an adverse effect on the system, such as the stick–slip of the cylinder, which is not easily overcome by the load, and it will also cause bad vibration of the compensator, so there are still some control problems in the heave compensation system based on many practical applications. In the AHC system as a typical electro-hydraulic servo system, the external disturbance and parameter uncertainty are unavoidable. Load disturbance force and unmodeled nonlinear friction force for modeling will also significantly reduce the system position tracking performance [7].

Focusing on the disturbance problem of the above system, Do et al. designed a nonlinear control system based on Lyapunov's law and a disturbance observer for the electro-hydraulic system driven by a two-link actuator [8]. Shaara and Egeland proposed a parallel force/position controller control system for offshore operations [9]. Neupert et al. designed a disturbance decoupling controller to track the desired trajectory to manipulate the crane winch [10]. In addition, in order to improve the position tracking performance of the electro-hydraulic servo system, researchers have designed more diverse control methods. Sliding mode control is used to solve the instability of a electro-hydraulic system [11], and adaptive control [12], pole placement and other control techniques are used to solve the local linearization problem of a nonlinear dynamic system. Yao et al. proposed nonlinear adaptive robust control for trajectory tracking of hydraulic actuators with nonlinear and uncertain parameters [13]. Since the disturbance is unmeasurable in practical applications, it is necessary to design a disturbance observer to estimate the disturbance force in the system. Han developed an extended state jammer (ESO), which can estimate generalized disturbances online without requiring a specific model [14].

Waves in extreme weather affect the load of the crane by various nonlinear factors, including nonlinear drag force, viscous damping force and inertia force caused by the vessel heave motion, friction between the mechanical structures and nonlinear disturbance generated by the hydraulic circuit itself [15]. In this paper, cascade control of an active heave compensation nonlinear system for a marine hydraulic crane is carried out. The displacement of an hydraulic cylinder is compensated by reverse control to realize the effective compensation of the load with vessel heave motion. For uncertain parameters such as drag force and viscous damping force, this paper uses an adaptive control method to estimate them. For the nonlinear friction generated by the hydraulic circuit and the mechanical structure, a disturbance observer is proposed in this paper to estimate the unmodeled friction. Furthermore, we built the active heave compensation nonlinear controller based on an asymmetric hydraulic cylinder-driven electro-hydraulic servo system. Nonlinear control can ensure the accuracy of the system model [16]. The main control strategy of the controller is to derive the control law according to the mathematical model of the crane system based on the backstepping method. The adaptive nonlinear cascade controller (DOB-ANCC) with a disturbance observer is designed according to Lyapunov's law to ensure the stability of the closed-loop system. In the designed cascade control structure, complete control is divided into inner and outer loop control parts: outer loop control is the displacement controller and inner loop control is the pressure controller.

2. Dynamic Model of AHC System

2.1. AHC System Description

Marine cranes will be affected by wind, waves and current when working at sea, resulting in complex spatial motion. The load movement of crane hoisting will be affected by the vessel's heave movement, which will seriously affect the operation efficiency of the hoisting load and also cause safety accidents [17]. The active heave compensation system designed in this paper takes the four-level sea state as the working sea state, and the rated load is 50 t. In this paper, the ITTC two-parameter spectrum is used to describe the spectral density of waves. It is known that the significant wave height H = 2.5 m, the average period T = 8.8 s, as shown in Figure 1. It can be seen from the diagram that in the four-level sea condition, the energy generated by the wave motion is mainly concentrated in the frequency range of 0.50 to 0.58 rad/s, and the wave energy is the highest when the frequency is about 0.56 rad/s. In order to simulate the waves in the harsh four-level sea conditions, this paper chooses a sinusoidal signal with an amplitude of ±3 m and a frequency of 0.1 Hz as the heave displacement of the vessel. The image of the vessel's heave motion is shown in Figure 2.

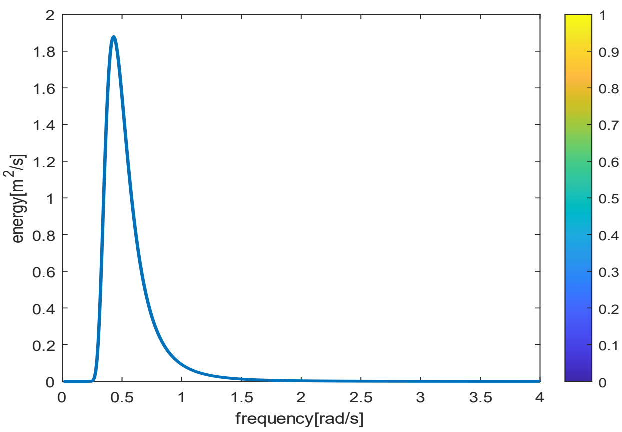

Figure 1. ITTC wave spectrum of Level 4 sea state.

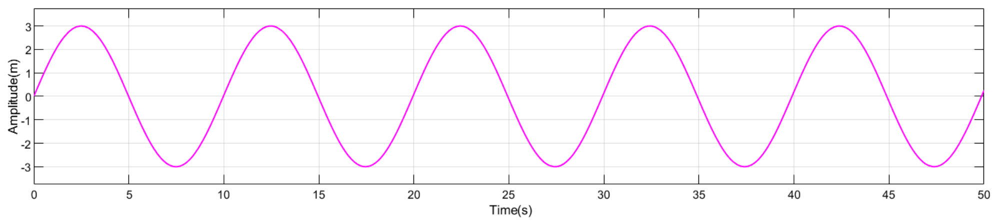

Figure 2. Vessel heave motion image.

The installation arrangement of the active heave compensation system device of the marine crane on the vessel is shown in Figure 3, and the active heave compensation equipment and load are lifted by the crane. The sensor detects the heave of the wave, and after the controller calculates, it controls the extension and retraction of the cylinder to compensate for the influence of wave heave on the lifting load, which can greatly improve the safety of offshore operations and the lifting capacity of the crane. The control goal of the active heave compensation system is to use the displacement of the hydraulic compensation cylinder x_p to reversely track and compensate the heave motion of the vessel x_h, so as to keep the altitude of the load x_l relative to the geocentric coordinate system unchanged. In the following, the initial altitude of the load is relative to the geocentric coordinate.

$$x_l = x_{l0} + x_h - x_p \tag{1}$$

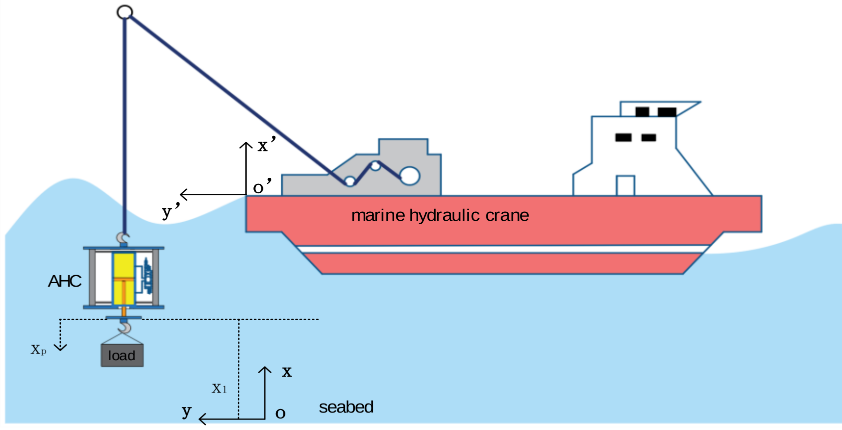

Figure 3. Layout of AHC system on marine crane.

2.2. Mathematical Model of AHC System

The AHC system can offset the uncertain parameters and friction of the compensator, which is usually more effective than PHC. The active heave compensation system designed in this paper is shown in Figure 4. A single piston hydraulic cylinder is mainly used as the active compensation cylinder, proportional servo valve and displacement sensor to compensate the load displacement. The high-frequency servo valve connects the two chambers of the piston hydraulic cylinder to form an active heave compensation hydraulic servo system. The compensation principle of the active heave compensation system is that the displacement sensor detects the heave displacement of the mother vessel, uses this signal to control the current of the servo valve, controls the oil flow in the hydraulic cylinder chamber to reversely control the telescopic displacement of the piston rod, and compensates the lifting load with the change of the vessel's heave displacement to realize the heave compensation of the whole system. In the following, the nonlinear mathematical model of the active heave compensation hydraulic servo system will be established.

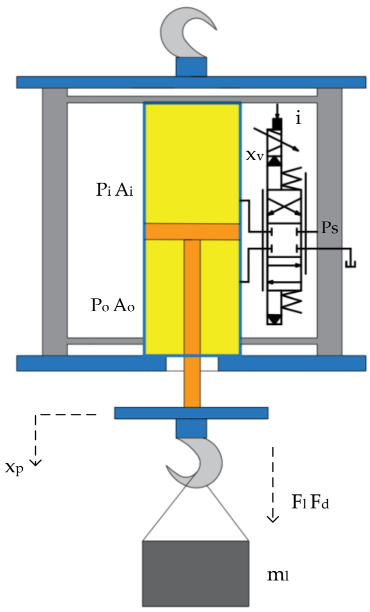

Figure 4. Schematic diagram of the AHC system.

When the crane is working at sea, the movement of the waves will cause various nonlinear factors to interfere with the lifting load of the crane, including the drag force, viscous damping force, buoyancy, inertial force, nonlinear parameters and disturbances generated by the hydraulic system itself. The disturbance term of the load is added to the force balance equation of the active heave compensation system as follows:

$$\begin{array}{l} m_l \ddot{x}_p = P_i A_i - P_o A_o - F_l \\ F_l = F_d + b_h \dot{x}_p + \Delta \end{array} \tag{2}$$

The equivalent mass of the load is m_l, x_p is the displacement of the hydraulic compensation cylinder, and P_i, A_i, P_o, A_o represents the pressure and piston area of the non-rod cavity and the rod cavity, respectively. Where F_l is the total disturbance force of the load, F_d is the nonlinear drag force generated due to the relative motion of the load and the fluid. b_h is the damping and viscous friction coefficient of the hydraulic cylinder, and Δ is the other unmodeled disturbance force on the load, which is mainly the nonlinear friction disturbance force in the hydraulic system. This paper does not ignore the nonlinear friction disturbance force Δ. In the following work, a disturbance observer will be designed to estimate it to achieve the control goal of the controller.

The drag force F_d on the relative motion of the load in water can be expressed as:

$$F_d = -\frac{1}{2} \cdot \rho_w \cdot A_l \cdot C_d \cdot \dot{x}_l \cdot \left| \dot{x}_l \right| - d_r \cdot \dot{x}_l \tag{3}$$

In the formula, ρ_w is the density of seawater, A_l is the cross-sectional area of the load along the heave direction, C_d is the drag coefficient of the load in the water and d_r is the load viscosity coefficient. For the actual application scenario, in order to facilitate the modeling and analysis of the hydraulic cylinder continuity equation, the following reasonable assumptions are made:

1. The hydraulic pipeline is short, and the friction loss and dynamic of the pipeline are neglected;
2. The pressure of each working chamber of the hydraulic cylinder is equal;
3. The temperature and bulk modulus of elasticity of hydraulic oil are constants;
4. The hydraulic cylinder ignores the external leakage, and the internal leakage is laminar flow [18].

The dynamic equation of the active compensation hydraulic cylinder is as follows:

$$\begin{array}{l} Q_i = A_i \dot{x}_p + C_{ip}(P_i - P_o) + \frac{V_i}{\beta_e} \dot{P}_i \\ Q_o = A_o \dot{x}_p + C_{ip}(P_o - P_i) - \frac{V_o}{\beta_e} \dot{P}_o \end{array} \tag{4}$$

Q_i is the flow of hydraulic oil into the non-rod chamber of the hydraulic cylinder, Q_o is the flow rate of the rod cavity of the hydraulic cylinder. β_e is the effective bulk elastic modulus of hydraulic oil, and C_{ip} is the internal leakage coefficient of the hydraulic cylinder. V_i and V_o are the working volume of the non-rod cavity and rod cavity, respectively, and their relationship with the initial volume $V_{i1} V_{o2}$ is:

$$\begin{array}{l} V_i = V_{i1} + A_i x_p \\ V_o = V_{o2} - A_o x_p \end{array} \tag{5}$$

The servo proportional valve is connected to the active compensation hydraulic cylinder which is controlled by hydraulic oil. The O-type median function servo valve is used. According to the valve port flow equation of the servo valve, the relationship between the flow of the hydraulic cylinder and the displacement x_v of the servo valve spool can be obtained. For the servo proportional valve, the slide valve structure is adopted, and the assumption is made to facilitate mathematical modeling:

1. The slide valve is an ideal four-side slide valve with zero opening, four throttle ports matching and symmetry;
2. The flow at the throttling window is turbulent;
3. Flow variation in response to valve spool displacement and valve pressure drop can occur instantaneously.

$$Q_i = \begin{cases} k_d x_v \sqrt{|P_s - P_i|} sgn(P_s - P_i)(x_v \geq 0) \\ k_d x_v \sqrt{|P_i|} sgn(P_i)(x_v < 0) \end{cases}$$
$$Q_o = \begin{cases} k_d x_v \sqrt{|P_s - P_o|} sgn(P_s - P_o)(x_v < 0) \\ k_d x_v \sqrt{|P_o|} sgn(P_o)(x_v \geq 0) \end{cases} \quad (6)$$

where $k_d = C_d \cdot w \cdot (2/\rho)^{1/2}$, ρ is the density of hydraulic oil. The flow coefficient C_d of the valve port and the area gradient w of the orifice are difficult to measure, so k_d can be obtained from the technical data of the valve. P_s is system oil supply pressure. x_v is servo valve spool displacement, the negative value indicates that the spool displacement is opposite to the positive direction of the regulation. Ignoring the second-order dynamic characteristics of the servo valve, the simplified relationship between the valve spool displacement x_v and the control current command of the servo valve u can be obtained as follows, where k_x is the proportional gain of the valve spool:

$$x_v = k_x u \quad (7)$$

The amplifier of the servo valve provides a specific current to the proportional electromagnet to control the displacement of the servo valve spool and controls the hydraulic oil flow at the inlet and outlet of the active hydraulic cylinder, thereby controlling the extension and retraction of the cylinder. Thus, the influence of the heave motion of the vessel with the wave on the lifting load of the crane is compensated. The working schematic diagram of the active heave compensation hydraulic servo system of the marine crane is as Figure 5:

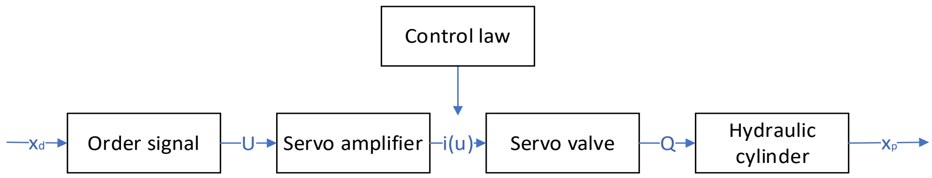

Figure 5. Working schematic diagram of crane AHC electro-hydraulic servo system.

For the actual active compensation, the compensation element is shown in Figure 6. In the active heave compensation system, the spool displacement of the servo valve is controlled by the industrial computer, and the hydraulic power is provided to the servo valve. The servo valve controls the motion of the active compensation hydraulic cylinder. The displacement sensor detects the displacement of the hydraulic cylinder and transmits the displacement signal of the hydraulic cylinder back to the industrial computer through the converter, forming a closed-loop control.

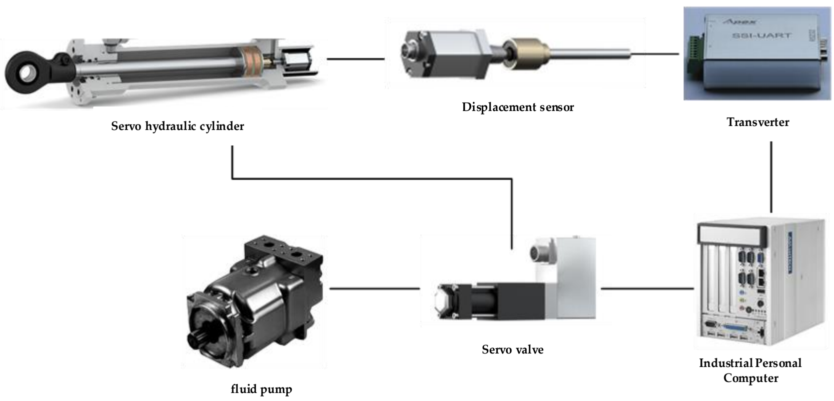

Figure 6. Actual control elements of active heave compensation system.

The valve port flow equation Formula (6) of the servo valve is brought into the dynamic equation Formula (5) of the hydraulic cylinder, and the dynamic equation of P_i, P_o

and the compensation displacement x_p of the hydraulic cylinder is solved. Combined with Formula (1), the mathematical model of the whole active compensation hydraulic servo system can be defined. The state variable of the system is defined as $[x_1, x_2, x_3, x_4] = [x_p, \dot{x}_p, P_i, P_o]$; then, the state space equation of the system is:

$$\begin{array}{l} \dot{x}_1 = x_2 \\ \dot{x}_2 = \frac{1}{m_l}\left[A_i x_3 - A_o x_4 - b_1 x_2 - b_2 f_1^2(x_2, \dot{x}_H) - \Delta\right] \\ \dot{x}_3 = h_1(x_1)\left[k_d x_v f_2(x_3, u) - A_i x_2 - C_{ip}(x_3 - x_4)\right] \\ \dot{x}_4 = h_2(x_1)\left[-k_d x_v f_3(x_3, u) + A_o x_2\right] \end{array} \tag{8}$$

where the damping and viscous friction coefficients of the hydraulic cylinder are $b_1 = b_h$, and the nonlinear drag coefficient of the load is $b_2 = \frac{1}{2} \cdot \rho_w \cdot A_l \cdot C_d$, Δ is the nonlinear friction disturbance force in the hydraulic system and

$$\begin{array}{l} h_1(x_1) = \beta_e/(V_{i1} + A_1 x_1), h_2(x_1) = \beta_e/(V_{o2} + A_2 x_1) \\ f_1(x_2, \dot{x}_h) = k(\dot{x}_h - x_2) \\ f_2(x_3, u) = \begin{cases} k_d x_v \sqrt{|P_s - P_i|} sgn(P_s - P_i)(x_v \geq 0) \\ k_d x_v \sqrt{|P_i|} sgn(P_i)(x_v < 0) \end{cases} \\ f_3(x_4, u) = \begin{cases} k_d x_v \sqrt{|P_s - P_o|} sgn(P_s - P_o)(x_v < 0) \\ k_d x_v \sqrt{|P_o|} sgn(P_o)(x_v \geq 0) \end{cases} \end{array} \tag{9}$$

The control objective of the system is to design a bounded control law such that x_1 can accurately track the expected value x_d when there are uncertain parameters and interference terms in the system, so that the load displacement remains unchanged relative to the altitude of the geocentric coordinate system. The following assumptions are made for the actual active heave compensation system:

Assumption 1. *The displacement of hydraulic cylinder x_p, velocity $\dot{x}_p$, acceleration $\ddot{x}_p$ and P_i, P_o are all bounded, i.e.,* $0 < P_i < P_s$, $0 < P_o < P_s$.

Assumption 2. *The unmodeled nonlinear friction force Δ is continuous and bounded and satisfies the following conditions:*

$$\begin{cases} \Delta_{\min} \leq \Delta \leq \Delta_{\max} \\ \left|\dot{\Delta}\right| \leq \lambda \end{cases} \tag{10}$$

where $\Delta_{\max}$ and $\Delta_{\min}$ are the maximum and minimum values of the disturbance force limit, respectively. λ is the maximum value of the change rate of the disturbance force. The system is simplified to a strict feedback form, and a new virtual state variable is defined, $\overline{x}_3 = x_3 - \alpha x_4$, where $\alpha = A_o/A_i$ represents the piston area ratio. Therefore, the dynamic equation of the rewritten marine hydraulic AHC system is:

$$\begin{array}{l} \dot{x}_1 = x_2 \\ \dot{x}_2 = \frac{1}{m_l}[A_i \overline{x}_3 - b_1 x_2 - b_2 g_1 - \Delta] \\ \dot{\overline{x}}_3 = -g_3 x_2 + g_4 u - h_1(x_1) C_{ip}(\beta\beta_e - \overline{x}_3) \\ g_1 = f_1(x_2, \dot{x}_h) \\ g_3 = h_1(x_1) A_i + \alpha h_2(x_1) A_o \\ g_4 = k_d k_x [h_1(x_1) f_2(x_3, u) + \alpha h_2(x_1) f_3(x_4, u)] \end{array} \tag{11}$$

When the AHC system of the vessel crane is lifting the load underwater, the lifting load is affected by the heave motion of the wave. In the motion equation of the active heave compensation system, the damping and viscous friction coefficient b_1 of the hydraulic cylinder will be affected by oil pressure and temperature, so it has time-variance and uncertainty. The nonlinear drag coefficient b_2 of the load is also uncertain. In addition, Δ is the nonlinear friction disturbance force in the hydraulic system, which will also have an interference effect on the system. Therefore, these uncertainties need to be estimated in the design of the system controller.

3. DOB-ANCC Controller Design

The system control is divided into an outer loop displacement controller and an inner loop pressure controller. Based on the backstepping method, the inner and outer loop control of the system is deduced and the control law u of the compensation system is designed to control the displacement output x_p of the hydraulic cylinder, so as to realize the compensation of the load heave motion in accordance with the nonlinear mathematical model derived above for the active heave compensation system. This paper designs the parameter adaptive rate to estimate the uncertain parameters $b_1 b_2$ in the system, for the influence of nonlinear friction disturbance force Δ, which is estimated by the designed disturbance observer. In this paper, an adaptive nonlinear cascade controller with a disturbance observer (DOB-ANCC) is designed. The control block diagram is as Figure 7:

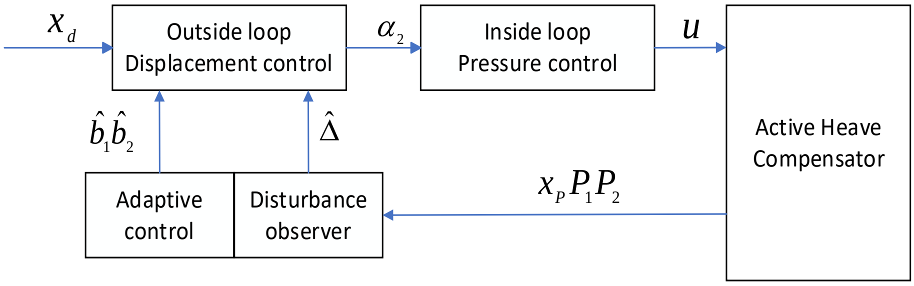

Figure 7. Block diagram of DOB-ANCC controller structure.

3.1. Adaptive Law Design for Uncertain Parameters

In the crane compensation hydraulic servo system, a change of oil pressure and temperature will lead to a change of the hydraulic cylinder damping coefficient b_1, which is uncertain. The drag force of the system will also be affected by the movement of the load in the water, so the drag force parameters b_2 are also time-varying. In this paper, the parameter adaptive control method is used to design the adaptive rate of uncertain parameters $b_1 b_2$ and estimate them. The purpose of the adaptive rate design is to identify these two unknown parameters in the controller online. The selected control type is the reference adaptive control based on the mathematical model. The design of the adaptive law of the parameters is based on the Lyapunov stability. According to the Lyapunov stability direct method, the adaptive control law of the uncertain parameters is obtained, so the obtained adaptive control law can guarantee the stability of the AHC system. The Lyapunov direct method does not judge the stability of the system by the solution of the system state equation but uses the Lyapunov function to judge the stability, and analyzes the stability from the perspective of generalized energy. The Lyapunov stability direct method is defined as follows:

For the system $\dot{x} = f[x, t]$, the equilibrium state is $x_e = 0$, and $f(x_e) = 0$. If there is a scalar function $V(x)$ such that it has a continuous first-order partial derivative for all x and $V(x)$ is positive definite, then the equilibrium state is asymptotically stable when $\dot{V}(x)$ is negative definite or semi-negative definite and not always zero for any state.

The uncertain parameters of the system exist in the second state equation of Formula (11). The expected value x_2 is designed for the state variable α_1, and the state equation of the state variable error $\tilde{x}_2$ is obtained as follows:

$$\begin{aligned} &\dot{x}_2 = \frac{1}{m_l}(A_i\overline{x}_3 - b_1x_2 - b_2g_1 - \Delta), \dot{\tilde{x}}_2 = x_2 - \alpha_1 \\ &\dot{\tilde{x}}_2 = \frac{1}{m_l}(A_i\overline{x}_3 - b_1x_2 - b_2g_1 - \Delta) - \dot{\alpha}_1 \end{aligned} \tag{12}$$

According to Formula (12), in order to ensure the stability of the whole system, the controller $\overline{x}_3$ is designed and brought into Formula (12):

$$\begin{array}{l} \overline{x}_3 = \frac{1}{A_i}\left[\hat{b}_1 x_2 + b_2 g_1 + \Delta + m_l\left(\dot{\alpha}_1 - K\widetilde{x}_2\right)\right] \\ \dot{\widetilde{x}}_2 = \frac{1}{m_l}\widetilde{b}_1 x_2 - K\widetilde{x}_2 \end{array} \tag{13}$$

where $\hat{b}_1$ is the estimated value, $\widetilde{b}_1 = \hat{b}_1 - b_1$, K is a positive stability term. The Lyapunov function is defined as follows:

$$\begin{array}{l} V\left(x_2, \hat{b}_1\right) = \frac{1}{2}\widetilde{x}_2^2 + \frac{1}{2\eta_1}\widetilde{b}_1^2 \\ = \widetilde{x}_2 \cdot \dot{\widetilde{x}}_2 + \frac{1}{\eta_1}\widetilde{b}_1 \cdot \dot{\hat{b}}_1 \\ = \widetilde{x}_2\left(\frac{1}{m_l}\widetilde{b}_1 x_2 - K\widetilde{x}_2\right) + \frac{1}{\eta_1}\widetilde{b}_1 \cdot \dot{\hat{b}}_1 \\ = -K\widetilde{x}_2 + \widetilde{b}_1\left(\frac{1}{m_l}x_2 \cdot \widetilde{x}_2 + \frac{1}{\eta_1} \cdot \dot{\hat{b}}_1\right) \end{array} \tag{14}$$

In order to make the system stable, therefore, $\dot{V}\left(x_2, \hat{b}_1\right) \leq 0$. The adaptive control law of uncertain parameters b_1 is designed as follows. The adaptive control rate for system uncertain parameter $\dot{\hat{b}}_1\dot{\hat{b}}_2$ design will be directly applied to the following controller design:

$$\dot{\hat{b}}_1 = -\eta_1 \frac{1}{m_l} \cdot x_2 \cdot \widetilde{x}_2 \tag{15}$$

For the same derivation design of parameters b_2, it can be obtained:

$$\dot{\hat{b}}_2 = -\eta_2 \cdot \frac{1}{m_l} \cdot g_1 \cdot \widetilde{x}_2 \tag{16}$$

3.2. Disturbance Force Observer Design

Focusing on the unmodeled nonlinear friction disturbance force Δ in the system, a disturbance observer is designed to estimate it. In order to facilitate the design of disturbance observer and intuitive numerical analysis verification, at the same time, this designed observer can be directly referred to when the controller is derived below. Thus, a simple system state equation with disturbance force is constructed, as shown in Formula (17):

$$\dot{x} = f(x) + u + \Delta(t, x) \tag{17}$$

where $f(x)$ represents the vector of the state variable x, in the following, there will be specific functional relationship references, u is the control input vector, $\Delta(t, x)$ represents the system interference force vector. In the above, it is assumed that the friction disturbance force Δ is bounded. In order to design the control law u to stabilize the system to the origin, it is necessary to design the disturbance observer $\hat{\Delta}(t, x)$ to accurately estimate the uncertain disturbance $\Delta(t, x)$. The control law can be intuitively designed as $u = -kx - f(x) - \hat{\Delta}(t, x)$ to ensure system stability, where k is the positive real number or positive definite matrix.

The principle of the disturbance observer is to ensure that the disturbance error $\Delta_e(t, x)$ converges exponentially to the ball centered at the origin, and the radius of the ball is arbitrarily small by adjusting the function $\rho(x)$. An observer is designed to estimate the nonlinear unmodeled friction Δ in this paper. The design form of the observer is as follows:

$$\begin{cases} \hat{\Delta}(t, x) = \xi + \rho(x) \\ \dot{\xi} = -K(x)\xi - K(x)(f(x) + u + \rho(x)) \end{cases} \tag{18}$$

In the above formula, $K(x)$ is positive definite for any x, and its relationship with $\rho(x)$ is $K(x) = \partial\rho(x)/\partial x$. The purpose of the disturbance observer is to ensure that the disturbance error of the disturbance observer converges exponentially to a ball centered at the origin. By adjusting the value of the function to make the radius of the ball arbitrarily small, the system is stabilized.

To verify the tracking performance of the designed disturbance observer, the numerical simulation method is used to test the estimation effect of the observer. Suppose the interference term is $\Delta(t,x) = \sum_{i=1}^{5}(\sin(it) + \sin(x)\sin(it))$, and consider selecting a scalar system $f(x) = \arctan(x + x^2)$, and a corresponding choice, the function $\rho(x) = 15(x + \frac{x^3}{3})$, $K(x) = 20(1 + x^2)$. In order to ensure the stability of the system, the control rate is designed as $u = -kx - f(x) - \hat{\Delta}(t,x)$, $k = 5$.

The simulation results are as follows. In Figure 8, the imaginary line represents the estimated value of the interference force $\hat{\Delta}$, and the real line represents the actual value of the interference force Δ. It can be seen from the figure that the estimated value obtained by using the disturbance observer has a good tracking performance for the actual disturbance force, so the disturbance observer achieves a good estimation of the disturbance term in the system.

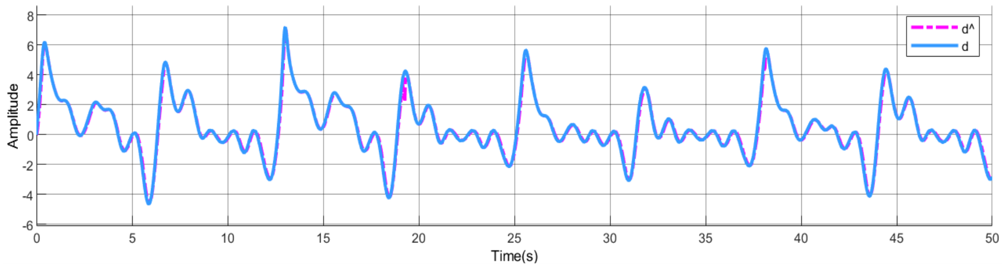

Figure 8. Tracking performance of disturbance observer.

For the variables x in the system, as shown in Figure 9, the red line with a large amplitude represents the system without the disturbance term, and the blue line floating at the zero point represents the system using the observer to estimate the disturbance term. It can be clearly seen from the figure that the system with disturbance observer converges to a ball with a smaller radius than the system without disturbance. It can be explained that even if the system has disturbance terms, the designed observer makes the whole system more stable. Therefore, it can be shown that the disturbance observer designed in this paper has a certain effect on the stability of the system.

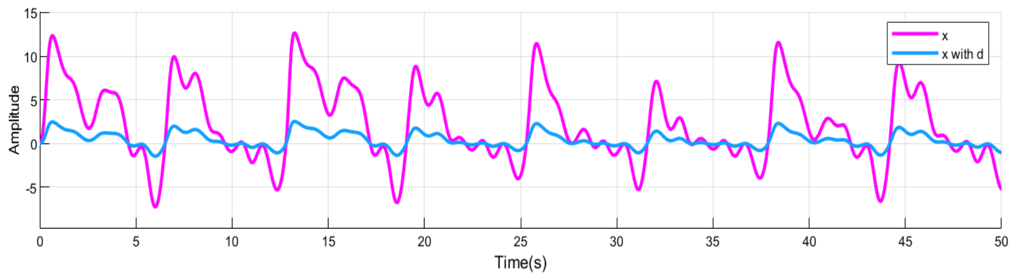

Figure 9. System variable curve with and without observer.

3.3. DOB-ANCC Nonlinear Cascade Control Based on Backstepping Method

Based on the adaptive rate $\hat{b}_1\hat{b}_2$ of the design of the uncertain parameter hydraulic cylinder damping coefficient b_1 and the drag force parameter b_2 of the load in the system, as well as the observer $\hat{\Delta}$ designed for the unmodeled nonlinear friction interference force in the system, they are directly applied to the backstepping derivation design of the control law. The design of the active heave compensation controller for the marine crane adopts the nonlinear cascade control algorithm. The displacement compensation control is used as the outer loop of the controller, and the pressure control is used as the inner loop of the controller. The backstepping theory is used to design the input u of the system controller.

3.3.1. Step 1

The first step of the cascade controller is to design the desired pressure for the outer loop displacement control. In this step, x_2 is the control law and α_1 is the virtual control variable of x_2. The control goal of this step is to stabilize the displacement error $\widetilde{x}_1$ by designing virtual control variables, where x_d is the expected displacement value and k_{11} is the positive compensation term. Define the displacement compensation error as follows:

$$\begin{aligned} \widetilde{x}_1 &= x_1 - x_d \\ \widetilde{x}_2 &= x_2 - \alpha_1 \\ \dot{\widetilde{x}}_1 &= \widetilde{x}_2 + \alpha_1 - \dot{x}_d \\ \alpha_1 &= -k_{11}\widetilde{x}_1 \\ \dot{\widetilde{x}}_1 &= -k_{11}\widetilde{x}_1 + \widetilde{x}_2 \end{aligned} \tag{19}$$

3.3.2. Step 2

Our goal in this step is to adjust $\widetilde{x}_2$ to the origin region to achieve stability, similar to the first step x_3 as the control law of the second step. The essence of introducing virtual control variables into the design of the backstepping method is a static model compensation. The control of the previous subsystem must rely on the virtual control of the subsequent subsystem to achieve stability. Therefore, a virtual control variable α_2 is designed for $\overline{x}_3$. Based on the system state equation and the formula of the first step, it can be concluded that:

$$\begin{aligned} \widetilde{\overline{x}}_3 &= \overline{x}_3 - \alpha_2 \\ \dot{\widetilde{x}}_2 &= \tfrac{1}{m_l}A_i(\widetilde{\overline{x}}_3 + \alpha_2) - \tfrac{1}{m_L}\widehat{b}_1(\widetilde{x}_2 + \alpha_1) - \Delta - \tfrac{\partial\alpha_1}{\partial\widetilde{x}_1}(-k_{11}\widetilde{x}_1 + \widetilde{x}_2) - \hat{b}_2 g_1 \end{aligned} \tag{20}$$

Ignoring $\widetilde{\overline{x}}_3$, the following virtual control variable α_2 is designed to stabilize the system, where k_{21} is a positive compensation term:

$$\alpha_2 = \frac{1}{\frac{1}{m_l}A_i}\left[-\widetilde{x}_1 - k_{21}\widetilde{x}_2 + \frac{1}{m_l}\hat{b}_1 + \hat{b}_2 g_1 + \frac{\partial\alpha_1}{\partial\widetilde{x}_1}(-k_{11}\widetilde{x}_1 + \widetilde{x}_2) + \hat{\Delta}\right] \tag{21}$$

The nonlinear friction disturbance term Δ in the above formula is estimated by using the disturbance observer designed above. If $\widetilde{\overline{x}}_3$ is neglected, it can be seen that the form of Formula (20) is the same as that of Formula (17). Therefore, according to the design of the disturbance observer proposed in Formula (20) and Section 3.2, the estimated value of the disturbance term $\hat{\Delta}$ can be designed, where k_{12} is the positive compensation term, and choosing $\rho(x) = k_{12}x$, the observer is designed as follows:

$$\begin{aligned} \hat{\Delta} &= \xi_1 + k_{12}\widetilde{x}_2 \\ \dot{\xi}_1 &= -k_{12}\xi_1 - k_{12}\left[\tfrac{1}{m_l}A_i(\overline{x}_3 + \alpha_2) - m_l\hat{b}_1(\widetilde{x}_2 + \alpha_1) - \tfrac{\partial\alpha_1}{\partial\widetilde{x}_1}(-k_{11}\widetilde{x}_1 + \widetilde{x}_2) - \hat{b}_2 g_1\right] \end{aligned} \tag{22}$$

Bring the designed virtual control variable Formula (21) into Formula (20) to obtain the following formula, where $\widetilde{\Delta} = \Delta - \hat{\Delta}$:

$$\dot{\widetilde{x}}_2 = -\widetilde{x}_1 - k_{21}\widetilde{x}_2 + \frac{1}{m_l}A_i\widetilde{\overline{x}}_3 + \widetilde{\Delta} \tag{23}$$

3.3.3. Step 3

This is the final step of the controller design. The virtual control variable α_2 designed in the previous step is used as the input of the inner loop pressure control. The goal of this step is to solve and design the control law u of the actual command output of the AHC system. The following equation is obtained by differentiating $\widetilde{\overline{x}}_3$:

$$\begin{aligned} \dot{\widetilde{\overline{x}}}_3 &= \dot{\overline{x}}_3 - \dot{\alpha}_2 \\ \dot{\widetilde{\overline{x}}}_3 &= -g_3x_2 + g_5u - h_1(x_1)C_{ip}(\beta\beta_e - \overline{x}_3) - \tfrac{\partial\alpha_2}{\partial\xi_1}\cdot\dot{\xi}_1 - \tfrac{\partial\alpha_2}{\partial\widetilde{x}_1}(\widetilde{x}_2 - k_{11}\widetilde{x}_1) - \tfrac{\partial\alpha_2}{\partial\widetilde{x}_2}(-\widetilde{x}_1 - k_{12}\widetilde{x}_2 + \tfrac{1}{m_l}A_i\widetilde{\overline{x}}_3 + \widetilde{\Delta}) \end{aligned} \tag{24}$$

In consequence, the control rate u which represents the input current command of the servo valve can be designed, and the expression is as follows:

$$u = \frac{1}{g_5}[-\frac{1}{m_l}A_i\widetilde{x}_2 - k_{13}\widetilde{\overline{x}}_3 + g_3x_2 + h_1(x_1)C_{ip}(\beta\beta_e - \overline{x}_3) + \frac{\partial\alpha_2}{\partial\xi_1}\dot{\xi}_1 + \frac{\partial\alpha_2}{\partial\widetilde{x}_1}(\widetilde{x}_2 - k_{11}\widetilde{x}_1) + \frac{\partial\alpha_2}{\partial\widetilde{x}_2}(-\widetilde{x}_1 - k_{12}\widetilde{x}_2 + \frac{1}{m_l}A_i\widetilde{\overline{x}}_3)] \quad (25)$$

where k_{13} is the positive compensation term. The whole control law of the AHC system is designed to realize the control of the input current of the servo valve. Thus, the reverse control of the displacement of the hydraulic cylinder is realized to compensate the load with the displacement of the vessel.

4. Simulations

To verify the effectiveness of the controller designed in this paper and the compensation effect of the load in the active heave compensation system of the marine crane, the designed control algorithm and the compensation effect of the AHC system of the marine crane on the load motion are simulated and analyzed, respectively, in this section. In order to verify the compensation performance of the system, the simulation parameters required for the active heave compensation system of the marine hydraulic crane are shown in Table 1.

Table 1. Crane AHC system model simulation parameter list.

Parameter	Value	Unit	Parameter	Value	Unit
A_1	1.113×10^{-2}	m^2	V_{01}	1.15×10^{-2}	m^3
A_2	5.6×10^{-3}	m^2	V_{02}	0.57×10^{-3}	m^3
C_{ip}	1.4×10^9	$m^3/(s \cdot Pa)$	k_d	1.25×10^{-3}	m/v
β_e	1.4×10^9	Pa	β	4.956×10^{-1}	

The tracking effect of the controller designed in this paper is verified. The DOB-ANCC controller used in the simulation is debugged and the parameters are set, as shown in Table 2.

Table 2. Control algorithm parameter list.

Parameter	Value
k_{11}	2
k_{12}	15
k_{13}	4
K_{21}	8
η_1	4×10^9
η_2	5×10^9

The effectiveness of the controller is verified by comparative simulation analysis. The adaptive nonlinear cascade controller with disturbance observer DOB-ANCC proposed in this paper is compared with the traditional PID controller for path tracking. The traditional PID controller is a linear controller, in which the input of the control law of the system is the deviation between the expected signal and the actual signal. The proportional coefficient of the PID controller is 100, the integral term is 2.5 and the differential term coefficient is 15. The traditional PID controller is designed as follows:

$$u = 100(x_d - x_1) + 2.5\int_0^t (x_d - x_1) + 15\dot{x}_d \quad (26)$$

Figure 10 compares the tracking performance of the linear control using the traditional PID controller and the adaptive nonlinear cascade controller with the disturbance observer (DOB-ANCC) controller designed in this paper for the expected value x_d of the vessel heave motion tracking signal. Because the DOB-ANCC controller designed in this paper estimates and compensates the uncertain parameters $b_1 b_2$ and the disturbance term Δ of the system, it can be seen from the diagram that the DOB-ANCC controller shows better displacement tracking performance than the PID controller under four-level sea conditions. From Figure 11, the maximum tracking error of the traditional PID controller is about 0.2 m. The tracking error of the controller (DOB-ANCC) proposed in this paper is within 0.035; the system robustness is preferably higher. For the verification of the tracking performance of the disturbance observer designed in this paper, it can be seen from Figure 12 that the observer still shows good tracking performance for the disturbance term even in a short time.

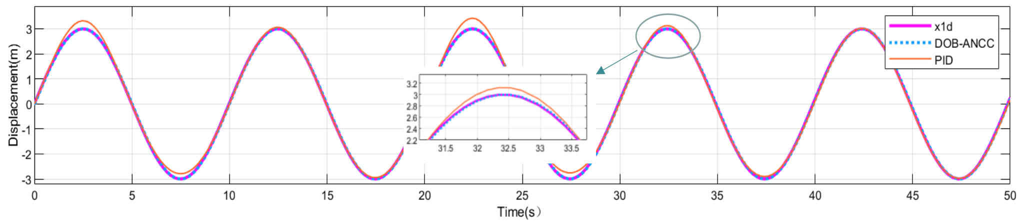

Figure 10. Tracking performance of different controllers.

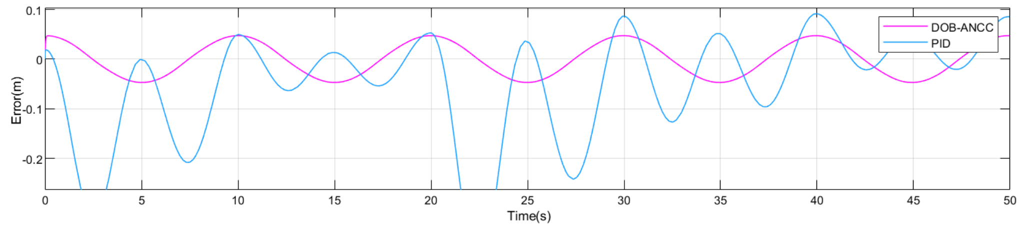

Figure 11. Tracking error of different controllers.

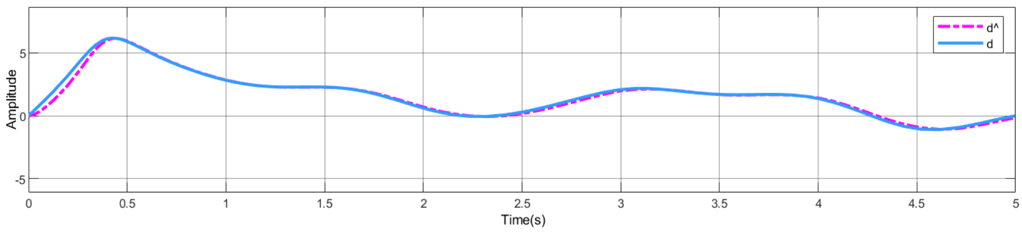

Figure 12. Tracking performance of disturbance observer in short time.

In the simulation of the load displacement compensation of the active heave compensation system, as shown in Figure 13, it can be seen that the designed control law reversely compensates the displacement of the hydraulic cylinder. For the displacement compensation of the 50-t load under the four-level sea condition. The heave displacement of the load is about 0.3 m; thus, the compensation efficiency is better, and the fluctuation of the vessel is small. The compensation ability reaches more than 95%, which can ensure the stability of the heave system. For the irregular motion of the actual wave, the randomly generated wave signal is used as the input, and the heave displacement of the load is obtained as shown in Figures 13 and 14. It can be seen from the figure that the compensation effect is still above 90%.

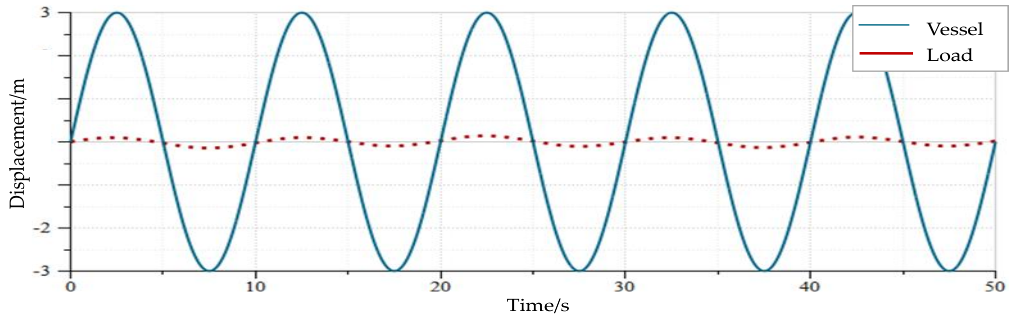

Figure 13. Displacement curves of vessel and load under four sea conditions.

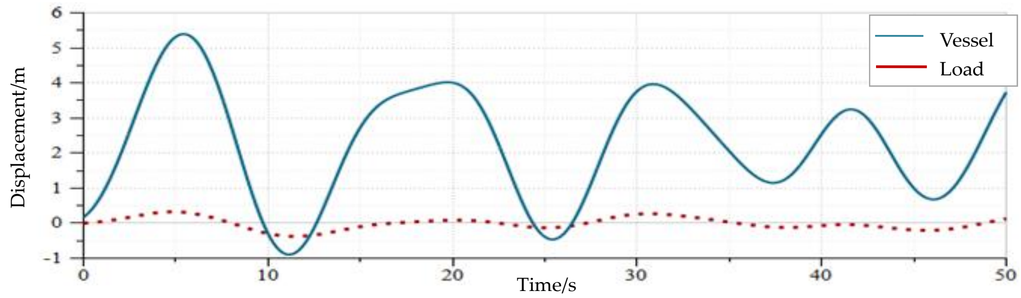

Figure 14. Displacement curves of vessel and load under a random wave.

5. Discussion

The nonlinear model of the hydraulic crane active heave compensator is carried out. The observer is designed to estimate the nonlinear friction force in the system, and the adaptive control is used to estimate the uncertain parameters in the mathematical model of the system. Although the linear model of the active compensator studied by the predecessors can completely decouple the motion of the ship and the load, the nonlinear model established in this paper by considering the drag force and nonlinear friction of the load makes the system accurate, and the inner and outer loop cascade control designed in this paper is robust to other research control schemes.

However, the study still has limitations. In practical applications, due to the incomplete accuracy of sensors and drive devices, the compensation system has a time delay problem. Moreover, the wave selected in this paper is an ideal sinusoidal wave, while the actual sea condition is an irregular uncertain wave, which requires predictive control to be added to the system to achieve perfect compensation as much as possible. Therefore, in the following research, it is necessary to design predictive controller and control algorithm in the control system.

6. Conclusions

In this paper, an active heave compensator control method for the marine crane is proposed to compensate the underwater lifting load operation of the vessel with the wave heave motion. By establishing a nonlinear mathematical model of the crane active heave compensator and taking into account the disturbance force of the load in the water, this paper designs an adaptive nonlinear cascade controller (DOB-ANCC) with a disturbance observer. The design of the active heave compensator controller is based on the backstepping method. The inner loop control adopts pressure control and the outer loop control adopts displacement control. The simulation results show that the compensation efficiency of the designed controller (DOB-ANCC) for the heave motion of the load can reach more than 95%, and the maximum displacement tracking error of the controller can reach $\pm$ 0.035 m.

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