Constant Tension Control of Hybrid Active-Passive Heave Compensator Based on Adaptive Integral Sliding Mode Method

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ABSTRACT Heave compensation systems are of great importance to the safety and efficiency of marine operations subject to irregular-wave excitation. While many efforts have been made to improve the displacement compensation control, only a few of researchers pay their attention on tension compensation control. This paper presents an adaptive robust integral sliding mode control (ARISMC) based on the back-stepping method to realize constant tension control of hybrid active-passive heave compensator (HAHC) applied to heavy deep-sea towing systems. The proposed ARISMC is intended to overcome the effects of parametric uncertainties, uncertain nonlinearities, and external disturbances in the electro-hydraulic system of HAHC. The stability of the whole system is proved using common Lyapunov method. To verify the effectiveness of the proposed controller, we carried out a number of simulations and experiments using the measured heave motion data under different sea conditions. The results demonstrate that the ARISMC controller presented in this paper has better advantages than the traditional PI controller in the accuracy and robustness of tension compensation.

INDEX TERMS Adaptive robustness, constant tension, heave compensation, heavy tow, integral sliding mode.

I. INTRODUCTION

Ships heave with waves on the sea, which will drive the payloads and cables accordingly. Harsh sea conditions cause drastic changes of payload depth and cable tension, which seriously affects the efficiency and safety of marine operations [1]. To solve this problem, the heave compensation system has been developed and applied to many marine operation equipment in the past 50 years, such as deep-sea oil and gas drilling, deep-sea towed detection, deep-sea salvage and rescue [2]–[4]. Heave compensation systems could eliminate the heave movement of the operating platform relative to the payload and decouple the payload movement from the ship’s movement, which can avoid severe rope slack and reduce the peak rope tension [5], [6]. Generally, heave compensation systems can be classified as passive heave compensation system (PHC), active heave compensation system (AHC) and hybrid active-passive heave compensation system (HAHC).

The passive heave compensation (PHC) system is mainly composed of a pulley, a hydraulic cylinder, and an oil-gas separator, which is equivalent to a gas spring. PHC works with simple principle and has no energy consumption. However, the compensation accuracy of PHC is low, which cannot satisfy the requirement under certain conditions. The power of the active heave compensation (AHC) system comes from hydraulic pumps and electric motors. AHC uses sensors to detect the current movement states of the ship and the load, which will be transferred to the controller for active compensation. AHC has fast compensation speed, high compensation precision, and strong anti-interference ability. However, the disadvantage is that AHC needs to consume extra power [2], [7]. For light-load systems, active heave compensators can be used to achieve good compensation performance, but the application of heavy-load systems is restricted...
due to its large power consumption. HAHC combines the advantages of AHC and PHC. The passive part balances the static load and the active part cancels the dynamic motion. HAHC is an ideal heave compensation system, especially for heavy load systems. Therefore, this article takes HAHC as the research object.

One of the main functions of the heave compensation system is to achieve constant tension control of the cable during marine operations. Wu et al. designed a passive compensator to reduce the fluctuation of tension and ensure the operation safety of a 4.5 km remotely operated vehicle [8]. Mitchell et al. used a remotely controlled adjustable relief valve to adjust the tension generated by the hydraulic winch in the multi-network plankton tow system and achieved constant tension control of the towing winch. The experimental results showed that the method significantly reduced the peak load of cable tension and improved the drag performance of the system [9].

To obtain better tension compensation performance, some scholars have proposed tension control algorithms including commonly used PID control, feedforward control, and feedback control. Souterland used proportional control with mechanical feedback in a payload transfer situation [10]. Korde, Gu, Hateskog, and Dunnigan et al. used feedforward control, feedforward PD feedback control and p-pi control respectively to realize the complete decoupling of the load and the mothership in theory. However, as compensation systems are strongly nonlinear, it is necessary to estimate system parameters and adjust control parameters online in real-time to obtain better control performance in actual systems [11]–[14]. In order to solve the load fluctuation of the payload across the splash zone, Skaare B et al. proposed a pressure - position parallel controller [15]. Li and Zhang et al. used a linear quadratic Gaussian controller to reduce the axial dynamic stress of deep-water risers [16].

In deep-sea heavy towing systems, armored cables and towed payloads are often disturbed by various non-linear factors, including non-linear towing forces caused by wave heave motion, viscous damping forces, buoyancy, inertial forces, frictional forces of mechanical systems, the non-linear dynamics and parameter uncertainty of the hydraulic system itself. Traditional control strategies are difficult to achieve ideal control performance under different sea conditions, and it is difficult to ensure the stability of the system in the full range [17].

The application of advanced non-linear control strategies to overcome the effects of system parameter uncertainty, uncertain nonlinearity, system uncertain delayed, and external interference is a current research focus and development direction [18]–[22].

To solve the strong load disturbance of HAHC, Do et al. studied a non-linear control algorithm based on Lyapunov and disturbance observer for the electro-hydraulic servo system of AHC [23]. Yuan et al. proposed the concept of active damping heave compensation (ADHC), in which active damping technology was used to significantly improve the effective damping of cable payload dynamics. Compared with the traditional active heave compensator, the underwater components controlled by ADHC can be stabilized much faster, especially when the external interference is unknown [24].

The simulation results of Li et al. showed that with the second-order active disturbance rejection controller, the compensation efficiency based on tension feedback control was up to 98% [25]. Li and Ma et al. presented an ADRC-ESMPC double-loop controller for a hydraulic valve-based active heave compensation system, which was proved to be effective through simulation [26]. However, the above studies have not been experimentally verified based on real sea condition data, and can not be directly applied to HAHC of heavy deep-sea towing systems.

Recently, Niu et al. presented the design and full-scale experimental results of a 3000-m HAHC for a 200-T winch. Full-scale factory tests at different wave periods and amplitudes showed that the displacement compensation efficiency of HAHC is up to 92.9 % [27]. Shi et al. applied the BP neural network PID control algorithm to a scale model of HAHC and the displacement compensation rate reached more than 90% [28]. Yu et al. proposed a robust state constrained variable structure controller for an active heave compensation system to cope with the situation that the ship’s heave motion exceeds the pre-designed compensation range or speed range, and the effectiveness of the controller was verified by experiments [29]. However, the above experimental researches only focus on the displacement compensation of HAHC, and neglects the performance of tension compensation. At present, the constant tension compensation efficiency of the actual marine equipment is about 70%. Therefore, an efficient tension compensation method which can be directly applied to HAHC of heavy deep-sea towing systems is urgently needed. To deal with those problems, ARISMC is proposed and its effectiveness is proved by experiments in this paper.

The rest of this paper is organized as follows. In section II, the mathematical model of hybrid active-passive heave compensator (HAHC) is presented. Section III presents the proposed ARISMC controller design. In section IV, simulation results and performance analysis are presented. Section V shows the experimental setup and results. The final section is the conclusion.

**II. MATHEMATICAL MODEL**

In order to facilitate the design of the controller of the composite tension compensation system, some necessary simplifications of the system are carried out first, and the following assumptions are made:

1. Ignore the influence of pressure loss and pressure dynamic characteristics of system pipeline;
2. Compared with the system flow, the compensation leakage flow and servo valve leakage flow are smaller, and the influence is ignored;
3. The hydraulic system is open-loop system, and the pressure of hydraulic oil tank is considered as 0.
Figure 1 depicts the principle of the HAHC. The dynamic equation of pressure in each chamber of hybrid active-passive compensation hydraulic cylinder can be expressed as:

\[
\dot{P}_1 = \frac{\beta_e}{V_{01} + A_1 x_p} [Q_1 - A_1 \dot{x}_p - C_i (P_1 - P_3)] \\
\dot{P}_2 = \frac{\beta_e}{V_{02} - A_2 x_p} [-Q_2 + A_2 \dot{x}_p] 
\]  

(1)

where, \( V_{01} \) is the initial volume of piston cylinder with rod chamber, \( V_{02} \) is the initial volume of piston cylinder with rod chamber, \( \beta_e \) is the effective bulk elastic modulus of hydraulic oil, \( C_i \) is the total internal leakage coefficient of hydraulic cylinder caused by pressure difference, \( x_p \) is the compensation cylinder displacement, \( Q_1 \) is the inflow of piston cylinder with rod chamber, and \( Q_2 \) is the outflow flow of piston cylinder.

The forward and return flow through the cylinder related to the high-response servo valve described as:

\[
Q_1 = k_q \cdot x_v \cdot [s_q(x_v) \cdot \sqrt{P_3 - P_1} + s_q(-x_v) \cdot \sqrt{P_1 - P_3}] \\
Q_2 = k_q \cdot x_v \cdot [s_q(x_v) \cdot \sqrt{P_2 - P_1} + s_q(-x_v) \cdot \sqrt{P_1 - P_2}] 
\]  

(2)

Now, by definition as:

\[
s_q(\bullet) = \begin{cases} 
1, & \bullet \geq 0 \\
0, & \bullet < 0 
\end{cases}
\]  

(3)

where \( k_q \) is the flow coefficient of servo valve; \( P_1 \) is the oil supply pressure of the system; \( P_t \) is the oil tank and \( P_t = 0 \); \( x_v \) is the displacement of the valve core of servo valve; \( s_q(\bullet) \) is a symbolic function.

In the servo valve controlled hydraulic system, the frequency response of the servo valve is significantly higher than that of the actuator. Therefore, the dynamic response process of the servo valve can be regarded as the first-order inertia link [30], that is

\[
\dot{x}_v = -\frac{1}{\tau_v} \cdot x_v + \frac{k_v}{\tau_v} \cdot u 
\]  

(4)

where \( \tau_v \) is the time constant of unit step response of servo valve, \( k_v \) is the command gain of servo valve, and \( u \) is the control input voltage of servo valve.

Define the state variables \( x \) form as:

\[
x = [x_1, x_2, x_3]^T = [P_1, P_2, x_v]^T
\]  

(5)

The entire system, equation (1) to (5) can be expressed in the state space form

\[
\begin{align*}
\dot{x}_1 &= \beta_e h_1(x_p)[-C_i \cdot x_1 + k_q g_1(x_1, u) \cdot x_3 + C_i P_3 - A_1 \dot{x}_p] \\
\dot{x}_2 &= \beta_e h_2(x_p)[-k_q g_2(x_2, u) \cdot x_3 + A_2 \dot{x}_p] \\
\dot{x}_3 &= -\frac{1}{\tau_v} x_3 + \frac{k_v}{\tau_v} u 
\end{align*}
\]  

(6)

in which

\[
\begin{align*}
 h_1(x_p) &= 1/(V_{01} + A_1 x_p) \\
 h_2(x_p) &= 1/(V_{02} - A_2 x_p) \\
 g_1(x_1, u) &= s_q(u) \sqrt{P_3 - x_1} + s_q(-u) \sqrt{P_1 - x_1} \\
 g_2(x_2, u) &= s_q(u) \sqrt{P_2 - x_2} + s_q(-u) \sqrt{P_1 - x_2} 
\end{align*}
\]  

(7)

The load pressure of HAHC is

\[
P_L = -P_1 + \alpha_1 P_2 + \alpha_2 P_3
\]  

(8)

where \( \alpha_1, \alpha_2 \) are the area ratios, \( \alpha_1 = A_2/A_1, \alpha_2 = A_3/A_1 \).

Thus, the output equation of the system is

\[
y = -x_1 + \alpha_1 x_2 + \alpha_2 P_3
\]  

(9)

For the electro-hydraulic system of HAHC, there are parameter uncertainties in the equivalent bulk modulus, servo valve flow coefficient, and system leakage coefficient. In this chapter, the influence on the system pressure control performance will be mainly considered, and the following uncertain parameters are defined:

\[
\begin{align*}
\theta_1 &= \beta_e C_i \\
\theta_2 &= \beta_e k_q \\
\theta_3 &= \beta_e A_1 
\end{align*}
\]  

(10)

Then define the state variables form as: \( \tilde{x}_1 = -x_1 + \alpha_1 x_2 \), and the state equation of the system can be expressed as:

\[
\begin{align*}
\dot{\tilde{x}}_1 &= h_1(x_p) \theta_1 x_1 - f_1(x_1, x_2, u) \theta_2 x_3 + f_2 \theta_3 \dot{x}_p - h_1(x_p) \theta_1 P_3 \\
\dot{x}_3 &= -\frac{1}{\tau_v} x_3 + \frac{k_v}{\tau_v} u \\
\dot{y} &= \tilde{x}_1 + \alpha_2 P_3 
\end{align*}
\]  

(11)

where

\[
\begin{align*}
 f_1(x_1, x_2, u) &= h_1(x_p) g_1(x_1, u) + \alpha_1 h_2(x_p) g_2(x_2, u) \\
f_2 &= h_1(x_p) + h_2(x_p) \alpha_1^2 
\end{align*}
\]  

(12)

III. CONTROLLER DESIGN

The control objective of HAHC is to make the load pressure track the given desired trajectory in different sea conditions, and to ensure the stability of the whole closed-loop control system.

Step 1: Define the system tracking error:

\[
z_1 = y - y_d
\]  

(13)

Design an integral sliding surface:

\[
s_1 = z_1 + \lambda \int z_1 dt
\]  

(14)

where \( \lambda \) is a positive constant.
Noting (11), the time derivative of (13) can be obtained:
\[
\dot{x}_1 = \theta_1 h_1(x_p)x_1 - \theta_2 f_1 \cdot x_3 + \theta_3 f_2 x_p - h_1(x_p)\theta_1 P_3 + \alpha_2 \dot{P}_3 - \dot{\gamma}_d + \lambda z_1 \tag{15}
\]
where \(c_1\) could be considered as the input of the equation (15).

The virtual control input \(x_{3d}\) is designed for system tracking error \(z_1\) converges to zero gradually at the specified speed:
\[
x_{3d} = \frac{1}{\theta_2 f_1} [\dot{\theta}_1 h_1(x_p)x_1 + \dot{\theta}_3 f_2 x_p - h_1(x_p)\dot{\theta}_1 P_3 + \alpha_2 \dot{P}_3 - (-\dot{\gamma}_d - c_1 z_1 + k_{11} s_1)] \tag{16}
\]
where \(c_1\) is equal to \(-\lambda\), \(k_{11}\) is a positive constant, and \(\dot{\theta}_1, \dot{\theta}_2, \dot{\theta}_3\) are the estimations of \(\theta_1, \theta_2, \theta_3\).

The parameter estimation error is defined as:
\[
\dot{\tilde{\theta}}_1 = \dot{\theta}_1 - \dot{\hat{\theta}}_1 \\
\dot{\tilde{\theta}}_2 = \dot{\theta}_2 - \dot{\hat{\theta}}_2 \\
\dot{\tilde{\theta}}_3 = \dot{\theta}_3 - \dot{\hat{\theta}}_3 \tag{17}
\]

Tracking error of servo valve spool displacement is designed as:
\[
s_2 = x_3 - x_{3d} \tag{18}
\]
From (15), (16) and (17), we can get
\[
\dot{x}_1 = \dot{\tilde{\theta}}_1 h_1(x_p)(x_1 - P_3) - \dot{\theta}_2 f_1 x_3 + \theta_3 f_2 x_p - \dot{\theta}_2 f_1 x_3 - k_{11} s_1 \tag{19}
\]

Define a positive semi-definite function
\[
V_1 = \frac{1}{2} x_1^2 + \frac{1}{2} \Gamma_1^{-1} \tilde{\theta}_1^2 + \frac{1}{2} \Gamma_2^{-1} \tilde{\theta}_2^2 + \frac{1}{2} \Gamma_3^{-1} \tilde{\theta}_3^2 \tag{20}
\]
where \(\Gamma_1^{-1}, \Gamma_2^{-1}, \Gamma_3^{-1}\) are positive constants.

Noting (19), the time derivative of (20) is given by
\[
\dot{V}_1 = -k_{11} s_1^2 + \tilde{\theta}_1 [s_1 h_1(x_p)(x_1 - P_3) - \Gamma_1^{-1} \dot{\tilde{\theta}}_1] + \dot{\tilde{\theta}}_2 [-s_1 f_1 x_3 + \Gamma_2^{-1} \dot{\tilde{\theta}}_2] + \tilde{\theta}_3 s_1 f_2 x_p - \Gamma_3^{-1} \dot{\tilde{\theta}}_3 - \dot{\theta}_2 f_1 s_1 s_2 \tag{21}
\]

The adaptation law is designed as follows:
\[
\begin{cases}
\dot{\tilde{\theta}}_1 = \text{Pr}_{\frac{s_1}{\theta_1}} [\Gamma_1 [s_1 h_1(x_p)(x_1 - P_3)]]
\dot{\tilde{\theta}}_2 = \text{Pr}_{\frac{s_1}{\theta_2}} [-\Gamma_2 s_1 f_1 x_3]
\dot{\tilde{\theta}}_3 = \text{Pr}_{\frac{s_1}{\theta_3}} [\Gamma_3 s_1 f_2 x_p]
\end{cases} \tag{22}
\]

The simple discontinuous projection mapping has the form as
\[
\text{Pr}_{\frac{s_1}{\theta}} (\bullet) = \begin{cases} 0, & \text{if} \quad \frac{\dot{\tilde{\theta}}_i}{\tilde{\theta}_i} = \tilde{\theta}_{i\text{max}} \& \bullet_i > 0 \\
\bullet_i, & \text{otherwise}
\end{cases} \tag{23}
\]

Substituting (22) into (21) yields
\[
\dot{V}_1 = -k_{11} s_1^2 + \dot{\theta}_2 f_1 s_1 s_2 \tag{24}
\]

**Step 2:** In the first step, the virtual control input is designed for the displacement of the valve core \(x_3\). In this step, the actual control input \(u\) is determined.

Differentiating (18) with respect to time gives
\[
\dot{s}_2 = \dot{x}_3 - \dot{x}_{3d} \tag{25}
\]

Substituting system dynamics equation (22) into (24) yields
\[
\dot{s}_2 = -\frac{1}{\tau_v} x_3 + \frac{k_v}{\tau_v} u - \dot{x}_{3d} \tag{26}
\]
Define a positive semi-definite function
\[
V_2 = V_1 + \frac{1}{2} s_2^2 \tag{27}
\]
The time derivative of (26) can be obtained
\[
\dot{V}_2 = \dot{V}_1 + s_2 \dot{s}_2 \tag{28}
\]
Substituting (23) and (25) into (27) yields
\[
\dot{V}_2 = -k_{11} s_1^2 - \dot{\theta}_2 f_1 s_1 s_2 + s_2(-\frac{1}{\tau_v} x_3 + \frac{k_v}{\tau_v} u - \dot{x}_{3d}) \tag{29}
\]
Design the controller input as
\[
u = \frac{x_3}{k_v} + \frac{\tau_v}{k_v} \dot{x}_{3d} + \frac{\tau_v}{k_v} \dot{\theta}_2 f_1 s_1 - \frac{\tau_v}{k_v} k_{21} s_2 \tag{30}
\]
where \(k_{21}\) is a positive constant.

Substituting (30) and (20) into (29) yields
\[
\dot{V}_2 = -k_{11} s_1^2 - k_{21} s_2^2 \leq 0 \tag{31}
\]
Equation (31) indicates that the stability of the closed-loop system is guaranteed.

Fig. 2 shows the control block diagram of the control algorithm of HAHC. In the actual deep-sea towing system, the tension of the cable fluctuates with the change of the mass of the towed body and the weight of the cable. Therefore, the low-pass filtered load pressure is taken as the expected cable tension in the actual system. And the controller receipts the displacement signal of HAHC and the pressure signal of the passive compensation cylinder at the same time.
Due to the maximum stroke limit of HAHC, the control algorithm will judge whether it is within the compensation stroke according to the stroke displacement of HAHC. When the set threshold of compensation stroke is exceeded, the controller will stop or switch to the displacement compensation control to ensure the system safety.

IV. SIMULATION RESULTS

To validate the performance of the proposed ARISMC controller, the simulations of HAHC in the deep-sea towing process are carried out based on MATLAB/AMESim co-simulation platform. The main parameters in the simulation model of HAHC are shown in Table 1. In the process of deep-sea towing, the mass, depth and speed of towing body and cable often vary greatly under different sea conditions, which have a great impact on the control of HAHC, so these factors must be considered in the simulation. According to the characteristics of the towing payload of the heave compensator in the deep-sea towing process, different damping coefficients $b_H$, $d_r$, $C_{ds}$ and different time-varying payload interference forces $d_1$, $d_2$ are designed. The different payload environment parameters are shown in Table 2. The simulation curve of interference force under different payload environment parameters is shown in Fig. 3.

TABLE 1. Main parameters of simulation model.

| Parameter | Value | Unit |
|-----------|-------|------|
| $A_1$     | $5.6 \times 10^3$ | m$^2$ |
| $A_2$     | $5.6 \times 10^3$ | m$^2$ |
| $A_3$     | $1.13 \times 10^2$ | m$^2$ |
| $\beta_4$ | $1.4 \times 10^6$ | Pa |
| $V_1$     | $5.7 \times 10^3$ | m$^3$ |
| $V_2$     | $5.7 \times 10^3$ | m$^3$ |
| $V_L$     | $1.14 \times 10^2$ | m$^3$ |
| $C_s$     | $1.4 \times 10^6$ | m$^3$/N Pa |

TABLE 2. Different payload environment parameters.

| m (kg)     | d (N) | $b_H$ (N/(m/s)) | $d_r$ (N/(m/s)) | Cd |
|------------|-------|-----------------|-----------------|----|
| Payload1   | 3000  | $5 \times 10^4$ | $7 \times 10^4$ | 0.6 |
| Payload2   | 1000  | $2 \times 10^6$ | $4 \times 10^6$ | 0.5 |

The parameters of the ARISMC controller used in the simulation are shown in Table 3.

TABLE 3. ARISMC controller parameters.

| Parameter | Value | Parameter | Value |
|-----------|-------|-----------|-------|
| $k_{11}$  | 50    | $k_{21}$  | 150   |
| $\lambda$ | 0.5   | $\Gamma_1$ | $1 \times 10^4$ |
| $\Gamma_3$ | $2 \times 10^4$ | $k_r$ | 1 |
| $\Gamma_4$ | $2 \times 10^4$ | $k_4$ | 0.4 |

A. SIMULATION ANALYSIS OF ARISMC TENSION COMPENSATION UNDER SINUSOIDAL INTERFERENCE

A 0.05 Hz sinusoidal motion trajectory is used to simulate the heave motion of the payload, and the performance of PI and ARISMC controllers under different payloads is studied. Fig. 4 shows that under payload 1 conditions, both PI and ARISMC controllers can stabilize the tension control system. The compensation accuracy of PI controller is $\pm 0.6$ bar, and that of ARISMC controller is about $\pm 0.2$ bar. Under the conditions of payload 2 (shown in Fig. 5), the control error of PI control algorithm is much larger than that of payload 1, from $\pm 0.6$ bar to $-2.5$ to $+3.6$ bar, while the control error of ARISMC is increased from $\pm 0.2$ bar to $\pm 0.5$ bar, which shows that ARISMC shows higher compensation accuracy and stronger robustness.

B. SIMULATION ANALYSIS OF ARISMC TENSION COMPENSATION UNDER DIFFERENT SEA CONDITIONS

The speed and acceleration of the heave disturbance have a great influence on the tracking performance of the controller. Fig. 6-9 show the comparative tension compensation performance of the two controllers under the level of 3-6 sea condition disturbance. When $t = 200$ sec, HAHC is turned on. It can be seen from the figure that HAHC has a very obvious tension compensation performance.
It can be seen from Fig. 6 that under the irregular heave motion disturbance of high frequency component, the PI controller jitters at 278 s, while the compensation process of ARISMC is stable. As shown in Fig. 9, under level 6 sea condition, PI controller is unstable again due to the influence of unknown strong disturbance ($t = 268$ s), while ARISMC can still compensate stably.

Summarizing the maximum steady-state tracking errors in Fig. 6 to Fig. 9 into Table 4, we can draw the following conclusions, ARISMC controller has better advantages than the traditional PI controller in the accuracy and robustness of tension compensation.

| TABLE 4. Comparison of the tracking error for PI and ARISMC controllers under the sea conditions of 3-6 levels. |
|---------------------------------------------------------------|
| Sea conditions | PI Error(max) | Unit | ARISMC Error(max) | Unit |
|----------------|---------------|------|-------------------|------|
| Level 3        | 10.66         | bar  | 3.14              | bar  |
| Level 4        | 4.47          | bar  | 2.71              | bar  |
| Level 5        | 7.83          | bar  | 2.12              | bar  |
| Level 6        | 6.07          | bar  | 4.58              | bar  |
V. EXPERIMENTAL SETUP AND RESULTS

A. EXPERIMENTAL SETUP

The diagram and photo of the experimental setup are shown in Fig.10 and Fig.11 respectively. The experimental system consists of a simulated winch device, an HAHC scale prototype, a cable, a payload mass block, a hydraulic pump station, a Simulink real-time measurement and control system, a PLC controller, a motion reference unit (MRU-H of Kongsberg), a displacement sensor (MTS), a tension sensor, a winch encoder, etc. The main technical parameters of the experimental system are shown in Table 5.

![FIGURE 10. Experimental system structure framework.](image1)

![FIGURE 11. The test prototype and measurement and control system of HAHC.](image2)

HAHC scale prototype is mainly composed of accumulators, two plunger type cylinders, two piston cylinders and a high response servo valve (REXROTH: 4WRPEH10C4B100P). And the control system of HAHC is constructed based on the Simulink Real-time technique (XPC), which includes a host computer, a target computer and a data acquisition (DAQ) card. The data acquisition equipment is NI PCI-6259, of which the ADC resolution is 16 bits and the sample rate can be 1.00 MS/s (16 channel).

B. EXPERIMENTAL RESULTS

Fig. 12 is the experimental comparison curve of the tension compensation performance of two control algorithms under the measured heave disturbance. It can be seen that the load pressure of the supporting cylinder fluctuates from 70 bar to 130 bar without HAHC. With HAHC PI controlled compensation, the load pressure is stable at 91.5-105.7 bar, and the steady-state compensation accuracy is ±7.1 bar. Besides, with ARISMC controlled compensation, the load pressure is negative load pressure range is 96.9-98.7 bar, and the steady-state accuracy is about ±0.9 bar. The compensation accuracy of ARISMC is much higher than the traditional PI controller.

Extensive experiments are conducted for testing the tension compensation performance of the proposed ARISMC controller and the traditional PI controller under the sea conditions of 3-6 levels. Experimental results are illustrated in Fig.13-Fig.16. It can be seen from Fig.13 that in the initial 0-100 sec, HAHC does not work, and the fluctuation range of the load pressure of the supporting cylinder is 77.1-121.3 bar. With PI tension controller, the load pressure is stable at 90.0-109.2 bar, while the load pressure changes in the range of 97.2-99.4 bar with the ARISMC controller.

As shown in Fig. 14, without HAHC, the load pressure disturbance is about 53-150 bar under level 4 sea condition. With PI controller, the load pressure is stable at 93 bar-103.8 bar.
However, with the ARISMC controller, the load pressure is stable at 96.7-99.3 bar.

Figure 15 shows that the load pressure disturbance is about 75 bar to 133.5 bar without HAHC under level 5 sea condition. After the compensation of PI controller, the load pressure is stable at 91.2-104 bar, while the load pressure changes in the range of 97.3 to 100 bar by using the ARISMC controller.

As shown in Fig. 16, without HAHC, the load pressure disturbance is about 23.4 bar to 175.5 bar under level 6 sea condition. After the compensation of PI controller, the load pressure is stable at 96.8-100.8 bar. However, with the ARISMC controller, the load pressure is stable at 97.8-99.1 bar.

A measure of the effectiveness of HAHC is provided by the load pressure fluctuation reduction ratio. In the present study, the tension compensation efficiency of HAHC (HAHC-TEC) can be defined as

\[
HAHC - TEC = \left[1 - \frac{\Delta P_{HAHC}}{\Delta P_{withoutHAHC}}\right] \times 100\% \tag{33}
\]

where \(\Delta P_{HAHC}\) is the load pressure disturbance after HAHC and \(\Delta P_{withoutHAHC}\) is the load pressure disturbance without HAHC.

HAHC-TEC of PI and ARISMC controllers under the sea conditions of 3-6 levels are compared in Fig.17, where the
tension compensation efficiency of ARISMC controller are always higher than that of PI controller and are not less than 95%.

To sum up the experimental analysis, the ARISMC controller presented in this paper has better advantages than the traditional PI controller in the accuracy and robustness of tension compensation.

VI. CONCLUSION

In this paper, to reduce the adverse effect of time-varying irregular heave disturbance load on the towing body and cable tension in the deep-sea towing system, HAHC scale prototype is developed, which mainly composed of accumulators, two plunger type cylinders, two piston cylinders and a high response servo valve.

To compensate the parameter uncertainties and disturbance in HAHC, a nonlinear adaptive robust integral sliding mode control algorithm for constant tension control of the heavy deep-sea towing system is designed. And based on the measured heave motion data of the mothership in the South China Sea and the simulated heave motion data of the mothership under level 3-6 sea conditions, the effectiveness of the proposed controller is verified by both simulation and experiment. The results show that the proposed ARISMC has more than 95% tension compensation efficiency and stronger robustness under level 3-6 sea conditions.

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