

Research on Active Heave Compensation Control Method and Simulation Considering the Influence of Time-delay

Hui Zhang^{1,2}, Lei Li^{1,2}, Jian-cheng liu^{1,2}, Xiu-Zhan Zhang^{1,2}, Kai-ye Hu³, Ling Cai³, Shi-Peng Wang^{1,2}, Feng Jiang^{1,2}

¹China merchants marine and offshore research institute co.ltd, Shenzhen 518054, China;

²China merchants deepsea research institute(sanya) co.,ltd, Sanya 572025, China;

³College of Shipbuilding Engineering, Harbin Engineering University, Harbin 150001, China

Corresponding Author: Kaiye Hu

ABSTRACT :As an indispensable auxiliary equipment for offshore operations such as offshore supply, ocean exploration, offshore drilling and subsea equipment installation, the crane of a floating crane is one of the core equipment of ocean engineering and plays an important role in offshore oil and gas exploitation. Installing an active heave compensation device on the crane of a floating crane can effectively reduce the heave motion of the lifted load and improve the safety and efficiency of offshore lifting operations. In order to solve the problems of time-varying parameters and serious time delay in the current active heave compensation control system. This paper proposes a new control scheme for the active heave compensation system, that is, a control method combining classical PID + Smith prediction + Extreme short-term prediction. The simulation shows that this control method can improve the stability and dynamic and static performance of the compensation control system, and can solve the problem of unstable closed-loop control caused by untimely feedback signals due to the time-delay behaviours of the system. It has good control accuracy.

KEYWORDS: active heave compensation system, PID control, time-delay, smith prediction, extreme short-term prediction

Date of Submission: 02-11-2025

Date of acceptance: 11-11-2025

I. INTRODUCTION

In order to develop and utilize marine resources to a greater extent, the development of marine engineering equipment is the primary link. Among them, ship-mounted cranes are essential supporting equipment for offshore replenishment, deep-sea drilling, seabed exploration and equipment installation. Since most of the marine oil, mines and other resources are stored in the deep sea, ship-mounted cranes must adapt to deep-sea operating conditions. However, most deep-sea operating environments are relatively harsh. During deep-sea operations, due to the interference of external factors such as wind, waves and currents, ship-mounted cranes will inevitably follow the ship and produce violent movements. For example, when lifting operations are carried out, the lifted cargo will inevitably move up and down with the ship, making it difficult to complete the lifting task safely and smoothly[1-3]. This requires that deep-sea operation ship-mounted cranes themselves have heave compensation function in order to ensure the safe and smooth completion of deep-sea installation operations. The control unit of the active heave compensation system is key in related research[4-6]. Its good control performance and operability are essential for the system's safe and efficient running. Thus, studying this control unit for deep - sea cranes is highly significant in engineering.

Over the past few decades, the heave compensation system has witnessed rapid development and has been extensively applied in marine engineering. In the hydraulic AHC test with proportional feedback carried out by Jakubowski et al. [7], it was demonstrated that the P controller could achieve a satisfactory compensation effect. However, the hydraulic cylinder exhibited severe chattering, and the impact of harmonic or random input was not taken into account. Liu et al. [8] designed an enhanced fuzzy PID controller for the space capsule salvage under harsh conditions. The fuzzy rules facilitated the tuning of PID parameters, and the original 49 rules were streamlined to 14, thus reducing the reasoning time. Xu et al.[9] proposed an adaptive backstepping

sliding mode control strategy based on a disturbance observer to address the problem of the active heave compensation secondary control hydraulic system being affected by uncertainty factors such as external disturbances and internal parameter perturbations of the model. The simulation results show that the adaptive backstepping sliding mode control strategy based on the disturbance observer can quickly observe the external disturbances and internal parameter perturbations of the system and improve the robustness of the compensation system in the face of uncertain factors. Zhao et al.[10] presents a novel electric-hydraulic servo system with an active-passive lift compensation feature designed for a lifting crane used in offshore operations. For observing the uncertain disturbance terms in the model, a new type of expanded state sliding mode observer(ESSMO) is designed, and a control strategy of sliding mode with variable structure is designed using the method of hyperbolic convergence law.

However, numerous heave compensation systems encounter delays attributable to the hydraulic drive system or the sluggish communication between the Inertial Measurement Unit (IMU) and the control system [11]. Hatleskog et al.[12] had already determined that predictions could enhance the performance of active heave compensation. Neupert et al.[13] introduced an inversion - based method that utilized a prediction algorithm to surmount system delays. This prediction algorithm combined a Fast Fourier Transform with a peak - detection algorithm to identify the dominant modes of wave motion. Subsequently, a Kalman filter was initialized with the identified modes and was used to estimate the amplitude and phase of each mode in real - time. K  chler et al.[14] put forward a comparable prediction algorithm and demonstrated its effectiveness when assuming a system delay of 0.7 s. Wave motion prediction is also crucial for other applications. For instance, Fusco et al.[15] presented a variety of different short - term forecasting methods aimed at improving the real - time control of wave energy converters.

This paper focuses on the control problem of active heave compensation. Considering the characteristics of the active heave compensation system such as large and complex structure and serious time delay, a control method combining classical PID + Smith prediction + Extreme short-term prediction is proposed. The classical PID control method is used to realize closed-loop feedback control and improve the stability and dynamic and static performance of the compensation control system. The Smith prediction control algorithm is used to solve the problem of unstable feedback control caused by system time delay. In view of the problem that the compensation output response caused by the time delay of the compensation system always lags behind the real-time movement of the mother ship, the extreme short-term prediction method is introduced. The combination of the three algorithms can better solve the control problem of active heave compensation.

II. Overall scheme design of the active heave compensation

As shown in Fig.1, the structural diagram of the active heave compensation system is presented. The active heave compensation system mainly consists of the real-time detection unit for the heave motion of the mother ship, the control unit, the hydraulic drive unit, and the mechanical execution unit. These four units coordinate and cooperate with each other to jointly complete the task of active heave compensation.

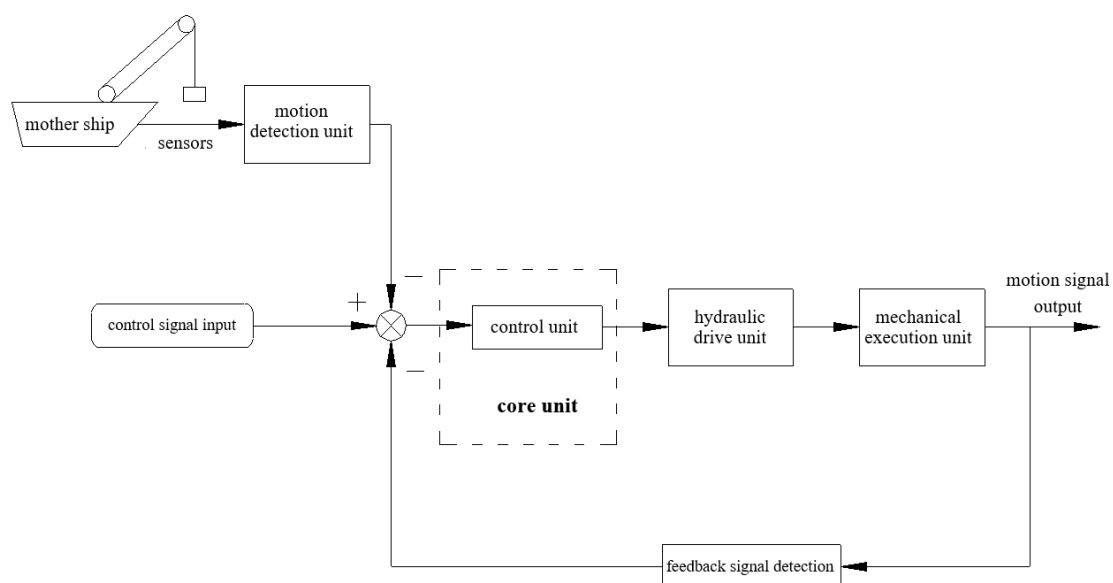


Fig.1 Active heave compensation system structure

The motion detection unit is composed of high-performance sensors and a data processing unit, which realizes the real-time detection of the heave motion of the ship. The Motion Reference Unit (MRU) can be used to detect the heave motion of the mother ship. Applying the principle of Micro Electro Mechanical Systems (MEMS) technology, it has high accuracy and good stability and is widely used in the field of ocean engineering.

The control unit consists of a hardware development platform and a software system. Based on the input, the ship's attitude motion amount, and the feedback from the replenishment device, it realizes the control of the hydraulic drive unit through certain control algorithms (such as PID, fuzzy PID, etc.). It is the core of the active heave compensation system and is required to possess high rapidity, stability, and accuracy.

The hydraulic drive unit is mainly composed of servo valves, servo amplifiers, and valve-controlled hydraulic motors. It is the power unit of the active heave compensation system and is mainly responsible for converting hydraulic energy into mechanical energy to drive the winch to complete the compensation.

The mechanical execution unit is the final actuator of the active heave compensation system and is mainly composed of planetary drive winches, wire ropes, pulley blocks, lifting frames, and so on.

For the active heave compensation system, the key lies in the control technology. The compensation system involves multiple fields such as machinery, electronics, and hydraulics. It is a large and complex system with multiple variables and time-varying characteristics, possessing severe nonlinearity and time delay, which poses higher requirements for the active heave compensation control system. The control performance of the compensation system is challenged. It is required that the control system not only has stability but also rapidity, real-time performance, and adaptability to different tasks and environments.

Under normal working conditions (in the absence of wind and waves), disconnect the active heave compensation controller and use the main control signal to complete the normal lifting/lowering operations. When the heaving motion amplitude of the crane ship is relatively large, starting the compensation controller will form an active heave compensation system.

III. MODELING OF THE CONTROL UNIT OF THE ACTIVE HEAVE COMPENSATION

In this paper, the valve-controlled hydraulic motor is selected as the hydraulic driving mechanism, and its basic principle is shown in Fig.2.

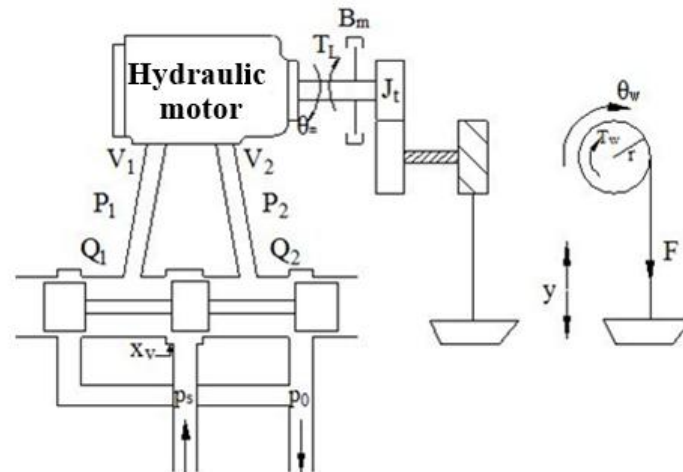


Fig.2 Hydraulic motor principle

Thus, three fundamental equations can be obtained:

(1) The linearized flow equation of the valve.

$$Q_L = K_q x_v - K_c p_L \quad (1)$$

(2) Continuity equation

$$Q_L = D_M \frac{d\theta_M}{dt} + C_{tM} p_L + \frac{V_t}{4\beta_e} \frac{dp_L}{dt} \quad (2)$$

(3) Moment equilibrium equation

$$T_g = J \frac{d^2\theta_M}{dt^2} + B_M \frac{d\theta_M}{dt} + G\theta_M + T_L \quad (3)$$

Performing Laplace transformation on the above three equations:

$$\begin{cases} Q_L = K_q X_v - K_c p_L \\ Q_L = D_M s \theta_M + C_{tM} p_L + \frac{V_t}{4\beta_e} s p_L \\ T_g = p_L D_M = J s^2 \theta_M + B_M s \theta_M + G \theta_M + T_L \end{cases} \quad (4)$$

Eliminate the intermediate variables:

$$\theta_M = \frac{\frac{K_q}{M_M} X_v - \frac{K_{ce}}{D_M^2} \left(\frac{V_t}{4\beta_e K_{ce}} s + 1 \right) T_L}{\frac{V_t J}{4\beta_e D_M^2} s^3 + \left(\frac{J K_{ce}}{D_M^2} + \frac{B_M V_t}{4\beta_e D_M^2} \right) s^2 + \left(1 + \frac{B_M K_{ce}}{D_M^2} + \frac{G V_t}{4\beta_e D_M^2} \right) s + \frac{G K_{ce}}{D_M^2}} \quad (5)$$

In the active heave compensation system, the elastic load is not taken into consideration, that is $G = 0$, and $B_M K_{ce} / D_M^2 \ll 1$, so the above formula can be simplified as follows:

$$\theta_M = \frac{\frac{K_q}{M_M} X_v - \frac{K_{ce}}{D_M^2} \left(\frac{V_t}{4\beta_e K_{ce}} s + 1 \right) T_L}{s \left(\frac{s^2}{\omega_h^2} + \frac{2\zeta_h}{\omega_h} s + 1 \right)} \quad (6)$$

Among them: ω_h represents the hydraulic natural frequency, and ζ_h represents the damping ratio of the hydraulic system.

$$\omega_h = \sqrt{\frac{4\beta_e D_M^2}{V_t J}} \quad (7)$$

$$\zeta_h = \frac{K_{ce}}{D_M} \sqrt{\frac{\beta_e J}{V_t}} + \frac{B_M}{4D_M} \sqrt{\frac{V_t}{\beta_e J}} \quad (8)$$

In the above equations: x_v represents the spool displacement, θ_M represents the rotation angle of the hydraulic motor shaft, D_M represents the theoretical displacement of the hydraulic motor, V_t represents the total volume of the valve chamber, the motor chamber and the connecting pipelines, J represents the total inertia of the hydraulic motor and the load (converted to the motor shaft), B_M represents the viscous damping coefficient of the load and the hydraulic motor, G represents the torsional spring stiffness of the load, T_L represents the external load torque acting on the motor shaft.

According to the compensation requirements and performance indicators of the active heave compensation system, the main necessary performance indicators of the control system are as follows:

- (1) Maximum vertical compensation displacement: $\pm 3\text{m}$;
- (2) Rated working load: 200t;
- (3) The accuracy of vertical displacement compensation shall not be less than 90%;
- (4) Maximum compensation speed: 62.5m/min;
- (5) Motor displacement: 355ml/rev, maximum rotational speed: 2000rpm.

IV. RESEARCH ON THE CONTROL STRATEGY OF ACTIVE HEAVE COMPENSATION

From the perspective of control strategies, the control system can be divided into feedback control, feedforward control, feedforward-feedback composite control, etc. Different control strategies can be adopted according to different control objects and control performance requirements. Feedback control is a common form of control, which corrects deviations by using deviations, that is, the deviation value serves as the basis for control, and the purpose of control is to eliminate deviations. The control system has good control accuracy and excellent adaptability.

The working environment of the active heave compensation system is complex and it is subject to many external interferences. In order to ensure the accuracy of active compensation, a feedback control strategy is

adopted. In the feedback control system, control algorithms such as classical PID, fuzzy control, and fuzzy adaptive PID can be used. Considering the real-time compensation characteristics of the active heave compensation system, as well as the characteristics of the classical PID algorithm such as small computational load and high control accuracy, this paper selects the classical PID control algorithm with a wider range of applicability. In view of the deterioration of the system control quality caused by the time delay of the compensation system, which seriously affects the stability of the control system, the Smith predictive control algorithm is added. To further address the problem that the control response output of the active heave compensation system always lags behind the real-time heave motion of the mother ship due to the time delay, the extremely short-term forecast technology is introduced.

4.1 Classical PID algorithm

PID (Proportional-Integrate-Differential) control involves performing simple proportional, integral, and differential operations on the deviation between the output signal after control and the input signal before control, so as to minimize the deviation and achieve the purpose of control. Here, P represents the proportional operation, I represents the integral operation, and D represents the differential operation.

The PID controller controls based on the deviation $e(t)$ between system output $c(t)$ and system input $r(t)$ until the deviation is eliminated or minimized. The ideal analog PID control algorithm is:

$$u(t) = K_P \left[e(t) + \frac{1}{T_I} \int_0^t e(t) dt + T_D \frac{de(t)}{dt} \right] \quad (9)$$

In the computer control process, it is necessary to approximately convert the analog PID control algorithm into a digital PID control algorithm. The following formula can be used to approximately convert the analog PID control algorithm to the digital PID control algorithm.

$$\begin{cases} \int_0^t e(t) dt = T_s \sum_{i=0}^k e(i) \\ \frac{de(t)}{dt} = \frac{e(k) - e(k-1)}{T_s} \end{cases} \quad (10)$$

where T_s is the sampling period of the system.

4.2 PID parameter tuning.

PID parameter tuning is to determine the specific values of the proportional coefficient K_P , integral time constant T_I , and derivative time constant T_D of the regulator in the optimal transition process.

This paper uses the empirical trial-and-error method to tune PID parameters. First, according to experience, the regulator parameters are preset to certain values. Then, directly under the closed-loop control system, according to the influence laws of K_P , T_I and T_D on the control system, the corresponding parameters are adjusted through on-site trial and error. The shape of the system output response curve is observed until satisfactory dynamic and static characteristics are achieved.

The process of trial and error is in the order of proportional (P), then integral (I), and finally derivative (D). The specific steps are as follows:

First set the integrator time $T_I = \infty$ and derivative time $T_D = 0$ of the regulator, and adjust the proportional coefficient K_P until a satisfactory 10:1 (servo control system) transient process response curve is obtained.

The second step introduce integral action, reduce the proportional coefficient K_P by 10% to 20%, and adjust T_I from large to small until satisfactory dynamic and static characteristics are achieved.

The third step, if derivative action needs to be introduced, preset or according $T_D = (1/3 \sim 1/4)T_I$ to empirical values and adjust it from small to large.

4.3 Smith predictive control

The active heave compensation system features a large and intricate structure, spanning multiple disciplines including mechanical, electrical, and hydraulic engineering. During the actual compensation process, system response delays inevitably occur. These delays lead to untimely feedback signals, degrading the system's control quality and severely undermining the stability of the feedback control system. As a consequence, it

becomes extremely challenging for conventional PID control to yield satisfactory results. In cases of significant delays, the control system can completely lose its stability and utterly fail to achieve the desired control effect. However, implementing Smith predictive control can effectively resolve the feedback stability issue of the PID controller within the heave compensation system.

Smith predictive control is a model-based solution proposed by O.J.M. Smith to address the time delay problem in feedback control systems. It involves connecting a compensation link in parallel with the generalized controlled object to eliminate the influence of time delay during the control process. The principle structure diagram is shown in Fig. 3 below.

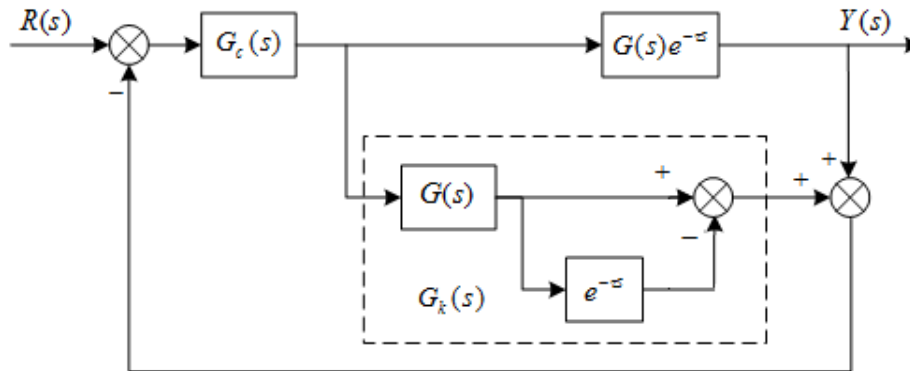


Fig.3 Block diagram of Smith predictive compensation control principle

In the fig.3, $G_c(s)$ is the system controller, $G(s)e^{-\tau}$ is the mathematical model of the controlled object (including the time-delay link $e^{-\tau}$), and $G_k(s)$ is the transfer function of the parallel compensation controller. It consists of two parts. Among them, $G(s)$ is the part without time delay in the prediction model, and $e^{-\tau}$ is the time-delay link in the prediction model. When the prediction compensator is not introduced, the closed-loop transfer function of the system is:

$$\phi_0 = \frac{Y(s)}{R(s)} = \frac{G_c(s)G(s)e^{-\tau}}{1 + G_c(s)G(s)e^{-\tau}} \quad (11)$$

It can be seen that the characteristic equation contains a lag link $e^{-\tau}$. After adding a predictive compensator to the system, the closed-loop transfer function of the system is:

$$\phi_1 = \frac{Y(s)}{R(s)} = \frac{G_c(s)G(s)e^{-\tau}}{1 + G_c(s)G(s)e^{-\tau} + G_c(s)G_k(s)} \quad (12)$$

In order to eliminate the time-delay link in the characteristic equation of the system, the following formula should hold, that is, it is required that:

$$\phi = \frac{Y(s)}{R(s)} = \frac{G_c(s)G(s)e^{-\tau}}{1 + G_c(s)G(s)} \quad (13)$$

Therefore, by combining Equation (12) and Equation (13), we can obtain:

$$\phi = \frac{Y(s)}{R(s)} = \frac{G_c(s)G(s)e^{-\tau}}{1 + G_c(s)G(s)} \quad (14)$$

In this way, the characteristic equation of the feedback control system no longer contains a time-delay link, that is, the influence of the delay of the feedback signal on the control stability of the control system is eliminated, and the stability problem of the PID control system is solved. However, the lag link $e^{-\tau}$ on the numerator of the closed-loop transfer function only delays the output response of the control system in time τ .

4.4 Extremely short-term prediction

The introduction of Smith predictive control can effectively solve the instability problem of the feedback control in the active heave compensation, improving the control quality of the system. However, due to the

influence of time delay $e^{-\tau}$, the output response of the active heave compensation control system is postponed in time. Moreover, the heave motion of the crane's mother ship has real-time characteristics, which leads to the fact that the output response of the compensation system always lags behind the real-time heave motion time of the mother ship. This seriously affects the compensation effect, and when the time delay is large, the compensation effect may not be achieved at all.

The extremely short-term prediction of ship motion is to use certain theories and technologies to predict the motion attitude of the ship within a certain period in the future. We select the AR model as the time series model to conduct extremely short-term forecasts of the heaving motion of ships. The ship motion can be regarded as a stationary random process, and it satisfies the statistical characteristics of a normal distribution. Therefore, by using the autoregressive AR model to predict the ship motion, a relatively satisfactory prediction effect can be obtained [16].

The general form of the $AR(p)$ model is:

$$x(k) = a_1x(k-1) + a_2x(k-2) + \cdots + a_px(k-p) + \zeta(k) \quad (15)$$

among them, the time series data $\{x(k), k=1,2,\cdots,N\}$ is known measurement data and is assumed to be a zero-mean stationary random sequence. The measurement error sequence $\{\zeta(k), k=1,2,\cdots,N\}$ is assumed to be a white noise sequence with zero mean and variance δ^2 . The order of the model is p . $\{a_j, j=1,2,\cdots,p\}$ are the coefficients in the model. The number of measurement data is N .

In equation (15), if $k = p+1, p+2, \cdots, N, (N \geq 2p)$, then

$$\begin{cases} x(p+1) = a_1x(p) + a_2x(p-1) + \cdots + a_px(1) + \zeta(p+1) \\ x(p+2) = a_1x(p+1) + a_2x(p) + \cdots + a_px(2) + \zeta(p+2) \\ x(N) = a_1x(N-1) + a_2x(N-2) + \cdots + a_px(N-p) + \zeta(N) \end{cases} \quad (16)$$

Definition:

$$X = [x(p+1)x(p+2)\cdots x(N)]^T \quad (17)$$

$$\Phi = \begin{bmatrix} x(p) & x(p-1) & \cdots & x(1) \\ x(p+1) & x(p) & \cdots & x(2) \\ \cdots & \cdots & \cdots & \cdots \\ x(N-1) & x(N-2) & \cdots & x(N-p) \end{bmatrix} \quad (18)$$

$$a = [a_1 \quad a_2 \quad \cdots \quad a_p]^T \quad (19)$$

$$a = [a_1 \quad a_2 \quad \cdots \quad a_p]^T \quad (20)$$

So the formula (17) can be written in the form of a vector equation.

$$X = \Phi a + \zeta \quad (21)$$

The determination of autoregressive models $AR(p)$ principally consists of two components. One is model parameter estimation, and the other is model order determination. Model parameter estimation aims to ascertain the values $\hat{a}_j$ in Equation (15). Similarly, model order determination also involves determining the relevant p values in Equation (15). Parameter estimation is the process of estimating model parameters when the model structure and order have already been defined. The objective of model order determination is to figure out the order of the model. Each distinct order corresponds to a different $AR(p)$ model configuration. Evidently, a change in the order will lead to a different model.

Least squares estimation is the best estimation method for parameter estimation of time series models. Let $\hat{a}$ be an estimate of a . The objective function J of the estimation is taken as:

$$J = (X - \Phi a)^T (X - \Phi a) \quad (22)$$

If the objective function J is minimized, then $\hat{a}$ is:

$$\hat{a}[N] = [\Phi_N^T \Phi_N]^{-1} \Phi_N^T X_N \quad (23)$$

Taking into account the real time performance of the compensation system, and on the other hand, with the aim of reducing the computer's memory footprint and enhancing the computing speed, by rearranging the above mentioned equation, the recursive form of the least squares estimation formula can be derived. Let $P_N = [\Phi_N^T \Phi_N]^{-1}$, if one more measurement of $X(N+1)$ is added later, then we can obtain:

$$\begin{aligned}
 P_N &= [\Phi_N^T \Phi_N]^{-1} \\
 &= \left[\begin{bmatrix} \Phi_N \\ \Phi_{N+1}^T \end{bmatrix}^T \begin{bmatrix} \Phi_N \\ \Phi_{N+1}^T \end{bmatrix} \right]^{-1} \\
 &= [P_N^{-1} + \Phi_{N+1}^T \Phi_{N+1}]^{-1} \\
 &= (I - \frac{P_N \Phi_{N+1} \Phi_{N+1}^T}{1 + \Phi_{N+1}^T P_N \Phi_{N+1}}) P_N
 \end{aligned} \tag{24}$$

where, I is the unit matrix.

The least squares estimation algorithm formula in recursive form is as follows:

$$\hat{a}[N+1] = \hat{a}[N] + M(N+1)[X(N+1) - \Phi_{N+1}^T \hat{a}(N)] \tag{25}$$

$$\text{where } N \geq 2p, M(N+1) = \frac{P_N \Phi_{N+1}}{1 + \Phi_{N+1}^T P_N \Phi_{N+1}}.$$

There are many methods for determining the order of a model. This paper uses the AIC criterion function method to determine the order of the model. Define $S_p(N)$ as the sum of the residual squares of the prediction model, and $I(p)$ represents the Akaike function value of the p-order model.

$$S_p(N) = (X_N - \Phi_N \hat{a}(N))^T (X_N - \Phi_N \hat{a}(N)) \tag{26}$$

$$I(p) = \log(S_p(N)/N) + 2p/N \tag{27}$$

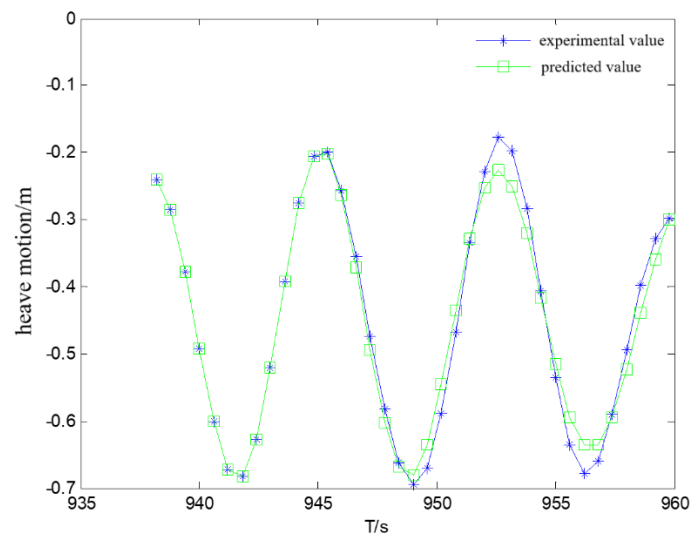
After the model parameters in Equation (22) have been optimally estimated, the forecast value for the future $k+1$ moment can be obtained:

$$\hat{x}(k+l) = \sum_{j=1}^p \hat{a}_j x(k-j) \tag{28}$$

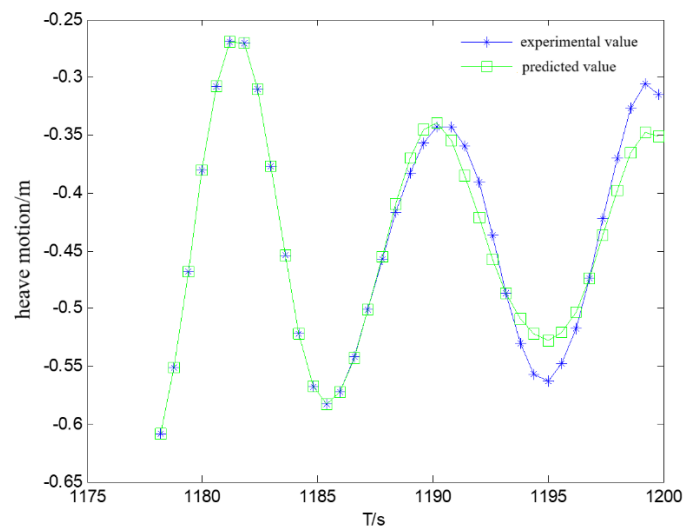
where $l = 1, 2, \dots$ is the forecast step. As long as the parameter estimation and model order determination are completed and the values of $\hat{a}_j$ and p are obtained, the forecast result of the forward forecast step can be obtained.

When the active heave compensation system of the ship crane is turned on, the data acquisition system pre-collects the heave data of the mother ship as the prediction data source for extremely short-term prediction, and updates the acquisition database of the mother ship's heave motion in real time.

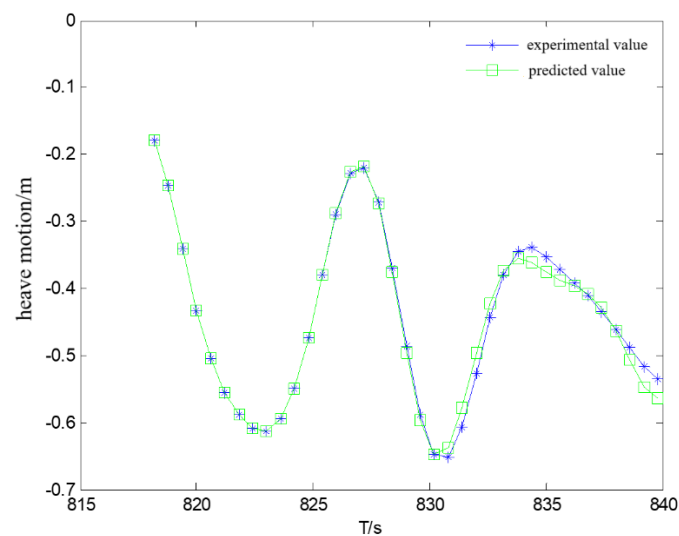
Fig. 4 is the extremely short-term prediction error chart. The training time is 100 seconds and the forecast time is 15 seconds. As can be seen from three typical examples, this forecast algorithm has high accuracy and the overall error is within 10%.



(a) Case A



(b) Case B



(c) Case C

Fig.4 Extremely short-term prediction error chart

V. SIMULATION RESULTS AND ANALYSIS

5.1 Simulation test of speed compensation control system

Using the control method combining classic PID + Smith prediction + extremely short-term prediction, a simulation model of the speed compensation control system is established as shown in Figure 5 below. Investigate the influence of the actual marine environment on the compensation performance of the speed compensation system. According to the ship type data of the crane mother ship and typical operating sea conditions, use hydrodynamic calculation software to calculate the heave velocity of the crane mother ship under actual sea conditions, and investigate the compensation performance of the speed compensation control system under actual working sea conditions.

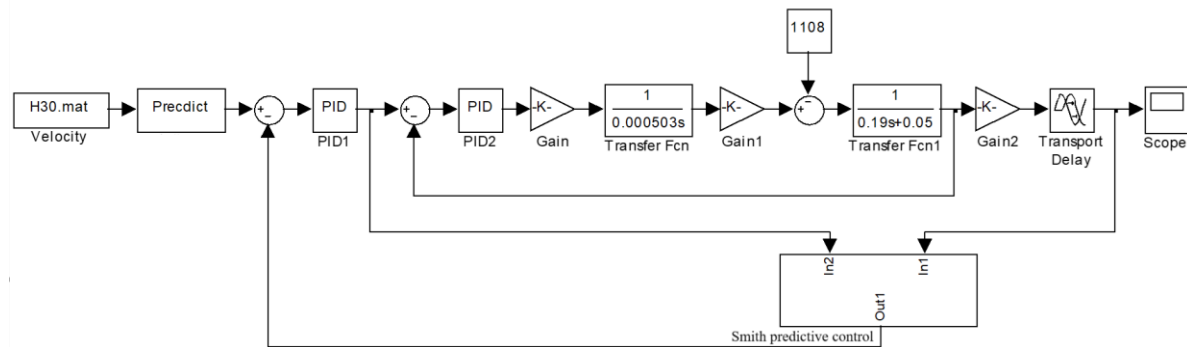


Fig.5 Simulation model of speed compensation control system

Fig.6 to 9 show the control effect diagrams of the speed compensation system by the control algorithm adopted in this paper under different sea conditions. As can be seen from the figures, the speed compensation system has a very good control effect.

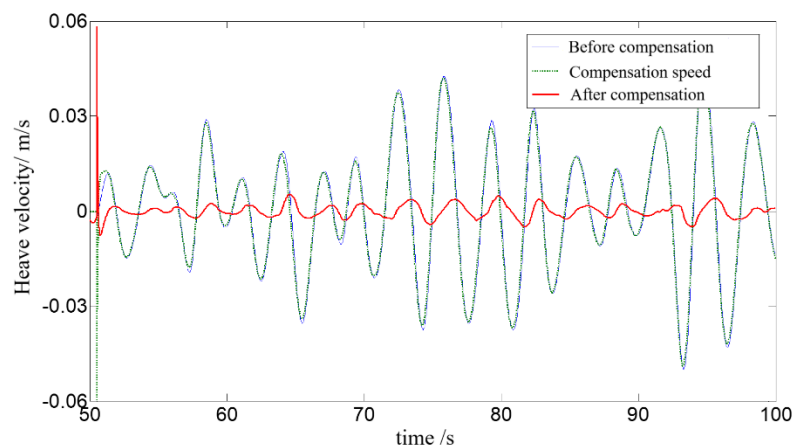


Fig.6 Simulation test of speed compensation control system ($\approx 0.5m$)

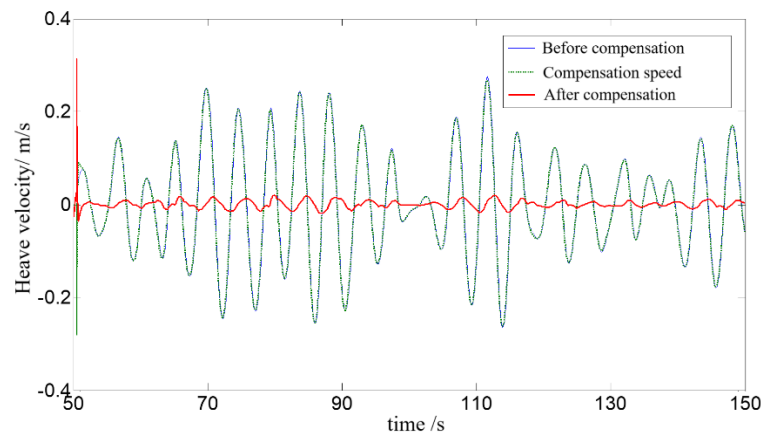


Fig.7 Simulation test of speed compensation control system ($\approx 1m$)

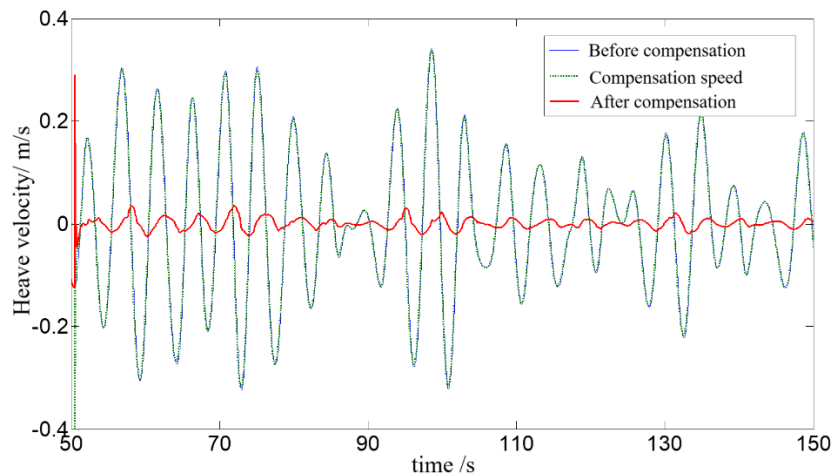


Fig.8 Simulation test of speed compensation control system ($h=1.25\text{m}$)

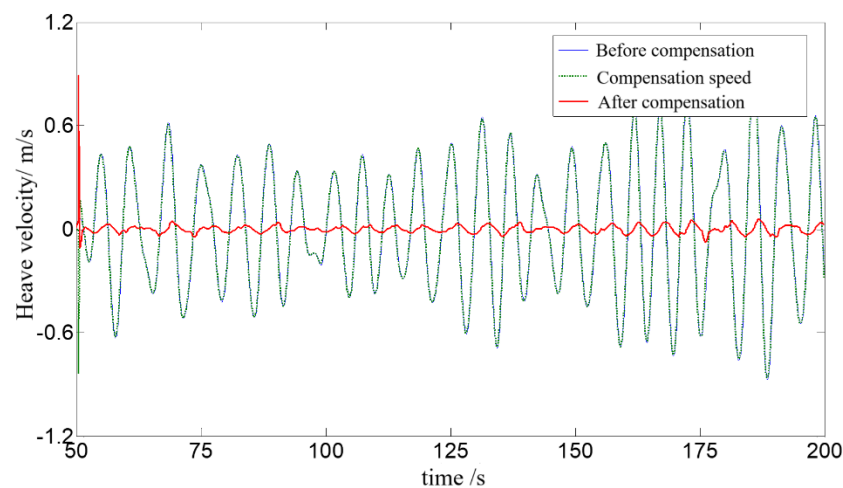


Fig.9 Simulation test of speed compensation control system ($h=2\text{m}$)

Define the accuracy of active heave compensation speed.

$$Hc = 1 - \frac{\sum_{i=1}^N |\hat{V}(i) - V(i)|}{\sum_{i=1}^N |V(i)|} \quad (29)$$

where $\hat{V}(i)$ represents the output response speed of the compensation system; $V(i)$ represents the actual heave speed of the key point P before compensation; N represents the total number of sampled data after the output becomes stable (data is taken after the response stabilizes).

Statistics of the compensation accuracy Hc of the speed compensation system under various calculated sea conditions are shown in the following table:

Table.1 Simulation results of speed compensation system						
$h_{1/3}$ (m)	0.5	1.0	1.25	2.0	2.5	3.0
Hc (%)	91.61	93.27	93.55	92.41	91.98	89.72

5.2 Simulation test of displacement compensation control system

Using the control method combining classic PID + Smith prediction + extremely short-term prediction, a simulation model of the displacement compensation control system is established as shown in Figure 10 below.

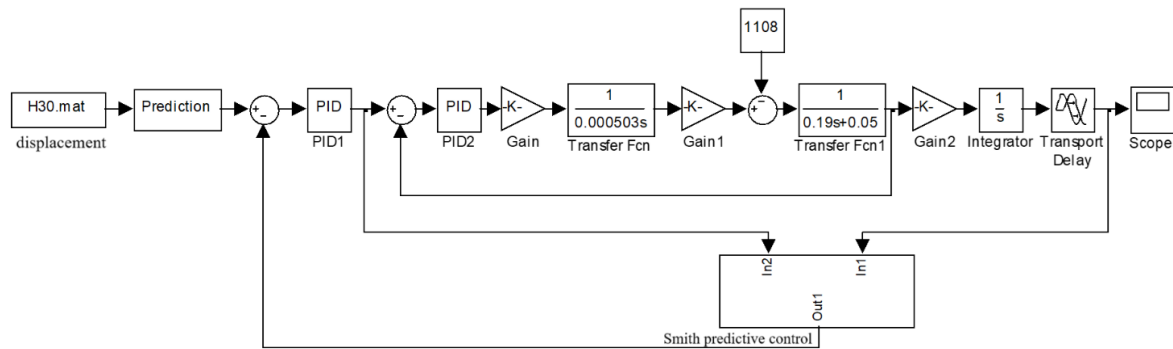


Fig.10 Simulation model of displacement compensation control system

Figures 11 to 14 show the control effect diagrams of the displacement compensation system by the control algorithm adopted in this paper under different sea conditions. As can be seen from the figures, the displacement compensation system has a very good control effect.

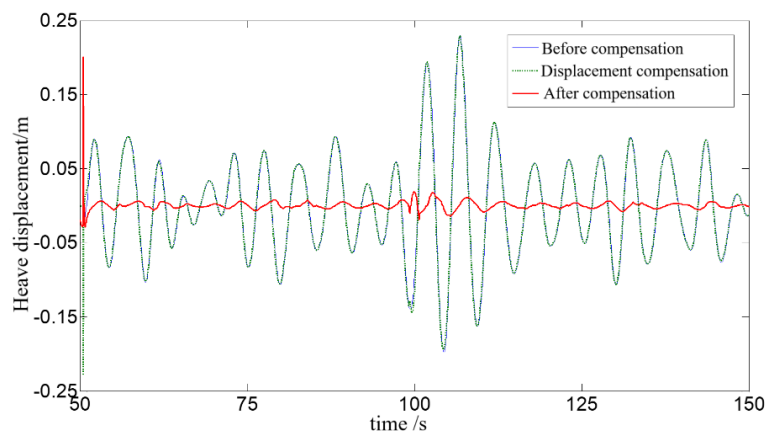


Fig.11 Simulation test of displacement compensation control system ($\Delta = 1.0\text{m}$)

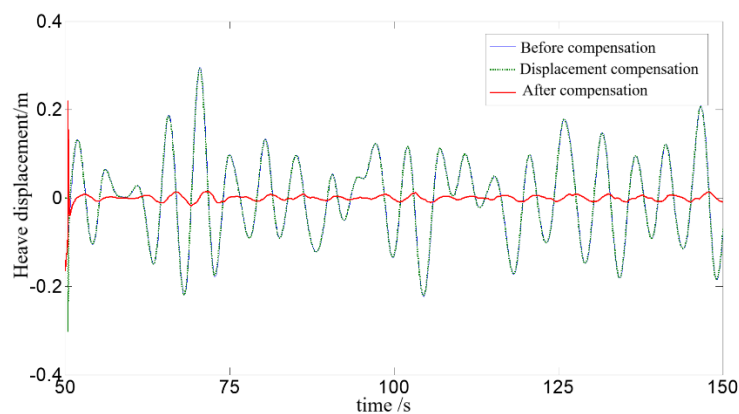


Fig.12 Simulation test of displacement compensation control system ($\Delta=1.25\text{m}$)

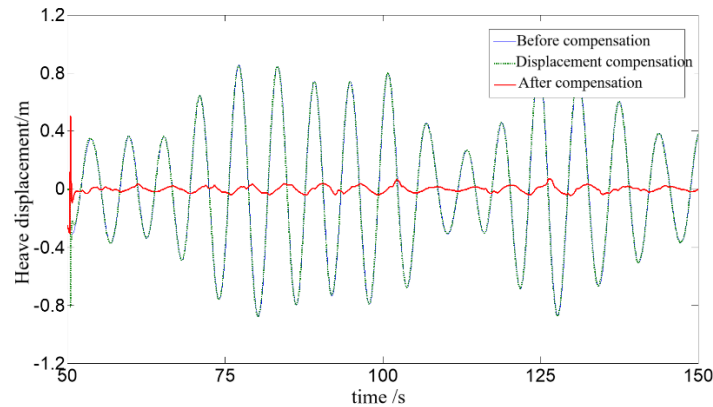


Fig.13 Simulation test of displacement compensation control system ($\Delta = 2.0\text{m}$)

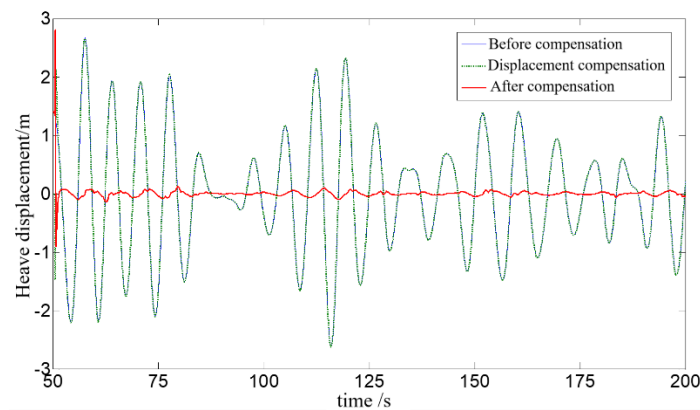


Fig.14 Simulation test of displacement compensation control system ($\Delta = 3.0\text{m}$)

Define the accuracy of active heave displacement compensation.

$$Pc = 1 - \frac{\sum_{i=1}^N |\hat{H}(i) - H(i)|}{\sum_{i=1}^N |H(i)|} \quad (30)$$

where $\hat{H}(i)$ is the output response displacement of the displacement compensation system; $H(i)$ is the real-time heave displacement of the lifting mother ship; N is the total number of sampled data after the output is stabilized (take the data after the response is stabilized).

Statistics of the compensation accuracy Hc of the displacement compensation system under various calculated sea conditions are shown in the following table:

Table2 Simulation results of displacement compensation system

$h_{1/3}$ (m)	0.5	1.0	1.25	2.0	2.5	3.0
Hc (%)	90.87	93.51	94.46	94.88	93.72	93.88

From the statistical results in the table, it can be seen that the displacement compensation system designed in this paper has high compensation performance. The compensation accuracy is basically above 90%, which can meet the technical index requirements of the active heave compensation system.

VI. CONCLUSION

This paper conducts research on the active heave compensation control model and control method. Through simulation, the following conclusions are obtained:

1. A control method combining classic PID + Smith prediction + extremely short-term prediction is proposed. In particular, the extremely short-term prediction method is used to solve the problem of large hysteresis of the active heave compensation system. The simulation test results show that this control method can enable the active heave compensation system to achieve better compensation effects and can meet the performance index requirements of active heave compensation in this paper. It provides a new control method for solving the control problem of active heave compensation.

2. Through the simulation test research of active heave compensation, the correctness of the research work on the active heave compensation control model in this paper is verified. The compensation accuracy can be maintained at more than 90%, providing a reference for the selection of PID parameters for active heave compensation control under actual sea conditions, and also providing a reference value for the development of the active heave compensation principle prototype and the design and manufacture of the actual compensation system.

REFERENCES

- [1]. Fossen, T. I. Guidance and control of ocean vehicles. *John Wiley & Sons*. West Sussex, United Kingdom. (1994)
- [2]. Chakrabarti, S. K. Handbook of offshore engineering. (2nd ed.). *Elsevier*, Amsterdam, Netherlands. (2008)
- [3]. Fossen, T. Handbook of marine craft hydrodynamics and motion control. *John Wiley and Sons*. West Sussex, United Kingdom. (2011)
- [4]. Duan, Y.X., Ren, Z.R., Zhou, L. Current status and development tendency of active heave compensation system. *Ship Science and Technology*, 42(11), 76-82. (2020)
- [5]. Woodacre, J.K., Bauer, R.J., Irani, R.A. A review of vertical motion heave compensation systems. *Ocean Engineering*. 104, 140-154. (2015)
- [6]. Xu, J., Yi, B., Zhan, Y. Review of Heave Compensation Systems: Design and Control Strategies. *2023 IEEE 11th International Conference on Computer Science and Network Technology (ICCSNT)*, Dalian, China. (2023)
- [7]. Jakubowski, A., Kubacki, A., Owczarek, P. Development of Experimental Design for Hydraulic Active Heave Compensation Systems. In: Hamrol, A., Ciszak, O., Legutko, S., Jurczyk, M. (eds) *Advances in Manufacturing. Lecture Notes in Mechanical Engineering*. Springer, Cham. (2018)
- [8]. Liu, X., Li, W., Wang, W., and Xu, Z. Control for the New Harsh sea Conditions Salvage Crane Based on Modified Fuzzy PID. *Asian Journal of Control*, 20: 1582–1594. (2018)
- [9]. Xu, J., Liu, H., Liu, H., Zhang, W., Zhan, Y. Active heave compensation control strategy for LARS based on SDOB-ABSMC, *Journal of Harbin Engineering University*. (2025)
- [10]. Zhao, F.S., Zhong, Y.G., Fu, Z.Y. Active and passive heave compensation system based on feedback linearization sliding mode variable structure control. *Ocean Engineering*, 305, 117962. (2024)
- [11]. Woodacre, J., Bauer, R., & Irani, R. A review of vertical motion heave compensation systems. *Ocean Engineering*, 104, 140–154. (2015)
- [12]. Hatleskog, J., Dunnigan, M. Active heave crown compensation sub-system. *OCEANS 2007 - Europe*, Aberdeen, UK, 1-6, (2007)
- [13]. Neupert, J., Mahl, T., Haessig, B., Sawodny, O., & Schneider, K. A heave compensation approach for offshore cranes. In *American control conference*, 538-543. (2008)
- [14]. Küchler, S., Mahl, T., Neupert, J., Schneider, K., & Sawodny, O. Active control for an offshore crane using prediction of the vessel's motion. *IEEE/ASME Transactions on Mechatronics*, 16(2), 297–309. (2011)
- [15]. Fusco, F., Ringwood, J. V. Short-term wave forecasting for real-time control of wave energy converters. *IEEE Transactions on Sustainable Energy*, 1(2), 99–106. (2010)
- [16]. Wu, W.; Liu, X.; Guo, Z.; Wang, H. Real Time-delay Control of Active Heave Compensation System for Marine Crane. *Chinese Hydraulics & Pneumatics*, 45(4), 167–174. (2020)