International Journal of Engineering Research and Development
e- ISSN: 2278-067X, p-ISSN: 2278-800X, www.ijerd.com
Volume 21, Issue 11 (November 2025), PP 134-140

Control and Simulation for Active Heave Compensation Crane Based on Fuzzy Self-adaptive PID

[1] [2] Hui Zhang, [1] [2] Jiancheng liu, [1] [2] Lei Li, [1] [2]XiuZhan Zhang, [3] Kaiye Hu, [4] Ling Cai, [1] [2]Shaoyuan Peng, [1] [2Feng Jiang

[1]. China merchants marine and offshore research institute co.ltd, Shenzhen 518054, China
[2]. China merchants deepsea research institute(sanya) co.,ltd, Sanya 572025, China
[3]. College of Shipbuilding Engineering, Harbin Engineering University, Harbin 150001, China
Corresponding Author: Kaiye Hu

ABSTRACT: *In order to solve the problem of poor control effect in the active heave compensation (AHC) system of marine cranes using traditional control algorithms, this paper studies the active heave compensation system of marine cranes and proposes a fuzzy self-adaptive PID control algorithm. Firstly, the principle of active heave compensation is introduced. Secondly the active heave compensation system is designed and its mathematical model is established. Thirdly a heave compensation algorithm is designed based on fuzzy self-adaptive PID. The results of system simulation indicate that the control algorithm can meet control requirements of the active heave compensation system, which provides a certain reference value for manufacturing a prototype of active heave compensation crane.*

KEY WORDS: *active heave compensation; fuzzy self-adaptive PID; crane; control; simulation*

Date of Submission: 05-11-2025 Date of acceptance: 15-11-2025

I. INTRODUCTION

The working environments in the deep sea are mostly harsh. During deep-sea operations, due to the interference of external factors such as wind, waves and currents, the ship-mounted crane will inevitably move violently along with the ship[1]. For example, when carrying out lifting operations, the suspended cargo will unavoidably move up and down together with the ship, making it difficult to complete the lifting task safely and smoothly. Therefore, the ship-mounted crane for deep-sea operations needs to have a heave compensation system to ensure the safe and successful completion of deep-sea installation operations [2–6].

The research on the active heave compensation control system is particularly important for the development of marine cranes for deep - sea operations. The quality of the control performance of the control system directly affects the effect of heave compensation and is the core content of the research on active heave compensation technology. Good control performance and operability of the control system are the prerequisites for the safe and efficient operation of the heave compensation system.

Michel et al. presented an active heave compensation approach based on heave measurements and a constant tension approach based on the measured rope tension[7]. Wu et al. present a cascade PID control strategy, combined with an extended state observer, to optimize the heave compensation performance of the offshore SCHD cranes[8]. Zinage and Somayajula investigate the response of proportional-derivative (PD), model predictive control (MPC), linear quadratic integral control (LQR/LQI), and sliding mode control (SMC) to disturbance and noise based on linear modeling[9]. Consider the conventional PID segment, proves insufficient for achieving satisfactory control efficiency in complex nonlinear systems. Therefore, the fuzzy PID controller [10], back propagation (BP) neural network PID controller[11], and an improved PID controller based on the radial basis function neural network method[12] are subsequently introduced.

Considering the nonlinearity and time-delay characteristics of the active heave compensation system, it is difficult to achieve the ideal control effect by using the traditional control algorithm. For this reason, this paper conducts research on the active heave compensation control of the fuzzy adaptive PID.

II. PRINCIPLE OF ACTIVE HEAVE COMPENSATION

When a working ship conducts lifting operations in the deep sea with relatively harsh conditions, due to the action of wind, waves and currents, the ship will generate violent swaying motions, and the crane on the ship will inevitably move along with the mother ship. At this time, the lifted object cannot be lifted and lowered at the previously set speed, and thus cannot be accurately landed on the platform. In addition, since the movement of the lifted object is subject to interference from the external environment at any time, the tension of the lifting wire rope will inevitably change greatly, which is likely to cause damage to working equipment such as winches, and wire rope breakage may also occur.

The first consideration in the research of the active heave compensation system is the motion of the mother ship. This paper mainly studies the compensation of the lifted object in the vertical direction, so the horizontal motion is not within the scope of this research. Among them, the roll, pitch and heave motions of the mother ship directly affect the vertical motion of the lifted object. Therefore, this paper decomposes the roll and pitch motions of the mother ship into the vertical direction and conducts research on active heave compensation based on the heave motion of the mother ship.

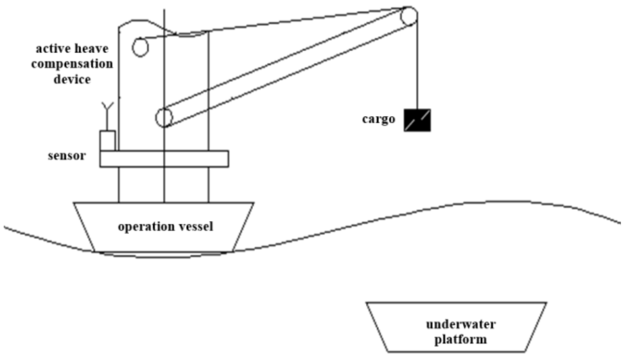

Figure 1: Schematic diagram of the principle of active heave compensation.

Suppose that the real - time heave speed of operation vessel in the deep sea is V_A, the hoisting rope retracting and releasing speed of the crane is V_S, the speed of the lifted object relative to the underwater platform without heave compensation is V_M, and the speed of the lifted object relative to the underwater platform with heave compensation is V'_M, as shown in Figure 1.

When the heave compensation is not turned on, there is

$$V_M = V_S + V_A \tag{1}$$

The goal of speed compensation is:

$$V'_M = V_S \tag{2}$$

Suppose the compensation speed is V_C, and the speed of the heavy object after compensation is V'_M. After the heave compensation is turned on, the speed of the heavy object is:

$$V'_M = V_C + V_S + V_A \tag{3}$$

According to the goal of speed compensation, the relationship between the compensation speed and the speed of the ship's heave motion can be deduced:

$$V_C = -V_A \tag{4}$$

III. DESIGN AND MODELING OF THE ACTIVE HEAVE COMPENSATION SYSTEM

3.1 Overall Design of the Active Heave Compensation System

The active heave compensation system consists of four subsystems, namely the detection system, the control system, the drive system and the mechanical execution system.

The detection system is composed of high - performance sensors and data processing units, realizing the real - time detection of signals such as ship attitude motion, wire rope tension, and displacement and speed of heavy objects. The control system consists of a hardware development platform and a software system. According to the input quantity of the replenishment device, the ship attitude motion quantity and the feedback quantity, the control of the drive system is achieved through a certain control algorithm. The drive system is divided into the main drive system and the auxiliary drive system according to different control quantities, and the main control signal (that is, the input quantity of the replenishment device) and the compensation signal are input respectively. The mechanical execution system is composed of a planetary - drive winch, wire ropes, pulley blocks and a lifting frame, etc., and is the final device for realizing material replenishment.

3.2 Modeling of the mechanical execution system

In this paper, the function of the mechanical execution system is simulated and realized through the differential planetary gear train model. Planetary gear transmission has good differential speed effect, high speed - regulating accuracy, stable operation and good speed - regulating characteristics.

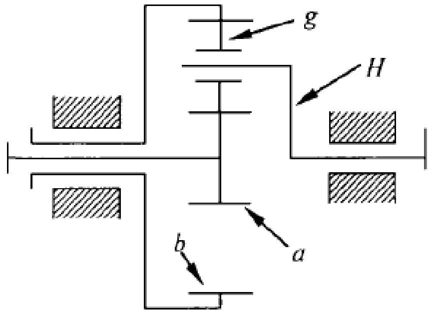

Figure 2 NGW - type differential planetary gear train

The 2K - H (NGW) planetary gear transmission is shown in Figure 2. Establish the 2K - H planetary gear dynamic model. A lumped - mass model is used during modeling. The generalized coordinates of the model are as follows: the angular displacement of the sun gear s is θ_s , the angular displacement of the i-th planet gear pi is θ_{pi} (i=1，2，3) , and the angular displacement of the planet carrier H is θ_H . The dynamic differential equations of the mathematical model of the differential planetary gear train can be derived from the Lagrange equation:

$$\begin{cases} J_s\ddot{\theta}_s + \sum_{i=1}^{3}(P_{spi} + D_{spi})r_{bs} = T_D \\ J_{pi}\ddot{\theta}_{pi} - (D_{spi} - D_{rpi} + P_{spi} - P_{rpi})r_{bps} = 0 \\ (J_H + \sum_{i=1}^{3} m_{pi}{r_H}^2)\ddot{\theta}_H - \sum_{i=1}^{3}(D_{spi} + D_{rpi} + P_{spi} + P_{rpi})r_H \cos\alpha = -T_L \end{cases} \tag{5}$$

In the equation, i=1, 2, 3; J is the moment of inertia of each central gear and planet carrier; r_b is the base circle radius of the gear; r_H is the radius of the planet carrier, m is the mass; α is the meshing angle of the gear pair; T_D is the input torque; T_L is the load torque; P and D are the elastic meshing force and viscous meshing force of the gear pair respectively.

3.3 Modeling of the drive system

For the convenience of the research of the experimental system, the function of the drive system in the design scheme of the active heave compensation system is simulated and realized through a stepping motor. The stepping motor has good operating characteristics and is a typical digitally - controlled actuating element. Selecting a stepping motor as the servo - drive has little impact on the research of the control system.

The force transfer function of the stepping motor in the control system can be directly expressed by the rotor motion equation:

$$G_T(s) = \frac{\omega(s)}{T_r(s)} = \frac{1/J}{s + B/J} = \frac{b}{s+a} \tag{6}$$

where $a = B/J$, $b = 1/J$, and $\tau_L = 1/a$ is the torque time constant.

Considering the magnetic - field - controlled motor, the approximate linear relationship between the motor torque and the winding current is

$$T_e(s) = K_m \cdot I_f(s) \tag{7}$$

where K_m is the motor constant.

If the back electromotive force is neglected, the relationship between the exciting current and the magnetic field voltage can be obtained as

$$I_f(s) = \frac{1}{(R_f + L_f s)} \cdot V_f(s) = \frac{d}{s+c} \cdot V_f(s) \tag{8}$$

where $c = R_f/L_f$, $d = 1/L_f = c/R_f$ and $\tau_f = 1/c$ is magnetic field time constant

Therefore, the transfer function of the stepping motor control model is

$$G_m(s)=\frac{\omega(s)}{V_f(s)}=\frac{d}{s+c}\cdot K_m\cdot\frac{b}{s+a}=\frac{K_m/(J\cdot L_f)}{(s+B/J)\cdot(s+R_f/L_f)} \tag{9}$$

III. Design of Fuzzy Adaptive PID Control Algorithm

PID control is the most commonly used method in industrial process control because the PID controller has a simple structure, is easy to implement, and can effectively control quite a number of industrial objects (or processes). However, the limitation of conventional PID control lies in the fact that when the controlled object has complex nonlinear characteristics, it is difficult to establish an accurate mathematical model, and due to the uncertainty of the object and the environment, it is often difficult to achieve satisfactory control results. Fuzzy - adaptive PID is a control strategy proposed to address the above - mentioned problems. The traditional PID control algorithm is expressed as:

$$u(k)=K_P e(k)+K_I\sum_{j=0}^{k}e(j)+K_D\left[e(k)-e(k-1)\right] \tag{10}$$

where $e(k)$ is the deviation value of the input at the k-th sampling moment; K_P is the proportional coefficient; K_I is the integral coefficient, $K_I=K_P T/T_I$; K_D is the differential coefficient, ; T is the sampling period, T_I is the integral time constant; T_D is the differential time constant.

The two input variables of the fuzzy - adaptive PID controller are error(e) and error change(ec) respectively. The fuzzy self - tuning of PID parameters is to find out the fuzzy relationship between the three PID parameters K_P 、 K_I 、 K_D, with e and ec. During operation, based on fuzzy reasoning, according to e and ec at different times, the fuzzy control rules are used to make on - line corrections to K_P 、 K_I and K_D, to meet the different requirements of different e and ec for control parameters, so that the controlled object has good dynamic and static performances.

On the basis of the PID algorithm, the fuzzy self - tuning PID calculates the current system error e and the error change rate etc., uses fuzzy rules for fuzzy reasoning, and queries the fuzzy matrix table for parameter adjustment. The core of fuzzy control design is to summarize the technical knowledge and practical operation experience of engineering designers, establish an appropriate fuzzy rule table, and obtain fuzzy rule tables for tuning the three parameters K_P 、 K_I and K_D respectively.

Table 1 Fuzzy rule table of K_P

ΔK_P (e \ ec)	NB	NM	NS	ZO	PS	PM	PB
NB	PB	PB	PM	PM	PS	ZO	ZO
NM	PB	PB	PM	PS	PS	ZO	NS
NS	PM	PM	PM	PS	ZO	NS	NS
ZO	PM	PM	PS	ZO	NS	NM	NM
PS	PS	PS	ZO	NS	NS	NM	NM
PM	PS	ZO	NS	NM	NM	NM	NB
PB	ZO	ZO	NM	NM	NM	NB	NB

Table 2 Fuzzy rule table of K_I

ΔK_I (e \ ec)	NB	NM	NS	ZO	PS	PM	PB
NB	NB	NB	NM	NM	NS	ZO	ZO
NM	NB	NB	NM	NS	NS	ZO	ZO
NS	NB	NM	NS	NS	ZO	PS	PS
ZO	NM	NM	NS	ZO	PS	PM	PM
PS	NM	NS	ZO	PS	PS	PM	PB

PM	ZO	ZO	PS	PS	PM	PB	PB
PB	ZO	ZO	PS	PM	PM	PB	PB

Table 3 Fuzzy rule table of K_D

ΔK_D (e \ ec)	NB	NM	NS	ZO	PS	PM	PB
NB	PS	NS	NB	NB	NB	NM	PS
NM	PS	NS	NB	NM	NM	NS	ZO
NS	ZO	NS	NM	NM	NS	NS	ZO
ZO	ZO	NS	NS	NS	NS	NS	ZO
PS	ZO	ZO	ZO	ZO	ZO	ZO	ZO
PM	PB	NS	PS	PS	PS	PS	PB
PB	PB	PM	PM	PM	PS	PS	PB

After the fuzzy rule tables of K_P、K_I and K_D are established, the adaptive correction of K_P、K_I and K_D can be carried out according to the following methods.

The variation ranges of the system error e and the error change rate ec are defined as the universes of discourse on the fuzzy set.

$$e, ec = \{-5,-4,-3,-2,-1,0,1,2,3,4,5\} \tag{11}$$

Its fuzzy subsets are

$$e, ec = \{NB, NM, NS, O, PS, PM, PB\} \tag{12}$$

The elements in the subsets represent negative large, negative medium, negative small, zero, positive small, positive medium, and positive large respectively. Suppose e、ec, K_P、K_I and K_D all follow normal distribution. Therefore, the membership degrees of each fuzzy subset can be obtained. According to the membership degree assignment table of each fuzzy subset and the fuzzy control model of each parameter, the fuzzy matrix table of PID parameters is designed by applying fuzzy synthetic reasoning, and the correction parameters are looked up and substituted into the following formula for calculation:

$$K_P = K_{P0} + \{e_i, ec_i\}_P \tag{13}$$

$$K_I = K_{I0} + \{e_i, ec_i\}_I \tag{14}$$

$$K_D = K_{D0} + \{e_i, ec_i\}_D \tag{15}$$

During the online operation process, the control system completes the online self - calibration of the PID parameters through the processing of the results of the fuzzy logic rules, look - up tables and operations.

IV. ANALYSIS OF THE SIMULATION OF THE ACTIVE HEAVE COMPENSATION SYSTEM

4.1 Simulation of the Control System

Use Matlab to simulate the control system. According to the simplified model structure and the actual system, the system parameters determined are as follows, see Table.4 and Table.5.

Table 4 Parameters of the winch system

Item	Parameter value
Number of teeth of the sun gear z_a	19
Number of teeth of the planetary gear z_H	29
Number of teeth of the ring gear z_b	77
Number of teeth of the planetary gear N	3
Diameter of the drum (mm)	164
Module	1

Table 5 Parameters of the stepper motor

Item	Parameter value
Moment of inertia $J(N \cdot m \cdot s^2)$	1.13×10^{-3}
Viscous damping coefficient $B(N \cdot m \cdot s)$	3.62×10^{-3}
Motor constant $K_m(N \cdot m \cdot A)$	50×10^{-3}
Magnetic - field time constant $\tau_f(ms)$	1
Winding resistance $R(\Omega)$	0.8
Step angle $\theta_b(°)$	0.9/1.8

4.2 Analysis of Simulation Results

Apply the PID control algorithm, the fuzzy control algorithm, and the fuzzy - adaptive PID control algorithm respectively to simulate the active heave compensation control system in terms of step response and sinusoidal signal tracking. The system sampling time T = 0.01s, and the simulation results are shown in Figure 3 and Figure 4.

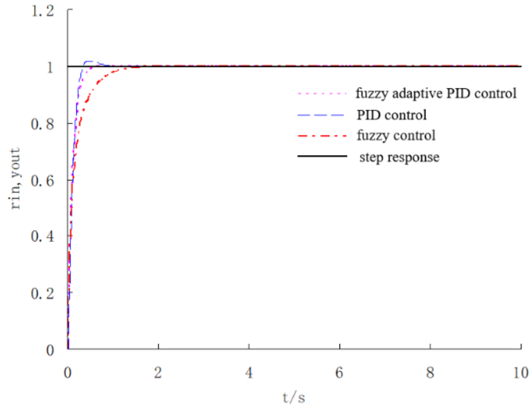

Figure 3: Simulation of Step - response of the Control System

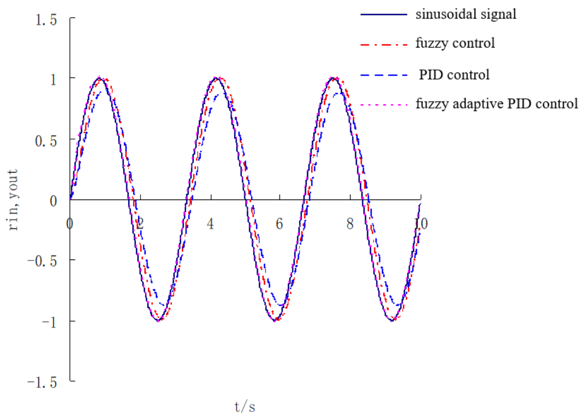

Figure 4: Simulation of Sinusoidal Tracking Performance of the Control System

Figure 3 presents the step - response of the control system. It can be seen that with the above - mentioned three control algorithms, the control system can respond to the change of the reference input relatively quickly. However, the PID control algorithm has a certain overshoot in the initial stage. Although the fuzzy control algorithm has a small steady - state error, its response time is relatively long. The fuzzy - adaptive PID control algorithm has a short response time, a small steady - state error, and good rapidity and stability.

Figure 4 presents the sinusoidal tracking results of the control system. It can be seen that with the above - mentioned three control algorithms, the control system can follow the sinusoidal input signal relatively well. However, the PID control algorithm has a slow response speed and the largest following error. The fuzzy control algorithm has a certain oscillation in the initial stage and relatively poor stability. The fuzzy - adaptive PID control algorithm has a fast response speed, stable tracking, and the smallest following error, and has good following performance and dynamic adaptability.

By comparing the above three control algorithms, it can be seen that the fuzzy - adaptive PID control algorithm has better control performance than the other two control algorithms. It not only has good rapidity and stability but also high control precision, and can well meet the control requirements of the active heave compensation system.

V. CONCLUSION

The fuzzy - adaptive PID control algorithm is highly practical, has good rapidity and excellent dynamic performance, and can well meet the control requirements of the active heave compensation system. The simulation results in this paper verify the feasibility of this control algorithm and have certain reference value for the manufacture of the prototype of the active heave - compensation crane.

ACKNOWLEDGEMENTS

The authors would like to acknowledge support from the National Key Research and Development Program projects of China(Grant No.2021YFC2801500)

REFERENCES

[1]. Lloyd ARJM. Seakeeping Ship Behaviour in Rough Weather. Ellis Horwood Ltd; 1998

[2]. Aker MH. Active heave compensator. Technical Bulletin, Kristiansand, Norway; 1998

[3]. Fossen TI. Handbook of marine craft hydrodynamics and motion control. John Wiley & Sons Ltd; 2011.

[4]. Xu JA, Yi BL, Zhan Y. Review of Heave Compensation Systems:Design and Control Strategies, 2023, p.406–413. https://doi.org/10.1109/ICCSNT58790.2023.10334591.

[5]. Korde UA. Active heave compensation on drill-ships in irregular waves.Ocean Engineering, 1998, 25(7), p.541-561. https://doi.org/10.1016/S0029-8018(97)00028-0.

[6]. Li, Zhi, Ma, Xin, Li, Yibin, Meng, Qinglin, Li, Jing, 2019. ADRC-ESMPC active heave compensation control strategy for offshore cranes. Ships Offshore Struct. https://doi. org/10.1080/17445302.2019.1703388.

[7]. Michel, A., Kemmetmller, W, & Kugi, A. (2012). Modellierung und Regelung eines aktiven Wellenkompensationssystems für Tiefseekräne. At- Automatisierungstechnik Methoden Und Anwendungen Der Steuerungs-, Regelungs- Und Informationstechnik 2012,60, p.39–52. https://doi.org/10.1524/auto.2012.0968

[8]. Wu JB., Wu DL. Integrated design of an active heave compensation crane with hydrostatic secondary control. *OCEANS 2018 MTS/IEEE Charleston*, Charleston, SC, USA, 2018, pp. 1-7, https://10.1109/OCEANS.2018.8604859.

[9]. Zinage S., Somayajula A., A comparative study of different active heave compensation approaches. Ocean Systems Engineering 2020,10 (4), 27. https://doi.org/ 10.12989/ose.2020.10.4.373

[10]. Yan X, Yan F, Zhao D, Li S. Speed control of secondary regulation heave compensation based on fuzzy PID. J. Phys.: Conf. Ser 2023. 2419. https://10.1088/1742-6596/2419/1/012058

[11]. Shi M, Guo S, Jiang L, Huang Z, 2019. Active-passive combined control system in crane type for heave compensation. IEEE Access 2019, 99, p.159960-159970. https://10.1109/ACCESS.2019.2950703.

[12]. Liu J., Chen X., 2022. Adaptive control based on neural network and beetle antennae search algorithm for an active heave compensation system. Int. J. Control Autom.Syst 2022. 20 (2), p.515–525. https://doi.org/10.1007/s12555-020-0615-2.