

## Article

# Energy-Based Coupling Control for 5-DOF Marine Cranes with Fuzzy Observation and Adaptive Gravity Compensation

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## Abstract

With the rapid development of offshore engineering, marine cranes are widely deployed in critical maritime operations, such as the precision installation of wind turbine blades. However, their highly coupled three-dimensional spatial dynamics, uncertain payload mass, and susceptibility to severe external sea wave disturbances pose significant challenges in achieving fast, accurate payload transportation and rapid anti-swing performance. To address these issues, this paper proposes a novel energy-based intelligent coupling control strategy utilizing fuzzy logic and adaptive gravity compensation for 5-DOF (5-Degrees of Freedom) marine cranes. Firstly, to handle the severe underactuation and facilitate natural energy dissipation, a novel set of error coupling variables is constructed, organically linking the actuated crane structure with the unactuated payload swing dynamics. Then, an adaptive gravity compensation mechanism is designed to dynamically estimate the uncertain payload mass in real time, eliminating the need for precise prior mathematical models. Subsequently, to counteract complex external environmental disturbances and unmodeled internal dynamics, a targeted fuzzy observer is developed based on the universal approximation theorem, providing robust, real-time perturbation compensation. The sufficient conditions for the asymptotic stability of the closed-loop system are rigorously proven based on the Lyapunov method and LaSalle's invariance principle. Finally, extensive comparative simulations are conducted, demonstrating that the proposed method significantly improves the operational accuracy, anti-swing capability, and safety of marine cranes under varying load conditions and persistent wave disturbances.

**Keywords:** marine cranes; underactuated systems; energy-based coupling control; fuzzy observer; adaptive gravity compensation; anti-swing control



Academic Editor: João Falcão Carneiro

Received: 25 May 2026

Revised: 4 July 2026

Accepted: 6 July 2026

Published: 9 July 2026

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## 1. Introduction

Marine cranes represent a highly specialized and indispensable category of heavy machinery designed specifically for cargo transfer operations and structural assembly within harsh and unpredictable marine environments. They are primarily employed for critical maritime tasks such as inter-ship cargo transfer, marine replenishment, and the precise deployment of large-scale engineering equipment [1,2]. Particularly in contemporary offshore engineering endeavors, marine cranes are increasingly tasked with the multi-machine coordinated anti-swing and precision positioning of non-coupled irregular heavy loads, such as the delicate lifting and installation of wind turbine blades. Unlike subsea deployment operations that involve complex cross-medium hoisting conditions,

the atmospheric installation of wind turbine blades relies entirely on highly precise aerial trajectory tracking and rapid vibration suppression [3,4]. Most marine cranes in use today are rotary cranes, which offer distinct advantages such as a compact footprint, structural robustness, and ease of spatial control coordination [5,6]. However, the marine environment introduces complex external disturbances such as wind, waves, and ocean currents, which inevitably induce persistent ship roll and yaw motions, substantially complicating the control tasks [7,8]. In the presence of these unknown external environmental disturbances and uncertain mechanical model parameters, achieving fast and accurate payload positioning alongside rapid anti-swing suppression remains a pressing theoretical control challenge with profound practical significance [9,10].

The dynamic behavior of a 5-DOF (5-Degrees of Freedom) marine crane incorporates base rotational degrees of freedom alongside boom luffing and hoisting mechanisms, resulting in highly coupled three-dimensional (3D) spatial motion [11]. Unlike conventional 2D overhead cranes or simplified planar systems, the 5-DOF architecture must account for simultaneous yaw and roll disturbances induced by unpredictable sea waves, thereby introducing additional, severely underactuated challenges for payload positioning and rapid anti-swing [12,13]. Existing control methods for 5-DOF marine cranes, while foundational, remain relatively limited in addressing the full scope of spatial coupling. For instance, several online trajectory planning methods for 3D offshore boom cranes have been proposed, where preliminary reference trajectories are meticulously designed for different crane motions to deliver the payload to the target position smoothly [14,15]. Furthermore, various output feedback control approaches and robust tracking controllers have been developed to address the regulation of 5-DOF offshore cranes operating under continuous wave perturbations [16–18]. While these existing methods have made significant progress in basic trajectory tracking and sway reduction, they often rely on simplified assumptions regarding the physical coupling between actuated crane structures and unactuated payload swing dynamics [19,20]. Consequently, designing advanced, energy-based intelligent coupling control methodologies that fully exploit the natural physical properties of the underactuated system is crucial for achieving superior synchronized positioning and anti-swing performance [21].

Beyond the fundamental challenges of trajectory planning and dynamic coupling, model-based control methods for marine cranes face a ubiquitous and critical issue: control performance may severely degrade due to model parameter uncertainties [22,23]. This problem arises because theoretical dynamic models derived from Lagrangian mechanics cannot perfectly match the actual physical hardware platforms operating in real-world oceans. In practical offshore assembly tasks, such as handling different types of wind turbine blades or variable rigging equipment, the exact mass of the payload is often uncertain and subject to abrupt changes [24]. Furthermore, variations in joint friction, unmodeled actuator dynamics, and shifting centers of gravity are prevalent, as extensively observed in studies concerning underactuated tower cranes, bridge cranes, and surface vessels [25–27]. If a controller relies strictly on nominal model parameters, these variations can lead to significant steady-state errors, excessive overshoot, and dangerous payload oscillations. To bridge this sim-to-real gap, recent studies have attempted to address parameter mismatches using adaptive control mechanisms or neural network approximations [28,29]. However, continuously and accurately compensating for the gravitational and inertial variations caused by unknown payload masses remains a challenging prerequisite for achieving high-precision offshore lifting operations.

Equally detrimental to the stability of marine cranes are the severe and unpredictable external environmental disturbances. Continuous sea waves and gusting winds subject the ship base to constant roll and yaw motions, which act as unmatched disturbances that prop-

agate directly to the unactuated payload, exciting violent multi-dimensional swings [30]. To mitigate these effects, recent literature has explored various active compensation mechanisms and advanced modeling techniques (e.g., feedforward compensation methodologies explored in complex multi-axis mechanical systems like MEMS gyroscopes[31]) to estimate and counteract external perturbations. Traditional linear feedback controllers often fail to maintain robustness under such persistent excitation. To mitigate these effects, recent literature has explored the integration of disturbance observers, active disturbance rejection control (ADRC), and sliding mode control to estimate and compensate for wave-induced perturbations [32,33]. While these disturbance estimation techniques have shown promise, they can be highly sensitive to tuning parameters and often struggle to approximate the highly nonlinear, high-frequency unmodeled dynamics inherent to a heavy-duty 5-DOF crane system operating in rough seas. Therefore, developing an intelligent approximation mechanism, such as a fuzzy logic system based on the universal approximation theorem, is highly desirable. A well-designed fuzzy observer can adaptively estimate and compensate for complex, aggregated external disturbances and unmodeled dynamics in real time, thereby ensuring that the control inputs remain robust and chatter-free.

While existing control strategies have made significant progress, they exhibit specific limitations when applied to complex offshore environments. Traditional linear feedback controllers, such as Proportional-Derivative (PD) and Linear Quadratic Regulator (LQR) methods [6], often fail to maintain robustness and exhibit significant steady-state errors under persistent, unmatched wave excitation. Meanwhile, conventional Energy-Based Error (EE) controllers heavily rely on precise physical parameters and may lack instantaneous adaptability to handle sudden uncertain payload masses or unmodeled dynamics. Consequently, designing an intelligent energy-based coupling control methodology—enhanced by fuzzy observation and adaptive gravity compensation—is crucial to providing a robust, adaptive, and chatter-free solution for superior synchronized positioning and anti-swing performance.

To tackle the compounding challenges of highly coupled 3D dynamics, unknown and variable payload parameters, and complex external sea wave disturbances, this paper proposes a novel energy-based intelligent coupling control strategy utilizing fuzzy logic and adaptive gravity compensation for 5-DOF marine cranes. The main innovations and technical contributions of this paper are concisely summarized as follows:

- (1) **Construction of a Novel Error Coupling Variable:** To address the severe underactuation of the 5-DOF marine crane, a novel set of error coupling variables is designed. This construct mathematically links the actuated structural state variables with the unactuated payload swing variables, facilitating natural energy dissipation and actively utilizing the suppression of payload swing to assist in precise trajectory tracking.
- (2) **Design of an Adaptive Gravity Compensation Mechanism:** Recognizing the uncertainty of payload mass in practical offshore assembly, an adaptive gravity compensation term is integrated into the control law. This mechanism dynamically estimates the payload mass parameters in real time, seamlessly adapting to varying load conditions and eliminating the need for precise prior mathematical models.
- (3) **Development of a Dedicated Fuzzy Observer:** To ensure robustness against unpredictable external environmental disturbances and unmodeled internal dynamics, a targeted fuzzy observer is developed. Based on the universal approximation theorem, this observer accurately estimates and actively compensates for complex perturbations in real time, significantly enhancing the anti-interference capabilities of the closed-loop system.

The remainder of this paper is organized as follows: Section 2 presents the derivation of the 5-DOF marine crane dynamic model. Section 3 details the core design of the energy-

based coupling controller, the fuzzy observer, the adaptive gravity compensation law, and the rigorous Lyapunov stability analysis. Section 4 provides extensive comparative simulation results. Finally, Section 5 concludes the paper.

## 2. Theoretical Backgrounds

As shown in Figure 1, the physical configuration of the marine crane system operating in complex offshore environments is established. To accurately describe the spatial kinematics, two fundamental reference frames are introduced:  $I_G$  represents the Inertial coordinate system, and  $I_S$  denotes the Ship coordinate system, which dynamically rotates with the wave-induced ship roll angle  $\alpha$  and pitch angle  $\beta$ . In this model,  $\alpha$  represents the ship's roll angle, and  $\beta$  represents the ship's pitch angle induced by continuous ocean waves. For the crane structure,  $\phi_1$  is the pitch angle of the boom, and  $\phi_2$  is the rotation angle of the boom. Regarding the heavy payload,  $\theta_1$  denotes its transverse swing angle, and  $\theta_2$  denotes its longitudinal swing angle. The physical parameters of the mechanical structure are defined as follows:  $m$  is the mass of the boom,  $M$  is the mass of the payload,  $l$  is the length of the boom.

To facilitate mathematical clarity and ensure consistent readability throughout the dynamic modeling and controller derivation, all fundamental physical parameters, coordinate frames, and state variables are comprehensively defined in Table 1.

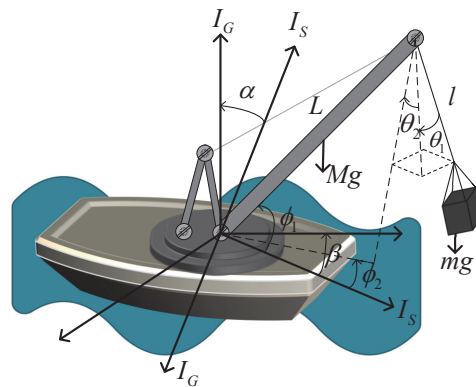
**Table 1.** List of Fundamental Notations and Variables.

| Symbol                                                   | Physical Meaning                                                          |
|----------------------------------------------------------|---------------------------------------------------------------------------|
| $I_G$                                                    | Inertial coordinate system                                                |
| $I_S$                                                    | Ship coordinate system                                                    |
| $\alpha$                                                 | Ship roll angle induced by continuous ocean waves                         |
| $\beta$                                                  | Ship pitch angle induced by continuous ocean waves                        |
| $\phi_1$                                                 | Pitch angle of the crane boom                                             |
| $\phi_2$                                                 | Rotation angle of the crane boom                                          |
| $\theta_1$                                               | Transverse swing angle of the unactuated payload                          |
| $\theta_2$                                               | Longitudinal swing angle of the unactuated payload                        |
| $m$                                                      | Mass of the rigid crane boom                                              |
| $M$                                                      | Mass of the suspended payload                                             |
| $l$                                                      | Length of the hoisting rope                                               |
| $\eta_1$                                                 | Actuated rotational angle of the boom ( $\phi_1 - \alpha$ )               |
| $\eta_2$                                                 | Actuated luffing angle of the boom ( $\phi_2 - \beta$ )                   |
| $\eta_3$                                                 | Actuated length of the hoisting rope ( $l$ )                              |
| $\eta_4$                                                 | Unactuated transverse swing angle of the payload ( $\theta_1 - \alpha$ )  |
| $\eta_5$                                                 | Unactuated longitudinal swing angle of the payload ( $\theta_2 - \beta$ ) |
| $\mathbf{M}(\boldsymbol{\eta})$                          | Symmetric and positive-definite inertia matrix of the system              |
| $\mathbf{V}(\boldsymbol{\eta}, \dot{\boldsymbol{\eta}})$ | Coriolis and centrifugal force matrix of the system                       |

To facilitate the subsequent controller design and mathematical analysis, we introduce the following auxiliary generalized coordinate variables:

$$\eta_1 = \phi_1 - \alpha, \eta_2 = \phi_2 - \beta, \eta_3 = l, \eta_4 = \theta_1 - \alpha, \eta_5 = \theta_2 - \beta \quad (1)$$

where  $\eta_1$  represents the rotational angle of the boom,  $\eta_2$  represents the luffing angle of the boom,  $\eta_3$  corresponds to the length of the hoisting rope, while  $\eta_4$  and  $\eta_5$  denote the payload's transverse and longitudinal swing angles, respectively.



**Figure 1.** Actual model of Marine crane.

For the convenience of geometric description, the center point of the payload is defined as  $P$ . By treating the boom as a uniform rigid body with an evenly distributed mass and a negligible cross-sectional area, the spatial coordinates of point  $P$  can be derived through standard kinematic transformations as follows:

$$\mathbf{P} = [x_p, y_p, z_p]^T \quad (2)$$

To enhance readability and prevent visual fatigue while maintaining mathematical transparency, we have explicitly introduced a unified trigonometric shorthand rule right before the Lagrangian derivation. We have clarified that:  $C_i = \cos(\eta_i)$ ,  $S_i = \sin(\eta_i)$ ,  $C_{i-j} = \cos(\eta_i - \eta_j)$ ,  $S_{i-j} = \sin(\eta_i - \eta_j)$  where  $i, j \in \{1, 2, 3, 4, 5\}$ . For instance, the notation  $C_{1-4}$  is explicitly defined as  $\cos(\eta_1 - \eta_4)$ . All scalar differential equations (Equations 5–9) have been rigorously checked to conform strictly to this standardized notation rule.

Based on analytical mechanics, the total kinetic energy of the system,  $T$ , is composed of the rotational kinetic energy of the boom and the translational kinetic energy of the payload, which can be formulated as:

$$T = \frac{1}{2}mV_m^2 + \frac{1}{6}ML^2(\dot{\eta}_1^2 + \dot{\eta}_2^2C_1^2) \quad (3)$$

Similarly, by establishing a zero-potential reference plane, the total gravitational potential energy of the system,  $P$ , is modeled as:

$$P = \frac{1}{2}MgLS_1 + mg(LS_1 - \eta_3C_4C_5) \quad (4)$$

The total energy of the system is given by  $E = T + P$ . By defining the Lagrangian of the system and applying Lagrange's theorem, the highly coupled nonlinear dynamic equations of the 5-DOF marine crane can be systematically derived as follows [19,34]:

$$\begin{aligned} & \left(mL^2 + \frac{1}{3}ML^2\right)\ddot{\eta}_1 + mL\eta_3\ddot{\eta}_2S_1S_5 - mL\ddot{\eta}_3C_{1-4}C_5 \\ & - mL\eta_3\ddot{\eta}_4S_{1-4}C_5 + mL\eta_3\ddot{\eta}_5C_{1-4}S_5 + \left(mL^2 + \frac{1}{3}ML^2\right)\ddot{\eta}_2^2S_1C_1 \\ & + mL\eta_3\ddot{\eta}_2^2S_1S_4C_5 + 2mL\ddot{\eta}_2\ddot{\eta}_3S_1S_5 + 2mL\eta_3\ddot{\eta}_2\ddot{\eta}_5S_1C_5 \\ & - 2mL\ddot{\eta}_3\ddot{\eta}_4S_{1-4}C_5 + 2mL\ddot{\eta}_3\ddot{\eta}_5C_{1-4}S_5 + mL\eta_3\ddot{\eta}_4^2C_{1-4}C_5 \\ & + 2mL\eta_3\ddot{\eta}_4\ddot{\eta}_5S_{1-4}S_5 + mL\eta_3\ddot{\eta}_5^2C_{1-4}C_5 + \left(m + \frac{1}{2}M\right)gLC_1 \\ & = f_1 - f_1^* \end{aligned} \quad (5)$$

$$\begin{aligned}
& mL\eta_3\ddot{\eta}_1S_1S_5 + \left(\frac{1}{3}ML^2C_1^2 + mL^2C_1^2 + m\eta_3^2S_4^2C_5^2 + m\eta_3^2S_5^2 + 2mL\eta_3C_1S_4C_5\right)\ddot{\eta}_2 \\
& + mL\ddot{\eta}_3C_1S_5 - m\eta_3^2\ddot{\eta}_4C_4S_5C_5 + \left(m\eta_3^2S_4 + mL\eta_3C_1C_5\right)\ddot{\eta}_5 \\
& - 2\left(mL^2 + \frac{1}{3}ML^2\right)\dot{\eta}_1\dot{\eta}_2S_1C_1 - 2mL\eta_3\dot{\eta}_1\dot{\eta}_2S_1S_4C_5 + mL\eta_3\dot{\eta}_1^2C_1S_5 \\
& + 2m\eta_3^2\dot{\eta}_2\dot{\eta}_4S_4C_4C_5^2 + 2mL\eta_3\dot{\eta}_2\dot{\eta}_4C_1C_4C_5 + 2m\eta_3^2\dot{\eta}_2\dot{\eta}_5C_4^2S_5C_5 \\
& - 2mL\eta_3\dot{\eta}_2\dot{\eta}_5C_1S_4S_5 + 2m\eta_3\dot{\eta}_2\dot{\eta}_3(S_4^2C_5^2 + S_5^2) + 2mL\dot{\eta}_2\dot{\eta}_3C_1S_4C_5 \\
& - 2m\eta_3\dot{\eta}_3\dot{\eta}_4C_4S_5C_5 + 2mL\dot{\eta}_3\dot{\eta}_5C_1C_5 + 2m\eta_3\dot{\eta}_3\dot{\eta}_5S_4 \\
& + m\eta_3^2\dot{\eta}_4^2S_4S_5C_5 + 2m\eta_3^2\dot{\eta}_4\dot{\eta}_5C_4S_5^2 - mL\eta_3\dot{\eta}_5^2C_1S_5 \\
& = f_2 - f_2^*
\end{aligned} \tag{6}$$

$$\begin{aligned}
& - mL\ddot{\eta}_1C_{1-4}C_5 + mL\ddot{\eta}_2C_1S_5 + m\ddot{\eta}_3 + mL\dot{\eta}_1^2S_{1-4}C_5 \\
& - 2mL\dot{\eta}_1\dot{\eta}_2S_1S_5 - m\eta_3\dot{\eta}_2^2(S_4^2C_5^2 + S_5^2) - mL\dot{\eta}_2^2C_1S_4C_5 \\
& + 2m\eta_3\dot{\eta}_2\dot{\eta}_4C_4S_5C_5 - 2m\eta_3\dot{\eta}_2\dot{\eta}_5S_4 - m\eta_3\dot{\eta}_4^2C_5^2 - m\eta_3\dot{\eta}_5\dot{\eta}_5 - mgC_4C_5 \\
& = f_3 + f_3^*
\end{aligned} \tag{7}$$

$$\begin{aligned}
& - mL\eta_3\ddot{\eta}_1S_{1-4}C_5 - m\eta_3^2\ddot{\eta}_2C_4S_5C_5 + m\eta_3^2\ddot{\eta}_4C_5^2 - mL\eta_3\dot{\eta}_1^2C_{1-4}C_5 \\
& - m\eta_3^2\dot{\eta}_2^2S_4C_4C_5^2 - mL\eta_3\dot{\eta}_2^2C_1C_4C_5 - 2m\eta_3^2\dot{\eta}_2\dot{\eta}_5C_4C_5^2 \\
& - 2m\eta_3\dot{\eta}_2\dot{\eta}_3C_4S_5C_5 + 2m\eta_3\dot{\eta}_3\dot{\eta}_4C_5^2 - 2m\eta_3^2\dot{\eta}_4\dot{\eta}_5S_5C_5 + mg\eta_3S_4C_5 \\
& = f_4^*
\end{aligned} \tag{8}$$

$$\begin{aligned}
& mL\eta_3\ddot{\eta}_1C_{1-4}S_5 + \left(m\eta_3^2S_4 + mL\eta_3C_1C_5\right)\ddot{\eta}_2 + m\eta_3^2\ddot{\eta}_5 - mL\eta_3\dot{\eta}_1^2S_{1-4}S_5 \\
& - 2mL\eta_3\dot{\eta}_1\dot{\eta}_2S_1C_5 - m\eta_3^2\dot{\eta}_2^2C_4^2S_5C_5 + mL\eta_3\dot{\eta}_2^2C_1S_4S_5 + 2m\eta_3\dot{\eta}_2\dot{\eta}_3S_4 \\
& + 2m\eta_3^2\dot{\eta}_2\dot{\eta}_4C_4C_5^2 + 2m\eta_3\dot{\eta}_3\dot{\eta}_5 + m\eta_3^2\dot{\eta}_4^2S_5C_5 + mg\eta_3C_4S_5 \\
& = f_5^*
\end{aligned} \tag{9}$$

To synthesize the control law, these differential equations are transformed into a standard, compact matrix form:

$$\mathbf{M}(\boldsymbol{\eta})\ddot{\boldsymbol{\eta}} + \mathbf{V}(\boldsymbol{\eta}, \dot{\boldsymbol{\eta}})\dot{\boldsymbol{\eta}} + \mathbf{G}(\boldsymbol{\eta}) = \mathbf{F}_c + \mathbf{F}^* \tag{10}$$

where  $\mathbf{M}(\boldsymbol{\eta})$  represents the symmetric and positive-definite inertia matrix, and  $\mathbf{V}(\boldsymbol{\eta}, \dot{\boldsymbol{\eta}})$  denotes the Coriolis and centrifugal force matrix, which inherently relies on both the position vector  $\boldsymbol{\eta}$  and the velocity vector  $\dot{\boldsymbol{\eta}}$ .  $\mathbf{G}(\boldsymbol{\eta})$  is the gravitational vector,  $\mathbf{F}_c = [f_1, f_2, f_3, 0, 0]^T$  represents the control input vector acting on the actuated degrees of freedom, and  $\mathbf{F}^* = [f_1^*, f_2^*, f_3^*, d_f l^2 \ddot{\eta}_4, d_f l^2 \ddot{\eta}_5]^T$  accounts for the generalized external disturbances such as wind loads and sea waves.

The established 5-DOF marine crane dynamic system inherently possesses several fundamental mechanical properties. Most notably, for the subsequent energy-based controller design and stability proof, the system satisfies the following critical property [19]:

**Property 1.** The matrix  $\dot{\mathbf{M}}(\boldsymbol{\eta}) - 2\mathbf{V}(\boldsymbol{\eta}, \dot{\boldsymbol{\eta}})$  is a skew-symmetric matrix. That is, for any non-zero vector  $\boldsymbol{\zeta}$  of appropriate dimensions, the following relation strictly holds:

$$\boldsymbol{\zeta}^T [\dot{\mathbf{M}}(\boldsymbol{\eta})/2 - \mathbf{V}(\boldsymbol{\eta}, \dot{\boldsymbol{\eta}})] \boldsymbol{\zeta} = 0, \quad \forall \boldsymbol{\zeta} \in \mathbb{R}^5 \tag{11}$$

This skew-symmetric property reflects the fundamental principle of energy conservation within the internal nonlinear dynamics of the crane system, ensuring that the fictitious Coriolis and centrifugal forces do no actual work. This serves as a vital mathematical foundation for the novel

energy-based coupling controller designed in Section 3, facilitating natural energy dissipation and robust trajectory tracking during precision lifting operations.

### 3. Controller Design and Stability Analysis

To simultaneously achieve the dual control objectives of precise spatial trajectory tracking for the actuated marine crane structures and rapid swing elimination for the unactuated payload, an energy-based intelligent coupling control framework is proposed. The synthesis of this controller integrates novel error coupling variables, adaptive parameter estimation, and fuzzy logic approximation to ensure robust performance under variable payload masses and complex sea wave disturbances. In practical offshore lifting operations, the motion of the actuated variables (boom luffing, base rotation, and rope length) and the unactuated payload swing angles are highly coupled. Standard linear tracking errors are insufficient to regulate this complex underactuated dynamic behavior. First, we define the standard state tracking error variables  $e(t)$  for the actuated degrees of freedom as:

$$e_1(t) = \eta_1 - \eta_{1d}, e_2(t) = \eta_2 - \eta_{2d}, e_3(t) = \eta_3 - \eta_{3d}, e_4(t) = \eta_4, e_5(t) = \eta_5 \quad (12)$$

To organically link the trajectory tracking of the crane structure with the swing suppression of the payload, a novel set of error coupling signals  $\xi$  is introduced. This design explicitly exploits the physical energy transfer mechanisms within the crane system, defined as:

$$\xi_1 = \dot{e}_1 + \lambda_1 \phi(e_4), \xi_2 = \dot{e}_2 + \lambda_2 \phi(e_5), \xi_3 = \dot{e}_3, \xi_4 = \dot{e}_4, \xi_5 = \dot{e}_5 \quad (13)$$

where  $\lambda_1, \lambda_2$  are positive adjustment gains, and  $\phi(e_4), \phi(e_5)$  represent continuous feedback functions related to the unactuated payload swing variables, whose specific linear forms will be detailed in the subsequent energy dissipation design. Taking the time derivative of Equation (13) yields:

$$\begin{aligned} \dot{\xi}_1 &= \ddot{e}_1 + \lambda_1 \frac{\partial \phi(e_4)}{\partial e_4} \xi_4, \dot{\xi}_2 = \ddot{e}_2 + \lambda_2 \frac{\partial \phi(e_5)}{\partial e_5} \xi_5, \\ \dot{\xi}_3 &= \ddot{e}_3, \dot{\xi}_4 = \ddot{e}_4, \dot{\xi}_5 = \ddot{e}_5 \end{aligned} \quad (14)$$

By substituting Equations (12)–(14) into Equation (10), the error dynamic model based on these novel coupling variables is obtained as follows:

$$\mathbf{M}\dot{\xi} + \mathbf{V}\xi = \mathbf{F}_c + \mathbf{F}^* - \mathbf{G} - \mathbf{M}\dot{\Lambda} + \mathbf{V}\Lambda \quad (15)$$

where  $\Lambda = [\lambda_1 \phi(e_4), \lambda_2 \phi(e_5), 0, \dot{e}_4, \dot{e}_5]^T \in \mathbb{R}^5$  represents the artificially constructed coupling variable vector. In this vector, the continuous feedback functions  $\phi(e_4)$  and  $\phi(e_5)$  explicitly inject the unactuated payload swing states into the actuated tracking error dynamics, thereby driving the natural energy dissipation mechanism of the entire closed-loop system.

The construction of these coupling variables ensures that any residual payload oscillation actively generates a corrective feedback signal to the actuated joints, facilitating a natural energy dissipation mechanism that rapidly suppresses the swing angles. First, based on the control objectives and the constructed coupling error signals, a novel energy storage function is formulated to capture the coupled dynamics of the system:

$$E = \frac{1}{2} \xi^T \mathbf{M}(\eta) \xi + mg\eta_3(1 - \cos \eta_4 \cos \eta_5) \quad (16)$$

Taking the time derivative of  $E$  and substituting the system dynamics along with the skew-symmetric property of the inertia matrix yields:

$$\begin{aligned}\dot{E} &= \dot{\boldsymbol{\zeta}}^T (\mathbf{M}\dot{\boldsymbol{\zeta}} + \frac{1}{2}\dot{\mathbf{M}}\dot{\boldsymbol{\zeta}}) + mg\dot{\eta}_3(1 - C_4C_5) + mg\eta_3(\dot{\eta}_4S_4C_5 + \dot{\eta}_5C_4S_5) \\ &= \dot{\boldsymbol{\zeta}}^T (\mathbf{F}_c + \boldsymbol{\Omega}) + mg\dot{\xi}_3(1 - C_4C_5) + mg\eta_3(\dot{\xi}_4S_4C_5 + \dot{\xi}_5C_4S_5) \\ &= \dot{\boldsymbol{\zeta}}^T \mathbf{F}_c + \dot{\boldsymbol{\zeta}}^T \boldsymbol{\Omega} + \dot{\xi}_1G_1 + \dot{\xi}_3mg\end{aligned}\quad (17)$$

where  $\boldsymbol{\Omega} = \mathbf{F}^* - \mathbf{M}\dot{\boldsymbol{\Lambda}} + \mathbf{V}\boldsymbol{\Lambda}$ . To facilitate energy dissipation, linear error feedback is selected for the functions to be designed follows:

$$\phi(e_4) = -e_4, \quad \phi(e_5) = -e_5 \quad (18)$$

In actual maritime environments, the aggregated disturbance vector  $\boldsymbol{\Omega} = \mathbf{F}^* - \mathbf{M}\dot{\boldsymbol{\Lambda}} + \mathbf{V}\boldsymbol{\Lambda}$  is highly nonlinear, time-varying, and inherently unknown. To robustly compensate for this lumped perturbation, a fuzzy logic system is developed.

**Explicit Input Space Dimension:** We have explicitly defined the comprehensive state input vector  $\mathbf{x} = [\eta_1, \dots, \eta_5, \dot{\eta}_1, \dots, \dot{\eta}_5]^T$ , mapping it strictly to its compact operating domain in the tenth-dimensional Euclidean space ( $\mathbf{x} \in \mathbb{R}^{10}$ ).

**Strict Error Boundedness:** The lumped unknown nonlinear function vector  $\mathbf{F}(\mathbf{x})$  is now formulated with an explicit ideal weight matrix  $\mathbf{W}^*$  and a fuzzy basis function vector  $\boldsymbol{\Phi}(\mathbf{x})$  as follows:

$$\mathbf{F}(\mathbf{x}) = \mathbf{W}^{*T} \boldsymbol{\Phi}(\mathbf{x}) + \sigma \quad (19)$$

Crucially, we have supplemented the mathematical boundary condition for the inherent approximation error vector  $\boldsymbol{\sigma}$ , explicitly specifying that  $|\sigma_i| \leq \sigma_i^*$ , which serves as a rigorous prerequisite for the subsequent switching gain  $\bar{\delta} \geq \bar{\sigma}$  selection in our robust control law.

Finally, driven by the constructed energy coupling error  $\xi_i$ , the membership functions are defined as Equation (20), and the adaptive weight update laws are rigorously derived as Equation (21):

$$\zeta_j(\mathbf{x}) = \frac{\prod_{i=1}^8 \mu_i^j(x_i)}{\sum_j \prod_{i=1}^8 \mu_i^j(x_i)} \quad (20)$$

where  $j$  is the number of fuzzy rules, and  $\mu_i^j(x_i)$  is the membership function of the fuzzy set. The adaptive update law for the fuzzy weights is designed as follows:

$$\dot{\hat{\mathbf{W}}}_i = \Gamma_i \boldsymbol{\Phi}_i(\mathbf{x}) \xi_i, \quad i = 1, \dots, 5 \quad (21)$$

Additionally, to address the uncertainty in the payload mass, we isolate the gravitational parameters by defining the auxiliary gravity vector:

$$\mathbf{g}_r(\mathbf{j}) = [G_1, 0, mg, 0, 0]^T \quad (22)$$

$$\theta_1 = \left(m + \frac{1}{2}M\right)gL, \quad \theta_2 = mg \quad (23)$$

Thus, the gravity vector can be linearly parameterized via the regression matrix  $\mathbf{Y}(\mathbf{j})$  and the unknown parameter vector  $\boldsymbol{\theta} = [\theta_1, \theta_2]^T$ :

$$\mathbf{g}_r(\boldsymbol{\eta}) = \mathbf{Y}(\boldsymbol{\eta})\boldsymbol{\theta} \quad (24)$$

where we have mathematically spelled out the exact matrix dimensions of  $\mathbf{Y}(\boldsymbol{\eta}) \in \mathbb{R}^{5 \times 2}$ :

$$\mathbf{Y}(\boldsymbol{\eta}) = \begin{bmatrix} \cos \eta_1 & 0 \\ 0 & 0 \\ 0 & 1 \\ 0 & 0 \\ 0 & 0 \end{bmatrix} \quad (25)$$

The adaptive update law for the unknown mass parameters is designed as:

$$\dot{\hat{\boldsymbol{\theta}}} = -\Gamma_{\theta} \mathbf{Y}^T(\boldsymbol{\eta}) \boldsymbol{\xi} \quad (26)$$

where it is clearly emphasized that  $\Gamma_{\theta}$  is a symmetric, positive-definite design matrix  $\Gamma_{\theta} = \Gamma_{\theta}^T > 0$ . This addition reinforces the mathematical precision of our mass estimation scheme.

Based on the above derivations, the complete energy-based coupling controller is synthesized as:

$$\mathbf{u} = -\mathbf{K}_p \boldsymbol{\xi} - \mathbf{K}_i \int_0^t \boldsymbol{\xi} d\tau - \mathbf{Y}(\boldsymbol{\eta}) \hat{\boldsymbol{\theta}} - \widehat{\mathbf{W}}^T \boldsymbol{\Phi}(\mathbf{x}) - \bar{\delta} \operatorname{sgn}(\boldsymbol{\xi}) \quad (27)$$

where  $\mathbf{K}_p = \operatorname{diag}(k_{p1}, k_{p2}, k_{p3}, 0, 0)$  and  $\mathbf{K}_i = \operatorname{diag}(k_{i1}, k_{i2}, k_{i3}, 0, 0)$  are positive-definite diagonal gain matrices, with their diagonal elements satisfying  $k_{pj} > 0$  and  $k_{ij} > 0$  ( $j = 1, 2, 3$ ). Here,  $\hat{\boldsymbol{\theta}} = [\hat{\theta}_1, \hat{\theta}_2]^T$  is the online estimation of  $\boldsymbol{\theta}$ , and  $\bar{\delta} \geq \bar{\sigma}$  is the robust switching gain vector.

**Remark 1.** Due to the highly coupled nonlinear dynamics and unpredictable wave disturbances, the controller parameters are tuned using a systematic trial-and-error method. Specifically, the observer and adaptive gains  $\Gamma_i, \Gamma_{\theta}$  are first adjusted to ensure stable state estimation without amplifying noise. Subsequently, the feedback and coupling gains  $\mathbf{K}_p, \mathbf{K}_i, \lambda_1, \lambda_2$  are incrementally increased to achieve an optimal balance between precise trajectory tracking, rapid swing suppression, and physical actuator limits.

This section provides a rigorous proof for the stability and state convergence of the closed-loop system. The main theoretical results are summarized in the following theorem.

**Theorem 1.** Under the proposed nonlinear energy-based coupling control laws defined above, the marine crane system can accurately transport the payload to the target position while effectively suppressing and eliminating the payload swing. Mathematically, the tracking errors satisfy:

$$\lim_{t \rightarrow \infty} [e_1(t) \quad \dot{e}_1(t) \quad e_2(t) \quad \dot{e}_2(t) \quad e_3(t) \quad \dot{e}_3(t)]^T \rightarrow [0 \quad 0 \quad 0 \quad 0 \quad 0 \quad 0]^T \quad (28)$$

**Proof.** To prove this, we construct the following Lyapunov candidate function:

$$V = E + \frac{1}{2} \sum_{i=1}^3 k_i \left( \int_0^t \xi_i d\tau \right)^2 + \frac{1}{2} \boldsymbol{\theta}^T \Gamma_{\theta}^{-1} \boldsymbol{\theta} + \frac{1}{2} \sum_{i=1}^5 \widetilde{\mathbf{W}}_i^T \Gamma_i^{-1} \widetilde{\mathbf{W}}_i \quad (29)$$

where  $\boldsymbol{\theta} = \hat{\boldsymbol{\theta}} - \boldsymbol{\theta}^*$  and  $\widetilde{\mathbf{W}}_i = \widehat{\mathbf{W}}_i - \mathbf{W}_i^*$ .

Taking the time derivative of  $V$  and substituting the system dynamics and adaptive laws, we obtain:

$$\begin{aligned}\dot{V} &= \xi^T \mathbf{u} + \xi^T \mathbf{F} + \xi^T \mathbf{g}_r + \sum_{i=1}^3 k_i \xi_i \int_0^t \xi_i d\tau + \theta^T \Gamma_\theta^{-1} \hat{\theta} + \sum_{i=1}^5 \tilde{\mathbf{W}}_i^T \Gamma_i^{-1} \hat{\mathbf{W}}_i \\ &= \xi^T \left( -\mathbf{K}_p \xi - \mathbf{K}_i \mathbf{z} - \mathbf{Y} \hat{\theta} - \hat{\mathbf{W}}^T \Phi - \bar{\delta} \operatorname{sgn}(\xi) \right) + \xi^T \left( \mathbf{W}^{*T} \Phi + \sigma \right) \\ &\quad + \xi^T \mathbf{Y} \theta^* + \xi^T \mathbf{K}_i \mathbf{z} + \theta^T \Gamma_\theta^{-1} \left( -\Gamma_\theta \mathbf{Y}^T \xi \right) + \sum_{i=1}^5 \tilde{\mathbf{W}}_i^T \Phi_i \xi_i \\ &= -\xi^T \mathbf{K}_p \xi - \xi^T \bar{\delta} \operatorname{sgn}(\xi) + \xi^T \sigma \\ &\leq -\xi^T \mathbf{K}_p \xi - \|\xi\| (\bar{\delta} - \bar{\sigma}) \\ &\leq -\xi^T \mathbf{K}_p \xi \leq 0.\end{aligned}\quad (30)$$

Given  $\dot{V} \leq -\xi^T \mathbf{K}_p \xi$  and  $\mathbf{K}_p > 0$ , we have  $\dot{V} = 0$  if and only if  $\xi_1 = \xi_2 = \xi_3 = \xi_4 = \xi_5 = 0$ . Because  $V(t) \geq 0$  and  $\dot{V} \leq 0$ , it follows that  $V(t) \in L_\infty$ . Consequently,  $\xi(t)$ ,  $\int_0^t \xi(\tau) d\tau$ ,  $\theta(t)$ ,  $\mathbf{W}_i(t) \in L_\infty$ . From the definition of the sliding surfaces  $\xi_1 = \dot{e}_1 - \lambda_1 e_4$ ,  $\xi_2 = \dot{e}_2 - \lambda_2 e_5$ , and  $\xi_3 = \dot{e}_3$ , it can be derived that  $\dot{e}_1(t), \dot{e}_2(t), \dot{e}_3(t) \in L_\infty$ . Furthermore, from the boundedness of the error integrals, it can be deduced that  $e_1(t), e_2(t), e_3(t) \in L_\infty$ .

According to LaSalle's invariance principle, consider the set:

$$\Omega = \{(\mathbf{e}, \xi, \hat{\theta}, \hat{\mathbf{W}}) \mid \dot{V} = 0\} \quad (31)$$

Substituting  $\xi_i = 0$  into the closed-loop dynamics, we obtain:

$$e_1 = 0, \quad e_2 = 0, \quad e_3 = 0 \quad (32)$$

Therefore, the largest invariant set contains solely the desired equilibrium point of the system:

$$\lim_{t \rightarrow \infty} e_1(t) = 0, \quad \lim_{t \rightarrow \infty} e_2(t) = 0, \quad \lim_{t \rightarrow \infty} e_3(t) = 0, \quad \lim_{t \rightarrow \infty} e_4(t) = 0, \quad \lim_{t \rightarrow \infty} e_5(t) = 0 \quad (33)$$

□

## 4. Simulation Experiments

In this section, extensive MATLAB2024B simulation experiments are conducted to rigorously verify the correctness, effectiveness, and superiority of the proposed energy-based intelligent coupling controller under complex offshore conditions. To objectively evaluate the positioning accuracy and anti-swing performance, the proposed method is systematically compared with three widely recognized baseline control strategies in recent literature [19]: the PD controller, the LQR controller, and the EE controller.

The physical parameters of the 5-DOF marine crane system are configured based on a scaled laboratory hardware experimental platform detailed in existing literature [35], ensuring the physical validity and practical relevance of the simulation settings. The initial controller gains are configured as follows:

$$\begin{aligned}M &= 4.5 \text{ kg}, m_2 = 0.5 \text{ kg}, g = 9.8 \text{ m/s}^2, L = 1.7 \text{ m} \\ \eta_{1d} &= 20 \text{ deg}, \eta_{2d} = 20 \text{ deg}, \eta_{3d} = 1 \text{ m}, \\ \mathbf{K}_p &= [7, 7, 6], \mathbf{K}_i = [5, 5, 3], \bar{\delta} = [2, 2, 1]\end{aligned}\quad (34)$$

**Remark 2.** It is worth noting that the physical parameters listed in Equation (34) (e.g., a payload mass of 1 kg and a boom length of 1.7 m) are systematically scaled down from full-scale industrial marine cranes. These parameters correspond to a specialized laboratory-level hardware experimental platform detailed in recent benchmark offshore crane control studies [35]. This dynamic scaling practice is essential and widely adopted to safely and rigorously validate the transient performance and stability bounds of advanced nonlinear controllers prior to highly hazardous full-scale maritime deployment.

To emulate a realistic and harsh maritime environment, continuous sea wave disturbances acting on the ship hull are introduced into the dynamic model as  $\alpha = 2 \cos(3t)$ . The comparative evaluation is divided into four distinct operational scenarios: a baseline comparative test, a variable payload mass test, a variable control target test, and a severe external disturbance rejection test.

The expressions of the compared methods are shown in the following equations, respectively:

(1) PD controller: The PD controller is designed as follows

$$\begin{aligned} f_1 &= -k_{p1}e_1 - k_{d1}\dot{e}_1 \\ f_2 &= -k_{p2}e_2 - k_{d2}\dot{e}_2 \\ f_3 &= -k_{p3}e_3 - k_{d3}\dot{e}_3 - mg \end{aligned} \quad (35)$$

(2) LQR controller: The LQR controller is designed as follows

$$\begin{aligned} f_1 &= -k_{11}e_1 - k_{12}\dot{e}_1 - k_{13}\eta_4 - k_{14}\dot{\eta}_4 \\ f_2 &= -k_{21}e_2 - k_{22}\dot{e}_2 - k_{23}\eta_5 - k_{24}\dot{\eta}_5 \\ f_3 &= -k_{31}e_3 - k_{32}\dot{e}_3 - mg \end{aligned} \quad (36)$$

(3) EE controller: The EE controller is designed as follows

$$\begin{aligned} f_1 &= -p_{11}(e_1 - k_a \sin(\eta_3)) - p_{12}(\dot{e}_1 - k_a \dot{\eta}_4 \cos(\eta_3)) \\ f_2 &= -p_{21}(e_2 - k_b \sin(\eta_5)) - p_{22}(\dot{e}_2 - k_b \dot{\eta}_5 \cos(\eta_5)) \\ f_3 &= -p_{31}e_3 - p_{32}\dot{e}_3 - mg \end{aligned} \quad (37)$$

To ensure a fair and objective comparison, the control gains of the baseline controllers (PD, LQR, and EE) were rigorously tuned using the same trial-and-error method applied to the proposed controller, guaranteeing that each method achieved its respective optimal transient and steady-state performance under identical physical constraints.

The continuous-time crane dynamics and the discrete-time digital controller are simulated using the *ode3 (Bogacki-Shampine)* fixed-step solver in MATLAB/Simulink, with a fundamental sampling step of 0.001 s. To ensure complete transparency and enable readers to easily reproduce the simulation experiments, the digital execution flow of the proposed energy-based intelligent coupling control strategy is summarized in Tables 2 and 3.

**Table 2.** Execution Flow of the Proposed Controller.

| Execution Phase | Step-by-Step Logic and Equations                                                                                                                                                                                                                                 |
|-----------------|------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| Initialization  | Set initial crane states $\eta(0), \dot{\eta}(0)$ ;<br>Initialize fuzzy weights $\hat{\mathbf{W}}(0) \leftarrow \mathbf{0}$ ;<br>Initialize adaptive mass estimation $\hat{\theta}(0) \leftarrow \mathbf{0}$ ;<br>Load controller parameters ( $T_s = 0.001$ s). |

Table 2. Cont.

| Execution Phase      | Step-by-Step Logic and Equations                                                                                                                                                                             |
|----------------------|--------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| <b>Loop Start</b>    | <b>for</b> each sampling step $k = 1, 2, 3, \dots$ <b>do</b>                                                                                                                                                 |
| 1. Measurement       | Measure actual spatial states $\eta_1$ to $\eta_5$ , and derivatives $\dot{\eta}_1$ to $\dot{\eta}_5$ .                                                                                                      |
| 2. Error Calculation | Calculate standard tracking errors: $e_i = \eta_i - \eta_{id}$ .                                                                                                                                             |
| 3. Coupling Signals  | Construct coupling errors based on Equation (13):<br>$\zeta_1 = \dot{e}_1 - \lambda_1 e_4$ , $\zeta_2 = \dot{e}_2 - \lambda_2 e_5$ , $\zeta_3 = \dot{e}_3$ , $\zeta_4 = \dot{e}_4$ , $\zeta_5 = \dot{e}_5$ . |
| 4. Fuzzy Basis       | Compute fuzzy basis functions $\Phi(\mathbf{x})$ using Equation (20).                                                                                                                                        |
| 5. Observer Update   | Update fuzzy observer weights via Equation (21).                                                                                                                                                             |
| 6. Gravity Matrix    | Calculate gravity regression matrix $\mathbf{Y}(\boldsymbol{\eta})$ using Equation (25).                                                                                                                     |
| 7. Mass Estimation   | Update adaptive mass estimation via Equation (26).                                                                                                                                                           |
| 8. Integral Action   | Compute integral of coupling error: $\mathbf{z} \leftarrow \mathbf{z} + \boldsymbol{\zeta} \cdot T_s$ .                                                                                                      |
| 9. Control Input     | Synthesize final control input vector $\mathbf{u}$ using Equation (27).                                                                                                                                      |
| 10. Execution        | Apply inputs ( $u_1, u_2, u_3$ ) to the actuated crane physical joints.                                                                                                                                      |
| <b>Loop End</b>      | <b>end for</b>                                                                                                                                                                                               |

Table 3. Controller Parameters Used in Simulation Experiments.

| Controller      | Control Parameters                                                                                                                         |
|-----------------|--------------------------------------------------------------------------------------------------------------------------------------------|
| <b>Proposed</b> | $\mathbf{K}_p = \text{diag}(7, 7, 6)$ , $\mathbf{K}_i = \text{diag}(5, 5, 3)$ , $\bar{\boldsymbol{\delta}} = [2, 2, 1]^T$                  |
| <b>PD</b>       | $k_{p1} = 4, k_{d1} = 7$ ; $k_{p2} = 4, k_{d2} = 8$ ; $k_{p3} = 4, k_{d3} = 7$                                                             |
| <b>LQR</b>      | $k_{11} = 4.3, k_{12} = 7, k_{13} = 1, k_{14} = 2$ ;<br>$k_{21} = 4.2, k_{22} = 6, k_{23} = 2, k_{24} = 1$ ;<br>$k_{31} = 4.3, k_{32} = 7$ |
| <b>EE</b>       | $p_{11} = 4.5, p_{12} = 7, k_a = 1$ ;<br>$p_{21} = 4.5, p_{22} = 8, k_b = 1$ ;<br>$p_{31} = 4.5, p_{32} = 7$                               |

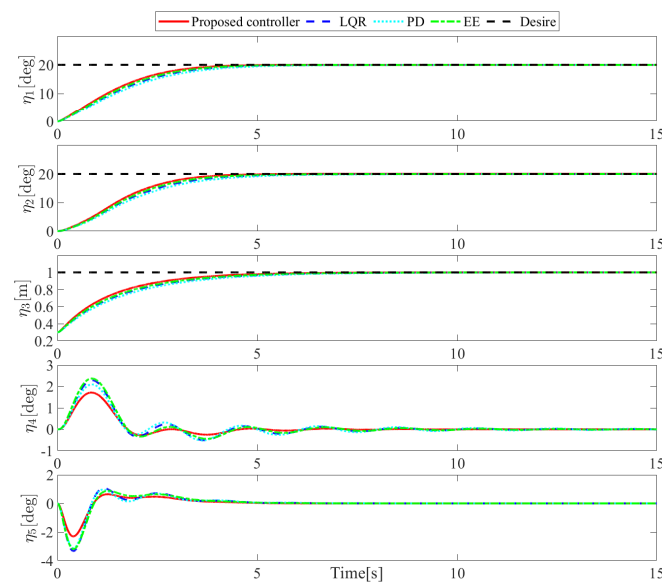
#### 4.1. Case 1: Baseline Comparative Test

In the first set of baseline experiments, the initial payload mass is set to  $m = 0.5$  kg and the control target is predefined as  $\eta_{1d} = 20$  deg,  $\eta_{2d} = 20$  deg,  $\eta_{3d} = 1$  m. The transient tracking waveforms of the five state variables are illustrated in Figure 2, while Figure 3 depicts the corresponding control input variations. As observed from the state response curves, the proposed controller drives the actuated crane structures (boom luffing, base rotation, and rope hoisting) to their target positions significantly faster and smoother than the PD, LQR, and EE methods. Concurrently, the unactuated payload swing angles are rapidly eliminated without exhibiting excessive overshoot, demonstrating the superior coordination of the proposed energy-coupling scheme.

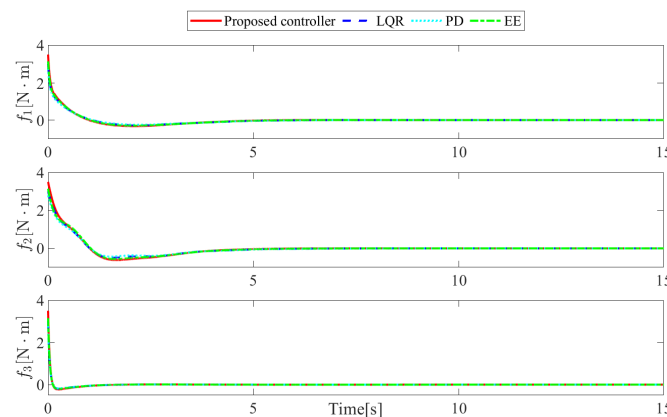
Table 4 demonstrates that during the baseline comparative test, the proposed controller achieves the smallest maximum swing angles among all tested methods, ensuring superior payload stability and operation safety under external wave disturbances.

Table 4. Comparison of Maximum Swing Angles in Case 1 (Baseline Comparative Test).

| Controller      | Max Transverse Swing $\eta_4$ [deg] | Max Longitudinal Swing $\eta_5$ [deg] |
|-----------------|-------------------------------------|---------------------------------------|
| <b>Proposed</b> | <b>1.71</b>                         | <b>2.30</b>                           |
| <b>PD</b>       | 2.09                                | 3.10                                  |
| <b>LQR</b>      | 2.28                                | 3.20                                  |
| <b>EE</b>       | 2.37                                | 3.40                                  |



**Figure 2.** The dynamic tracking waveforms of the five state variables are vividly illustrated in Figure 2, explicitly demonstrating the transient convergence superiority of the proposed framework.



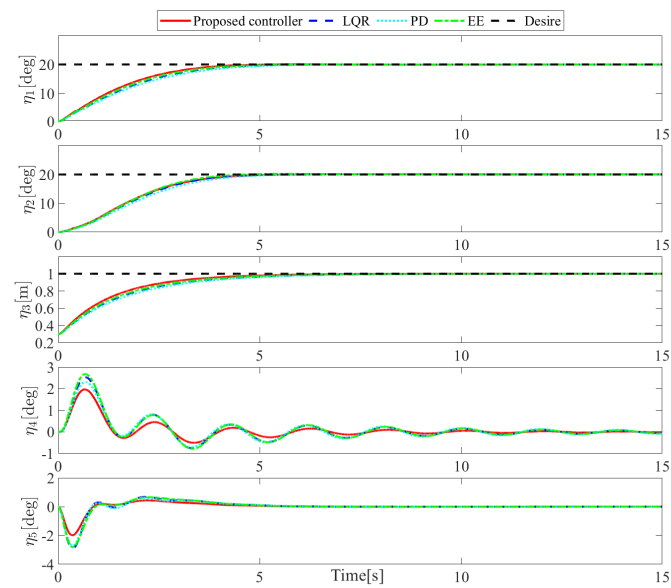
**Figure 3.** The corresponding control input variations.

#### 4.2. Case 2: Variable Payload Mass Test

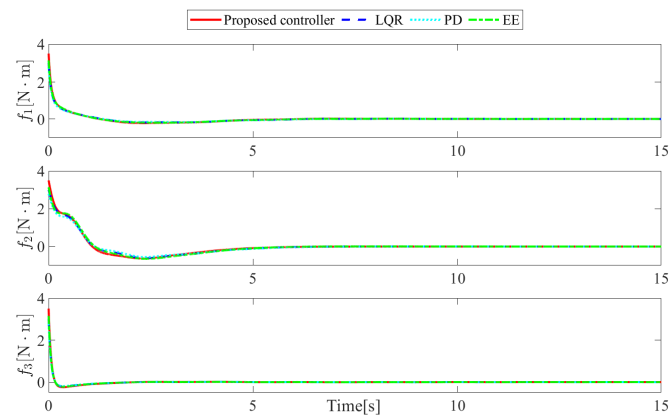
The second set of experiments evaluates the robustness of the controllers against payload mass variations. The transient tracking waveforms of the five state variables are illustrated in Figure 4, while Figure 5 depicts the corresponding control input variations. Here, the payload mass is abruptly changed to  $0.5 \text{ kg} \rightarrow 1 \text{ kg}$ . Despite this significant parameter mismatch, the proposed adaptive gravity compensation mechanism dynamically updates the mass estimation, ensuring that the controller maintains high-precision trajectory tracking and rapid swing suppression. In contrast, the linear LQR and conventional PD controllers suffer from noticeable steady-state errors and prolonged residual oscillations.

#### 4.3. Case 3: Variable Control Target Test

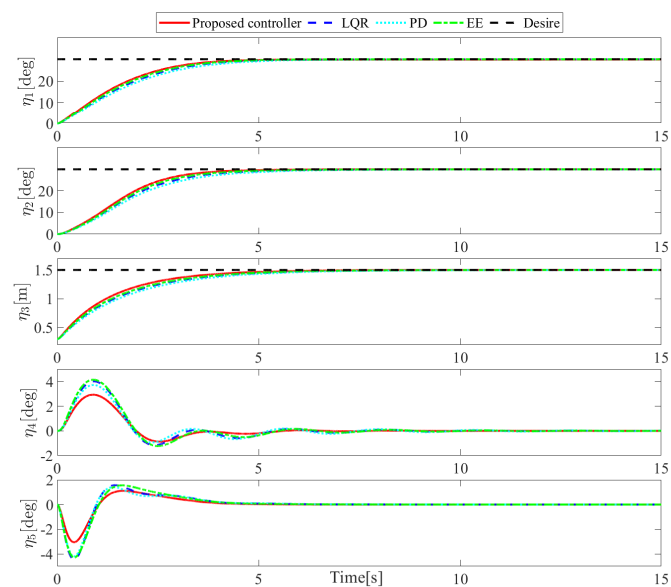
The third set of experiments further tests the system's adaptability by shifting the initial control target to  $\eta_{1d} = 20 \text{ deg}, \eta_{2d} = 20 \text{ deg}, \eta_{3d} = 1 \text{ m} \rightarrow \eta_{1d} = 30 \text{ deg}, \eta_{2d} = 30 \text{ deg}, \eta_{3d} = 1.5 \text{ m}$ . The transient tracking waveforms of the five state variables are illustrated in Figure 6, while Figure 7 depicts the corresponding control input variations. The resulting state responses and control inputs demonstrate that the proposed nonlinear coupling controller seamlessly adapts to the new trajectory demands, maintaining stringent error convergence and optimal transient performance, whereas the alternative methods exhibit sluggish response times.



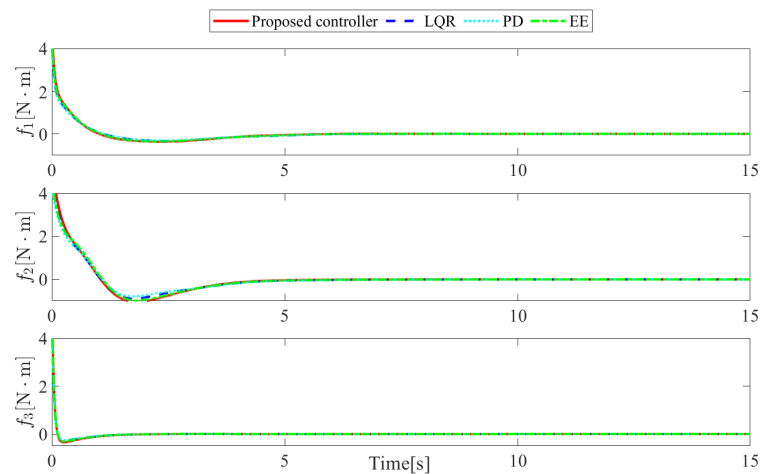
**Figure 4.** The transient tracking waveforms of the five state variables.



**Figure 5.** The corresponding control input variations.



**Figure 6.** The transient tracking waveforms of the five state variables.

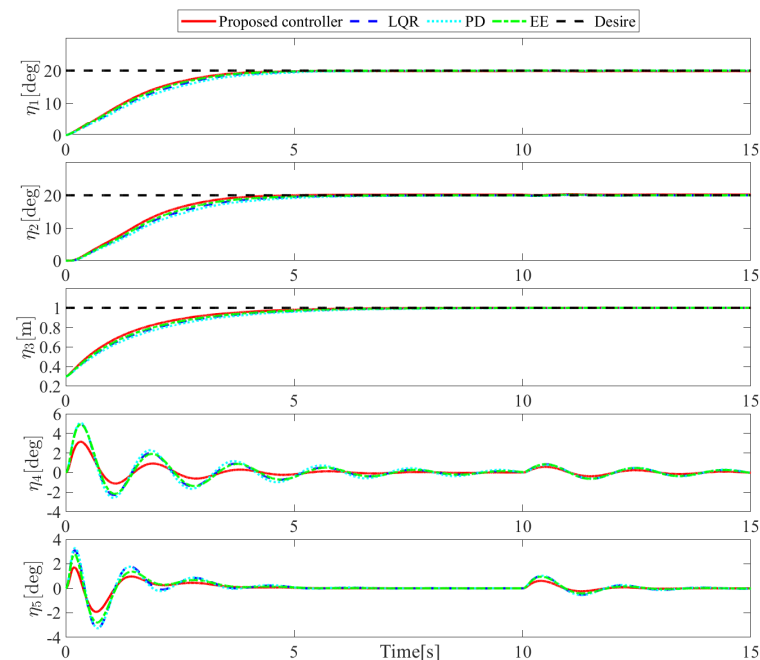


**Figure 7.** The corresponding control input variations.

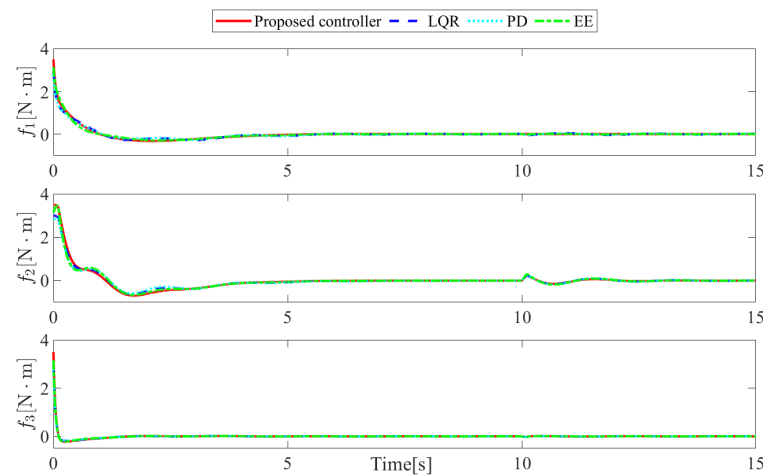
#### 4.4. Case 4: External Disturbance Rejection Test

Finally, to validate the robust anti-interference capabilities of the closed-loop system, the fourth set of experiments introduces sudden and severe external perturbations at  $t = 0$  and again at  $t = 10$  s, after the initial payload swing has settled. The transient tracking waveforms of the five state variables are illustrated in Figure 8, while Figure 9 depicts the corresponding control input variations. The dynamic waveforms reveal that the embedded fuzzy observer instantaneously approximates and actively compensates for these unknown lumped disturbances.

Collectively, these four groups of comparative experiments fully demonstrate that the proposed control framework significantly improves the operational accuracy, anti-swing capability, and overall structural safety of marine cranes operating under complex sea conditions.



**Figure 8.** The transient tracking waveforms of the five state variables.



**Figure 9.** The corresponding control input variations.

## 5. Conclusions

This paper proposed a novel energy-based intelligent coupling control framework for 5-DOF marine cranes to address the severe challenges of highly coupled 3D spatial dynamics, uncertain payload masses, and complex external sea wave disturbances. By innovatively constructing a set of error coupling variables and integrating them with an adaptive gravity compensation mechanism and a dedicated fuzzy observer, the proposed framework explicitly resolved the difficulties of underactuated state regulation and eliminated the traditional reliance on exact prior mathematical models. Extensive simulation experiments rigorously validated that the designed controller achieves high-precision trajectory tracking and rapid payload anti-swing, while maintaining robust stability and superior control performance under variable load conditions and persistent external perturbations.

Despite the demonstrated performance advantages, this study primarily relies on a rigid-body dynamic model and currently lacks full multivariable state constraints to guarantee strict physical motion limits and collision avoidance in complex environments. Future work will focus on incorporating flexible-body dynamics to account for boom and rope elasticity, as well as integrating strict state constraints into the control framework. These advancements will aim to ensure safe, reliable, and efficient deployment on full-scale maritime vessels, particularly for the precise aerial lifting and synchronized installation of offshore wind turbine blades under harsh sea conditions.

**Author Contributions:** Conceptualization, W.P.; formal analysis, T.L.; methodology, W.P.; software, J.Z.; validation, H.Z.; visualization, L.T.; writing-original draft, T.L.; writing-review and editing T.L.; supervision, W.P.; project administration, T.L. All authors have read and agreed to the published version of the manuscript.

**Funding:** This research received no external funding.

**Institutional Review Board Statement:** Not applicable.

**Informed Consent Statement:** Not applicable.

**Data Availability Statement:** The raw data supporting the conclusions of this article will be made available by the authors on request.

**Conflicts of Interest:** Authors Tao Liang, Hui Zhang, Jixiang Zhao were employed by Shandong Electric Power Engineering Consulting Institute Corporation Limited. The remaining authors declare that the research was conducted in the absence of any commercial or financial relationships that could be construed as a potential conflict of interest.

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