

Article

Prescribed Performance Control for Electro-Hydrostatic Erecting System Based on Dual-RISE Scheme

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Abstract

Unmodeled uncertainties, such as friction and stage-change collision of the hydraulic cylinder, along with system disturbances, exist in the multi-link erecting system and impede high-precision erecting angle tracking. To tackle these challenges, this study develops a novel control framework characterized by asymptotic prescribed performance based on a distributed dual robust integral of the sign of the error (Dual-RISE) for the electro-hydrostatic multi-link erecting system. First, a precise system model is established by integrating complex multi-link kinematics with the pressure-flow dynamics of the two-stage hydraulic cylinder. A prescribed performance function (PPF) and nonlinear error transformation are then introduced to strictly constrain the tracking error within predefined transient and steady-state boundaries. The proposed framework integrates a distributed dual-loop RISE architecture to simultaneously reject matched and unmatched uncertainties, mathematically enforcing semi-global asymptotic convergence of the tracking error to zero without requiring infinite high-gain feedback. Comparative experiments with Dual-RISE and VFPI controllers demonstrate superior tracking accuracy and boundary protection under different erecting conditions.

Keywords: electro-hydrostatic erecting system; Dual-RISE control; prescribed performance; high-precision tracking

1. Introduction

Characterized by high power density, substantial output force, and superior dynamic performance, the electro-hydrostatic actuator (EHA) serves as a representative pump-controlled hydraulic servo mechanism. Consequently, it is widely utilized in heavy-duty drive scenarios [1–5], particularly in the load-lifting mechanisms of heavy construction machinery. In addition, owing to the prominent advantage of offering an ultra-long driving stroke within a compact initial structural size [6], multi-stage cylinders play a vital and irreplaceable role in the special vehicle erecting system under investigation. Nevertheless, the high-precision control of the electro-hydrostatic erecting system is quite challenging due to not only strong nonlinearities, such as mechanism nonlinearity (i.e., the nonlinear geometric transformation between cylinder displacement and erecting rotation angle), friction nonlinearity, and pressure-flow nonlinearity, but also unmodeled uncertainties (i.e., unmodeled internal leakage of the hydraulic system, nonlinear friction and external disturbances) [7–10]. Hence, classical linear control strategies often encounter

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significant bottlenecks in attaining the desired servo control accuracy of pump-controlled hydraulic systems. To cope with these issues and enhance the control performance, it is essential to explore some advanced control methodologies for pump-controlled hydraulic systems.

For decades, a variety of sophisticated control methods have been designed to handle the inherent uncertain factors and nonlinear features of electro-hydraulic actuators. As demonstrated in [11,12], dynamic nonlinearities can be effectively mitigated through the application of feedback linearization control, which incorporates a feed-forward compensation strategy. However, feedback linearization suffers from poor robustness against unmodeled dynamics and external disturbances due to its reliance on precise model cancellation. With the aim of enhancing robustness against lumped uncertainties and achieving superior control accuracy, backstepping control, especially when augmented with adaptive strategies [9,13–15], offers a robust approach to performance enhancement by effectively leveraging available model-based information. In [16], an active disturbance rejection adaptive control framework was developed for hydraulic lifting systems subject to valve dead-zones and uncertainties, which integrates backstepping-based parameter adaptation laws with extended state disturbance observers to compensate for parametric uncertainties and unmodeled disturbances, thereby enhancing the tracking accuracy and avoiding high-gain feedback issues. In [17], an enhanced adaptive control architecture incorporating integral sliding mode and backstepping techniques based on incomplete differentiation was adopted to mitigate the tracking accuracy degradation caused by unmodeled uncertainties in hydraulic servo systems. By synthesizing the RISE-based feedback technique with the backstepping design methodology, a RISE-based backstepping controller was formulated in [18] to realize asymptotic tracking accuracy despite the impact of external disturbances. In addition, RISE-based feedback control schemes were devised in [19] to enhance tracking performance, manage mismatched uncertainties [20], and compensate for lumped disturbances through the integration of neural networks [21]. These aforementioned nonlinear control strategies provide a robust theoretical basis for the subsequent control architecture of the EHA system.

In recent years, the security, reliability, and robust monitoring of complex engineering systems have attracted intensive research interest. Specifically, addressing anomaly detection, attack identification, and resilient state estimation under uncertain environments and network constraints has become critical. Chen et al. investigated watermarking-based attack detection for sensor networks with intermittent observations under stealthy attacks, highlighting the necessity of active detection schemes against sophisticated deception strategies in [22,23]. Furthermore, to handle prolonged output anomalies and disturbances in perturbed nonlinear systems, advanced observer and cascade prediction architectures have been developed to guarantee fast anomaly response and joint state estimation accuracy. Meanwhile, it is widely recognized that control schemes must guarantee stringent performance indices, such as transient convergence velocity, overshoot limitations, and final steady-state accuracy. Nevertheless, achieving prescribed performance for electro-hydraulic systems remains a formidable challenge, particularly when confronted with the combined effects of uncertainties and disturbances. Researchers in [24] developed an adaptive prescribed performance control (PPC) strategy that effectively compensates for lumped disturbances to ensure desired convergence rates. To address the challenges of output feedback subjected to both structural and non-structural uncertainties, a barrier Lyapunov function-based adaptive PPC scheme was proposed in [25], guaranteeing that all system states remain within pre-defined boundaries. In [26], to handle complex nonlinearities and large-scale uncertainties, funnel function-based control strategy was investigated to provide more flexible performance constraints for hydraulic actuators. The authors of [27,28] synthesized neural-network-driven adaptive mechanisms

with predefined transient and steady-state performance specifications to handle the severe nonlinearities and complex coupling dynamics of hydraulic manipulators, thereby guaranteeing both stringent transient response and high-precision steady-state tracking performance. However, compared to the aforementioned general hydraulic control strategies, the controller design for the multi-link erecting system considered in this paper is significantly more challenging due to the complex interaction between multi-link kinematics and the non-smooth dynamics of the two-stage hydraulic cylinder. Consequently, how to effectively achieve both stringent transient performance and high-precision asymptotic tracking for such a highly nonlinear erection mechanism remains an open issue that warrants further investigation.

Motivated by the aforementioned challenges, this study proposes an asymptotic prescribed performance control framework that incorporates Dual-RISE technique for the electro-hydrostatic multi-link erecting system. This approach aims to achieve both superior transient response characteristics and high-precision asymptotic tracking for the erecting angle. The pivotal contributions and technical merits of this research are delineated as follows:

- (1) An advanced prescribed performance control scheme, incorporated with Dual-RISE technique, is proposed to regulate the multi-link erecting system. It is formally demonstrated that all closed-loop variables remain finite, and the tracking error converges asymptotically to the origin.
- (2) By integrating complex multi-link kinematics with the pressure-flow coupling characteristics of the two-stage hydraulic cylinder, a precise mathematical model is established. This model captures the non-smooth dynamics and highly nonlinear coupling inherent in the erecting mechanism, providing a robust foundation for high-precision control design.
- (3) Unlike existing controllers that often require precise modeling of nonlinearities or struggle with non-smooth collisions, the proposed Dual-RISE control framework effectively compensates for lumped disturbances, such as friction and stage-change impact forces. By mitigating the necessity for aggressive high-gain feedback, this methodology substantially bolsters the stability and disturbance-rejection capabilities of the erecting process.

To clearly delineate the theoretical and practical novelty of the developed framework from existing prescribed performance control (PPC), standard RISE, adaptive robust control (ARC), and disturbance-observer-based (DOB) electro-hydraulic control methodologies, the substantive differences are distinguished across three dimensions:

- (1) Unlike standard PPC or funnel control which feed transformed error signals into conventional backstepping or sliding-mode loops, the proposed framework integrates a nonlinear error transformation layer with a distributed dual-loop RISE configuration. While standard RISE approaches deploy a single integral signum block to handle matched uncertainties, this architecture splits and distributes separate robust integral signum components across both virtual and actual control channels to simultaneously compensate for unmatched and matched uncertainties.
- (2) Traditional DOB frameworks and conventional ARC strategies typically achieve only practical stability (uniform ultimate boundedness) when exposed to high-frequency, non-smooth structural events like multi-stage cylinder transitions. ARC schemes handle parametric uncertainties well but are vulnerable to parameter drift under non-smooth structural impacts, while DOB methods introduce phase lag that degrades dynamic accuracy during abrupt cylinder area mutations. The proposed PPC-Dual-RISE strategy mathematically guarantees semi-global asymptotic tracking

precision under non-smooth disturbances without relying on online parameter estimation loops, high-gain feedback amplification, or observer dynamics.

- (3) While existing methods frequently rely on empirical trial-and-error parameter tuning, this work establishes a systematic, hierarchical tuning procedure based on the bandwidth separation principle. This calibration method isolates the outer-loop error convergence from inner-loop motor and pump frequency response limitations, physically preventing control voltage saturation and mechanical resonance during severe stage-change cylinder impacts.

The structure of the remainder of this paper is outlined as follows: Section 2 covers the mathematical modeling of the system, while the synthesis of the Dual-RISE-based prescribed performance controller is detailed in Section 3. Subsequent to this, Section 4 offers experimental results and performance evaluations, with final conclusions drawn in Section 5.

2. Problem Description and System Modeling

2.1. Dynamic Modeling of the Multi-Link Erecting System

The structure of the multi-link erecting system is illustrated in Figure 1. The launch canister features a symmetrical configuration, with two-stage hydraulic cylinders and associated linkage mechanisms positioned on either side. Load erection and retraction are accomplished through the synchronized action of the dual cylinders. Hinge point A is the common hinge point for the top-mounted pivot of the two-stage hydraulic cylinder, link AB , and link AD , which is suspended and not connected to the launch canister or the chassis. Hinge point B is the center of rotation for link AB , fixed to the chassis base and has no connection with the launch canister. Hinge point C is the erecting rotation center of the launch canister, where the canister is hinged to another base on the chassis. Hinge point D is fixed on the launch canister and serves as the hinge point between link AD and the canister. Hinge point E is fixed to the chassis and serves as the hinge point between the bottom-mounted pivot of the hydraulic cylinder and the chassis.

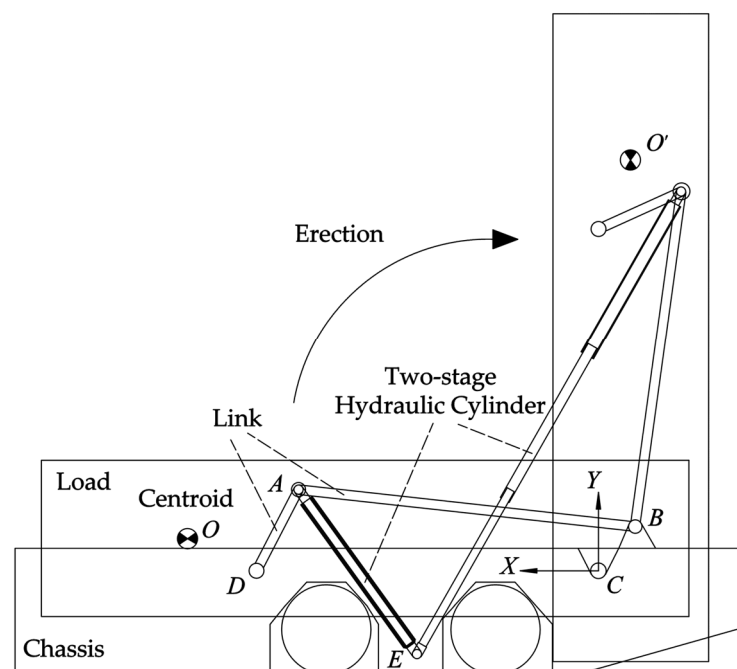


Figure 1. Multi-link erecting system.

During the erecting process, the two-stage hydraulic cylinders on both sides are extended synchronously. Under the kinematic relationship of linkage mechanism ABE , link AB is driven to rotate around hinge point B . Simultaneously with the extension of the hydraulic cylinders, the launch canister is driven to erect by linkage mechanism $ABCD$ through link AC and hinge point C , which is actuated by the rotation of link AB about hinge point B . Consequently, rotation of the canister about hinge point D is induced until the predetermined launch angle is achieved.

The layout of the multi-link erecting mechanism is depicted in Figure 2. A fixed coordinate system CXY is established in the motion plane, with the load rotation point defined as the origin. The positive X -axis is oriented from the rear of the vehicle to the front, while the positive Y -axis is directed perpendicular to the ground pointing upward. Let H be the extension point of $\overline{CB}$. An auxiliary line $\overline{DI} \parallel \overline{CH}$ is drawn. The lengths are defined as $|\overline{CD}| = L_1$, $|\overline{AD}| = L_2$, $|\overline{AB}| = L_3$, $|\overline{BC}| = L_4$, $|\overline{BE}| = L_5$, $|\overline{AE}| = L_6$, $|\overline{CE}| = L_7$, $|\overline{CO}| = L_8$. The angles are defined as $\angle CBE = \alpha$, $\angle OCX = \beta_0$, $\angle BCD = \varphi_1$, $\angle ADI = \varphi_2$, $\angle ABH = \varphi_3$. G is the load gravity, and F is the thrust of single two-stage hydraulic cylinder.

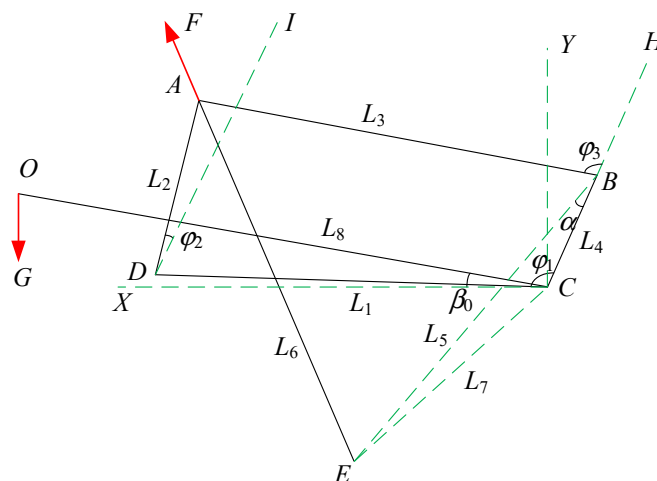


Figure 2. Layout of the multi-link erecting mechanism.

By utilizing the complex vector method to solve the planar linkage mechanism and performing Euler expansion, the vector equation of the mechanism is derived as follows:

$$L_1 (\cos \varphi_1 + i \sin \varphi_1) + L_2 (\cos \varphi_2 + i \sin \varphi_2) = L_3 (\cos \varphi_3 + i \sin \varphi_3) + L_4 \quad (1)$$

By equating the real and imaginary parts of Equation (1) respectively and performing variable elimination, we obtain:

$$a \cos \varphi_3 + b \sin \varphi_3 + c = 0 \quad (2)$$

where $a = L_4 - L_1 \cos(\varphi_1 - \theta)$, with θ being the load erecting angle. $b = -L_1 \sin(\varphi_1 - \theta)$, $c = 0.5(a^2 + b^2 - L_2^2 + L_3^2) / L_3$. The angular displacements of link AB and link AD are solved as:

$$\varphi_3 = 2 \arctan \frac{b + \sqrt{a^2 + b^2 - c^2}}{a - c} \quad (3)$$

$$\varphi_2 = \arctan \frac{b + L_3 \sin \varphi_3}{a + L_3 \cos \varphi_3} \quad (4)$$

Based on the Law of Cosines, the time-varying extension of the two-stage hydraulic cylinder can be formulated as:

$$x_p(t) = \sqrt{L_3^2 + L_5^2 - 2L_3L_5 \cos(\pi - \alpha - \varphi_3)} - L_6 \quad (5)$$

Based on the Law of Sines, the moment arm at any time can be given as:

$$\tau(\theta) = L_7 \sin(\angle AEB + \angle BEC) \quad (6)$$

$$\text{where } \angle AEB = \arccos \left[\frac{(L_5^2 + L_6^2 - L_3^2)}{(2L_5L_6)} \right], \quad \angle BEC = \arccos \left[\frac{(L_5^2 + L_7^2 - L_4^2)}{(2L_5L_7)} \right].$$

The dynamic model of the multi-link erecting system is formulated as follows:

$$J\ddot{\theta} = 2\tau(\theta)F - mgL_8 \cos(\beta_0 + \theta) - T_f + d(t) \quad (7)$$

where J denotes the moment of inertia of the load, m is the mass of the load and g is the gravitational acceleration. $T_f = B\dot{\theta} + A_f S_f(\dot{\theta})$ represents the total friction torque, where B is the viscous friction coefficient and $A_f S_f(\dot{\theta})$ denotes the Coulomb friction torque with amplitude A_f and a continuous approximation shape function $S_f(\dot{\theta})$. $d(t)$ accounts for unmodeled disturbances, such as neglected friction components and stage-change collision of the two-stage hydraulic cylinders.

Remark 1. The dynamic model formulated in Equation (7) utilizes a single equivalent thrust F multiplied by a factor of 2, which implicitly assumes ideal synchronization between the dual hydraulic cylinders. The validity range of this symmetric modeling simplification is bounded by two physical conditions: (i) the structural rigidity of the heavy-duty launch canister, which mechanically ties the dual kinematic chains and restricts large independent displacement deviations, and (ii) the bounded nature of the hydraulic and mechanical asymmetries. In practical operation, minor parameter mismatches, asymmetric payload distributions, and localized leakage variations inevitably introduce cross-coupled synchronization errors and secondary torsional loads. Within our state-space architecture, these asymmetric dynamic disturbances are explicitly mapped into the lumped unmatched disturbance torque $d(t)$ and the dual-cavity fluid modeling errors $q_1(t)$ and $q_2(t)$. Consequently, the burden of handling physical synchronization imbalances is shifted to the robust feedback loops, avoiding over-parameterization of the nominal model.

2.2. Modeling of the Electro-Hydrostatic Actuation System

The layout of the electro-hydrostatic actuation system designed for the erecting system is depicted in Figure 3. During the erecting process, the permanent magnet synchronous motor (PMSM) serves as the prime mover to run a bidirectional closed fixed-displacement piston pump, thereby supplying pressurized fluid into the piston cavity of the two-stage hydraulic cylinder. Consequently, the actuator extends stage by stage in alignment with the targeted command velocity.

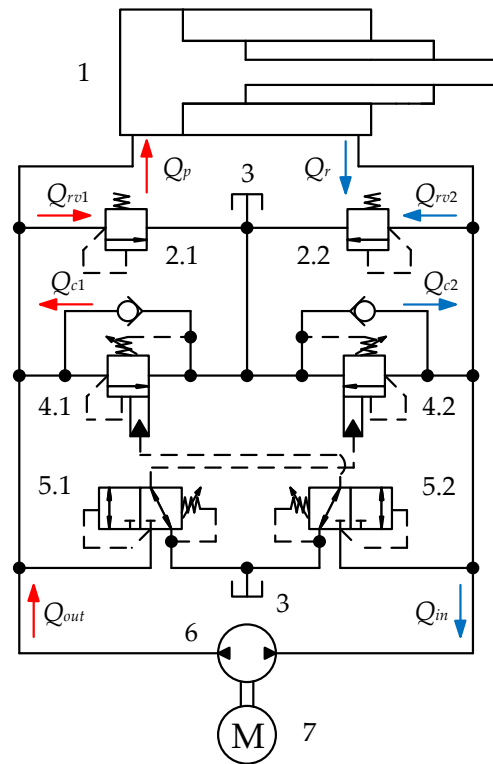


Figure 3. Layout of the electro-hydrostatic actuation system. 1—Two-stage hydraulic cylinder; 2—Relief valve; 3—Reservoir; 4—Counterbalance valve; 5—Pilot-operated directional valve; 6—Bidirectional piston pump; 7—PMSM.

According to the expansion and contraction principles of the inverted two-stage hydraulic cylinder, the effective acting areas of the two cavities are given by:

$$A_p = \begin{cases} A_{1p}, & \text{if } 0 < x_p \leq s_{1M} \\ A_{2p}, & \text{if } s_{1M} < x_p \leq s_{1M} + s_{2M} \end{cases} \quad (8)$$

$$A_r = \begin{cases} A_{1r}, & \text{if } 0 < x_p \leq s_{1M} \\ A_{2r}, & \text{if } s_{1M} < x_p \leq s_{1M} + s_{2M} \end{cases} \quad (9)$$

where A_p and A_r denote the acting areas of the piston-side and rod-side cavities, respectively. A_{1p} and A_{2p} denote the piston cavity areas of the first stage and second stage, while A_{1r} and A_{2r} are the corresponding rod cavity areas. s_{1M} and s_{2M} are the maximum strokes of the first stage and second stage, respectively.

The flow rate relationships connecting the bidirectional piston pump and two cavities of the hydraulic actuator are formulated as:

$$\begin{cases} Q_p = Q_{out} + Q_{c1} - Q_{rv1} \\ Q_r = Q_{in} - Q_{c2} + Q_{rv2} \end{cases} \quad (10)$$

where Q_p and Q_r denote the inlet and outlet flow rates of the two-stage hydraulic cylinder, respectively. Q_{out} and Q_{in} are the discharge and suction flow rates of the bidirectional piston pump, satisfying $Q_{out} = Q_{in} = Q_{ap}$, in which Q_{ap} is the working flow rate of the pump. Q_{c1} and Q_{c2} represent the flow rates passing through the counterbalance

valves. Q_{rv1} and Q_{rv2} are the flow rates through the relief valves, which remain normally closed during standard operation (i.e., $Q_{rv1} = Q_{rv2} = 0$).

Defining the extension of the two-stage hydraulic cylinder as positive motion (i.e., the extension velocity satisfies $\dot{x}_p > 0$), the flow rates Q_{c1} and Q_{c2} can be represented as:

$$\begin{cases} Q_{c1}(\dot{x}_p) = \dot{x}_p (A_p - A_r) \operatorname{sgn}(-\dot{x}_p) \\ Q_{c2}(\dot{x}_p) = \dot{x}_p (A_p - A_r) \operatorname{sgn}(\dot{x}_p) \end{cases} \quad (11)$$

where the signum function $\operatorname{sgn}(\bullet)$ is defined as:

$$\operatorname{sgn}(\bullet) = \begin{cases} 1 & \text{if } \bullet > 0 \\ 0 & \text{if } \bullet \leq 0 \end{cases} \quad (12)$$

The flow rate of the bidirectional fixed-displacement piston pump is determined by its rotational speed and leakage, we can obtain:

$$Q_{ap} = D_p k_\omega u - C_p (P_p - P_r) \quad (13)$$

where D_p denote the pump displacement. k_ω and u specify the rotational speed gain and the applied input voltage corresponding to the servo motor, respectively. C_p denotes the leakage factor of the piston pump, and P_p and P_r are the pressures of the piston and rod cavities, respectively.

By combining Equations (10)–(13), neglecting the external leakage in the closed fluid circuit, and applying the flow continuity theorem, the pressure-flow dynamic relationships governing the two-stage hydraulic cylinder are derived as follows:

$$\begin{cases} \frac{V_p}{\beta_e} \dot{P}_p = Q_{ap} - A_p \frac{\partial x_p}{\partial \theta} \dot{\theta} - C_t (P_p - P_r) + \frac{\partial x_p}{\partial \theta} \dot{\theta} (A_p - A_r) \operatorname{sgn}\left(-\frac{\partial x_p}{\partial \theta} \dot{\theta}\right) + q_1(t) \\ \frac{V_r}{\beta_e} \dot{P}_r = -Q_{ap} + A_r \frac{\partial x_p}{\partial \theta} \dot{\theta} + C_t (P_p - P_r) + \frac{\partial x_p}{\partial \theta} \dot{\theta} (A_p - A_r) \operatorname{sgn}\left(\frac{\partial x_p}{\partial \theta} \dot{\theta}\right) - q_2(t) \end{cases} \quad (14)$$

where $V_p = V_{0p} + A_{1p}x_{10} + A_{2p}x_{21}$ and $V_r = V_{0r} - A_{1r}x_{10} - A_{2r}x_{21}$ represent the total fluid volumes of the piston and rod cavities, while V_{0p} and V_{0r} characterize the corresponding original volumes, respectively. x_{10} is the displacement of the first-stage cylinder relative to the fixed piston rod, and x_{21} is the displacement of the second-stage cylinder relative to the first stage. β_e denote the equivalent bulk modulus of the hydraulic fluid, C_t is the internal leakage factor of the actuator, and $q_1(t)$ and $q_2(t)$ signify the modeling errors of the pressure-flow dynamics within the dual cavities.

2.3. State-Space Equation Derivation

According to the mathematical models of the electro-hydrostatic erecting system characterized by Equations (7) and (14), the state vectors are chosen as $x = [x_1, x_2, x_3]^T = [\theta, \dot{\theta}, F]^T$, in which $F = A_p P_p - A_r P_r$. The state-space model governing the erecting mechanism can be established as:

$$\begin{cases} \dot{x}_1 = x_2 \\ f_1 \dot{x}_2 = x_3 - f_2 - f_3 + \Delta_1(t) \\ \dot{x}_3 = g_1 u - f_4 - f_5 - f_6 + \Delta_2(t) \end{cases} \quad (15)$$

where $f_1, f_2, f_3, f_4, f_5, f_6, g_1, \Delta_1$ and Δ_2 are expressed as follows:

$$\begin{aligned}
 f_1 &= \frac{J}{2\tau(x_1)} \\
 f_2 &= \frac{mgL_8 \cos(x_1 + \beta_0)}{2\tau(x_1)} \\
 f_3 &= \frac{Bx_2 + A_f S_f(x_2)}{2\tau(x_1)} \\
 \Delta_1 &= \frac{d}{2\tau(x_1)} \\
 g_1 &= \left(\frac{A_p}{V_p} + \frac{A_r}{V_r} \right) \beta_e D_p k_\omega \\
 f_4 &= \beta_e \left(\frac{A_p^2}{V_p} + \frac{A_r^2}{V_r} \right) \frac{\partial x_p}{\partial x_1} x_2 \\
 f_5 &= \beta_e \left(\frac{A_p}{V_p} + \frac{A_r}{V_r} \right) (C_p + C_t) (P_p - P_r) \\
 f_6 &= -\beta_e \frac{\partial x_p}{\partial x_1} x_2 (A_p - A_r) \left[\frac{A_p}{V_p} \operatorname{sgn} \left(-\frac{\partial x_p}{\partial x_1} x_2 \right) - \frac{A_r}{V_r} \operatorname{sgn} \left(\frac{\partial x_p}{\partial x_1} x_2 \right) \right] \\
 \Delta_2 &= \beta_e \left(\frac{A_p q_1}{V_p} + \frac{A_r q_2}{V_r} \right)
 \end{aligned} \tag{16}$$

The state-space architecture defined in Equation (15) relies on the following structural simplifications: synchronized dual-cylinder operation, normally closed relief valves, negligible external leakage, and piecewise continuous effective acting areas. The validity of ignoring external leakages and relief valve flows is maintained by the high-pressure sealing rings and standard system safety thresholds. However, the stage-changing process introduces non-smooth step discontinuities into the system state space due to the sudden transition between the first-stage and second-stage piston areas and the sudden changes in friction profiles and internal pressures. In this manuscript, these piecewise step variations are handled within the state-space model by splitting them into nominal deterministic jumps and time-varying residual dynamics. The nominal area changes are directly evaluated via the piecewise expressions in Equations (8) and (9), whereas the resulting high-frequency pressure shocks, localized friction mutations, and impact forces are mathematically absorbed into the lumped matched and unmatched uncertainty channels. The distributed Dual-RISE controller implementation resolves these discontinuities by utilizing its high-order integral filtering chains to continuously smooth the control voltage, while the nonlinear error transformation layer traps any transient tracking spikes inside the safety limits.

This research aims to formulate a control strategy characterized by bounded input u for the electro-hydrostatic erecting mechanism. This design guarantees that the actual erecting angle x_1 maintains high-precision tracking performance relative to the reference trajectory x_{1d} , while strictly enforcing the asymptotic convergence of the tracking error within the designated boundaries dictated by the performance function.

Assumption 1. The desired angle command x_{1d} for the erecting system is third-order continuously differentiable, and x_{1d} along with its derivatives through the third order are all guaranteed to be bounded.

Assumption 2. The unmatched and matched disturbances are sufficiently smooth, and their first and second-order time derivatives are constrained by bounded values

$$|\dot{\Delta}_1| \leq \zeta_{11}, |\ddot{\Delta}_1| \leq \zeta_{12}, |\dot{\Delta}_2| \leq \zeta_{21}, |\ddot{\Delta}_2| \leq \zeta_{22} \quad (17)$$

where $\zeta_{11}, \zeta_{12}, \zeta_{21}, \zeta_{22}$ are unknown positive constants.

Remark 2. In practical engineering systems, hydraulic stage-change impacts are often categorized as non-smooth or impulsive. However, from a physical perspective, the actual collision force transmitted to the control loop is fundamentally mediated by the structural elasticity of the multi-link mechanism, the compliance of the cylinder seals, and the compressibility of the hydraulic oil. These physical properties act as natural high-frequency low-pass filters, meaning the impact energy is transferred over a finite, non-zero time interval rather than an instantaneous mathematical delta impulse. Consequently, the lumped disturbances Δ_1 and Δ_2 remain continuous and second-order differentiable, preserving the mathematical validity of Assumption 2.

3. Controller Design and Synthesis

3.1. Prescribed Performance Function Design

Defining the erecting angle tracking error as $e_1(t) = x_1 - x_{1d}$. With the aim of optimizing the transient response of the electro-hydrostatic actuation system, a prescribed performance function (PPF) $\mu(t)$ is employed to constrain the tracking error $e_1(t)$ within a predefined boundary, such that $-\underline{\delta}\mu(t) < e_1(t) < \bar{\delta}\mu(t)$. The PPF satisfies:

$$\begin{cases} \mu(t) = (\mu_0 - \mu_\infty)e^{-kt} + \mu_\infty \\ \lim_{t \rightarrow \infty} \mu(t) = \mu_\infty > 0 \end{cases} \quad (18)$$

where $\mu(t)$ is established as a strictly positive, monotonically decreasing scalar function. $\underline{\delta}, \bar{\delta}, \mu_0, \mu_\infty, k$ are positive adjustable parameters. $-\underline{\delta}\mu_0$ and $\bar{\delta}\mu_0$ delineate the respective lower and upper thresholds governing the initial tracking error $e_1(0)$. k signifies the convergence velocity, and μ_∞ represents the prescribed steady-state error range. A nonlinear mapping is introduced as:

$$e_1(t) = \mu(t)S(z_1) \quad (19)$$

which possesses the following properties:

$$\begin{cases} -\underline{\delta} < S(z_1) < \bar{\delta} \\ \lim_{z_1 \rightarrow -\infty} S(z_1) = -\underline{\delta} \\ \lim_{z_1 \rightarrow +\infty} S(z_1) = \bar{\delta} \end{cases} \quad (20)$$

The increasing function $S(z_1)$ is given as:

$$S(z_1) = \frac{\bar{\delta}e^{z_1} - \underline{\delta}e^{-z_1}}{e^{z_1} + e^{-z_1}} = \frac{e_1(t)}{\mu(t)} = w \quad (21)$$

The transformed error z_1 obtained via inverse function transformation is:

$$z_1 = \frac{1}{2} \ln \frac{w + \bar{\delta}}{\bar{\delta} - w} \quad (22)$$

The parameters $\underline{\delta}, \bar{\delta}, \mu_0, \mu_\infty, k$ are positive adjustable design values. Crucially, to ensure that the tracking error $e_1(t)$ remains strictly inside the prescribed performance boundaries for all $t \geq 0$, the initial system states must satisfy the strict initial-condition

constraint $-\underline{\delta}\mu_0 < e_1(0) < \bar{\delta}\mu_0$. This constraint guarantees that the initial normalized tracking error $S(z_1(0)) = e_1(0)/\mu(0)$ is well-defined within the open interval $(-\underline{\delta}, \bar{\delta})$, thereby ensuring that the initial transformed error $z_1(0)$ calculated via the inverse mapping is finite.

Differentiating Equation (22) yields:

$$\dot{z}_1 = \frac{1}{2} \left(\frac{1}{w + \underline{\delta}} - \frac{1}{w - \bar{\delta}} \right) \left(\frac{\dot{e}_1}{\mu} - \frac{e_1 \dot{\mu}}{\mu^2} \right) = \rho \left(\dot{e}_1 - \frac{e_1 \dot{\mu}}{\mu} \right) = \rho \left(x_2 - \dot{x}_{1d} - \frac{e_1 \dot{\mu}}{\mu} \right) \quad (23)$$

where $\rho = \frac{1}{2\mu} \left(\frac{1}{w + \underline{\delta}} - \frac{1}{w - \bar{\delta}} \right)$ satisfies $0 < \rho < \rho_M = (\underline{\delta} + \bar{\delta}) / (\mu_\infty \underline{\delta} \bar{\delta})$.

To ensure that the nonlinear error transformation layer does not generate control commands that exceed physical limitations, the selection of the PPF parameters must be mathematically coupled to the system's hardware constraints. The physical boundaries of the electro-hydrostatic erecting system are defined by the maximum input voltage, maximum pump speed, maximum flow rate, and the structural pressure limits of the cylinder cavities. To prevent the transformed error z_1 from approaching infinity, the maximum control effort required to track the reference trajectory $\ddot{x}_{1d}$ and reject the nominal disturbances must strictly reside within the physical capacity of the actuator. Based on the system dynamics defined in Equation (15), the maximum acceleration capacity of the erection mechanism under peak control voltage is governed by $\ddot{\theta}_{\max} \approx f_1^{-1} g_1 u_{\max}$. To guarantee bound feasibility, the predefined tracking envelope must satisfy the following inequality condition:

$$\max \left| \ddot{x}_{1d} + \frac{d}{dt} \left(\frac{e_1 \dot{\mu}}{\mu} \right) \right| + \Psi_{\max} < \ddot{\theta}_{\max} \quad (24)$$

where $\Psi_{\max}$ represents the upper bound of the lumped mechanical and volumetric uncertainties. By configuring the performance function decay rate k and steady-state bound μ_∞ conservatively relative to the trajectory acceleration $\ddot{x}_{1d}$, the tracking error e_1 is physically restricted from ever approaching the predefined safety thresholds $-\underline{\delta}\mu(t)$ and $\bar{\delta}\mu(t)$. This explicit parameter constraint prevents the nonlinear mapping layer from entering the high-gain saturation region during standard operation.

Remark 3. As long as the transformed error z_1 maintains its boundedness through the implementation of the control law, the prescribed performance condition $-\underline{\delta}\mu(t) < e_1(t) < \bar{\delta}\mu(t)$ is guaranteed to hold based on the properties of $S(z_1)$. The claim that the tracking error $e_1(t)$ asymptotically converges to exactly zero requires careful coordination with the symmetry of the chosen boundary parameters. Evaluating the error transformation mapping at the origin yields:

$$S(0) = \frac{\bar{\delta}e^0 - \underline{\delta}e^0}{e^0 + e^0} = \frac{\bar{\delta} - \underline{\delta}}{2} \quad (25)$$

If the upper and lower boundary parameters are asymmetric, then $S(0) \neq 0$. Consequently, driving the transformed error $z_1(t)$ to zero as $t \rightarrow \infty$ would cause the tracking error to converge to a non-zero steady-state offset defined by $\lim_{t \rightarrow \infty} e_1(t) = \mu_\infty (\bar{\delta} - \underline{\delta}) / 2$. To ensure that the stabilization of the transformed error z_1 at the origin serves as a necessary and sufficient condition for the

true tracking error e_1 to vanish asymptotically, a symmetric boundary configuration where $\bar{\delta} = \underline{\delta}$ must be strictly enforced. It is worth noting that this stability property represents a significant theoretical departure from existing funnel control and conventional PPC methods. While traditional approximation-based or funnel strategies typically guarantee only practical stability—where the tracking error is driven into a bounded residual set unless feedback gains are increased to infinity—the integration of the nonlinear mapping layer with the subsequent Dual-RISE architecture mathematically ensures that the tracking error asymptotically approaches exactly zero ($e_1(t) \rightarrow 0$ as $t \rightarrow \infty$) while maintaining strict boundary envelopes throughout the transient phase.

3.2. Controller Design

Drawing upon the established dynamic equations of the electro-hydrostatic actuation system and the transformed error z_1 , a prescribed performance control strategy based on dual integral robust control is formulated. Differentiating Equation (23) yields:

$$\ddot{z}_1 = \dot{\rho} \left(\dot{e}_1 - \frac{e_1 \dot{\mu}}{\mu} \right) + \rho \left[\dot{x}_2 - \ddot{x}_{1d} - \left(\frac{\dot{e}_1 \dot{\mu} + e_1 \ddot{\mu}}{\mu} - \frac{e_1 \dot{\mu}^2}{\mu^2} \right) \right] \quad (26)$$

Define the error variables as follows:

$$\begin{cases} z_2 = \dot{z}_1 + k_1 z_1 \\ r_1 = \dot{z}_2 + k_2 z_2 \\ z_3 = x_3 - \alpha_2 \\ r_2 = \dot{z}_3 + k_3 z_3 \end{cases} \quad (27)$$

where r_1 and r_2 are auxiliary error variables. k_1, k_2, k_3 are positive feedback gains.

From Equations (23), (26) and (27), we can derive:

$$\begin{aligned} f_1 r_1 &= f_1 \dot{z}_2 + k_2 f_1 z_2 = f_1 \ddot{z}_1 + k_1 f_1 \dot{z}_1 + k_2 f_1 z_2 \\ &= f_1 (k_1 \dot{z}_1 + k_2 z_2) + f_1 \left[\dot{\rho} \left(\dot{e}_1 - \frac{e_1 \dot{\mu}}{\mu} \right) - \rho \left(\frac{\dot{e}_1 \dot{\mu} + e_1 \ddot{\mu}}{\mu} - \frac{e_1 \dot{\mu}^2}{\mu^2} \right) \right] - \rho f_1 \ddot{x}_{1d} \\ &\quad + \rho z_3 + \rho \alpha_2 - \rho f_2 - \rho f_3 + \rho \Delta_1 \end{aligned} \quad (28)$$

The synthesized virtual control law α_2 is formulated through the following expression:

$$\begin{cases} \alpha_2 = (\alpha_{2a} + \alpha_{2s1} + \alpha_{2s2}) / \rho \\ \alpha_{2a} = \rho (f_1 \ddot{x}_{1d} + f_2 + f_3) \\ \alpha_{2s1} = -(k_{r1} + k_s) z_2 \\ \alpha_{2s2} = -\int_0^t [(k_{r1} + k_s) k_2 z_2 + \beta_1 \text{sign}(z_2)] d\nu \end{cases} \quad (29)$$

where α_{2a} serves as the feedforward compensation term derived from the system model. α_{2s1} denotes the linear robust control component for stabilizing the nominal system dynamics, while α_{2s2} represents the nonlinear RISE control law aimed at attenuating unmodeled disturbances. k_{r1}, k_s, β_1 are positive gains.

Substituting Equation (29) into Equation (28) yields:

$$f_1 r_1 = \Phi + \rho z_3 - (k_{r1} + k_s) z_2 - \int_0^t [(k_{r1} + k_s) k_2 z_2 + \beta_1 \text{sign}(z_2)] d\nu + \rho \Delta_1 \quad (30)$$

where Φ is defined as:

$$\Phi = f_1 (k_1 \dot{z}_1 + k_2 z_2) + f_1 \left[\dot{\rho} \left(\dot{e}_1 - \frac{e_1 \dot{\mu}}{\mu} \right) - \rho \left(\frac{\dot{e}_1 \dot{\mu} + e_1 \ddot{\mu}}{\mu} - \frac{e_1 \dot{\mu}^2}{\mu^2} \right) \right] \quad (31)$$

Differentiating both sides of Equation (30) leads to:

$$\begin{aligned} f_1 \dot{r}_1 &= \dot{\Phi} + \dot{\rho} z_3 + \rho \dot{z}_3 - (k_{r1} + k_s) \dot{z}_2 - (k_{r1} + k_s) k_2 z_2 - \beta_1 \text{sign}(z_2) - \dot{f}_1 r_1 + \rho \dot{\Delta}_1 + \dot{\rho} \Delta_1 \\ &= \bar{N} - (k_{r1} + k_s) r_1 - \beta_1 \text{sign}(z_2) - \frac{1}{2} \dot{f}_1 r_1 + N_d + \rho \dot{z}_3 \end{aligned} \quad (32)$$

$$\bar{N} \triangleq \dot{\Phi} + \dot{\rho} z_3 - \frac{1}{2} \dot{f}_1 r_1 \quad (33)$$

$$N_d \triangleq \rho \dot{\Delta}_1 + \dot{\rho} \Delta_1 \quad (34)$$

By applying the Mean Value Theorem to the continuously differentiable function $\bar{N}$, it is easily verified that $\bar{N}$ has an upper bound:

$$\|\bar{N}\| \leq \eta(\|z\|)\|z\| \quad (35)$$

where $z \triangleq [z_1, z_2, r_1, z_3, r_2]^T$, and the bounding function $\eta(\cdot)$ is characterized as a globally invertible, strictly positive, and monotonically non-decreasing mapping.

In accordance with Assumption 2, the following inequalities can be derived:

$$\|N_d\| \leq \zeta_{31}, \|\dot{N}_d\| \leq \zeta_{32} \quad (36)$$

where ζ_{31} and ζ_{32} are positive constants.

From Equations (15) and (27), the expanded expression of r_2 can be obtained as:

$$r_2 = \dot{x}_3 - \dot{\alpha}_2 + k_3 z_3 = g_1 u - f_4 - f_5 - f_6 + \Delta_2 - \dot{\alpha}_2 + k_3 z_3 \quad (37)$$

Similar to Equation (29), the control input u is synthesized through the following formulation:

$$\begin{cases} u = (u_a + u_{s1} + u_{s2}) / g_1 \\ u_a = f_4 + f_5 + f_6 + \dot{\alpha}_2 - k_3 z_3 \\ u_{s1} = -k_{r2} z_3 \\ u_{s2} = -\int_0^t [k_{r2} k_3 z_3 + \beta_2 \text{sign}(z_3)] dv \end{cases} \quad (38)$$

where u_a serves as the feedforward compensation term derived from the system model.

u_{s1} denotes the linear robust control component, and u_{s2} represents the nonlinear RISE control law, which can suppress the impact of matched uncertainties on system control performance. k_{r2} , β_1 are positive gains.

Substituting Equation (38) into Equation (37) yields:

$$r_2 = -k_{r2} z_3 - \int_0^t [k_{r2} k_3 z_3 + \beta_2 \text{sign}(z_3)] dv + \Delta_2 \quad (39)$$

Differentiating both sides of Equation (39) leads to:

$$\dot{r}_2 = \dot{\Delta}_2 - k_{r2} r_2 - \beta_2 \text{sign}(z_3) \quad (40)$$

The model parameters utilized by the feedforward terms α_{2a} and u_a are nominal physical parameters, including effective cylinder areas, linkage lengths, fluid bulk modulus, internal leakage factor, and friction coefficients. In practical electro-hydrostatic erecting system, parameter sensitivity is divided into two distinct categories: (i) Geometry and effective areas are measured offline via high-precision CAD models and coordinate measuring machines, through which only negligible variations are introduced. (ii) Fluid bulk modulus and oil viscosity vary dynamically during operation, while friction coefficients are impacted by operating speed and temperature.

Mathematically, any mismatch between nominal model parameters and physical realities is explicitly absorbed into the lumped unmatched and matched uncertainty channels. Because these parameter mismatches are formulated as continuous and smooth functions of system states and time, Assumption 2 is strictly satisfied. Nominal dynamic anticipation is provided by the feedforward model terms to reduce tracking lag, while exact asymptotic cancellation of the residual parameter errors is mathematically guaranteed by the distributed RISE integral feedback blocks without loss of stability or the requirement of online parameter adaptation loops. Consequently, closed-loop tracking precision and boundary protection are rendered highly insensitive to parameter drift.

The schematic of the proposed control architecture is illustrated in Figure 4.

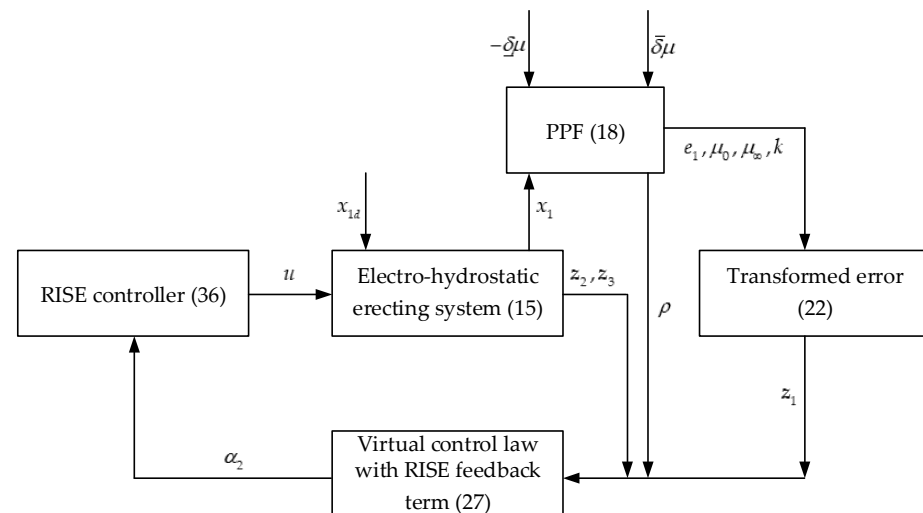


Figure 4. The schematic of the proposed controller.

3.3. Theoretical Result and Stability Verification

Lemma 1. Define the auxiliary functions as follows:

$$L_1(t) \triangleq r_1 [N_d - \beta_1 \text{sign}(z_2)] \quad (41)$$

$$L_2(t) \triangleq r_2 [\dot{N}_d - \beta_2 \text{sign}(z_3)] \quad (42)$$

$$P_1(t) \triangleq \beta_1 |z_2(0)| - z_2(0) N_d(0) - \int_0^t L_1(v) dv \quad (43)$$

$$P_2(t) \triangleq \beta_2 |z_3(0)| - z_3(0) \dot{N}_d(0) - \int_0^t L_2(v) dv \quad (44)$$

where $z_2(0)$ and $z_3(0)$ denote the initial values of z_2 and z_3 , while $N_d(0)$ and $\dot{N}_d(0)$ represent the initial values of N_d and $\dot{N}_d$, respectively. As long as the subsequent conditions are satisfied, $P_1(t)$ and $P_2(t)$ are always positive:

$$\beta_1 > \zeta_{31} + \frac{\zeta_{32}}{k_2} \quad (45)$$

$$\beta_2 > \zeta_{21} + \frac{\zeta_{22}}{k_3} \quad (46)$$

The rigorous derivation supporting Lemma 1 is provided in [29].

Let D denote a domain containing $\varepsilon(t) = 0$, where $\varepsilon(t)$ is defined as follows:

$$\varepsilon(t) \triangleq [z^T, \sqrt{P_1}, \sqrt{P_2}]^T \quad (47)$$

Theorem 1. By selecting sufficiently large control gains k_1 , k_2 , k_3 , k_{r1} and k_{r2} to ensure the matrix Λ in Equation (51) is positive definite, the synthesized RISE-based asymptotic prescribed performance control law (38) ensures that all signals within the closed-loop architecture remain bounded and strictly enforces the prescribed tracking performance constraints. Furthermore, the proposed control scheme guarantees that semi-global asymptotic tracking precision is effectively realized.

Proof of Theorem 1. Choosing the Lyapunov function $V(\varepsilon, t)$ as:

$$V(\varepsilon, t) = \frac{1}{2}z_1^2 + \frac{1}{2}z_2^2 + \frac{1}{2}z_3^2 + \frac{1}{2}f_1r_1^2 + \frac{1}{2}r_2^2 + P_1(t) + P_2(t) \quad (48)$$

For any $\varepsilon \in D$, we can bound $V(\varepsilon, t)$ by

$$U_1(\varepsilon) \leq V(\varepsilon, t) \leq U_2(\varepsilon) \quad (49)$$

where $U_1(\varepsilon) \triangleq \sigma_1 \|\varepsilon\|^2$ and $U_2(\varepsilon) \triangleq \sigma_2 \|\varepsilon\|^2$ are continuous positive definite functions, in which $\sigma_1 = \min\{1, f_{1\min}\}/2$, $\sigma_2 = \max\{2, f_{1\max}\}/2$, and $f_{1\min}$ and $f_{1\max}$ are the minimum and maximum values of function f_1 , respectively.

Differentiating $V(\varepsilon, t)$ yields:

$$\begin{aligned} \dot{V}(\varepsilon, t) &= z_1\dot{z}_1 + z_2\dot{z}_2 + z_3\dot{z}_3 + \frac{1}{2}\dot{f}_1r_1^2 + f_1r_1\dot{r}_1 + r_2\dot{r}_2 + \dot{P}_1 + \dot{P}_2 \\ &= z_1(z_2 - k_1z_1) + z_2(r_1 - k_2z_2) + z_3(r_2 - k_3z_3) + \frac{1}{2}\dot{f}_1r_1^2 \\ &\quad + r_1\left[-\frac{1}{2}\dot{f}_1r_1 + \bar{N} + N_d - (k_{r1} + k_s)r_1 - \beta_1\text{sign}(z_2) + \rho(r_2 - k_3z_3)\right] \\ &\quad + r_2\left[\dot{\Delta}_2 - k_{r2}r_2 - \beta_2\text{sign}(z_3)\right] - r_1\left[N_d - \beta_1\text{sign}(z_2)\right] - r_2\left[\dot{\Delta}_2 - \beta_2\text{sign}(z_3)\right] \\ &= -k_1z_1^2 - k_2z_2^2 - k_3z_3^2 - k_{r1}r_1^2 - k_{r2}r_2^2 + z_1z_2 + z_2r_1 + z_3r_2 + r_1r_2\rho \\ &\quad - r_1k_3z_3\rho - k_sr_1^2 + r_1\bar{N} \\ &= -z^T\Lambda z - k_sr_1^2 + r_1\bar{N} \end{aligned} \quad (50)$$

where the matrix Λ is defined as:

$$\Lambda = \begin{bmatrix} k_1 & -\frac{1}{2} & 0 & 0 & 0 \\ -\frac{1}{2} & k_2 & -\frac{1}{2} & 0 & 0 \\ 0 & -\frac{1}{2} & k_{r1} & \frac{\rho k_3}{2} & -\frac{\rho}{2} \\ 0 & 0 & \frac{\rho k_3}{2} & k_3 & -\frac{1}{2} \\ 0 & 0 & -\frac{\rho}{2} & -\frac{1}{2} & k_{r2} \end{bmatrix} \quad (51)$$

To provide clear guidance for parameter calibration and guarantee the positive definiteness of the matrix Λ , explicit sufficient inequalities can be derived by applying Young's inequality to the cross-coupling terms in Equation (50). Let $c_1, c_2, c_3, c_4, c_5 > 0$ be arbitrary scaling constants. The matrix Λ is guaranteed to be strictly positive definite if the selected controller gains satisfy the following sufficient conditions:

$$k_1 > \frac{c_1}{2}, k_2 > \frac{1}{2c_1} + \frac{c_2}{2}, k_3 > \frac{c_3}{2} + \frac{\rho_M k_3 c_5}{2}, k_{r1} > \frac{1}{2c_2} + \frac{\rho_M c_4}{2} + \frac{\rho_M k_3}{2c_2}, k_{r2} > \frac{1}{2c_3} + \frac{\rho_M}{2c_4} \quad (52)$$

Substituting Equation (35) into Equation (50) yields:

$$\begin{aligned} \dot{V}(\varepsilon, t) &\leq -\kappa_{\min}(\Lambda) \|z\|^2 - k_s \|r_1\|^2 + \eta (\|z\|) \|z\| \|r_1\| \\ &\leq -\kappa_{\min}(\Lambda) \|z\|^2 - k_s \left(\|r_1\| - \frac{\eta (\|z\|) \|z\|}{2k_s} \right)^2 + \frac{\eta^2 (\|z\|) \|z\|^2}{4k_s} \\ &\leq -\left(\kappa_{\min}(\Lambda) - \frac{\eta^2 (\|z\|)}{4k_s} \right) \|z\|^2 \leq -\zeta \|z\|^2 \triangleq -U(\varepsilon) \end{aligned} \quad (53)$$

where $\kappa_{\min}(\Lambda)$ stands for the minimum eigenvalue of matrix Λ , and ζ represents a positive constant derived under the satisfaction of the following condition:

$$\|z\| \leq \eta^{-1} \left(2\sqrt{\kappa_{\min}(\Lambda) \cdot k_s} \right) \quad (54)$$

$U(\varepsilon)$ serves as a non-negative definite function established within the domain D defined as below:

$$D \triangleq \left\{ \varepsilon \mid \|\varepsilon\| \leq \eta^{-1} \left(2\sqrt{\kappa_{\min}(\Lambda) \cdot k_s} \right) \right\} \quad (55)$$

From Equation (20), it can be inferred that $V(\varepsilon, t) \in L_\infty$ and $U(\varepsilon) \in L_2$. Based on Equation (48), all error signals including z_1, z_2, z_3, r_1 and r_2 are bounded. Utilizing Assumption 1, the states x_1, x_2 and x_3 are also bounded. According to Equations (27), (32) and (40), we can obtain that the time derivatives of z_1, z_2, z_3, r_1 and r_2 are bounded, which implies that $U(\varepsilon)$ exhibits the property of uniform continuity within the domain D .

Define a set $\Xi \in D$ as:

$$\Xi \triangleq \left\{ \varepsilon \in D \mid U_2(\varepsilon) \leq \sigma_1 \left(\eta^{-1} \left(2\sqrt{\kappa_{\min}(\Lambda) \cdot k_s} \right) \right)^2 \right\} \quad (56)$$

By invoking Theorem 8.4 as presented in [30], one can deduce that

$$\zeta \|z\|^2 \rightarrow 0 \text{ as } t \rightarrow \infty \quad \forall \varepsilon(0) \in \Xi \quad (57)$$

According to the definition of the transformed error z_1 and Remark 3, the system tracking error is effectively constrained within the predefined performance boundaries, while simultaneously achieving asymptotic tracking performance, i.e.,

$$e(t) \rightarrow 0 \text{ as } t \rightarrow \infty \quad \forall \varepsilon(0) \in \Xi \quad (58)$$

To further evaluate system behavior under extreme operating regimes where physical hardware saturates, the control input entering the system is modeled using a standard saturation operator

$$u_{sat} = \text{sat}(u) = \text{sign}(u) \min(|u|, u_{\max}) \quad (59)$$

Define the input saturation error as $\tilde{u} = u_{sat} - u$. When input saturation occurs, the actual third-order state derivative becomes

$$\dot{x}_3 = g_1 u + g_1 \tilde{u} - f_4 - f_5 - f_6 + \Delta_2(t) \quad (60)$$

To prevent the non-smooth derivative of the hard saturation function from invalidating the stability proof, the error dynamics are separated into a nominal control path and an actual state path. The nominal inner-loop error is defined as

$$r_{2n} = g_1 u - f_4 - f_5 - f_6 + \Delta_2 - \dot{\alpha}_2 + k_3 z_3 \quad (61)$$

The derivative of r_{2n} can be obtained as

$$\dot{r}_{2n} = \dot{\Delta}_2 - k_{r2} r_{2n} - \beta_2 \text{sign}(z_3) \quad (62)$$

The actual state tracking error derivative then absorbs the saturation mismatch:

$$\dot{z}_3 = r_{2n} - k_3 z_3 + g_1 \tilde{u} \quad (63)$$

Incorporating Equations (60)–(63) into the time derivative of the composite Lyapunov function $V(\varepsilon, t)$ yields:

$$\dot{V}(\varepsilon, t) = -z^T \Lambda z - k_s r_1^2 + r_1 \bar{N} + (z_3 + \rho r_1) g_1 \tilde{u} \quad (64)$$

Under nominal conditions, $\tilde{u} = 0$, preserving the semi-global asymptotic stability established in Theorem 1. During severe stage-change impacts where the motor voltage or hydraulic pump speed briefly saturates, $\tilde{u} \neq 0$. Because the mechanical pressures and motor power are physically capped by system relief valves ($P_{\max}$) and drive limitations ($u_{\max}$), the saturation term is strictly bounded by an upper scalar constant $\Omega_{\max} \geq \|(z_3 + \rho r_1) g_1 \tilde{u}\|$. This modifies the energy derivative during transient saturation to:

$$\dot{V}(\varepsilon, t) \leq -\zeta \|z\|^2 + \Omega_{\max} \quad (65)$$

This guarantees that $\dot{V}(\varepsilon, t) < 0$ outside the compact residual set $\Omega_R \triangleq \{z \mid \|z\| \leq \sqrt{\Omega_{\max} / \zeta}\}$, proving that the closed-loop system is uniformly ultimately bounded under input saturation. Because the performance bounds are chosen according to the feasibility criteria, the tracking error e_1 remains safely trapped inside the invariant set of the PPF boundary $-\bar{\delta}u(t) < e_1(t) < \bar{\delta}u(t)$ during the brief saturation interval. Once the transient impact decays and the controller exits saturation, asymptotic convergence is restored seamlessly. $\square$

During the exact microsecond of the stage-change collision, the higher-order derivatives $\ddot{\Delta}_1$ and $\ddot{\Delta}_2$ exhibit localized high-amplitude spikes. If these transient peaks briefly exceed the conservative theoretical bounds ζ_{12} and ζ_{22} assumed for the ideal asymptotic proof, the system's stability properties transiently transition from asymptotic convergence to semi-global practical boundedness. Crucially, during this brief practical stability phase, the nonlinear error transformation layer of the PPF acts as an impact-absorbing safety envelope. It natively restricts the tracking error from escaping, enforcing compliance with the predefined transient boundary $-\bar{\delta}u(t) < e_1(t) < \bar{\delta}u(t)$. Once the structural impact oscillations decay and the disturbance derivatives return within their nominal bounds, the distributed Dual-RISE integral signum terms immediately eliminate the residual tracking error, fully restoring the semi-global asymptotic convergence verified in Theorem 1.

In the electro-hydrostatic multi-link erecting system, disturbance rejection is achieved through a distributed dual-loop structure designed to cancel both unmatched and matched uncertainty channels. The total system uncertainty is partitioned into an unmatched channel Δ_1 and a matched channel Δ_2 . The proposed strategy employs a RISE mechanism. The disturbance estimation effort is distributed across two continuous integral signum channels α_{2s2} and u_{s2} . Because the signum operator is placed inside time

integrals, the control signals generated remain continuous. As proven in Theorem 1, if the time derivative of the lumped disturbances satisfies C^2 smoothness, the RISE integral channels continuously reconstruct the exact inverse profile of the time-varying disturbances, driving tracking errors asymptotically to zero.

Remark 4. To facilitate practical engineering implementation and ensure the theoretical stability requirements are met without triggering actuator saturation or sensor noise amplification, the controller parameters are systematically configured via a step-by-step hierarchical procedure rather than empirical trial-and-error: (i) The PPF parameters are selected to reflect physical design specifications: the initial bound μ_0 is scaled with the expected alignment offset to prevent transient voltage saturation, μ_∞ dictates steady-state tracking accuracy, k governs the settling time, and $\underline{\delta}$, $\bar{\delta}$ define transient safety envelopes. This establishes the baseline parameter ρ_M for the sufficient inequalities. (ii) The cascaded tracking gains k_1 , k_2 and k_3 are configured to satisfy the strict bandwidth separation principle $k_1 > k_2 > k_3$. A large outer-loop gain k_1 forces rapid error convergence within the boundary. The velocity-loop gain k_2 and pressure-loop tracking gain k_3 are sequentially scaled down to match the physical frequency response limits of the motor and pump dynamics, preventing high-frequency loop interactions. (iii) Rather than driving the linear robust feedback terms k_{r1} and k_{r2} to extreme values to reject disturbances—which would directly amplify high-frequency measurement noise and induce severe control voltage chattering— k_{r1} and k_{r2} are intentionally kept at low, conservative levels, while the integral signum gain k_s eliminates lumped disturbances and is bounded to prevent control input chattering and mechanical resonance. Structurally, this parameter hierarchy isolates the control channels into a distributed framework: the virtual control law component α_{2s2} utilizes an integral signum loop to eliminate unmatched kinematics-induced disturbances, while the actual voltage input component u_{s2} deploys a separate integral signum loop to reject matched pressure-flow anomalies. This dual-channel distribution prevents high-gain feedback amplification, ensuring that when the system experiences severe energy shocks during cylinder stage-change transitions, the control inputs remain smooth and continuous without inducing voltage saturation or mechanical resonance.

4. Experimental Validation and Results Analysis

To validate the performance of the proposed control architecture, comparative experiments are conducted on the multi-link erecting platform depicted in Figure 5. The established experimental platform is driven via the electro-hydrostatic power source as shown in Figure 6. Components and specifications of the experimental platform are listed in Table 1. The physical parameters of the experimental platform are summarized in Table 2. The prime mover is driven by a REALLAND ETM25C PMSM powered by a Baumüller BM5563 digital servo drive. The motor features a linear speed-voltage gain of 300 r/min/V with a linearity error of less than $\pm 0.1\%$ across the operating envelope. The integrated electronic drive circuitry utilizes space vector pulse width modulation (SVPWM) operating at a high switching frequency of 8 kHz, ensuring rapid current-loop execution and precise torque-speed linearity. A Rexroth A4VG180EP4DT1 bidirectional closed-loop piston pump with a displacement of $1.8 \times 10^{-4} \text{ m}^3/\text{r}$ is employed. The swash-plate displacement control circuit incorporates an integrated electro-hydraulic proportional amplifier electronic card with closed-loop spool position feedback, achieving high flow linearity and volumetric precision. The erecting angle is measured via a Heidenhain ERN180 incremental optical rotary encoder. It provides a high system precision of 16-bit resolution,

translating to an angular measurement accuracy of ± 5 arcsec. The electronic interface circuitry utilizes differential RS-422 line-receiver signals connected directly to a Heidenhain IK-220 high-speed quadrature counter card, completely eliminating high-frequency electromagnetic interference (EMI) in industrial environments. The hydraulic working pressures in the piston and rod chambers are monitored using Huba Control 511.954602745 piezoresistive pressure transmitters. These sensors exhibit a static linearity better than $\pm 0.3\%$ FS, a total error band of $\pm 0.5\%$ FS across the compensated temperature range, and an analog output electronic circuit generating a filtered 0–10 V signal fed directly into the Advantech PCI-1716 high-speed A/D conversion board.

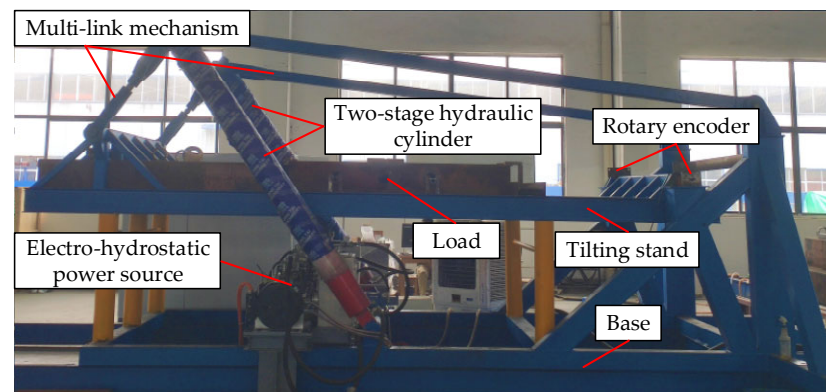


Figure 5. Multi-link erecting platform.

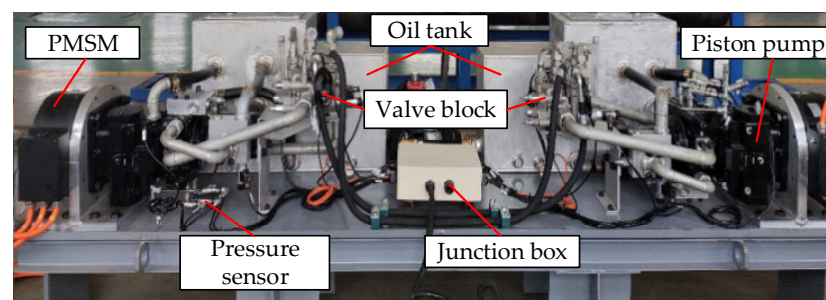


Figure 6. Electro-hydrostatic power source.

Table 1. Components and specifications of the experimental platform.

Component	Specification	Component	Specification
PMSM	REALLAND ETM25C	A/D card	Advantech PCI-1716
Servo drive	Baum Mueller BM5563	D/A card	Advantech PCI-1723
Piston pump	Rexroth A4VG180EP4DT1	Counter card	Heidenhain IK-220
Pressure sensor	Huba Control 511.954602745	Host computer	LabWindows/CVI
Rotary encoder	Heidenhain ERN180	Operating system	RTX+VS 2005

Table 2. Physical parameters of the experimental platform.

Parameter	Value	Parameter	Value
m	1.05×10^4 kg	L_4	0.848 m
J	2.6×10^5 kg $\cdot$ m ²	L_5	3.731 m
k_ω	300 r/min/V	L_6	2.988 m
D_p	1.8×10^{-4} m ³ /r	L_7	2.947 m
A_{1p}	2.69×10^{-2} m ²	L_8	6.087 m
A_{1r}	6.8×10^{-3} m ²	V_{0p}	1.075×10^{-4} m ³

A_{2p}	$1.43 \times 10^{-2} \text{ m}^2$	V_{0r}	$2.83 \times 10^{-2} \text{ m}^3$
A_{2r}	$4.8 \times 10^{-3} \text{ m}^2$	C_t	$4.82 \times 10^{-13} \text{ m}^3/\text{s}/\text{Pa}$
s_{1M}	2.442 m	C_p	$2.56 \times 10^{-11} \text{ m}^3/\text{s}/\text{Pa}$
s_{2M}	2.442 m	B	$8 \times 10^4 \text{ N} \cdot \text{m} \cdot \text{s} / \text{rad}$
L_1	5.050 m	A_f	3500 N·m
L_2	1.347 m	β_e	700 MPa
L_3	5.005 m	T_s	1 ms

To substantiate the performance of the control framework formulated in this work, comparative experiments were conducted against three distinct controllers:

- (1) PPC-Dual-RISE: This controller represents the novel strategy formulated in this manuscript. The associated parameters are selected as: $k_1 = 600$, $k_2 = 350$, $k_3 = 200$, $k_{r1} = 15$, $k_{r2} = 5$, $k_s = 250$, $\mu_0 = 0.3$, $\mu_\infty = 0.15$, $k = 1.5$, $\underline{\delta} = \bar{\delta} = 1$.
- (2) Dual-RISE: This strategy represents the Dual-RISE-based approach formulated without the prescribed performance function mapping layer. To establish a rigorous ablation study that isolates the precise performance contributions of the PPF block, the Dual-RISE controller retains the exact same tracking and robust gains k_1 , k_2 , k_3 , k_{r1} , k_{r2} , k_s utilized in PPC-Dual-RISE. It should be noted that while standard Dual-RISE could be independently retuned with alternative aggressive gains to modify its transient curve, keeping the core control parameters constant serves the specific scientific purpose of demonstrating the exact mathematical and physical effect of the PPF mapping layer.
- (3) VFPI: This refers to the velocity feed-forward proportional-integral (PI) controller, a frequently adopted architecture in engineering practice. The proportional and integral control gains are selected as $k_p = 1900$, $k_i = 650$, respectively. The velocity feed-forward gain is assigned a value of $k_v = 35$.

Moreover, to conduct a comprehensive comparative assessment of the three control schemes mentioned above, three statistical indices—namely, the maximum value (Max), the arithmetic mean (Mean), and the standard deviation (Std) of the absolute tracking error—are employed. Two experimental scenarios were carried out to validate the robust performance and efficacy of the control approach introduced in this study.

Case 1: Initially, a 30 s medium-speed erecting command generated via constant power trajectory planning is utilized to evaluate the performance of the developed control architecture. The derived results are illustrated in Figures 7–11. The tracking performance achieved by PPC-Dual-RISE is illustrated in Figure 7, which depicts that the proposed control strategy maintains a high level of tracking precision. Figure 8 presents the trajectory tracking error responses for the three evaluated controllers, while the corresponding quantitative performance indices are summarized in Table 3. Synthesizing the data presented in Figure 8 and Table 3, it is obvious that PPC-Dual-RISE achieves the highest tracking precision. In addition, the tracking error of PPC-Dual-RISE is always constrained within the set safety boundaries by incorporating the prescribed performance function into the controller design. Conversely, the Dual-RISE controller exhibits tracking errors that violate the designated performance boundaries, which stems from the non-incorporation of a prescribed performance function. Furthermore, the control performance of both PPC-Dual-RISE and Dual-RISE is significantly superior to that of VFPI, thereby validating the efficacy of the presented model-based nonlinear robust control law in suppressing

unmodeled disturbances and uncertainties. Figure 9 illustrates the working pressure of the two-stage hydraulic cylinder. It is observed that the hydraulic cylinder exhibits no severe chattering or crawling phenomena. Figure 10 exhibits the control input of PPC-Dual-RISE. The observed smoothness and continuity of the input signal confirm its suitability for mechanical and industrial scenarios.

Table 3. Performance indices of the erecting angle tracking in Case 1.

Indices	Max (°)	Mean (°)	Std (°)
PPC-Dual-RISE	0.1496	0.0487	0.0418
Dual-RISE	0.3575	0.0667	0.0611
VFPI	0.8477	0.1890	0.1847

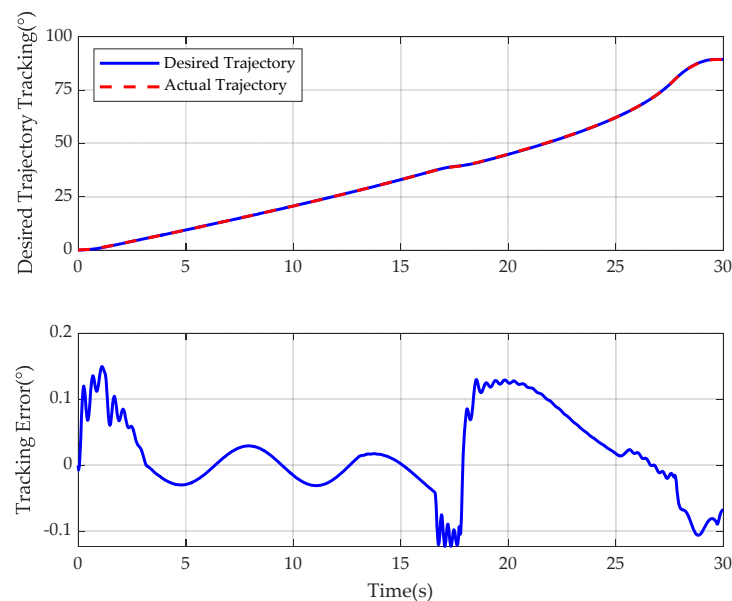


Figure 7. Tracking performance of PPC-Dual-RISE under Case 1.

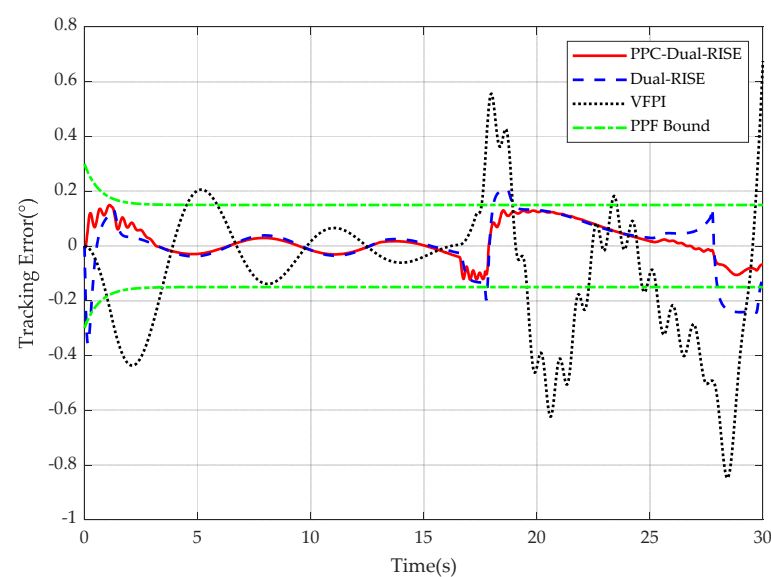


Figure 8. Comparative evaluation of tracking error performance under Case 1.

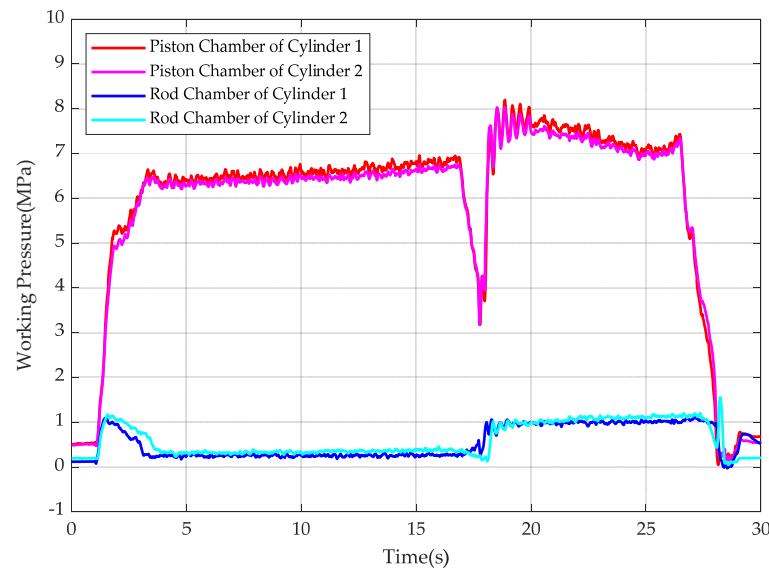


Figure 9. Working pressure under PPC-Dual-RISE under Case 1.

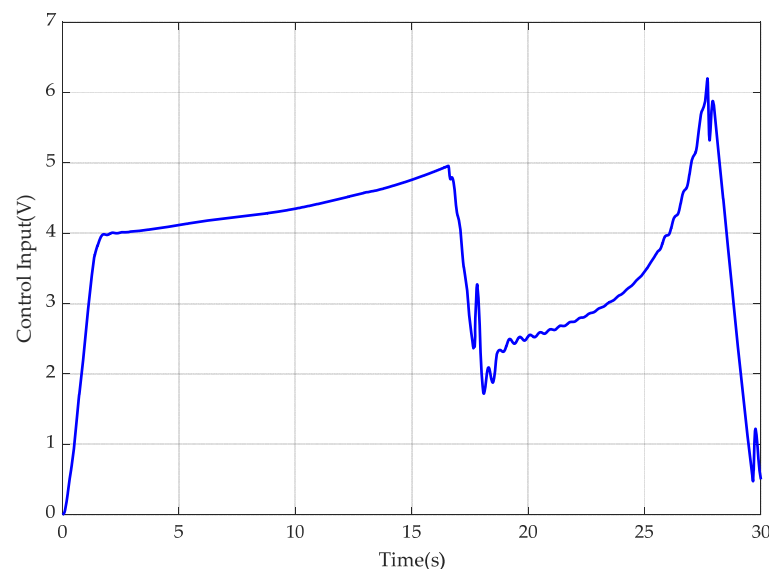


Figure 10. Control input of PPC-Dual-RISE under Case 1.

Case 2: To evaluate the dynamic response and boundary protection capabilities of the proposed strategy under increased operational speeds, a 15 s rapid erecting trajectory is implemented. In this scenario, the operating velocity is doubled relative to Case 1. Consequently, higher dynamic loads, amplified transient impact forces during cylinder stage-change transitions, and increased friction nonlinearities are introduced into the electro-hydrostatic actuator. This test condition evaluates the disturbance rejection and error-bounding properties of the control law under elevated dynamic demands.

The derived experimental results are illustrated in Figures 11–14. Under this operating condition, the erecting speed and acceleration increase significantly, posing higher requirements for the controller's response speed and robustness. As can be seen from Figure 11, PPC-Dual-RISE maintains superior angle tracking accuracy. Figure 12 displays the comparative tracking error curves for the three controllers under this case, while Table 4 summarizes the corresponding quantitative performance indices. It is evident that the tracking error of PPC-Dual-RISE is consistently maintained within the designated performance envelopes throughout the entire erecting process. Furthermore, all performance indices of PPC-Dual-RISE are superior to those of Dual-RISE and VFPI. At the stage-

change instant ($t \approx 8.5\text{s}$), doubling the erecting velocity magnifies the non-smooth collision force and abrupt area variations into a severe energy shock. Consequently, the unconstrained standard Dual-RISE controller suffers a massive transient deviation that violates the performance boundaries. In sharp contrast, the proposed PPC-Dual-RISE controller consistently bounds the tracking error within the PPF envelope. This demonstrates that the nonlinear error transformation layer serves as an effective impact-absorbing barrier, seamlessly suppressing high-frequency transient mutations under high-speed switching constraints. The working pressure in this case are shown in Figure 13, which demonstrates that the output of the hydraulic cylinder can still remain stable without significant oscillations under high-speed operating conditions. Figure 14 presents the control input of PPC-Dual-RISE, which remains smooth and continuous throughout the operation. Hence, Case 2 further validates the efficacy of the developed control scheme.

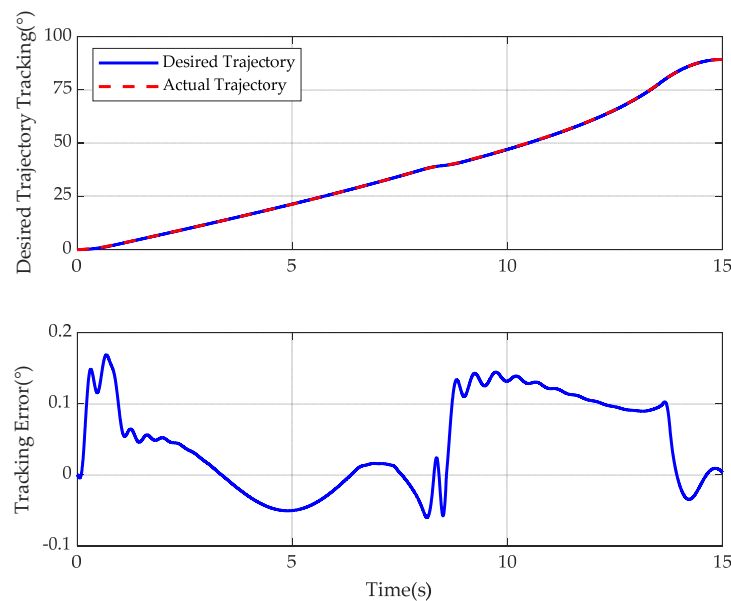


Figure 11. Tracking performance of PPC-Dual-RISE under Case 2.

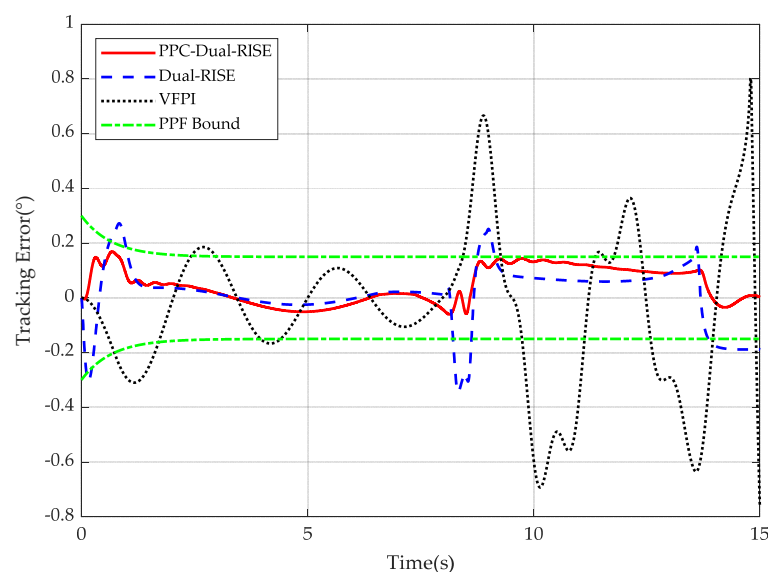


Figure 12. Comparative evaluation of tracking error performance under Case 2.

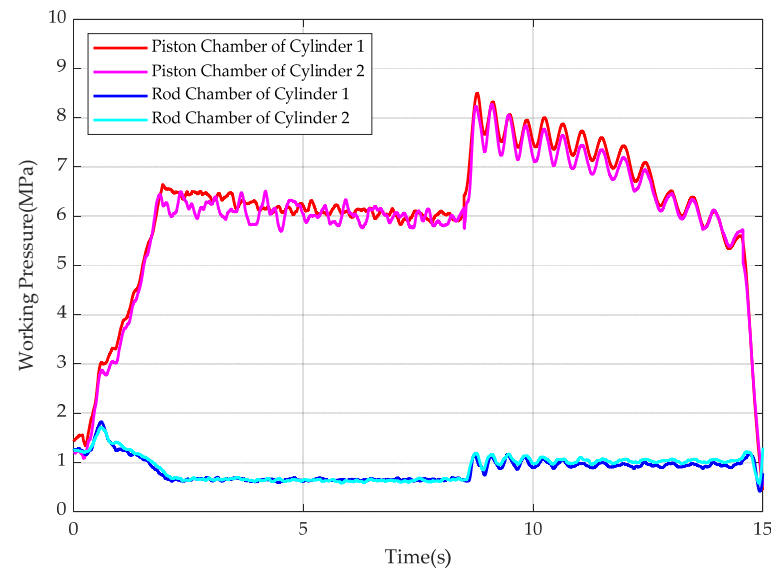


Figure 13. Working pressure under PPC-Dual-RISE under Case 2.

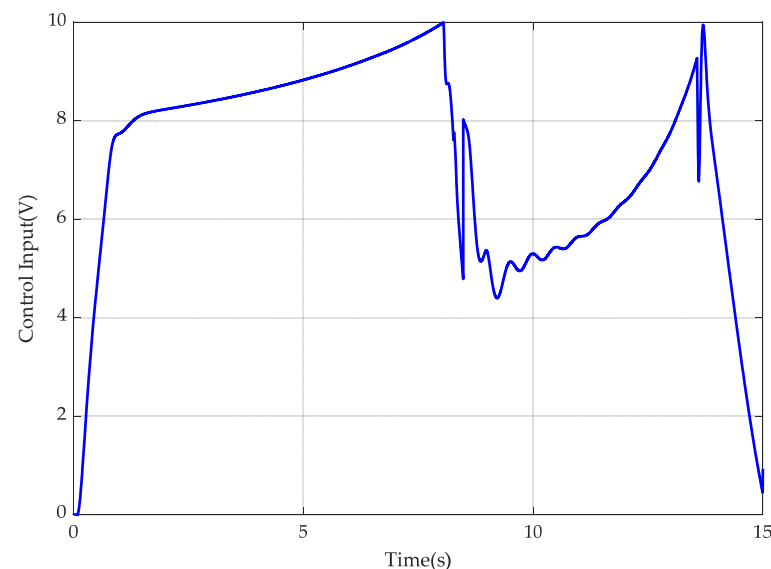


Figure 14. Control input of PPC-Dual-RISE under Case 2.

Table 4. Performance indices of the erecting angle tracking in Case 2.

Indices	Max (°)	Mean (°)	Std (°)
PPC-Dual-RISE	0.1689	0.0609	0.0470
Dual-RISE	0.3635	0.0764	0.0697
VFPI	0.8034	0.2109	0.1860

To evaluate the validity of the synchronized modeling simplification under real-world conditions, the physical differences between the dual cylinders were continuously recorded across both test scenarios. Geometric synchronization was strictly maintained by the rigid mechanical structure. In terms of fluid dynamics, the maximum pressure difference between the dual piston chambers remained under 0.25 MPa during the steady erecting phases of Case 1. During the high-speed velocity mutation in Case 2, this differential pressure spiked transiently to 0.67 MPa at the non-smooth stage-change instant due to minor internal leakage and area transition mismatches. Because these asymmetric loads and differential pressures are entirely bounded, the distributed integral signum loops successfully dominated the resulting cross-coupled torsional disturbances. This experimental

confirmation proves that the single equivalent thrust simplification and the lumped handling of stage-change discontinuities remain accurate and effective for high-precision erecting control applications.

In previous studies focusing on adaptive robust control [9,15] and disturbance-observer-based hydraulic systems [16,26], tracking errors typically converge to a residual steady-state band under standard operating loads, often exhibiting significant transient spikes exceeding 1.0° during non-smooth mechanical transitions. As evidenced by quantitative indices in Tables 3 and 4, the proposed PPC-Dual-RISE framework achieves a significantly superior steady-state tracking precision, with an arithmetic mean error of 0.0487° in Case 1 and 0.0609° in Case 2, accompanied by standard deviations well below 0.05° . Furthermore, while tracking errors are successfully restricted within predefined envelopes by standard prescribed performance control (PPC) and funnel-based strategies reported in [24,26], high-frequency control input chattering or transient boundary violations are frequently suffered when fast velocity mutations and non-smooth cylinder stage-change impacts are encountered. By integrating the nonlinear error transformation layer with a distributed dual-loop RISE structure, semi-global asymptotic stability is mathematically guaranteed, and severe mechanical energy shocks are successfully absorbed without the safety envelope being violated. Superior disturbance rejection, stricter transient boundary protection, and smoother control effort relative to existing state-of-the-art nonlinear hydraulic controllers are confirmed by this comparative evaluation.

5. Conclusions

In this work, an asymptotic prescribed performance control strategy based on the Dual-RISE scheme is proposed for the electro-hydrostatic multi-link erecting system. First, the precise system dynamic model was established by analyzing the kinematic relationships of the complex multi-link mechanism and the pressure-flow dynamics of the two-stage hydraulic cylinder. Next, by integrating a prescribed performance function (PPF) into the dual integral robust control framework, the erecting tracking error was effectively enforced to remain within predefined boundaries, ensuring semi-global asymptotic stability. Finally, the superiority of the developed controller was substantiated through comparative experiments under both medium-speed and high-speed operating conditions. It is evidenced from the experimental results that the PPC-Dual-RISE scheme significantly outperforms standard Dual-RISE and VFPI methods in suppressing unmodeled disturbances and uncertainties. However, the tracking accuracy and robust capabilities of the presented control architecture currently rely on the selection of multiple positive feedback gains and prescribed boundary parameters. In the future, integrating an adaptive parameter adjustment mechanism to reduce the complexity of manual tuning may be a valuable direction for study.

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