Valve Deadzone/Backlash Compensation for Lifting Motion Control of Hydraulic Manipulators

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Abstract: In this paper, a novel nonlinear model and high-precision lifting motion control method of a hydraulic manipulator driven by a proportional valve are presented, with consideration of severe system nonlinearities, various uncertainties as well as valve backlash/deadzone input non-linearities. To accomplish this mission, based on the independent valve orifice throttling process, a new comprehensive pressure-flow model is proposed to uniformly indicate both the backlash and deadzone effects on the flow characteristics. Furthermore, in the manipulator lifting dynamics, considering mechanism nonlinearity and utilizing a smooth LuGre friction model to describe the friction dynamics, a nonlinear state-space mathematical model of hydraulic manipulation system is then established. To suppress the adverse effects of severe nonlinearities and uncertainties in the system, a high precision adaptive robust control method is proposed via backstepping, in which a projection-type adaptive law in combination with a robust feedback term is conducted to attenuate various uncertainties and disturbances. Lyapunov stability analysis demonstrates that the proposed control scheme can acquire transient and steady-state close-loop stability, and the excellent tracking performance of the designed control law is verified by comparative simulation results.

Keywords: motion control; adaptive robust control; hydraulic proportional valve; modeling

1. Introduction

Hydraulic systems have been extensively applied in heavy manipulators [1–4] due to their good capabilities, such as high power/weight ratio, large output force, etc., especially in the operation of load-lifting with heavy load gravity. However, the control of electro-hydraulic systems is very challenging because the hydraulic system involves not only severe nonlinearities, such as mechanism nonlinearity, flow nonlinearity and friction nonlinearity, but also various modeling uncertainties (i.e., parametric uncertainties and uncertain nonlinearities). Thus, the traditional linear control methods have become more and more difficult to satisfy the demand of modern hydraulic systems. To cope with these issues, dozens of advanced nonlinear controllers have been proposed, such as feedback linearization control [5–7], nonlinear adaptive control [8–10] and adaptive robust control [11–13].

In most hydraulic control applications, nozzle-flapper-based servovalves are utilized as the control elements by virtue of fast response and high precision. Although servovalves have good controllability, their own defects are also obvious, that is, they are expensive and prone to malfunction due to fluid contamination. As a result, the electro-hydraulic systems controlled by servovalves have low reliability and poor economy. In recent decades, the proportional servo technology, using proportional valves as the control elements, has developed rapidly and received widespread attention in industrial fields. Compared to servovalves, proportional valves possess various advantages. They are much cheaper and have higher reliability because less accuracy is required in the manufacturing of spool and valve sleeve, and vulnerable nozzle-flapper structure is replaced by proportional...
elements. The electric feedback form is also employed in proportional valves to construct their internal closed-loop control, so they are less sensitive to oil contamination and simultaneously retain satisfactory servo control performance. Thereby, they are more suitable for industrial environments.

However, because of less precise manufacturing accuracy, proportional valves usually encounter backlash/deadzone input nonlinearities existing between the valve spool and the valve sleeve, which are one of the most typical features different from servovalves. Generally, the process of manufacturing proportional valves can be divided into two modes, namely, the backlash mode and the deadzone mode. In the backlash mode, there is a backlash between the valve spool and the valve sleeve, which may raise stability concerns due to the increased flow gain near neutral spool position. The deadzone mode, meaning that there exists overlapped part between the valve spool and the valve sleeve, may result in response to lag because a small control input cannot drive the spool in neutral spool position. Practice reveals that the existence of a backlash/deadzone may lead to reduced control performance, even instability. Furthermore, valve backlash/deadzone input nonlinearities, coupled with other system nonlinearities and various uncertainties, make it harder to control the hydraulic systems controlled by proportional valves. Thus, how to effectively compensate the influence of valve backlash/deadzone input nonlinearities in the presence of severe nonlinearities and modeling uncertainties has always been a practically important problem.

Regretfully, there are few works considering valve backlash nonlinearities, so the following discussion mainly focuses on the existing control method of valve deadzone compensation. As pointed out in [14], there are generally two kinds of methods to alleviate the effects of deadzone in the literature. One is to model the deadzone as a combination of a linear control input with a constant/time-varying gain (for symmetric/asymmetric deadzone, respectively) and a bounded disturbance-like term, examples like in [15–18]. In these control algorithms, the disturbance-like term was lumped into uncertain nonlinearities and suppressed by various robust feedback terms, which was likely to arise reduced control performance. Furthermore, when the practical hydraulic system encounters severe valve deadzone, high control gains have to be implemented in the robust feedback control, which might amplify the noise and even lead to instability. The other way of minimizing the deadzone effects is to construct a deadzone inverse based active compensation control, such as in [19–22]. In [19], an integrated direct/indirect adaptive robust controller was exploited for an electrohydraulic manipulator with an unknown valve deadzone via a deadzone inverse, but it is worth pointing out that its employed deadzone inverse was discontinuous, which might result in the control input chattering. Motivated by this issue, a smooth deadzone inverse function was developed to compensate deadzone effects and an adaptive controller was proposed for uncertain nonlinear systems [20]. Unfortunately, both Refs. [19,20] can only yield ultimately bounded tracking results. Hence, Deng et al. developed a robust adaptive control strategy for hydraulic systems with valve deadzone nonlinearities and modeling uncertainties, which can theoretically obtain asymptotic tracking performance [21]. However, the nonlinear characteristics of valve deadzone and backlash characteristics are not considered together in the modeling of valve pressure-flow characteristics in all above-mentioned deadzone compensation control approaches, which limits its application in electro-hydraulic proportional servo systems.

Based on the above analysis, it can be concluded that there still remain some open issues in the research of high-performance control methods for hydraulic manipulators controlled by proportional valves: (1) The mathematical model is an important foundation for the research of nonlinear control methods, whose accuracy directly determines the achievable precision of the closed-loop control. Thereby, for electrohydraulic proportional servo systems, how to establish a proper valve pressure-flow model with both backlash/deadzone input nonlinearities possibility in a unified framework, and how to employ a nonlinear dynamic friction model which also plays an important role in achieving high accuracy manipulation, and then set up an accurate system model, is still a pending
issue. (2) In addition to system nonlinearities and valve backlash/deadzone input nonlinearities, electrohydraulic proportional servo systems also encounter many modeling uncertainties, which further restrict the improvement of servo performance, so how to design an appropriate high-performance control strategy is a remaining unsettled problem. (3) The hydraulic manipulator has typical features of severe mechanism nonlinearity and eccentric load. These properties are coupled with the inherent nonlinearities and uncertainties in the electrohydraulic proportional servo systems, which further complicates the controller design.

In this paper, a novel nonlinear model and high-precision control strategy of hydraulic manipulator driven by a proportional valve is presented. Not only system nonlinearities and modeling uncertainties, but also valve backlash/deadzone input nonlinearities, are taken into account. By creatively utilizing the Boolean nonlinear function, a new comprehensive pressure-flow model is proposed to uniformly indicate both the backlash and deadzone effects of the proportional valve. Based on this new model, the pressure dynamics of the hydraulic actuator are established. In terms of the manipulator dynamics, the smooth dynamic LuGre friction model [23] is combined. Meanwhile, with consideration of mechanism nonlinearity and the eccentric load of the hydraulic manipulator, the manipulator dynamics is then built up. By selecting appropriate system states, the system state space equation is then obtained. In addition, to alleviate various modeling uncertainties, by referring to the adaptive robust control method [24], a projection-type adaptive law is designed to compensate parametric uncertainties, whose designing core is to limit the parameter estimate within the reset range and ensure the bounds of estimation errors are calculatable via projection mapping. Additionally, a robust feedback term is utilized to restrain the uncertain nonlinearities (e.g., modeling errors and external disturbances). Lyapunov stability analysis indicates the proposed control strategy can obtain transient and steady-state close-loop stability. Comparative simulation results are acquired to further demonstrate its priority.

2. Dynamic Model and Problem Formulation

In this paper, the considered lifting degree of the hydraulic manipulator is illustrated in Figure 1.

![Figure 1. The motion schematic diagram of hydraulic lifting manipulator.](image-url)
2.1. Manipulator Dynamics

Figure 1 shows two states of the manipulator’s arm in motion. In state 1, the manipulator is in a horizontal position and state 2 is any state during the motion. As shown in Figure 1, \( q \) represents the angular displacement of the robot arm, within the range of \( 0^\circ \sim 90^\circ \); \( O \) is the rotation center; \( O_1 \) and \( O_2 \) represent the rotation centers of the upper and lower ears of the hydraulic cylinder, respectively; \( O_3 \) is the center of gravity of the manipulator, and let \( OO_2 = L_1, O_1O_2 = L_2, OO_1 = L_3, OO_3 = L_4, O_1'0_2 = L_v, \angle O_1OO_3 = \beta_0, \angle O_1'O_2O = \theta_0, \angle O_1'O_1O_2 = \varphi \).

The existence of nonlinear friction seriously affects the system control performance. If only the linear friction model or the simple nonlinear friction model are used to describe the linear friction torques in the system are lumped into the total friction torque term \( f(t) \). The use of nonlinear friction seriously affects the system control performance. If only the linear friction model or the simple nonlinear friction model are used to describe the nonlinear friction torques in the system, the system is not accurate enough which will increase the time-varying disturbances. Given that the traditional LuGre dynamic friction model is non-differentiable, it is not applicable to the unmatched friction compensation control for hydraulic systems. Therefore, most of the literature uses a simplified approximate smooth friction model. To achieve better tracking performance, according to the analysis in [23], the following more accurate friction model is obtained:

\[
f(t) = \sigma_0 z + \sigma_1 \dot{z} + \sigma_2 \dot{q}
\]

where \( \sigma_0, \sigma_1 \) and \( \sigma_2 \) are friction force parameters, which can be physically interpreted as the stiffness of the bristles between two contact surfaces, damping coefficient of the bristles, and viscous coefficient, respectively; the unmeasurable internal friction state \( z \).
physically stands for the average deflection of the bristles between two contact surfaces, and its dynamics is given by

\[ \dot{z} = \dot{q} - N(\dot{q})z \]  

where the nonlinear function \(N(\dot{q})\) is modeled by

\[ N(\dot{q}) = \frac{\dot{q}}{(f_s - f_c)\tanh(c_1\dot{q}) - \tanh(c_2\dot{q})} + f_c\tanh(c_3\dot{q}) \]  

in which \(f_c\) and \(f_s\) denote the level of the normalized Coulomb friction and stiction force, respectively; \(c_1, c_2, c_3\) are various shape coefficients to approximate various friction effects.

The core of the aforementioned friction model is to construct the friction nonlinear Striebeck effect by a sufficiently smooth \(\tanh\) function \([23,25]\), which makes the dynamic friction model described in Equation (6) smooth enough while accurately describing various real friction effects. It is conducive to specifically compensate the nonlinear friction effects in the subsequent backstepping design.

**Remark 1.** In practical applications, the hydraulic valves are usually controlled with the dither technique, which compensates the nonlinear friction passively. Since the friction dynamics have been carefully considered in this paper, the nonlinear friction can be compensated actively. Therefore, the dither technique is not utilized in this work.

2.3. Pressure Dynamics

Considering the oil compressibility in the actuator, the pressure dynamics in both chambers can be written as

\[
\begin{align*}
\frac{V_1(q)}{\beta} \dot{P}_1 &= -A_1 \frac{\partial P}{\partial q} \dot{q} - C_i (P_1 - P_2) + Q_1 + d_{21}(t) \\
\frac{V_2(q)}{\beta} \dot{P}_2 &= +A_2 \frac{\partial P}{\partial q} \dot{q} + C_i (P_1 - P_2) - Q_2 - d_{22}(t)
\end{align*}
\]  

where \(V_1(q) = V_{01} + A_1 x_p\), \(V_2(q) = V_{02} - A_2 x_p\) are the control volume of the forward and return chambers, respectively; \(V_{01}\) and \(V_{01}\) are the original total volumes of the two chambers; \(\beta\) is the effective oil bulk modulus; \(C_i\) is the internal leakage coefficient; \(Q_1\) is the supplied oil flow to the forward chamber and \(Q_2\) is the return oil flow to the return chamber, and both of them are positive; \(d_{21}(t)\) and \(d_{22}(t)\) are the modeling errors in the dynamics of the two chambers, including complex unmodeled leakage, valve dynamics, etc.

2.4. Flow Characteristics

The valve backlash/deadzone characteristics of the proportional valve are a vital nonlinear factor to be reckoned with in hydraulic systems. If it is ignored in the modeling of valve flow characteristics and simply handled by robustness, reduced control accuracy will be yielded. To this end, a new comprehensive pressure-flow equation is exploited to uniformly indicate both the backlash and deadzone nonlinearities of the proportional valve in this paper.

Taking backlash in the valve ports as an example, the schematic diagram of the proportional valve flow in the valve ports is shown in Figure 2. According to the actual orifice flow equation proposed in [26], and combining with the results in [27], the supplied flow to the hydraulic forward chamber can be designed as

\[
Q_1 = \begin{cases} 
  k_q(x_0 + \varepsilon)\sqrt{|P_2 - P_1|\text{sgn}(P_2 - P_1)} & \text{if } x_0 > \varepsilon \\
  -k_q(-x_0 + \varepsilon)\sqrt{|P_1 - P_2|\text{sgn}(P_1 - P_2)} & \text{if } -\varepsilon \leq x_0 \leq \varepsilon \\
  -k_q(-x_0 + \varepsilon)\sqrt{|P_1 - P_2|\text{sgn}(P_1 - P_2)} & \text{if } x_0 < -\varepsilon 
\end{cases}
\]  

where

\[ k_q = C_d \sqrt{\frac{2}{\rho}} \]
To introduce the subsequent controller design more clearly, we assume that the backlashes of the proportional valve between the two orifice sides and their corresponding valve sleeve are identical. Denote the backlash as $\varepsilon > 0$. If the backlashes between the two sides of the proportional valve are unequal, the modeling approach is parallel. $P_s$ and $P_r$ are the supplied and return pressures, respectively; $k_q$ is the flow gain; $x_v$ is the spool displacement of the proportional valve; $C_d$ is the discharge coefficient; $w$ is the area gradient of the orifice; $\rho$ is the density of the oil.

To obtain a comprehensive pressure-flow model in the following controller design, two nonlinear functions are proposed:

$$R_1 = \sqrt{|P_s - P_1|} \text{sgn}(P_s - P_1)$$
$$R_2 = \sqrt{|P_1 - P_r|} \text{sgn}(P_1 - P_r)$$

(12)

and a nonlinear Boolean function is defined as

$$s(*) = \begin{cases} 1 & \text{if } * \geq 0 \\ 0 & \text{if } * < 0 \end{cases}$$

(13)

Hence, the supplied flow $Q_1$ can be uniformly expressed as

$$Q_1 = k_q x_v [R_1 s(x_v + \varepsilon) + R_2 s(-x_v + \varepsilon)] + k_q \varepsilon [R_1 s(x_v + \varepsilon) - R_2 s(-x_v + \varepsilon)]$$

(14)

To further synthesize the flow equation, the following function is defined.

$$W_1 = R_3 s(x_v + \varepsilon) + R_4 s(-x_v + \varepsilon)$$
$$W_2 = k_q \varepsilon [R_1 s(x_v + \varepsilon) - R_2 s(-x_v + \varepsilon)]$$

(15)

Then the resulting new comprehensive pressure-flow model can be obtained as

$$Q_1 = k_q W_1 x_v + W_2$$

(16)

The return flow $Q_2$ has a similar form as:

$$Q_2 = k_q W_3 x_v + W_4$$

(17)

where the four nonlinear functions are defined as

$$W_3 = R_5 s(x_v + \varepsilon) + R_6 s(-x_v + \varepsilon)$$
$$W_4 = k_q \varepsilon [R_3 s(x_v + \varepsilon) - R_4 s(-x_v + \varepsilon)]$$
$$R_3 = \sqrt{|P_2 - P_1|} \text{sgn}(P_2 - P_1)$$
$$R_4 = \sqrt{|P_3 - P_2|} \text{sgn}(P_3 - P_2)$$

(18)

Since the system frequency is much lower than the valve frequency, the proportional valve dynamics can be neglected. Hence, it can be assumed that the control applied to the
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3.1. Design Model and Issues to Be Addressed

In general, the considered system is subjected to parametric uncertainties due to the variations of \( \sigma_0, \sigma_1, \sigma_2, k_i, \beta, C_{i} \). Define an unknown parameter vector as \( \theta = [\theta_1, \theta_2, \theta_3, \theta_4, \theta_5, \theta_6]^T \), where \( \theta_1 = \sigma_0, \theta_2 = \sigma_1, \theta_3 = \sigma_2, \theta_4 = k_i \beta, \theta_5 = \beta, \theta_6 = C_i \beta \). Define a set of state variables as \( x = [x_1, x_2, x_3, x_4]^T = [q, \dot{q}, p_1, p_2]^T \). Utilizing these state variables, the system described by Equations (5), (6), (9), (19) can be expressed as

\[
\begin{align*}
\dot{x}_1 &= x_2 - N(x_2)z \\
\dot{x}_2 &= x_3 - \bar{A}_c x_4 - f_1 - \theta_1 f_2 z + \theta_2 f_4 z - \theta_3 f_5 + D_1(t) \\
\dot{x}_3 &= \frac{\dot{q} W_1}{V_2} u - \frac{\dot{p} W_1}{V_2} - A_1 \frac{\partial x_p}{\partial x_1} x_2 - \frac{\partial x_p}{\partial x_1} (x_3 - x_4) + q_1(t) \\
\dot{x}_4 &= -\frac{\dot{q} W_1}{V_2} u + \frac{\dot{p} W_1}{V_2} A_2 \frac{\partial x_p}{\partial x_1} x_2 + \frac{\partial x_p}{\partial x_1} (x_3 - x_4) - q_2(t)
\end{align*}
\]

where \( \bar{A}_c = \frac{A_z}{A_1} \), the nonlinear functions are written as

\[
\begin{align*}
f_1 &= \frac{f}{A_1} \left( \frac{\partial x_p}{\partial x_1} \right)^{-1} f_2 &= \left( \frac{\partial x_p}{\partial x_1} \right)^{-1} \frac{mg L_4}{A_1} \cos(q + \beta_0), f_3 &= \left( \frac{\partial x_p}{\partial x_1} A_1 \right)^{-1}, \\
f_4 &= \left( \frac{\partial x_p}{\partial x_1} \right)^{-1} N(x_2) A_1, f_5 &= \left( \frac{\partial x_p}{\partial x_1} \right)^{-1} \frac{x_3}{A_1}
\end{align*}
\]

The modeling errors are

\[
D_1(t) = \frac{\partial x_p}{\partial x_1} f_1(t) A_1, q_1(t) = \frac{d_{11}(t) \beta}{V_1}, q_2(t) = \frac{d_{22}(t) \beta}{V_2}.
\]

The control objective is: Given a desired motion trajectory \( q(t) = x_{id}(t) \), design a bounded control input \( u \) such that the system output \( x_1 \) can track \( x_{id} \) as closely as possible despite of system nonlinearities, various modeling uncertainties as well as valve deadzone/backlash nonlinearities.
Assumption 1. The desired motion trajectory $x_{id} \in C^3$ and bounded; in practical hydraulic systems under normal working conditions, $P_1$ and $P_2$ are always bounded by $P_s$ and $P_r$ (i.e., $0 \leq P_r < P_1 < P_s$, $0 \leq P_2 < P_s$).

Assumption 2. The unknown parameter set $\theta$ satisfies

$$\theta \in \Omega_\theta = \{\theta : \theta_{\min} \leq \theta \leq \theta_{\max}\}$$

where $\theta_{\max} = [\theta_{1\max}, \ldots, \theta_{6\max}]^T$, $\theta_{\min} = [\theta_{1\min}, \ldots, \theta_{6\min}]^T$ are the known upper and lower bounds; and the unmodeled disturbances $D_1(t), D_2(t)$ are bounded by

$$|D_1(t)| \leq \delta_1$$
$$|D_2(t)| \leq \delta_2$$

where $\delta_1, \delta_2$ are the upper bounds of $D_1(t)$ and $D_2(t)$, respectively; and

$$D_2(t) = q_1(t) + \overline{A}q_2(t)$$

3.2. Projection Mapping and Parameter Adaptation

In the following sections, $\cdot_i$ denotes the $i$th element of the vector $\cdot$, and the operation $<$ for two vectors is performed in terms of the corresponding elements of the vectors.

Define $\hat{\theta}$ as the estimate of $\theta$, and $\tilde{\theta} = \hat{\theta} - \theta$ as the estimation error. To ensure the stability of the adaptation law and limit the parameter estimation within the range defined in Equation (21), a discontinuous projection can be represented as [13]

$$\text{Proj}_{\theta_i}(\cdot) = \begin{cases} 
0 & \text{if } \hat{\theta}_i = \theta_{\max} \text{ and } \cdot_i > 0 \\
0 & \text{if } \hat{\theta}_i = \theta_{\min} \text{ and } \cdot_i < 0 \\
\cdot_i & \text{otherwise}
\end{cases}$$

where $i = 1, \ldots, 6$. Then the following adaptation law is given by

$$\dot{\hat{\theta}} = \text{Proj}_{\theta}(\Gamma \tau) \tilde{\theta}(0) \in \Omega_\theta$$

in which $\text{Proj}_{\theta}(\cdot) = [\text{Proj}_{\theta_1}(\cdot_1), \ldots, \text{Proj}_{\theta_6}(\cdot_6)]^T$; $\Gamma > 0$ is a positive diagonal adaptation rate matrix; $\tau$ is an adaptation function to be synthesized later. For any adaptation function $\tau$, the discontinuous projection used in Equation (24) satisfies [13]

$$\dot{\hat{\theta}} \in \Omega_\theta = \{\hat{\theta} : \theta_{\min} \leq \hat{\theta} \leq \theta_{\max}\}$$
$$\dot{\hat{\theta}}^T[\Gamma^{-1}\text{Proj}_{\theta}(\Gamma \tau) - \tau] \leq 0, \forall \tau.$$

Except for the unknown parameter set $\theta$, the friction state $z$ in Equation (20) is neither known nor measurable. Hence, a state observer has to be designed to estimate the value of $z$. In order to deal with the different characteristics of the unmeasurable friction state $z$ between $\theta_{1f}z$ and $\theta_{2f}z$, according to [23], the following dual state observer is constructed

$$\hat{z}_1 = \text{Proj}_{\eta_1}(\eta_1), \hat{z}_2 = \text{Proj}_{\eta_2}(\eta_2)$$

where $\hat{z}_1$ and $\hat{z}_2$ are the estimates of the two unmeasurable friction state $z$; $\eta_1$ and $\eta_2$ are two learning functions to be synthesized later; the projection mapping in Equation (27) is given as

$$\text{Proj}_{\eta_i}(\eta_i) = \begin{cases} 
0 & \text{if } \hat{z}_i = z_{\max} \text{ and } \eta_i > 0 \\
0 & \text{if } \hat{z}_i = z_{\min} \text{ and } \eta_i < 0 \\
\eta_i & \text{otherwise}
\end{cases}$$

where the observation range corresponds to the physical bounds of $z$ (i.e., $z_{\max} = f_s$, $z_{\min} = -f_s$).
Similar to the adaptation law, for any learning functions $\eta_1$ and $\eta_2$, the projection mapping in Equation (28) guarantees [23]

$$
\begin{align*}
z_{1\min} \leq \hat{z}_1 \leq z_{1\max}, & \quad \hat{z}_{2\min} \leq \hat{z}_2 \leq z_{2\max} \\
\hat{z}_1[\hat{z}_1 - \eta_1] \leq 0, & \quad \hat{z}_2[\hat{z}_2 - \eta_2] \leq 0
\end{align*}
$$

(29)

where $\hat{z}_1 = z_1 - z$, $\hat{z}_2 = z_2 - z$ are the estimation errors, which have the following dynamics:

$$
\begin{align*}
\dot{\hat{z}}_1 &= \hat{z}_1 - \hat{z} = \text{Proj}_{[\eta_1]}(\eta_1) - [x_2 - N(x_2) z] \\
\dot{\hat{z}}_2 &= \hat{z}_2 - \hat{z} = \text{Proj}_{[\eta_2]}(\eta_2) - [x_2 - N(x_2) z]
\end{align*}
$$

(30)

3.3. Controller Design

To handle parametric uncertainties and uncertain nonlinearities, the nonlinear robust controller is designed based on the classical backstepping technique [13].

Step 1: Define a set of variables as

$$
\epsilon_2 = \epsilon_1 + k_1 \epsilon_1 = x_2 - a_1, \quad a_1 = x_{1d} - k_1 \epsilon_1
$$

(31)

where $\epsilon_1 = x_1 - x_{1d}$ is the output tracking error; $k_1$ is any positive feedback gain; $a_1$ is the virtual control input of $x_2$, $\epsilon_2$ is the deviation between them; differentiating Equation (31) and noting Equation (20), we have

$$
f_1 \epsilon_2 = -f_1 \dot{\epsilon}_1 + x_3 - \overline{A}_c x_4 - f_2 - \theta_1 f_3 z + \theta_2 f_4 z - \theta_3 f_5 + D_1(t)
$$

(32)

Define the load pressure as $P_L = x_3 - \overline{A}_c x_4$. According to Equation (32), $P_L$ can be treated as the virtual control input and simultaneously design a virtual control function $a_2$ for it. Additionally, define the input discrepancy as $\epsilon_3 = P_L - a_2$ then the virtual control law $a_2$ can be designed as

$$
a_2 = a_{2d} + a_{2s}, \quad a_{2d} = a_{2d1} + a_{2d2}, \quad a_{2s} = f_1 \dot{\epsilon}_1 + f_2 + \hat{\theta}_1 f_3 \hat{z}_1 - \hat{\theta}_2 f_4 \hat{z}_2 + \hat{\theta}_3 f_5
$$

$$
a_{2s1} = -k_2 \epsilon_2
$$

(33)

where $k_2$ is a positive feedback gain; $a_{2d}$ is the model compensation with the parameter estimates $\hat{\theta}$; $a_{2s}$ is the robust feedback control consisting of the linear term $a_{2s1}$ and the nonlinear term $a_{2s2}$. Substituting Equation (33) and the expression of $\epsilon_3$ into Equation (32), we then acquire

$$
f_1 \epsilon_2 = -k_2 \epsilon_2 + \epsilon_3 + a_{2s2} - \hat{\theta}^T \varphi_2 + \theta_1 \hat{z}_1 f_3 - \theta_2 \hat{z}_2 f_4 + D_1(t)
$$

(34)

where the regressor $\varphi_2$ is written as

$$
\varphi_2 \triangleq [-\hat{z}_1 f_3, \hat{z}_2 f_4, -f_5, 0, 0, 0]^T
$$

(35)

To handle parametric uncertainties and uncertain nonlinearities, the nonlinear robust feedback term is constructed as

$$
a_{2s2} = -\left[\frac{||\theta_M|| ||\varphi_2|| + \theta_1 M f_3 |f_3| + \theta_2 M f_4 |f_4| + \delta_1}{4 \epsilon_2} \right]^2 e_2
$$

(36)

where $\theta_M = \theta_{\max} - \theta_{\min}$, $\theta_1 M = \theta_{1\max} - \theta_{1\min}$, $\theta_2 M = \theta_{2\max} - \theta_{2\min}$, $z_M = z_{\max} - z_{\min}$, $\epsilon_2$ is a positive design parameter. Then the above designed $a_{2s2}$ satisfies:

$$
a_{2s2} e_2 \leq 0
$$

$$
e_2 [-\hat{\theta}^T \varphi_2 + \theta_1 \hat{z}_1 f_3 - \theta_2 \hat{z}_2 f_4 + D_1(t) + a_{2s2}] \leq \epsilon_2
$$

(37)
Step 2: According to the definition of $z_3$, its dynamics can be represented as

$$\dot{e}_3 = \theta_4 f_6 u + \theta_5 f_7 - \theta_6 f_8 - \hat{a}_2 + D_2(t) \quad (38)$$

where

$$f_6 = \frac{\dot{W}_8}{V_4} + \frac{\dot{W}_9}{V_5} f_7 = \left( \frac{\dot{W}_8}{V_4} + \frac{\dot{W}_9}{V_5} \right) - \left( \frac{\dot{\theta}_4}{V_4} + \frac{\dot{\theta}_5}{V_5} \right) \frac{\partial x_2}{\partial x_1} x_2$$

$$f_8 = \frac{1}{V_7} + \frac{1}{V_8} (x_3 - x_4), \quad D_2(t) = q_1(t) + \tilde{A}_e q_2(t)$$

For a clearer presentation, in this paper, the first derivative of $\dot{z}_3$, its dynamics can be represented as

$$\dot{z}_3 = \theta_4 f_6 u + \theta_5 f_7 - \theta_6 f_8 - \hat{a}_2 + D_2(t) \quad (38)$$

where

$$u = u_a + u_s, \quad u_a = \frac{1}{\hat{a}_4 f_6} (\hat{a}_2 - \hat{\theta}_5 f_7 + \hat{\theta}_6 f_8)$$

$$u_s = \frac{1}{\hat{a}_4 f_6} (u_{s1} + u_{s2}), \quad u_{s1} = -k_3 e_3 \quad (39)$$

where $u_a$ is the model compensation term with online parameter adaptation; $u_{s1}$ is the negative linear feedback term with gain $k_3 > 0$ to stabilize the nominal model of the system; $u_{s2}$ is the nonlinear robust feedback term to handle modeling uncertainties in Equation (38). Combining Equation (39), we can rewrite Equation (38) as

$$\dot{e}_3 = -\frac{\theta_4}{\hat{a}_4 f_6} k_3 e_3 - \tilde{\theta}^T q_3 + \frac{\theta_4}{\hat{a}_4 f_6} u_{s2} + D_2(t) \quad (40)$$

where the regressor for the parameter adaptation $q_3$ is defined as

$$q_3 \triangleq [0, 0, 0, f_6 u_a, f_7, -f_8]^T \quad (41)$$

Based on Equation (40), the robust feedback term $u_{s2}$ is designed as

$$u_{s2} = -\left[ \frac{||\partial_M|| ||q_3|| + \tilde{\theta}_2}{4 \varepsilon_3} \right]^2 e_3 \quad (42)$$

where $\varepsilon_3 > 0$ is an arbitrarily small design parameter. Then $u_{s2}$ satisfies the following conditions:

$$e_3 [-\tilde{\theta}^T q_3 + \frac{\theta_4}{\hat{a}_4 f_6} u_{s2} + D_2(t)] \leq e_3 \quad (43)$$

3.4. Main Results

**Theorem 1.** If the unmodeled disturbances $D_1(t) = D_2(t) = 0$, namely, there only exists parametric uncertainties and friction nonlinearity in the system, choosing large enough feedback gains $k_1, k_2, k_3$, such that the following defined matrix $\Lambda$ is positive definite:

$$\Lambda = \begin{bmatrix}
    k_1 & -\frac{1}{2} & 0 \\
    -\frac{1}{2} & k_2 & -\frac{1}{2} \\
    0 & -\frac{1}{2} & k_3
  \end{bmatrix} \quad (44)$$

and utilizing the discontinuous projection-type adaptation law in Equation (25) and adaptation function $\tau = q_2 e_2 + q_3 e_3$, giving the dual state observer in Equation (27) and learning functions as

$$\eta_1 = x_2 - N(x_2) \tilde{z}_1 - \gamma_1 f_3 e_2$$

$$\eta_2 = x_2 - N(x_2) \tilde{z}_2 - \gamma_2 f_4 e_2 \quad (45)$$
where $\gamma_1$ and $\gamma_2$ are positive learning gains, then the proposed control method (39) can ensure that all the system signals are bounded under closed-loop operation, and asymptotic tracking performance is also obtained (i.e., $t \rightarrow \infty$, $e_1, e_2, e_3 \rightarrow 0$).

**Proof of Theorem 1.** See Appendix A. □

**Theorem 2.** If the system exists unmodeled disturbances (i.e., $D_1(t)$, $D_2(t)$ are not zero at the same time), then the designed control law (39) can guarantee that all system signals are bounded under closed-loop operation. Define the Lyapunov function as

$$V_2(t) = \frac{1}{2}e_1^2 + \frac{1}{2}f_1e_2^2 + \frac{1}{2}e_3^2$$  \hspace{1cm} (46)

According to the definition of the nonlinear function $f_1$, it can be inferred that $f_1$ is always positive within the motion angle of the manipulator. Hence, $V_2$ is positive definite. Additionally, it is bounded by

$$V_2(t) \leq V_2(0) \exp(-\kappa t) + \frac{\epsilon_f}{\kappa}[1 - \exp(-\kappa t)]$$  \hspace{1cm} (47)

where

$$\epsilon_f = \epsilon_e + \epsilon_\gamma, \kappa = \frac{2\lambda_{\min}(A)}{\max\{1, f_{1\max}\}}$$  \hspace{1cm} (48)

in which $f_{1\max}$ is the maximum of $f_1$ within the angular range of the manipulator motion.

In such a case, it can also be obtained that $t \rightarrow \infty$, $e_1, e_2, e_3$ will be bounded by $\sqrt{2\epsilon_f / \kappa}$.

**Proof of Theorem 2.** See Appendix B. □

### 4. Simulation Results and Discussion

To further demonstrate the effectiveness of the proposed control strategy, various simulations are carried out based on the MATLAB/Simulink software. The simulation model parameters of hydraulic manipulation system are chosen as: $P_s = 21$ MPa, $P_f = 0$ MPa, $A_1 = 3.14 \times 10^{-2}$ m$^2$, $A_2 = 1.6 \times 10^{-2}$ m$^2$, $V_{01} = 3.1416 \times 10^{-4}$ m$^3$, $V_{02} = 3.04 \times 10^{-2}$ m$^3$, $j = 1.5 \times 10^5$ kg-m$^2$, $m = 10$ t, $L_1 = 1.6$ m, $L_2 = 2$ m, $L_3 = 3.5$ m, $L_4 = 3$ m. The nonlinear function $N(x_2)$ is selected as $N(x_2) = x_2 / [2 \times 10^{-3} \times [\tanh(15 \times 2) - \tanh(1.5 \times 2)] + 3 \times 10^{-3} \times \tanh(900 \times 2)]$. The simulation step size is set to 0.5 ms and the applied disturbance torque $d_1(t) = 5000\sin(t)$ N-m. The following three controllers are compared:

(1) ARCBF: This is the proposed adaptive robust controller with valve backlash/deadzone compensation and dynamic friction compensation described in Section III. The following control gains are utilized: $k_1 = 200$, $k_2 = 5 \times 10^7$, $k_3 = 80$. The initial estimate of $\theta$ is chosen as $\hat{\theta}_0 = [5 \times 10^5, 1100, 2 \times 10^5, 60, 5 \times 10^8, 0]^T$. The initial estimate of $z$ is $\hat{z}_1(0) = \hat{z}_2(0)$. The bounds of $\theta$ are chosen as $\theta_{\text{max}} = [1 \times 10^7, 1 \times 10^4, 3 \times 10^6, 500, 2 \times 10^8, 0.01]^T$, $\theta_{\text{min}} = [0, 900, 0, 50, 2 \times 10^8, 0]^T$. The bounds of $z$ estimation are $z_{\text{max}} = 5 \times 10^{-3}$, $z_{\text{min}} = -5 \times 10^{-3}$. Friction state learning gains are set at $\Gamma = \text{diag}[2 \times 10^8, 500, 2 \times 10^5, 2 \times 10^{-13}, 100, 1.5 \times 10^{-20}]$. Friction state learning gains are $\gamma_1 = 5 \times 10^{-3}$, $\gamma_2 = 5 \times 10^{-3}$.

**Remark 3.** The simplification suitable for our simulation is made for the selection of the control gains $k_1, k_2$ and $k_3$. We may implement the needed robust control gains in the following two ways. The first method is to pick up a set of values for $k_1, k_2$, and $k_3$ rigorously to ensure the theoretical stringency with various prerequisites. However, it increases the complexity of the resulting control law considerably since it may need a significant amount of offline investigating work, sometimes even be impossible. Alternatively, a pragmatic approach is to simply choose $k_1, k_2$, and $k_3$ large enough without worrying about the specific prerequisites. In this way, prerequisites will be satisfied for a certain set of values of $k_1, k_2$, and $k_3$, at least locally around the desired trajectory to be tracked.
In this paper, the second approach is used since it facilitates the online tuning process of gains in implementation.

(2) ARCF: This is the adaptive robust controller same as the proposed ARCBF controller but without considering backlash/deadzone nonlinearities (i.e., let backlash/deadzone $\varepsilon = 0$ in the controller design). Thereby, it can be utilized to verify the significance of the new comprehensive pressure-flow equation compensated in ARCBF. To ensure fair comparison this controller selects the same control gains as ARCBF.

(3) ARCB: The adaptive robust controller without dynamic friction compensation (i.e., let $\hat{z}_1 = \hat{z}_2 = 0$ with $\gamma_1 = \gamma_2 = 0$). Hence, it will be used to illustrate the effectiveness of the smooth dynamic LuGre friction model proposed in ARCBF. The control parameters are also selected to be consistent with those of ARCBF.

Case 1: The desired position trajectory $x_{1d} = 5[1 - \cos(0.5t)][1 - \exp(-0.1t)]^\circ$ is first implemented, shown in Figure 3. In this case, the three controllers are tested for two kinds of proportional valves with different backlash values ($1 \times 10^{-6}$ m and $1 \times 10^{-5}$ m). It can be used to illustrate the influence of backlash/deadzone on control performance and the effectiveness of the new comprehensive pressure-flow equation proposed in ARCBF. The compared tracking errors of the three controllers controlled by the two valves are shown in Figures 4 and 5, respectively. As illustrated, the designed control strategy ARCBF has the best tracking performance among the compared controllers since the backlash nonlinearity and nonlinear friction have been compensated effectively. It is interesting to note that, without using the backlash compensation, the control accuracy of ARCF gets worse sharply as the valve backlash increases. Additionally, by comparing the tracking errors between ARCBF and ARCB, it can be inferred that the proposed dynamic friction compensation scheme of ARCBF can effectively suppress the nonlinear friction effects in electrohydraulic systems. Furthermore, the simulation results when $\varepsilon = 1 \times 10^{-6}$ are depicted in Figures 3 and 6–8. The position tracking performance of ARCBF is shown in Figure 3. From Figure 6, the convergence of the parameter estimation of ARCBF is rather good, which can demonstrate the validity of parameter adaptive law. The estimation of unknown friction states of ARCBF are presented in Figure 7 and the control input is in Figure 8.

Figure 3. The position tracking of ARCBF as $\varepsilon = 1 \times 10^{-6}$. 
the best tracking performance among the compared controllers since the backlash nonlinearity and nonlinear friction have been compensated effectively. It is interesting to note that, without using the backlash compensation, the control accuracy of ARCF gets worse sharply as the valve backlash increases. Additionally, by comparing the tracking errors between ARCBF and ARCB, it can be inferred that the proposed dynamic friction compensation scheme of ARCBF can effectively suppress the nonlinear friction effects in electrohydraulic systems. Furthermore, the simulation results when $\varepsilon = 1 \times 10^{-6}$ are depicted in Figures 3 and 6–8. The position tracking performance of ARCBF is shown in Figure 3. From Figure 6, the convergence of the parameter estimation of ARCBF is rather good, which can demonstrate the validity of parameter adaptive law. The estimation of unknown friction states of ARCBF are presented in Figure 7 and the control input is in Figure 8.

Figure 3. The position tracking of ARCBF as $\varepsilon = 1 \times 10^{-6}$.

Figure 4. The compared tracking errors of the three controllers as $\varepsilon = 1 \times 10^{-6}$.

Figure 5. The compared tracking errors of the three controllers as $\varepsilon = 1 \times 10^{-5}$.

Figure 6. The parameter adaptation of ARCBF as $\varepsilon = 1 \times 10^{-6}$.
Case 2: To further verify the control capability of the proposed control scheme, a higher frequency motion trajectory $x_{1d} = [1 - \cos(3.14t)][1 - \exp(-0.1t)]^\circ$ is tested and the valve backlash $\varepsilon = 1 \times 10^{-5}$. The position tracking performance of ARCBF is seen from Figure 9 and the compared tracking errors of the three controllers are shown in Figure 10. In this test stage, since the proportional valve switches more frequently, backlash has a greater impact on the tracking performance. However, even in such a high-frequency tracking test, the proposed control strategy is able to compensate for the unexpected effects and achieves the best performance among the three compared controllers, as shown in Figure 10. The parameter adaptation and the friction state estimation, as well as the control input of the proposed ARCBF, are omitted due to space limitation.
In this paper, the nonlinear modeling and control of a hydraulic manipulator driven by the proportional valve is studied. Not only system nonlinearities and modeling uncertainties, but also explicit valve backlash/deadzone input nonlinearities in proportional valves, are taken into consideration. To this end, a new comprehensive pressure-flow model is proposed which uniformly indicates both the backlash and deadzone input nonlinearities. Based on this, the pressure dynamics of the hydraulic actuator are established. Additionally, in terms of the manipulator dynamics, with consideration of mechanism nonlinearity and utilizing the smooth LuGre friction model to describe the friction dynamics, a more accurate nonlinear model of the electrohydraulic proportional servo manipulator is then set up. To effectively compensate the nonlinearities and various uncertainties, a high-precision adaptive robust control scheme according to [13,19] is proposed based on the backstepping technology, where a projection-type adaptive law in combination with a robust feedback term is conducted to attenuate parametric uncertainties and uncertain nonlinearities. Based on this, the pressure dynamics of the hydraulic actuator are established. Additionally, in terms of the manipulator dynamics, with consideration of mechanism nonlinearity and utilizing the smooth LuGre friction model to describe the friction dynamics, a more accurate nonlinear model of the electrohydraulic proportional servo manipulator is then set up. To effectively compensate the nonlinearities and various uncertainties, a high-precision adaptive robust control scheme according to [13,19] is proposed based on the backstepping technology, where a projection-type adaptive law in combination with a robust feedback term is conducted to attenuate parametric uncertainties and uncertain nonlinearities. Based on the Lyapunov stability analysis, the stability of the closed-loop system and the excellent control performance of the proposed controller are proved. Comparative simulation tests are finally obtained to illustrate the effectiveness and priority of the proposed control scheme. As future works, it is worth further testifying the advantages of this new comprehensive pressure-flow model in a practical experimental platform.
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Appendix A

Proof of Theorem 1. Define a Lyapunov function as

\[ V_1(t) = \frac{1}{2} e_1^2 + \frac{1}{2} f_1 e_2^2 + \frac{1}{2} e_3^2 + \frac{1}{2} \hat{\theta}^T \Gamma^{-1} \hat{\theta} + \frac{1}{2} \gamma_1^{-1} \theta_1 \hat{z}_1^2 + \frac{1}{2} \gamma_2^{-1} \theta_2 \hat{z}_2^2 \]  

(A1)

and its time derivative is

\[ \dot{V}_1 = e_1 \dot{e}_1 + f_1 e_2 \dot{e}_2 + e_3 \dot{e}_3 + \hat{\theta}^T \Gamma^{-1} \dot{\hat{\theta}} + \gamma_1^{-1} \theta_1 \dot{\hat{z}}_1 + \gamma_2^{-1} \theta_2 \dot{\hat{z}}_2 \]  

(A2)

Based on Equations (31), (34) and (40), and noting the condition \( D_1(t) = D_2(t) = 0 \), then we can infer

\[ \dot{V}_1 \leq -k_1 e_1^2 + e_1 e_2 - k_2 e_2^2 + e_2 e_3 - \frac{k_3}{\sigma_{\text{min}}} k_3 e_3^2 - \hat{\theta}^T \phi_2 e_2 + \theta_1 z_1 f_3 e_2 + \sigma_{\text{min}} k_2 e_2 - \hat{\theta}^T \phi_3 e_3 + \frac{k_4}{\sigma_{\text{min}}} k_4 e_3 \]  

(A3)

Combining the definition of \( \tau \), the property in Equation (26), and noting \( \frac{k_4}{\sigma_{\text{min}}} > 1 \), we can upper bound \( \dot{V}_1 \) by

\[ \dot{V}_1 \leq -k_1 e_1^2 + e_1 e_2 - k_2 e_2^2 + e_2 e_3 - k_3 e_3^2 + \theta_1 z_1 f_3 e_2 - \theta_2 z_2 f_4 e_2 + \theta_1 \gamma_1^{-1} z_1 \dot{\hat{z}}_1 + \theta_2 \gamma_2^{-1} \dot{\hat{z}}_2 \]  

(A4)

Noting the positive definite matrix defined in Equation (44) and the dynamics given in Equation (30), we obtain

\[ \dot{V}_1 \leq -e^T \Lambda e + \theta_1 \gamma_1^{-1} z_1 \left\{ \dot{\hat{z}}_1 - |x_2 - N(x_2) \hat{z}_1| - \gamma_1 f_3 e_2 \right\} + \theta_2 \gamma_2^{-1} \dot{\hat{z}}_2 \left\{ \dot{\hat{z}}_2 - |x_2 - N(x_2) \hat{z}_2| - \gamma_2 f_4 e_2 \right\} + \theta_1 \gamma_1^{-1} N(x_2) \dot{\hat{z}}_1^2 - \theta_2 \gamma_2^{-1} N(x_2) \dot{\hat{z}}_2^2 \]  

(A5)

where \( e = [e_1, e_2, e_3]^T \). Combining the definition of the learning functions \( \eta_1 \) and \( \eta_2 \), and the property in Equation (29), we can upper bound the above equation as

\[ \dot{V}_1 \leq -e^T \Lambda e - \theta_1 \gamma_1^{-1} N(x_2) \dot{\hat{z}}_1^2 - \theta_2 \gamma_2^{-1} N(x_2) \dot{\hat{z}}_2^2 \]  

(A6)

Noting that the nonlinear function \( N(x_2) \) is always positive, then

\[ \dot{V}_1 \leq -e^T \Lambda e - \lambda_{\text{min}}(\Lambda)(e_1^2 + e_2^2 + e_3^2) \leq -W \]  

(A7)

where \( \lambda_{\text{min}}(\Lambda) \) is the minimal eigenvalue of matrix \( \Lambda \). Therefore, \( V_1 \in L_\infty \) and \( W \in L_2 \). Based on the definition of \( V_1 \) in Equation (A1), it can be inferred that \( e_1, e_2, e_3, \hat{\theta}, \hat{z}_1, \hat{z}_2 \) are bounded; from assumptions 1 and 2, we can infer that \( x \) is bounded; based on Equation (34) and assumption 2, the boundness of \( e_3 \) is thus obtained; from Equation (33), we can easily obtain that \( \hat{\theta}_2 \) is bounded. Then the boundedness of the control input \( u \) can be concluded. Hence, all system signals are bounded under closed-loop operation.
From the above analysis, combining the dynamics of $e_1$, $e_2$, $e_3$, it is easy to check that the time derivative of $W$ is bounded, thus $W$ is uniformly continuous. By applying Barbalat’s lemma [28], $W \to 0$ as $t \to \infty$, which leads to the results in Theorem 1. □

Appendix B

Proof of Theorem 2. If there exists unmodeled disturbances in the system, the time derivative of $V_2$ defined in Equation (46) can be presented by

$$
\dot{V}_2 \leq -k_1 e_1^2 + e_1 e_2 - k_2 e_2^2 + e_2 e_3 - k_3 e_3^2 + e_1 \left[-\theta^T \phi_2 + \theta_1 \tilde{z}_1 f_3 - \theta_2 \tilde{z}_2 f_4 + D_1(t) + s_2 \right] + e_2 \left[-\theta^T \phi_3 + \theta_4 \tilde{u}_{S2} + D_2(t) \right]
$$

(A8)

Combining the conditions in Equations (37) and (43), we can infer that

$$
\dot{V}_2 \leq -k_1 e_1^2 + e_1 e_2 - k_2 e_2^2 + e_2 e_3 - k_3 e_3^2 + \varepsilon t
$$

(A9)

Based on the definition of $\Lambda$ in Equation (44), the inequality becomes

$$
\dot{V}_2 \leq -\lambda_{\min}(\Lambda) \left( e_1^2 + e_2^2 + e_3^2 \right) + \varepsilon t \leq -\kappa V + \varepsilon t
$$

(A10)

By integrating the aforementioned equation, the inequality in Equation (47) can be proved. Then, $V_2$ is bounded (i.e., $e_1$, $e_2$, $e_3$ are bounded). Similar to the proof in Theorem 1, the boundness of $u$ can also be obtained. □

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