Adaptive Robust Motion Control of a Pump Direct Drive Electro-hydraulic System with Meter-Out Pressure Regulation *

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Abstract: In industry, pump-controlled hydraulic systems are generally considered energy efficient but are not accurate due to their inherent characteristics such as low response frequencies, thus pump control hydraulic systems have been traditionally applied in situations that require high power but limited accuracy. Besides, for the conventional pump-controlled cylinder systems, especially open-circuit systems, the cylinder vibration and cavitation may result due to the negative load when the cylinder moves with a high deceleration trajectory. Therefore, it is difficult to simultaneously keep the advantages of energy-saving and high motion tracking accuracy. This paper proposes a novel control strategy for a servo motor-pump and proportional valves combination control system. The cylinder is directly driven by a variable speed pump to give full play to the advantage of energy-saving by pump control. The meter-out pressure is controlled by proportional valves to provide the required resistance for motion tracking. A pressure regulation method is proposed, consisting of an optimized meter-out pressure planning and a pressure tracking controller. Both the pump controller and the meter-out pressure controller use the adaptive robust control (ARC) algorithm to achieve high motion control accuracy.

Keywords: Motion control, direct drive, servo motor-pump, meter-out throttling, adaptive robust control.

1. INTRODUCTION

Nowadays, demands for highly accurate and energy-efficient hydraulic systems are rising. The pump-controlled hydraulic systems realize high energy efficiency through the volume control method, and technically avoid the throttling energy loss, which is inevitable for the valve control systems. However, the conventional pump-controlled hydraulic systems have long been considered inaccurate because of the inherent characteristics such as high order dynamics, low response frequency, and highly nonlinear and uncertain dynamics (Wang et al. (2012)). Thus pump-controlled hydraulic systems have traditionally been used in systems that require high power but limited accuracy.

To improve simultaneously the accuracy and efficiency of the hydraulic systems, the servo motor-pump technology has been developed for pump control systems with digital and highly integrated configurations. Compared to conventional variable displacement pump-controlled systems, the variable speed pump direct drive systems achieve a faster response and make it enable to exploit advanced control algorithms to achieve high control accuracy. However, for the conventional pump-controlled cylinder systems, especially open-circuit systems, a large tracking error may occur due to the negative load when the cylinder moves at a high deceleration trajectory. In this condition, the meter-out chamber can not provide sufficient resistance to the cylinder actuator. Meanwhile, the pressure of the pump side cylinder chamber reduces greatly and drops to a low value, causing cylinder actuator vibration, cavitation, and other nonlinear phenomena (Gogate and Kabadi (2009)). In addition, the cavitation in the cylinder chamber can damage the hydraulic components (Moholkar and Pandit (1997)). All these mentioned phenomena could limit the motion tracking accuracy. Therefore, to achieve high tracking performance, both the advanced nonlinear control strategy and the hydraulic principle design with meter-out pressure regulation are needed.

Among the recent studies on precision control of hydraulic systems, Lyu et al. (2019a) combines the independent metering valves control method with a variable displacement pump to reduce throttling losses. In (Tivay et al. (2014)), a servo valve is used for the cylinder motion control, besides, a proportional relief valve is used to adjust the pump pressure to match the load. In Xu et al. (2015), a three-level controller of an independent metering system is proposed to achieve energy-saving and precise control. Mengren and Qingfeng (2018) proposed a variable pump and meter-out combination control strategy to handle a time-varying negative load. In (Cheng et al. (2018)), a decoupling compensator is proposed for a multiple actuators system to deal with the low damping situations. Ding et al. (2019) proposed an energy-saving approach for a pump and valve combination control system with multiple actuators. Shen...
et al. (2018) proposed a robust controller for the integrated direct drive volume control steering systems to achieve accurate position tracking. In (Lyu et al. (2019b)), the independent metering valves control method is combined with the direct pump control method, achieving high energy efficiency and high motion control accuracy. Besides, the adaptive robust control (ARC) algorithm has been exploited in the motion control hydraulic systems, achieving high motion tracking accuracy and effective control performance in handling the parametric uncertainties and uncertain nonlinearities. Aiming at accurate motion tracking for a pump direct drive system, Helian et al. (2019) fully considered the nonlinear hydraulic dynamics, however, the cylinder chamber in the return circuit is directly connected to the tank, which disables the system to deal with the negative load and trajectory tracking with a high deceleration.

This paper proposes a novel control strategy for a servomotor-pump and proportional valves combination control system. A variable speed pump directly drives the cylinder, which gives full play to the energy-saving advantage of the pump control. The meter-out pressure of the cylinder is controlled by proportional valves to provide the required resistance for motion tracking. The control objective is to achieve high precision motion tracking accuracy for the variable pump driving system with a high deceleration trajectory. To achieve accurate motion tracking performance, the adaptive robust control algorithm is designed for the proposed pump and valve combination control system. Based on the high order dynamics, an backstepping ARC control structure is designed for the pump controller with two steps. In addition, a meter-out pressure regulation method is proposed in order to generate an appropriate resistance for the return side of the cylinder, and avoid a large drop in pressure and cavitation on the pump side. Besides, both the pump controller and the meter-out pressure controller use the adaptive robust control (ARC) algorithm for high control accuracy in the presence of nonlinear hydraulic dynamics and parametric uncertainties.

2. SYSTEM PRINCIPLE AND MODELING

In this paper, a motion tacking strategy is designed for a servomotor pump direct-driven electro-hydraulic system with meter-out proportional valves pressure regulation. As the hydraulic schematic shown in Fig. 1, the valves are chosen as two reversing valves and two proportional valves. The cylinder is directly driven by a pump while the reversing valves decide the direction of actuator motion. The two proportional valves are used to provide the resistant pressure as meter-out orifices. The working modes of this hydraulic system are detailed in Section 3.

The cylinder dynamics is modeled as

\[ m\ddot{x}_L = P_1A_1 - P_2A_2 - bx_L + d \]

where \( m \) is the inertia load mass, \( x_L \) is the displacement of the load, \( P_1 \) and \( P_2 \) represent the cylinder pressures. \( A_1 \) and \( A_2 \) are the areas of the two sides of the piston, \( b \) represents the viscous friction coefficient, \( d \) represents the lumped uncertain nonlinearities.

The pressure-flow dynamics can be described as

\[ \frac{V_1}{P_1} = -A_1\dot{x}_L + Q_1 \]
\[ \frac{V_2}{P_2} = A_2\dot{x}_L - Q_2 \]

where \( V_1 = V_{01} + A_1x_L \) and \( V_2 = V_{02} - A_2x_L \). \( V_{01} \) and \( V_{02} \) are the initial value of \( V_1 \) and \( V_2 \), considering the volume of the two chambers and pipes which are connected to the two sides of the cylinder. \( \beta_e \) represents the effective bulk modulus, \( Q_1 \) and \( Q_2 \) represent the flow rate of the two chambers, respectively.

The pump flow rate model can be described by

\[ Q_p = u_p k_o D - C_p (P_2 - P_1) \]

where \( Q_p \) and \( u \) represent the flow rate and the voltage input of the pump, respectively. \( k_o \) is the voltage input coefficient. \( D \) is the displacement of the pump, \( C_p \) is pump flow leakage coefficient. \( P_2 \) and \( P_1 \) represent the pressures of the pump and tank.

The flow rate model of the proportional valves is modeled as

\[ Q_{vi} = k_v u_{vi} \sqrt{\Delta P_{vi}}, i = 1, 2 \]

where \( k_v \) is a lumped coefficient for valve flow rate, \( \Delta P_{vi} \) represents the pressure difference. Besides, \( u_{vi} \) represents the valve voltage input for spool position Lyu et al. (2019a).

To sum up, defining state variables as \( x = [x_1, x_2, x_3, x_4]^T = [x_L, \dot{x}_L, P_1, P_2]^T \), the state-space equations are written as

\[ \dot{x}_1 = x_2 \]
\[ \dot{x}_2 = \frac{1}{m}(A_1x_3 - A_2x_4) - \frac{b}{m}x_2 + d(x_1, x_2, t) \]
\[ \dot{x}_3 = \frac{1}{V_1(x_1)}(-\beta_e A_1x_2 + \beta_e Q_1) \]
\[ \dot{x}_4 = \frac{1}{V_2(x_1)}(\beta_e A_2x_2 - \beta_e Q_2). \]

Considering the uncertainties \( m, b, \beta_e, C_p \), the nominal disturbance value \( d_o \), and the normal values of flow rate deviation \( Q_{1in} \) and \( Q_{2in} \), the uncertain parameters are defined as \( \theta_1 = 1/m, \theta_2 = b/m, \theta_1 = d_o, \theta_4 = \beta_e, \theta_5 = \beta_e C_p, \theta_6 = \beta_e Q_{2in}, \theta_7 = \beta_e Q_{1in} \).

The dynamical model can be rewritten as

![Fig. 1. Schematic of the proposed hydraulic system.](image-url)
Table 1. Mode Selection

<table>
<thead>
<tr>
<th>(x_d)</th>
<th>(x_L - x_d)</th>
<th>Configuration</th>
<th>Piston</th>
<th>Mode</th>
</tr>
</thead>
<tbody>
<tr>
<td>&gt; 0</td>
<td>/</td>
<td>(Q_1 = Q_p, Q_2 = Q_{v_2})</td>
<td>extend</td>
<td>1</td>
</tr>
<tr>
<td>&lt; 0</td>
<td>/</td>
<td>(Q_1 = -Q_{v_1}, Q_2 = -Q_p)</td>
<td>retract</td>
<td>2</td>
</tr>
<tr>
<td>= 0</td>
<td>&lt; (\varepsilon)</td>
<td>(Q_1 = Q_p, Q_2 = Q_{v_2})</td>
<td>extend</td>
<td>1</td>
</tr>
<tr>
<td>= 0</td>
<td>&gt; (\varepsilon)</td>
<td>(Q_1 = -Q_{v_1}, Q_2 = -Q_p)</td>
<td>retract</td>
<td>2</td>
</tr>
<tr>
<td>= 0</td>
<td>otherwise</td>
<td>(Q_1 = 0, Q_2 = 0)</td>
<td>/</td>
<td>3</td>
</tr>
</tbody>
</table>

Table 2. Meter-out Pressure Regulation

<table>
<thead>
<tr>
<th>Mode</th>
<th>(P_1)</th>
<th>(P_2)</th>
<th>(F_{Ld})</th>
<th>(u_{o_2})</th>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>(\geq F_{e_1})</td>
<td>/</td>
<td>/</td>
<td>10 V</td>
</tr>
<tr>
<td>1</td>
<td>(&lt; F_{e_1})</td>
<td>/</td>
<td>(&lt; 0)</td>
<td>(u_{o_2})</td>
</tr>
<tr>
<td>2</td>
<td>/</td>
<td>(\geq P_{2})</td>
<td>/</td>
<td>10 V</td>
</tr>
<tr>
<td>2</td>
<td>/</td>
<td>(&lt; P_{2})</td>
<td>&gt; 0</td>
<td>(u_{o_2})</td>
</tr>
</tbody>
</table>

\[
\dot{x}_1 = x_2 \\
\dot{x}_2 = \theta_1(A_1x_3 - A_2x_4) - \theta_2x_2 + \theta_3 + \Delta
\]

when cylinder extends:

\[
\dot{x}_3 = \frac{1}{V_1(x_1)}(-\theta_4A_1x_2 - \theta_5(x_3 - P_t) + \theta_6Dk_{o_2u_p})
\]

\[
\dot{x}_4 = \frac{1}{V_2(x_1)}(\theta_4A_2x_2 - \theta_5k_{o_2u_2}\sqrt{x_4} - \tilde{P}_t - \theta_6) + \Delta_{o_2}
\]

when cylinder retracts:

\[
\dot{x}_3 = \frac{1}{V_1(x_1)}(-\theta_4A_1x_2 - \theta_5k_{o_2u_2}\sqrt{x_3} - \tilde{P}_t - \theta_6) + \Delta_{o_1}
\]

\[
\dot{x}_4 = \frac{1}{V_2(x_1)}(\theta_4A_2x_2 - \theta_5(x_4 - \tilde{P}_t) + \theta_6Dk_{o_2u_p})
\]

where \(\Delta = d - d_n, \Delta_{o_1} = \beta_v(\tilde{Q}_{o_1} - \tilde{Q}_{o_1})/V_1, \Delta_{o_2} = \beta_v(\tilde{Q}_{o_2} - \tilde{Q}_{o_2})/V_2\).

3. METER-OUT PRESSURE PLANNING AND CONTROLLER DESIGN

For pump direct drive systems, when the cylinder decelerates significantly, the cylinder actuator sometimes is unable to accurately track the desired mode of cylinder vibration. Therefore, the pump side chamber pressure reduces greatly and can drop to a very low value, causing cavitation, cylinder vibration, and other nonlinear phenomena.

In this section, a cylinder meter-out pressure regulation method is designed to avoid cavitation and accurately track the cylinder. When the hydraulic cylinder actuator decelerates greatly, the pump side chamber pressure reduces greatly and can drop to a very low value, causing cavitation, cylinder vibration, and other nonlinear phenomena.

Besides, an ARC controller for meter-out proportional valves is designed to track the required meter-out pressure accurately. The working mode selection of the pump and valve combination control configuration is given by Table 1, which depends on the desired trajectory \(x_d\) introduced in Section 4, and the current tracking error. The pump is connected to the cylinder chamber through the reversing valve 3 or 4. The meter-out pressure is regulated by proportional valve 1 or 2. The voltage input (0-10V) is proportional to the valve opening.

3.1 Meter-out Pressure Planning

As the meter-out pressure regulation method shown in Table 2, when the cylinder extends, \(P_2\) and \(P_1\) are the meter-out pressure and pump side cylinder chamber pressure, respectively. A desired pump side cylinder chamber pressure \(P_{e_1}\) is designed to avoid cavitation and maintain the pump side pressure \(P_1\) positive when the hydraulic cylinder actuator decelerates greatly.

According to the desired cylinder piston load force \(F_{Ld} = P_1A_1 - P_2A_2\), \(P_1\) can be described as

\[
P_1 = \frac{F_{Ld} + P_eA_1}{A_1}.
\]

The regulation goal is to make \(P_1 \rightarrow P_e\). The desired pump side cylinder chamber pressure \(P_e\) is given by

\[
P_e = P_{e_0} + k_e(t - t_e)
\]

where \(k_e = -P_{e_0}/\Delta\) represents the gradient of \(P_e\) and can be regulated by \(\Delta, P_{e_0}\) is a preset small pressure value. \(t_e\) is the starting time of the pressure regulation, and \(t_e = t\) when the meter-out pressure control is not activated.

Based on (7) and (8), the desired meter-out pressure \(P_{2d}\) can be calculated by

\[
P_{2d} = -\frac{F_{Ld} + P_eA_1}{A_2}.
\]

Depending on the desired meter-out pressure \(P_{2d}\) and pump direct-driven chamber pressure feedback, \(P_1\), the meter-out valve input \(u_{o_2}\) planning is designed as

\[
u_{o_2} = \begin{cases} 10(V), & \text{when } P_1 \geq P_{e_0} \\ u_{o_2} : P_2 \rightarrow P_{2d}, & \text{when } P_1 < P_{e_0}. \end{cases}
\]

Similarly, when the cylinder retracts, \(P_1\) is the meter-out chamber, and the working mode is 2. To track the desired meter-out pressure \(P_{1d}\), the meter-out valve input \(u_{o_1}\) planning is given as

\[
u_{o_1} = \begin{cases} 10(V), & \text{when } P_2 \geq P_{e_0} \\ u_{o_1} : P_1 \rightarrow P_{1d}, & \text{when } P_2 < P_{e_0}. \end{cases}
\]

3.2 ARC Pressure Tracking Controller

An adaptive robust pressure control method is proposed in this section to tracking the desired meter-out pressure. As \(P_2\) is defined as the meter-out chamber pressure when the cylinder extends, and the pressure dynamics can be described as

\[
P_2 = \frac{1}{V_2(x_1)}(\beta_uA_2x_2 - \beta_kk_{o_2u_2}\sqrt{x_4} - \tilde{P}_t - \beta_v\tilde{Q}_{o_2}) + \Delta_{o_2}.
\]

We define \(e_{P2} = P_2 - P_{2d}\) as the pressure regulation error, and the valve 2 flow rate \(Q_{o_2} = k_{o_2u_2}\sqrt{x_4} - \tilde{P}_t\). Considering the uncertainties due to \(\beta_v\) and \(Q_{o_2}\), the uncertain parameter set is defined as \(\theta_p = [\theta_p^T, \theta_Q^T]\) in which \(\theta_p = \beta_v\) and \(\theta_Q = \beta_vQ_{o_2}\). From (9), the differential of \(P_{2d}\) can be calculated as \(P_{2d} = k_{eA_1}/A_2\). Then the pressure tracking error dynamics is given as

\[
e_{P2} = P_2 - P_{2d} = \frac{\theta_p}{V_2} (A_2x_2 - Q_{o_2}) - \frac{\theta_Q}{V_2} + \Delta_{o_2} - \frac{k_{eA_1}}{A_2}.
\]
Let $Q_{\text{v2}}$ be the control input of meter-out pressure controller, the ARC law $Q_{\text{v2d}}$ is given as

$$
Q_{\text{v2d}}(x_1, x_2, P_2, \hat{\theta}_p) = Q_{\text{v2a}} + Q_{\text{v2ds}}
$$

where $Q_{\text{v2a}}$ represents a model compensation term, $Q_{\text{v2ds}}$ represents the robust feedback term, $k_{p2}$ a positive stabilizing feedback gain for meter-out pressure control. $Q_{\text{v2ds}}$ represents a robust feedback term, and it satisfies

$$(i) \quad e_{p2} = -Q_{\text{v2ds}}(x_1, x_2, P_2, \hat{\theta}_p) \leq 0
$$

$$(ii) \quad e_{p2} \leq 0.
$$

With the adaptation law which will be given later by (19), the adaptation function and regression are designed as

$$
\tau_{p2} = \phi_{p2} e_{p2}
$$

$$
\phi_{p2} = \frac{A_{x2} x_2 - Q_{\text{v2d}}}{V_2}, \frac{1}{V_2}.
$$

Similarly, when the cylinder retracts, the pressure tracking error is defined as $e_{p1} = P_1 - P_{id}$. In this condition, the desired flow rate $Q_{\text{v1d}}$ (for valve 1) is designed to track the desired meter-out pressure $P_{id}$ and can be obtained by the above steps.

The valve i (i=1, 2) voltage input $u_{vi}$ can be reversely calculated by (4), and be given as

$$
u_{vi} = Q_{\text{v1d}}/(k_{q1} \sqrt{\Delta P_1})
$$

where i=1,2.

4. PUMP DIRECT DRIVE CONTROLLER DESIGN

In this section, an adaptive robust backstepping controller is designed for the servomotor pump. As the controller structure presented by Fig. 2, the control objective is to make the cylinder position accurately track a given trajectory. Considering the uncertain nonlinearities and parametric uncertainties of the proposed hydraulic system, the backstepping controller consists of two steps. Step 1 is designed for the position tracking step, and step 2 is designed for the piston force tracking.

As the state-space equations given by (6), the uncertain parameter set is defined as $\theta = [\theta_1, \theta_2, \theta_3, \theta_4, \theta_5, \theta_6]^T$ when cylinder extends. Define $\hat{\theta}$ and $\bar{\theta}$ as the estimation and estimation error of $\theta$, respectively, i.e. $\hat{\theta} = \theta - \bar{\theta}$. The uncertain parameters $\theta$ are bounded by $\theta_{\text{max}}$ and $\theta_{\text{min}}$.

The adaptation law is given by

$$
\dot{\hat{\theta}} = \text{Proj}_{\hat{\theta}}(\Gamma \tau)
$$

where $\Gamma > 0$ is a diagonal adaptation matrix, $\tau$ is an adaptation function, $\text{Proj}_{\hat{\theta}}(\bullet)$ is a discontinuous projection(Helian et al. (2017)).

4.1 Backstepping ARC controller design

Step 1 (Motion tracking) Define the position tracking error as $z_1 = x_1 - x_1(t)$. Then, $z_2$ is defined as

$$z_2 = z_1 + k_1 z_1 = x_2 - x_{2eq}, \quad x_{2eq} = x_{id} - k_1 z_1$$

where $x_{id}(t)$ is the given reference trajectory, and $k_1$ is a positive feedback gain.
\[ F_{Ldc} = \frac{\partial F_{Ld}}{\partial x_1} x_2 + \frac{\partial F_{Ld}}{\partial x_2} \dot{x}_2 + \frac{\partial F_{Ld}}{\partial t} \dot{L}_{du} = \frac{\partial F_{Ld}}{\partial x_2} (\dot{x}_2 - \dot{\hat{x}}_2) + \frac{\partial F_{Ld}}{\partial \hat{\theta}} \dot{\hat{\theta}}. \] (28)

The adaptive robust control input is designed as

\[ u_{pd}(x, \dot{x}, \dot{\hat{x}}, t) = u_{pda} + u_{pds} \]

\[ u_{pda} = \frac{V_1}{A_1 \theta_0 k_d D} A_1^2 \frac{\dot{x}_3 - P_l}{V_1} \dot{\theta}_1 + \frac{A_2^2}{V_2} \dot{\theta}_2 (x_3 - P_l - \frac{F_{Ld}}{\dot{x}_2} \Delta_3) \frac{\partial F_{Ld}}{\partial x_2} - \frac{A_2}{V_2} \dot{\theta}_0 + \frac{F_{Ld}}{\dot{x}_2} \Delta_3 \]

\[ u_{pds} = u_{pd1} + u_{pd2}, \quad u_{pd1} = -\frac{V_1}{A_1 \theta_0 k_d D} k_3 z_3 \] (29)

where \( k_3 > 0 \) is a feedback gain, and the nonlinear robust feedback \( u_{pd2} \) satisfies:

\[ (i) \quad z_3 \geq \beta^T \phi_3 + \frac{A_1}{V_1} \theta_0 k_d D \theta_d u_{pd2} - A_2 \Delta_3 \dot{\theta}_0 - \frac{\partial F_{Ld}}{\partial x_2} \Delta_3 \leq \varepsilon_3 \]

\[ (ii) \quad z_3 u_{pd2} \leq 0 \] (30)

with \( \varepsilon_3 > 0 \).

The adaptation function is designed as

\[ \tau_3 = \tau_2 + \phi_3 z_3 \] (31)

where \( \phi_3 = \frac{z_2}{\tau_2 - \phi_1 z_3} \frac{\partial F_{Ld}}{\partial x_2} - \frac{\partial F_{Ld}}{\partial x_2} \dot{x}_2, - \frac{\partial F_{Ld}}{\partial x_2} \frac{A_1^2}{V_1} x_2 + \frac{A_1}{V_1} \theta_0 + k_d u_{pd2} \]

\[ - \frac{A_2^2}{V_2} x_2 + \frac{A_2}{V_2} Q_{cd} - \frac{A_1}{V_1} x_3 - P_l, A_2^2 \frac{\partial F_{Ld}}{\partial \dot{x}_2} \frac{\partial F_{Ld}}{\partial \dot{x}_2} \frac{A_1}{V_1} \theta_0 + k_d u_{pd2} \] (32)

The pump control input \( u_{pd} \) of Step 2 can be similarly designed to track \( F_{Ld} \) when the cylinder piston retracts, and is omitted in this section.

Theoretically, the stability of the system and asymptotic motion tracking performance when parametric uncertainties exist only can be rigorously proved by Lyapunov functions (Helian et al. (2019); Lyu et al. (2019a)).

5. EXPERIMENTAL RESULTS

To validate the presented control strategy, comparative experiments are presented in this section.

The desired tracking trajectory \( x_d \) in this paper is a point-to-point curve shown in Fig. 3, in which the actuator position is between 0m to 0.6m, the max velocity is 0.25m/s, and the max acceleration is 2m/s^2.

To show the advantages of the proposed controller, two comparative control strategies were chosen as

1. C1: The proposed control strategy in which the ARC control strategy is designed for a pump and valve combination control system with meter-out pressure regulation.
2. C2: Similar to C1, in which ARC controller is designed for pump control, but the meter-out chamber is directly connected to the tank without meter-out pressure regulation.

The experimental equipment is with the hydraulic schematic shown in Fig. 1. The mass of the inertia load was 280kg. The cylinder piston diameter is 50mm, and the rod diameter is 36mm.
Fig. 6. Pressures of the Cylinder Chambers.

sation. C2 can also achieve high tracking accuracy when the cylinder actuator accelerates, but large errors occurred when the hydraulic cylinder decelerated at t = 3.5s and t = 8.7s. The meter-out chamber could not provide sufficient resistance to decelerate the cylinder, making the system cannot handle the negative load. The results show the effectiveness of the proposed meter-out pressure regulation method, and the system can achieve accurate tracking performance with a high deceleration trajectory.

6. CONCLUSION

In this paper, a nonlinear motion control strategy is proposed for the accurate motion tracking of a pump and valve combination control system. The cylinder is driven by a variable speed pump with an backstepping ARC controller. Besides, a meter-out pressure planning method with the proportional valves is proposed to avoid cylinder vibration and cavitation when the cylinder decelerates substantially. The uncertainties and nonlinearities are fully considered by both controllers. Experiments validated that the proposed control strategy can achieve an accurate tracking accuracy under high dynamical nonlinearities of the hydraulic system and a high deceleration trajectory.

REFERENCES


Appendix A. DESCRIPTIONS OF SYMBOLS

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Meaning</th>
</tr>
</thead>
<tbody>
<tr>
<td>$x_L$</td>
<td>Cylinder position (m).</td>
</tr>
<tr>
<td>$P_1$, $P_2$</td>
<td>Pressures of the cylinder chambers ($P_d$).</td>
</tr>
<tr>
<td>$A_1$, $A_2$</td>
<td>Areas of the two sides of the piston ($m^2$).</td>
</tr>
<tr>
<td>$b$</td>
<td>A combined coefficient of friction forces</td>
</tr>
<tr>
<td>$V_1(x_L)$, $V_2(x_L)$</td>
<td>Volume of the two chambers ($m^3$).</td>
</tr>
<tr>
<td>$V_{01}$, $V_{02}$</td>
<td>Initial values of $V_1(x_L)$, $V_2(x_L)$ ($m^3$).</td>
</tr>
<tr>
<td>$Q_1$, $Q_2$</td>
<td>Flow rate of the two chambers ($m^3$/s).</td>
</tr>
<tr>
<td>$Q_P$</td>
<td>Quantitative pump flow ($m^3$/s).</td>
</tr>
<tr>
<td>$D$</td>
<td>Quantitative pump displacement ($m^3$).</td>
</tr>
<tr>
<td>$C_P$</td>
<td>Pump leakage coefficient.</td>
</tr>
<tr>
<td>$k_{ea}$</td>
<td>Coefficient of voltage input.</td>
</tr>
<tr>
<td>$k_s$</td>
<td>Lumped coefficient of valve flow.</td>
</tr>
<tr>
<td>$k_L$</td>
<td>Gradient of tracking pressure $P_c$.</td>
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<tr>
<td>$P_s$, $P_t$</td>
<td>Pressures of the pump and tank ($P_0$).</td>
</tr>
<tr>
<td>$\beta_p$</td>
<td>Effective bulk modulus.</td>
</tr>
<tr>
<td>$u_p$</td>
<td>Voltage input of the servo motor pump ($V$).</td>
</tr>
<tr>
<td>$u_{d1}$, $u_{d2}$</td>
<td>Voltage inputs of the proportional valves ($V$).</td>
</tr>
<tr>
<td>$\Delta_{Q_1}$, $\Delta_{Q_2}$</td>
<td>Position of the desired trajectory ($m$).</td>
</tr>
<tr>
<td>$\Delta_{P_1}$, $\Delta_{P_2}$</td>
<td>Uncertain nonlinearities.</td>
</tr>
<tr>
<td>$F_2$</td>
<td>The cylinder load force (N).</td>
</tr>
<tr>
<td>$P_{d1}$, $P_{d2}$</td>
<td>The desired Meter-out pressure.</td>
</tr>
<tr>
<td>$F_{dls}$, $u_{dls}$, $Q_{dls}$</td>
<td>Robust feedback terms, where $i = 1, 2$.</td>
</tr>
<tr>
<td>$F_{lds}$, $u_{lds}$, $Q_{lds}$</td>
<td>Adaptive model compensation terms.</td>
</tr>
<tr>
<td>$k_1$, $k_2$, $k_3$, $k_p$</td>
<td>Stabilizing feedback gains.</td>
</tr>
<tr>
<td>$z_1$, $z_2$, $z_3$, $e_{pl}$</td>
<td>Errors.</td>
</tr>
<tr>
<td>$\theta$</td>
<td>Uncertain parameters.</td>
</tr>
<tr>
<td>$\hat{\theta}$</td>
<td>Estimation of uncertain parameters.</td>
</tr>
<tr>
<td>$\tau$</td>
<td>Adaptation law.</td>
</tr>
<tr>
<td>$\Gamma$</td>
<td>Adaptation rate matrix.</td>
</tr>
</tbody>
</table>