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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 chamber reduces greatly and drops to a low value, causing cylinder actuator vibration, cavitation, and other nonlinear phenomenons (Gogate and Kabadi (2009)). In addition, the cavitation in the cylinder chamber can damage the hydraulic components (Moholkar and Pandit (1997)). All these mentioned phenomenons 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

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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

$$m\ddot{x}_L = P_1A_1 - P_2A_2 - b\dot{x}_L + d \tag{1}$$

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

$$\begin{aligned}\frac{V_1}{\beta_e}\dot{P}_1 &= -A_1\dot{x}_L + Q_1\\ \frac{V_2}{\beta_e}\dot{P}_2 &= A_2\dot{x}_L - Q_2\end{aligned} \tag{2}$$

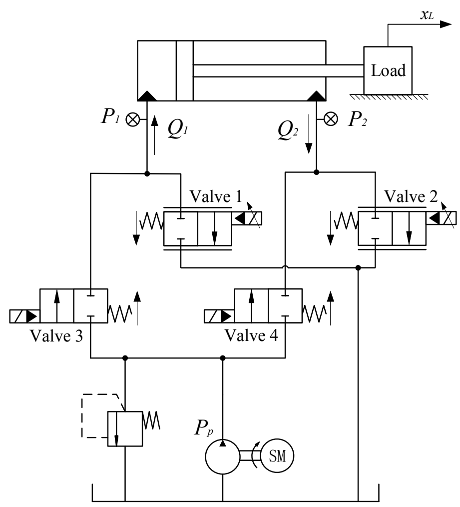

Fig. 1. Schematic of the proposed hydraulic system.

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. β_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

$$Q_p = u_pk_\omega D - C_p(P_s - P_t) \tag{3}$$

where Q_p and u represent the flow rate and the voltage input of the pump, respectively. k_ω is the voltage input coefficient. D is the displacement of the pump, C_p is pump flow leakage coefficient. P_s and P_t represent the pressures of the pump and tank.

The flow rate model of the proportional valves is modeled as

$$Q_{vi} = k_qu_{vi}\sqrt{\Delta P_{vi}},\ i = 1,2 \tag{4}$$

where k_q is a lumped coefficient for valve flow rate, Δp_{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 space-state equations are written as

$$\begin{aligned}\dot{x}_1 &= x_2\\ \dot{x}_2 &= \frac{1}{m}(A_1x_3 - A_2x_4) - \frac{b}{m}x_2 + d(x_1, x_2, t)\\ \dot{x}_3 &= \frac{1}{V_1(x_1)}(-\beta_eA_1x_2 + \beta_eQ_1)\\ \dot{x}_4 &= \frac{1}{V_2(x_1)}(\beta_eA_2x_2 - \beta_eQ_2).\end{aligned} \tag{5}$$

Considering the uncertainties m, b, β_e, C_p, the nominal disturbance value d_n, and the nominal values of flow rate deviation $\tilde{Q}_{v1n}$ and $\tilde{Q}_{v2n}$, the uncertain parameters are defined as $\theta_1 = 1/m,\ \theta_2 = b/m,\ \theta_3 = d_n,\ \theta_4 = \beta_e,\ \theta_5 = \beta_eC_i,\ \theta_6 = \beta_e\tilde{Q}_{v2n},\ \theta_7 = \beta_e\tilde{Q}_{v1n}$.

The dynamical model can be rewritten as

Table 1. Mode Selection

$\dot{x}_d$	$x_L - x_d$	Configuration	Piston	Mode
>0	/	$Q_1 = Q_p, Q_2 = Q_{v2}$	extend	1
<0	/	$Q_1 = -Q_{v1}, Q_2 = -Q_p$	retract	2
$=0$	$<-\varepsilon$	$Q_1 = Q_p, Q_2 = Q_{v2}$	extend	1
$=0$	$>\varepsilon$	$Q_1 = -Q_{v1}, Q_2 = -Q_p$	retract	2
$=0$	otherwise	$Q_1 = 0, Q_2 = 0$	/	3

Table 2. Meter-out Pressure Regulation

Mode	$P1$	$P2$	F_{Ld}	u_{vi}
1	$\geq P_{\varepsilon 1}$	/	/	$10\,V$
1	$< P_{\varepsilon 1}$	/	<0	u_{v2}
2	/	$\geq P_{\varepsilon 2}$	/	$10\,V$
2	/	$< P_{\varepsilon 2}$	>0	u_{v1}

$$
\begin{aligned}
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= \theta_1(A_1x_3 - A_2x_4) - \theta_2 x_2 + \theta_3 + \Delta \\
&\textit{when cylinder extends}: \\
\dot{x}_3 &= \frac{1}{V_1(x_1)}(-\theta_4 A_1 x_2 - \theta_5(x_3 - P_t) + \theta_4 D k_\omega u_p) \\
\dot{x}_4 &= \frac{1}{V_2(x_1)}(\theta_4 A_2 x_2 - \theta_4 k_q u_{v2}\sqrt{x_4 - P_t} - \theta_6) + \Delta_{Qv2} \\
&\textit{when cylinder retracts}: \\
\dot{x}_3 &= \frac{1}{V_1(x_1)}(-\theta_4 A_1 x_2 - \theta_4 k_q u_{v2}\sqrt{x_3 - P_t} - \theta_7) + \Delta_{Qv1} \\
\dot{x}_4 &= \frac{1}{V_2(x_1)}(\theta_4 A_2 x_2 - \theta_5(x_4 - P_t) + \theta_4 D k_\omega u_p) \qquad (6)
\end{aligned}
$$

where $\Delta = d - d_n, \Delta_{Qv1} = \beta_e(\tilde{Q}_{v1n} - \tilde{Q}_{v1})/V_1, \Delta_{Qv2} = \beta_e(\tilde{Q}_{v2n} - \tilde{Q}_{v2})/V_2$.

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 trajectory because the return side of the cylinder could not properly provide the resistance. At the same time, the load force could be negative. In this case, the pump side chamber pressure reduces greatly and can drop to a very low value, causing cavitation, cylinder vibration, and other nonlinear phenomenons.

In this section, a cylinder meter-out pressure regulation method is designed to avoid cavitation and to accurately track the given trajectory even when the hydraulic cylinder decelerates greatly. Based on the pump side pressure and cylinder dynamics (1), a meter-out pressure planning is designed to obtain an ideal meter-out pressure, so as to generate the appropriate resistance for the cylinder to maintain the cylinder load force positive. 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 tacking 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 $P_{\varepsilon 1}$ 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

$$P_1 = \frac{F_{Ld} + P_2A_2}{A_1}. \qquad (7)$$

The regulation goal is to make $P_1 \to P_\varepsilon$. The desired pump side cylinder chamber pressure P_ε is given by

$$P_\varepsilon = P_{\varepsilon 0} + k_\varepsilon(t - t_\varepsilon) \qquad (8)$$

where $k_\varepsilon = -P_{\varepsilon 0}/\Delta t$ represents the gradient of P_ε and can be regulated by Δt, $P_{\varepsilon 0}$ is a preset small pressure value. t_ε is the starting time of the pressure regulation, and $t_\varepsilon = t$ 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

$$P_{2d} = \frac{-F_{Ld} + P_\varepsilon A_1}{A_2}. \qquad (9)$$

Depending on the desired meter-out pressure P_{2d} and pump direct-driven chamber pressure feedback P_1, the meter-out valve input u_{v2} planning is designed as

$$u_{v2} = \begin{cases} 10(V),\ \textit{when}\ P_1 \geq P_{\varepsilon 0} \\ u_{v2}:\ P_2 \to P_{2d},\ \textit{when}\ P_1 < P_{\varepsilon 0}. \end{cases} \qquad (10)$$

Similarly, when the cylinder retracts, P_1 side 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_{v1} planning is given as

$$P_{1d} = \frac{F_{Ld} + P_\varepsilon A_2}{A_1} \qquad (11)$$

$$u_{v1} = \begin{cases} 10(V),\ \textit{when}\ P_2 \geq P_{\varepsilon 0} \\ u_{v1}:\ P_1 \to P_{1d},\ \textit{when}\ P_2 < P_{\varepsilon 0}. \end{cases} \qquad (12)$$

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

$$\dot{P}_2 = \frac{1}{V_2(x_1)}(\beta_e A_2 x_2 - \beta_e k_q u_{v2}\sqrt{x_4 - P_t} - \beta_e \tilde{Q}_{v2n}) + \Delta_{Qv2}. \qquad (13)$$

We define $e_{P2} = P_2 - P_{2d}$ as the pressure regulation error, and the valve 2 flow rate $Q_{v2} = k_q u_{v2}\sqrt{x_4 - P_t}$. Considering the uncertainties due to β_e and $\tilde{Q}_{v2n}$, the uncertain parameter set is defined as $\theta_p = [\theta_\beta, \theta_Q]^T$ in which $\theta_\beta = \beta_e$ and $\theta_Q = \beta_e\tilde{Q}_{v2n}$. From (9), the differential of P_{2d} can be calculated as $\dot{P}_{2d} = k_\varepsilon A_1/A_2$. Then the pressure tracking error dynamics is given as

$$
\begin{aligned}
\dot{e}_{p2} &= \dot{P}_2 - \dot{P}_{2d} \\
&= \frac{\theta_\beta}{V_2}(A_2x_2 - Q_{v2}) - \frac{\theta_Q}{V_2} + \Delta_{Qv2} - \frac{k_\varepsilon A_1}{A_2}. \qquad (14)
\end{aligned}
$$

Let Q_{v2} be the control input of meter-out pressure controller, the ARC law Q_{v2d} is given as

$$\begin{aligned} Q_{v2d}(x_1,x_2,P_2,\hat{\theta}_p) &= Q_{v2da}+Q_{v2ds} \\ Q_{v2da} &= -\frac{\hat{\theta}_Q}{\hat{\theta}_\beta}+A_2x_2-\frac{V_2}{\hat{\theta}_\beta}\frac{k_\varepsilon A_1}{A_2} \\ Q_{v2ds} &= Q_{v2ds1}+Q_{v2ds2},\ Q_{v2ds1}=k_{p2}\frac{V_2}{\hat{\theta}_{\beta min}}e_{P2} \end{aligned} \tag{15}$$

where Q_{v2da} represents a model compensation term, Q_{v2ds} represents the robust feedback term, k_{p2} a positive stabilizing feedback gain for meter-out pressure control. Q_{v2ds2} represents a robust feedback term, and it satisfies

$$\begin{aligned} &(i)\ \ e_{p2}(-Q_{v2ds2}\frac{\theta_\beta}{V_2}-\tilde{\theta}^T\phi_2+\Delta_{Qv2})\le\varepsilon_{p2} \\ &(ii)\ \ e_{p2}Q_{v2ds2}\le 0. \end{aligned} \tag{16}$$

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

$$\begin{aligned} \tau_{p2} &=\phi_{p2}e_{p2} \\ \phi_{p2} &=[\frac{A_2x_2-Q_{v2da}}{V_2},-\frac{1}{V_2}]^T. \end{aligned} \tag{17}$$

Similarly, when the cylinder retracts, the pressure tracking error is defined as $e_{P1}=P_1-P_{1d}$. In this condition, the desired flow rate Q_{v1d} (for valve 1) is designed to track the desired meter-out pressure P_{1d} 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

$$u_{vi}=Q_{vid}/(k_q\sqrt{\Delta P_i}) \tag{18}$$

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 $\tilde{\theta}$ as the estimation and estimation error of θ, respectively,i.e. $\tilde{\theta}=\hat{\theta}-\theta$). The uncertain parameters θ are bounded by θ_{max} and θ_{min}.

The adaptation law is given by

$$\dot{\hat{\theta}}=Proj_{\hat{\theta}}(\Gamma\tau) \tag{19}$$

where $\Gamma>0$ is a diagonal adaptation matrix, τ is an adaptation function, $Proj_{\hat{\theta}_i}(\bullet_i)$ is a discontinuous projection(Helian et al. (2017)).

4.1 Backstepping ARC controller design

Step1 (Motion tracking) Define the position tracking error as $z_1=x_1-x_d(t)$. Then, z_2 is defined as

$$z_2=\dot{z}_1+k_1z_1=x_2-x_{2eq},\ \ x_{2eq}=\dot{x}_{1d}-k_1z_1 \tag{20}$$

where $x_{1d}(t)$ is the given reference trajectory, and k_1 is a positive feedback gain.

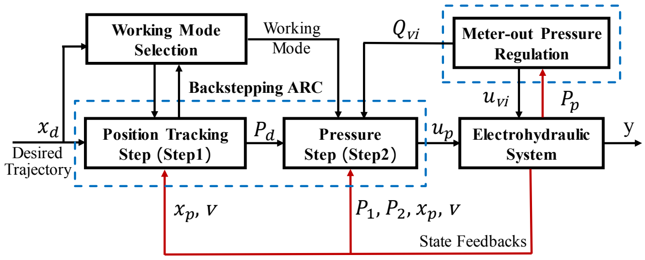

Fig. 2. Controller structure.

Define F_L as the hydraulic force on cylinder which is modeled as

$$F_L=x_3A_1-x_4A_2. \tag{21}$$

Then the error dynamics is given as

$$\dot{z}_2=\dot{x}_2-\dot{x}_{2eq}=\theta_1F_L-\theta_2x_2+\theta_3+\Delta-\dot{x}_{2eq} \tag{22}$$

where Δ is the uncertain nonlinearity.

The control input for step1 is defined as F_{Ld}, and the control law is given as

$$\begin{aligned} F_{Ld}(x_1,x_2,\hat{\theta}_1,\hat{\theta}_2,\hat{\theta}_3,t) &= F_{Lda}+F_{Lds} \\ F_{Lda} &= \frac{1}{\hat{\theta}_1}(\hat{\theta}_2x_2-\hat{\theta}_3+\dot{x}_{2eq}) \\ F_{Lds} &= F_{Lds1}+F_{Lds2},\ \ F_{Lds1}=-\frac{1}{\hat{\theta}_{1min}}k_2z_2 \end{aligned} \tag{23}$$

where the positive feedback gain k_2 is a given due to the closed-loop bandwidth of the system. P_{Lds2} satisfies

$$\begin{aligned} &(i)\ \ z_2(\theta_1F_{Lds2}-\tilde{\theta}^T\phi_2+\Delta)\le\varepsilon_2 \\ &(ii)\ \ z_2F_{Lds2}\le 0 \end{aligned} \tag{24}$$

with $\varepsilon_2>0$ and can be arbitrarily small. The adaptation function of step1 is

$$\begin{aligned} \tau_2 &= \phi_2z_2 \\ \phi_2 &= [F_{Lda},-x_2,1,0,0,0]^T. \end{aligned} \tag{25}$$

A positive-semidefinite (p.s.d.) function is defined as $V_2=\frac{1}{2}{z_2}^2$. It can be obtained that

$$\begin{aligned} \dot{V}_2 &=z_2\dot{z}_2=z_2[\theta_1(z_3+F_{Ld})-\theta_2x_2+\theta_3+\Delta-\dot{x}_{2eq}] \\ &=-\frac{\theta_1}{\hat{\theta}_{1min}}k_2{z_2}^2+\theta_1z_2z_3+z_2(\theta_1F_{Lds2}-\tilde{\theta}^T\phi_2+\Delta). \end{aligned} \tag{26}$$

Step2 (Pressure-flow) The input error of step 1 is defined as $z_3=F_L-F_{Ld}$. To make z_3 converge to zero or a small value, we define the motor-pump voltage input u_p as the control input of step 2

When the cylinder extends and works at mode 1, the error dynamics is

$$\begin{aligned} \dot{z}_3 &=\dot{F}_L-\dot{F}_{Ld}=(\dot{x}_3A_1-\dot{x}_4A_2)-(\dot{F}_{Ldc}+\dot{F}_{Ldu}) \\ &=-\frac{{A_1}^2}{V_1}x_2\theta_4-\frac{A_1}{V_1}(x_3-P_t)\theta_5+\frac{A_1}{V_1}\theta_4Dk_\omega u_{pd} \\ &\quad -\frac{{A_2}^2}{V_2}x_2\theta_4+\frac{A_2}{V_2}\theta_4Q_{v2d}+A_2\frac{\theta_6}{V_2}-A_2\Delta_{Qv2}-(\dot{F}_{Ldc}+\dot{F}_{Ldu}) \end{aligned} \tag{27}$$

where

$$\begin{aligned}\dot{F}_{Ldc} &= \frac{\partial F_{Ld}}{\partial x_1}x_2 + \frac{\partial F_{Ld}}{\partial x_2}\dot{\hat{x}}_2 + \frac{\partial F_{Ld}}{\partial t}\\ \dot{F}_{Ldu} &= \frac{\partial F_{Ld}}{\partial x_2}(\dot{x}_2 - \dot{\hat{x}}_2) + \frac{\partial F_{Ld}}{\partial \hat{\theta}}\dot{\hat{\theta}}.\end{aligned} \tag{28}$$

The adaptive robust control input is designed as

$$\begin{aligned}u_{pd}(x,\hat{\theta},t) &= u_{pda} + u_{pds}\\ u_{pda} =& \frac{V_1}{A_1\hat{\theta}_4 k_\omega D}\Big(\frac{{A_1}^2}{V_1}x_2\hat{\theta}_4 + \frac{A_1}{V_1}(x_3 - P_t)\hat{\theta}_5 + \frac{{A_2}^2}{V_2}x_2\hat{\theta}_4\\ &- \frac{A_2}{V_2}\hat{\theta}_4 Q_{v2d} - \frac{A_2}{V_2}\hat{\theta}_6 + \dot{F}_{Ldc} - \hat{\theta}_1 z_2\Big)\\ u_{pds} =& u_{pds1} + u_{pds2},\ \ u_{pds1} = -\frac{V_1}{A_1\hat{\theta}_{4min}k_\omega D}k_3 z_3\end{aligned} \tag{29}$$

where $k_3 > 0$ is a feedback gain, and the nonlinear robust feedback u_{pds} satisfies:

$$\begin{aligned}&(i)\ \ z_3[-\tilde{\theta}^T\phi_3 + \frac{A_1}{V_1}\theta_4 k_\omega D\theta_4 u_{pds2} - A_2\Delta_{Qv2} - \frac{\partial F_{Ld}}{\partial\, x_2}\Delta] \le \varepsilon_3\\ &(ii)\ \ z_3 u_{pds2} \le 0\end{aligned} \tag{30}$$

with $\varepsilon_3 > 0$.

The adaptation function is designed as

$$\tau_3 = \tau_2 + \phi_3 z_3 \tag{31}$$

where ϕ_3 is

$$\begin{aligned}\phi_3 =& [z_2 - \frac{\partial F_{Ld}}{\partial x_2}F_L, \frac{\partial F_{Ld}}{\partial x_2}x_2, -\frac{\partial F_{Ld}}{\partial x_2}, -\frac{{A_1}^2}{V_1}x_2 + \frac{A_1}{V_1}Dk_\omega u_{pda}\\ &- \frac{{A_2}^2}{V_2}x_2 + \frac{A_2}{V_2}Q_{v2d}, -\frac{A_1}{V_1}(x_3 - P_t), \frac{A_2}{V_2}]^T.\end{aligned} \tag{32}$$

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 $36mm$.

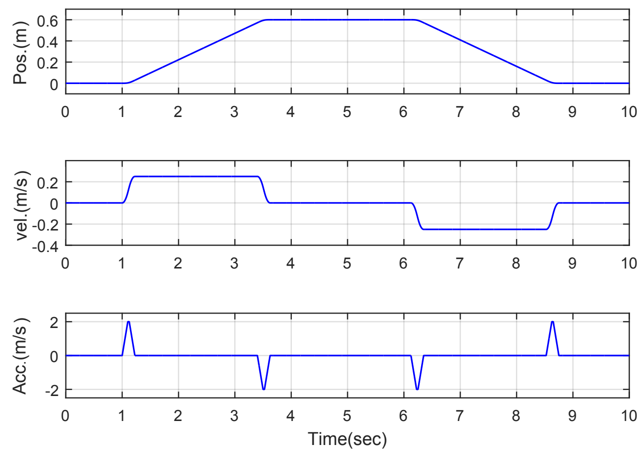

Fig. 3. The desired point-to-point trajectory.

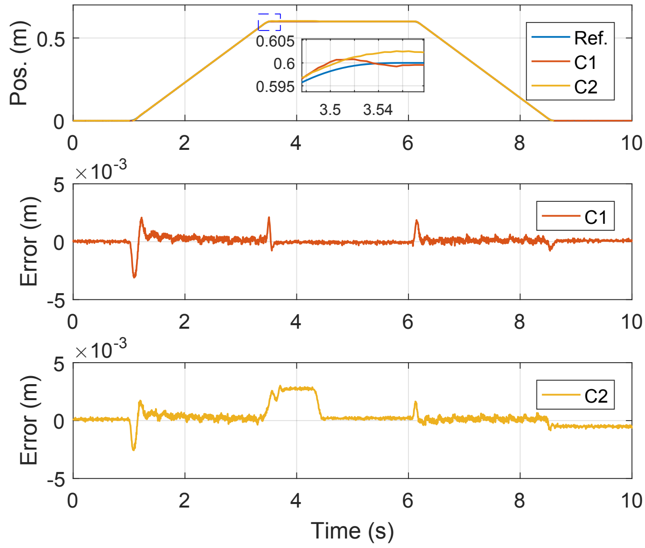

Fig. 4. Comparative tracking errors.

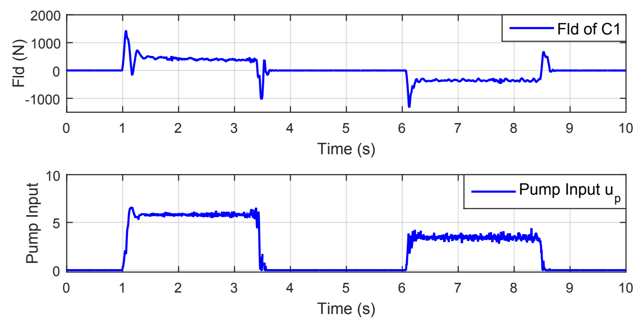

Fig. 5. Control Inputs.

The experimental tracking errors of both controllers are shown in Fig. 4, besides, the pump control input u_{pd} and the visual control input F_{Ld} are presented by Fig. 5. Pressures in each cylinder chamber are shown in Fig. 6.

High tracking accuracy is achieved with C1, stability and transient performance is guaranteed. The tracking error converges to zero or a very small value due to the adaptive model compen-

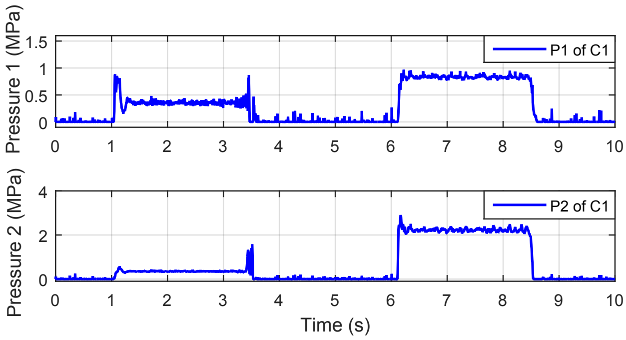

Fig. 6. Pressures of the Cylinder Chambers.

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.

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Appendix A. DESCRIPTIONS OF SYMBOLS

Symbol	Meaning
x_L	Cylinder position (m).
P_1, P_2	Pressures of the cylinder chambers (P_a).
A_1, A_2	Areas of the two sides of the piston (m^2).
b	A combined coefficient of friction forces
$V_1(x_L)$, $V_2(x_L)$	Volume of the two chambers (m^3).
V_{01}, V_{02}	Initial values of $V_1(x_L)$, $V_2(x_L)$ (m^3).
Q_1, Q_2	Flow rate of the two chambers (m^3/s).
Q_p	Quantitative pump flow (m^3/s).
D	Quantitative pump displacement (m^3).
C_p	Pump leakage coefficient.
k_ω	Coefficient of voltage input.
k_q	Lumped coefficient of valve flow.
k_ε	Gradient of tracking pressure P_ε.
P_s, P_t	Pressures of the pump and tank (P_a).
β_e	Effective bulk modulus.
u_p	Voltage input of the servo motor pump (V).
u_{v1}, u_{v2}	Voltage inputs of the proportional valves (V).
x_d	Position of the desired trajectory (m).
Δ, Δ_{Qv1}, Δ_{Qv2}	Uncertain nonlinearities.
F_L	The cylinder load force (N).
P_{2d}, P_{1d}	The desired Meter-out pressure.
F_{Lds}, u_{pds}, Q_{vids}	Robust feedback terms, where i = 1, 2.
F_{Lda}, u_{pda}, Q_{vida}	Adaptive model compensation terms.
k_1, k_2, k_3, k_{pi}	Stabilizing feedback gains.
z_1, z_2, z_3, e_{pi}	Errors.
θ	Uncertain parameters.
$\hat{\theta}$	Estimation of uncertain parameters.
τ	Adaptation law.
Γ	Adaptation rate matrix.