Disturbance observer–based nonlinear energy-saving control strategy for electro-hydraulic servo systems

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Abstract
Precise tracking performance and significant energy-saving effect are both important issues for electro-hydraulic servo system. Nevertheless, the electro-hydraulic servo system usually demonstrates inferior efficiency compared with other available actuation methods. To improve the efficiency of the electro-hydraulic servo system during the task of position tracking, a nonlinear energy-saving control strategy is proposed. The presented controller employs a variable supply pressure control to make the pump pressure adapt to the system demand and utilizes a nonlinear cascade controller for precise position tracking. Additionally, a disturbance observer is developed to compensate for the lumped uncertainties including parametric uncertainties and uncertain nonlinearities both in the design of variable supply pressure control and nonlinear cascade controller. Results indicate that the presented multi-input single-output system combined with the nonlinear energy-saving control strategy can save 45% energy in a harmonic trajectory tracking test and 68% energy in a multi-step trajectory tracking test, respectively, compared with its counterparts. Furthermore, the maximum tracking errors of both tests are 1.2 mm.

Keywords
Energy saving, position tracking, variable supply pressure, disturbance observer, electro-hydraulic system

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Introduction
The electro-hydraulic servo system (EHSS) has been demonstrated to be a promising choice for diverse applications due to its high power density, ease of implementation, good dynamic performance, and durability, despite the rapid development of electric power transmission. Owing to these characteristics, the EHSS has been widely used in aircraft control, machine tools and manufacturing, excavating, and automotive industries.¹

Many papers are devoted to solving the problem of high-performance position tracking control of EHSS, and numerous successful methods for specific purposes were proposed.²–⁵ However, the EHSS usually has a lower energy efficiency compared with other actuation methods, which may limit its further utilization in industrial and mechanical equipments.

Therefore, certain efforts have been made in the literature to improve the overall efficiency of the EHSS. The electro-hydrostatic actuation (EHA) does not require the usage of control valve, and this method can...
increase the system efficiency considerably. However, the overall achievable bandwidth of the closed-loop system is much lower than the valve-controlled EHSS due to the higher inertias. An integrated control strategy of clamping force and energy saving using a load-sensing variable pump was proposed in Chiang et al. Furthermore, the electrical adaptations of this method, electro-hydraulic load sensing (EHLS), which includes a constant displacement pump and an AC motor, was also utilized to improve the energy efficiency of the EHSS in Cho et al. and Lovrec et al. Nevertheless, the pressure difference across the valve, which is usually set as a constant value in these methods, is not optimized from an energy-saving perspective, and the slow response also put an obstacle for its utilization. As an alternative, energy regeneration is also an attractive research field for energy saving of EHSS. Specifically, in Lu et al., the independent valve metering control method for a hydraulic manipulator using five cartridge valves and one accumulator was proposed. Although a significant energy-saving capability was obtained, the energy loss in the cartridge valve is still considerable.

In order to reduce the energy consumption across the valve, the variable supply pressure control (VSPC) strategy, which can control the supply pressure according to the system demand, has gained more attention in recent years. A kind of VSPC, which utilized the optimal control method, was presented to improve the energy-saving capabilities of the EHSS in Tivay et al. Another VSPC, which was proposed in Baghestan et al., can also achieve energy-saving effect on the EHSS. Nevertheless, the discrepancy between the real values and the nominal parameters utilized in the aforementioned studies and the uncertain nonlinearities including unmodeled dynamics and external disturbance have not been fully considered in the design of the VSPC and the tracking controller. More recently, a load prediction–based VSPC controller was developed for the energy-saving purpose of a hydraulic actuated robot. But the VSPC in this study using the kinetics model to predict the external disturbance is not applicable for mobile machinery, since the external disturbances vary abruptly so that the load is difficult to predict.

As reviewed above, some of the researchers mainly focused on the precise tracking performance without considering the energy efficiency of the system. Some of the researchers developed the controller with VSPC for the purpose of energy saving. Nevertheless, the energy-saving performance and the achievable position tracking performance of these systems may degrade when the system suffers from the lumped uncertainties including parametric uncertainties and uncertain nonlinearities. Both precise tracking and significant energy-saving performance are critical problems of the EHSS.

Therefore, a nonlinear controller for the EHSS, which could solve the dual objective problem and take the lumped uncertainties into account explicitly, is necessary.

In this study, a solution for this multi-objective problem is proposed. To improve the system efficiency, a VSPC is developed in this study. Moreover, a nonlinear cascade control strategy is employed to achieve the task of position tracking. Unlike the existing VSPC, a disturbance observer (DOB) is presented to compensate for the lumped uncertainties both in the design of VSPC and nonlinear cascade controller. The effectiveness of the proposed controller is verified by tests.

**System modeling and problem formulation**

The EHSS configuration utilized in this article is presented in Figure 1. Specifically, a fixed displacement pump is used as a power unit. A proportional relief valve (PRV) is located in parallel with the pump to control the supply pressure, and a proportional directional valve (PDV) is utilized for the purpose of position tracking. The multi-objective problem can be addressed using this setup.

Since the natural frequency of the servo valve used here is much higher than that of a typical EHSS, the valve dynamics is often neglected without a significant reduction in the control performance. Therefore, the flow equations of the hydraulic valve can be written as

\[
Q_1 = k_q q \left[ s_\delta(u_\delta) \left( \frac{2}{\rho} (p_s - p_1) + s_\delta(-u_\delta) \right) \right]
\]

\[
Q_2 = k_q q \left[ s_\delta(u_\delta) \left( \frac{2}{\rho} (p_2 - p_1) + s_\delta(-u_\delta) \right) \right]
\]

(1)

**Figure 1.** Schematic diagram of the proposed electro-hydraulic system.
where $k_q$ is the flow gain coefficient of the servo valve, $k_r$ is a positive constant, $p_1$ and $p_2$ are the chamber pressures of the cylinder, $p_s$ is the supply pressure of the pump, $p_t$ is the tank pressure, $\rho$ is the oil density, $Q_1$ is the supply flow rate to the forward chamber, $Q_2$ is the return flow rate from the return chamber, and $u_d$ is the input signal of PDV.

The pressure dynamics of the actuator can be written as

$$
\begin{align*}
\dot{p}_1 &= h_1[Q_1 - A_1\dot{x}_p - C_ip_1 + C_jp_2] + \Delta_1 \\
\dot{p}_2 &= h_2[-Q_2 + A_2\dot{x}_p - C_ip_2 + C_jp_1] + \Delta_2
\end{align*}
$$

(3)

where $A_1$ and $A_2$ are the effective areas of the two chambers of the cylinder, $x_p$ is the piston position, $C_i = C_i + C_e$ is the total leakage coefficient, $C_e$ is the external leakage coefficient, $h_1 = \beta_e/V_1$, $h_2 = \beta_e/V_2$, $V_1$ and $V_2$ are the control volumes of the two chambers, $\beta_e$ is the effective bulk modulus of the system, and $\Delta_1$ and $\Delta_2$ denote the lumped uncertainties in the pressure dynamics.

The force balance equation of the system is expressed as

$$
m\ddot{x}_p = p_1A_1 - p_2A_2 - B_c\dot{x}_p - kx_p - F_L - F_f + \Delta_3
$$

(4)

where $m$ is the mass of the load, $B_c$ is the coefficient of viscous damping, $k$ is the environment stiffness, $F_L$ is the external load force, $F_f$ is the Coulomb friction force, $\Delta_3$ represents the lumped uncertainties including parametric uncertainties and uncertain nonlinearities.

Define the system state variables as equation (5)

$$
[x_1, x_2, x_3, x_4]^T = [x_p, \dot{x}_p, p_1, p_2]^T
$$

(5)

In order to make the system fall into the strict feedback form, the state variables are reconstructed as equation (6), where $a = A_2/A_1$

$$
[x_1, x_2, \dot{x}_3]^T = [x_p, \dot{x}_p, p_1 - ap_2]^T
$$

(6)

Therefore, the entire system can be expressed in a state space form as equation (7)

$$
\begin{align*}
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= \frac{A_1}{m}x_3 - \frac{B_c}{m}x_2 - \frac{k}{m}x_1 - \frac{F_L}{m} - \frac{F_f}{m} + \frac{\Delta_3}{m} \\
\dot{x}_3 &= -f_1x_2 - f_2x_3 + f_3x_4 + f_4u_d + \frac{\Delta_{1_2}}{a} \\
y &= x_1
\end{align*}
$$

(7)

where $f_1$ to $f_4$ are shown in equation (8).

The control task is summarized as follows: given the desired motion trajectory $x_d$, the objective is to synthesize a control input of the PDV such that the output $x_1$ tracks $x_d$ as closely as possible. Meanwhile, the controller generates a control input of PRV in order to reduce the energy consumption during the task of position tracking.

The whole control block diagram in this study is shown in Figure 2. The tracking performance is guaranteed by utilizing the proposed cascade tracking controller combined with DOB to synthesize a command signal $u_d$ for PDV. Meanwhile, the VSPC adopts the observer result of $d_1$, desired motion trajectory $x_d$, and other system parameters as the control input to generate a desired command signal $u_r$ to the PRV.

**Controller design**

**DOB design**

The dynamics (equation (7)) can be rewritten as

$$
\begin{align*}
d_1 &= \dot{x}_2 + \frac{1}{m}(B_cx_2 + kx_1 - A_1\dot{x}_3) \\
d_2 &= \dot{x}_3 + f_1x_2 + f_2x_3 - f_3x_4 - f_4u_d
\end{align*}
$$

(9)

Define the estimates of the disturbances as $\hat{d}_1$ and $\hat{d}_2$. The estimation errors are expressed as

$$
\begin{align*}
\hat{d}_1 &= d_1 - \hat{d}_1 \quad \text{and} \quad \hat{d}_2 = d_2 - \hat{d}_2
\end{align*}
$$

(10)

The dynamics of $\hat{d}_1$ and $\hat{d}_2$ are designed as

$$
\begin{align*}
\dot{\hat{d}}_1 &= \frac{1}{\varepsilon_1}(\dot{x}_2 + \frac{1}{m}(B_cx_2 + kx_1 - A_1\dot{x}_3) - \hat{d}_1) \\
\dot{\hat{d}}_2 &= \frac{1}{\varepsilon_2}(\dot{x}_3 + f_1x_2 + f_2x_3 - f_3x_4 - f_4u_d - \hat{d}_2)
\end{align*}
$$

(11)

where $1/\varepsilon_1$ and $1/\varepsilon_2$ are the observer gains.
Assumption. The disturbances and their derivatives are bounded such that \(|d_i| \leq d_{i,\text{max}}\) and \(|\dot{d}_i| \leq \dot{d}_{i,\text{max}}\) for \(i = 1, 2\), where \(d_{i,\text{max}}\) and \(\dot{d}_{i,\text{max}}\) are known constants.

In fact, the disturbance is not differentiable at zero velocity since it includes the Coulomb friction. However, the actual friction has finite jumps, so that the friction model can be chosen as any differentiable function that approximates the actual discontinuous Coulomb friction. Thus, the assumption is physically reasonable.4

In case the measurement noises are amplified by the high gains, the auxiliary state variables \(\xi_1\) and \(\xi_2\), which are proposed in Won et al.4, are utilized in this study.

Define the auxiliary state variables as

\[
\xi_1 = \tilde{d}_1 - \frac{x_2}{e_1} \quad \text{and} \quad \xi_2 = \tilde{d}_2 - \frac{x_3}{e_2}
\]  

(12)

The dynamics of the auxiliary state variables are

\[
\begin{align*}
\dot{\xi}_1 &= \frac{1}{e_1} \left( B \dot{x}_2 + k x_1 - A \dot{x}_3 \right) - \frac{1}{e_1} \left( \xi_1 + \frac{x_2}{e_1} \right) \\
\dot{\xi}_2 &= \frac{1}{e_2} \left( f_1 \dot{x}_2 + f_2 x_3 - f_3 x_4 - f u_d \right) - \frac{1}{e_2} \left( \xi_2 + \frac{x_3}{e_2} \right) 
\end{align*}
\]  

(13)

Then, \(|\tilde{d}_i| \leq e^{-\frac{(1/e_i)t}{m}} |\tilde{d}_i(0)| + e_i \rho_t(t)\) for an envelope function \(\rho_t(t)\), such that \(\rho_t(t) \geq |d_i|, \forall t \geq 0, \ i = 1, 2\).

In case \(d_i\) varies significantly, which may result in a large input voltage, \(\tilde{d}_i\) is updated using the following projection-type adaption law with a preset adaption rate limit \(d_{i,M}\) based on equation (12)

\[
\begin{align*}
\hat{d}_1 &= \text{sat}_{d_{i,M}} \left( \text{Proj}_{d_i} \left( \frac{\xi_1 + \frac{x_2}{e_1}}{\xi_2} \right) \right) \\
\hat{d}_2 &= \text{sat}_{d_{i,M}} \left( \text{Proj}_{d_i} \left( \frac{\xi_2 + \frac{x_3}{e_2}}{\xi_2} \right) \right) \\
\end{align*}
\]  

(14)

\[
\text{Proj}_{d_i}(\bullet) = \begin{cases} 
0 & \text{if } \hat{d}_i \geq d_{i,\text{max}} \text{ and } \bullet > 0 \\
\hat{d}_i & \text{otherwise} 
\end{cases}
\]  

(15)

Proof. From equations (9), (10), (12), and (13), the following disturbance estimation error dynamics is obtained as

\[
\dot{\tilde{d}}_i = -\frac{1}{e_i} \tilde{d}_i + \dot{d}_i \quad \text{for } i = 1, 2
\]  

(16)

Therefore, \(|\tilde{d}_i| \leq e^{-\frac{(1/e_i)t}{m}} |\tilde{d}_i(0)| + e_i \rho_t(t)\), and the upper bound of \(|\tilde{d}_i(\infty)|\) becomes smaller as \(e_i\) gets smaller.

Remark 1. Since the projection function and adaption rate limit are added in adaption law, two special working points should be discussed:

1. Assume that the \(\hat{d}_i\) reaches to the limit, that is, \(d_{i,\text{max}}\) (similar as the case for \(d_{i,\text{min}}\)), equation (10) can be rewritten as \(\hat{d}_i = d_i - d_{i,\text{max}}\). Furthermore, the \(\hat{d}_i\) in equation (11) can be substituted by \(d_{i,\text{max}}\). Differentiating equation (10) and noting equation (11), we can still draw a conclusion as equation (16).

2. Assume that the \(\hat{d}_i\) reaches to the limit, that is, \(d_{i,M}\), the dynamics of equation (10) can be written as \(\hat{d}_i = d_i - d_{i,M}\). Thus, both \(\hat{d}_i\) and \(d_i\) are bounded according to the assumption of the article and the projection function. Although equation (16) cannot be satisfied in this case, the stability of the entire controller can still be obtained by suitable values of \(k_1\) and \(k_2\), which will be introduced in the following section.

Nonlinear cascade tracking control

The nonlinear tracking controller is designed as a position tracking outer loop and a load pressure control inner loop. Using a recursive backstepping procedure, the controller can be expressed as follows.

Step 1. Define the tracking error as \(\tilde{x}_1 = x_1 - x_d\). Then a sliding surface \(s\) is defined as

\[
s = \hat{x}_1 + \lambda \tilde{x}_3
\]  

(17)

where \(\lambda\) is a positive constant. Since making \(\hat{x}_1\) small or converging to zero is equivalent to making \(s\) small or converging to zero, the rest work is to make \(s\) as small as possible.

The time derivative of \(s\) is obtained as

\[
\dot{s} = \frac{A_1}{m} \tilde{x}_3 - \frac{B_c}{m} \frac{k}{x_2} - \frac{k_1}{m} x_1 + d_i - \tilde{x}_d + \lambda (x_2 - \tilde{x}_d) - k_1 s
\]  

(18)

A virtual controller \(\bar{a}_3\) for \(\tilde{x}_3\) is designed as

\[
\bar{a}_3 = \frac{m}{A_1} \left[ \frac{B_c}{m} x_2 + \frac{k}{m} x_1 - d_i + \tilde{x}_d - \lambda (x_2 - \tilde{x}_d) - k_1 s \right]
\]  

(19)

where \(k_1\) is a positive constant.

Let \(\bar{s} = \tilde{x}_3 - \bar{a}_3\) denote the tracking error of load pressure, and substituting equation (19) into equation (18) yields

\[
\dot{s} = \frac{A_1}{m} \bar{s}_3 - k_1 s + \tilde{d}_1
\]  

(20)
Define the Lyapunov function as
\[ V_1 = \frac{1}{2} s^2 + \frac{1}{2} \bar{d}_1^2 \] (21)
Combining equations (16) and (20), the time derivative of \( V_1 \) is given by
\[
\dot{V}_1 = \frac{A_1}{m} s \ddot{s} - k_1 s^2 + s \ddot{d}_1 - \frac{1}{\bar{c}_1} \bar{d}_1^2 + \ddot{d}_1 \dot{d}_1 \\
\leq \frac{A_1}{m} s \ddot{s} - k_1 s^2 - \frac{1}{\bar{c}_1} \bar{d}_1^2 + s \ddot{d}_1 + |\bar{d}_1||\dot{d}_1| \\
\leq \frac{A_1}{m} s \ddot{s} - k_1 \left( s - \frac{1}{2 \bar{c}_1} \bar{d}_1 \right)^2 \\
- \gamma \left( |\bar{d}_1| - \frac{1}{2 \gamma} |\dot{d}_1| \right)^2 + \frac{1}{4 \gamma} \bar{d}_1^2 \max
\] (22)
where \( \gamma = 4k_1 - \bar{c}_1/4k_1\bar{c}_1 \) and could be turned to be positive by selecting the control parameters.

**Step 2.** In this step, an actual control law for \( u_d \) is determined. The time derivative of \( \ddot{s} \) is given by
\[
\ddot{s}_3 = -f_1 x_2 - f_2 x_3 + f_3 x_4 + f_4 u_d + d_2 - \ddot{a}_3 \] (23)
The actual control \( u_d \) is designed as
\[
u_d = \frac{1}{f_4} \left( f_1 x_2 + f_2 x_3 - f_3 x_4 - \ddot{a}_2 + \ddot{a}_3 - k_2 \ddot{s}_3 - \frac{A_1}{m} s \right)\] (24)
where \( k_2 \) is a positive constant. Substituting equation (24) into equation (23) yields the dynamics of \( \ddot{s}_3 \) as
\[
\ddot{s}_3 = \ddot{d}_2 - k_2 \ddot{s}_3 - \frac{A_1}{m} s
\] (25)
Define the Lyapunov function as
\[
V = V_1 + \frac{1}{2} \ddot{s}_3^2 + \frac{1}{2} \ddot{d}_2^2
\] (26)
Combining equations (16), (22), and (25), the time derivative of \( V \) is given by
\[
\dot{V} = \dot{V}_1 + \ddot{s}_3 \ddot{s}_3 + \ddot{d}_2 \ddot{d}_2 \\
\leq -k_1 \left( s - \frac{1}{2 \bar{c}_1} \bar{d}_1 \right)^2 - \gamma \left( |\bar{d}_1| - \frac{1}{2 \gamma} |\dot{d}_1| \right)^2 + \frac{1}{4 \gamma} \bar{d}_1^2 \max \\
- k_2 \left( \ddot{s}_3 - \frac{1}{2 \bar{c}_2} \ddot{d}_2 \right)^2 - \beta \left( |\ddot{d}_2| - \frac{1}{2 \bar{d}_2} |\bar{d}_1| \right)^2 + \frac{1}{4 \bar{d}_2} \bar{d}_2 \max
\] (27)
where \( \beta = 4k_2 - \bar{c}_2/4k_2\bar{c}_2 \) and could be turned to be positive by selecting the control parameters. Thus, the states of the closed-loop system enter into the bounded ball \( Br \) in finite time and stay within \( Br \), where \( Br \) could be expressed as
\[
Br = \left( k_1 \left( s - \frac{1}{2 \bar{c}_1} \bar{d}_1 \right)^2 + \gamma \left( |\bar{d}_1| - \frac{1}{2 \gamma} |\dot{d}_1| \right)^2 \right) \\
+ k_2 \left( \ddot{s}_3 - \frac{1}{2 \bar{c}_2} \ddot{d}_2 \right)^2 + \beta \left( |\ddot{d}_2| - \frac{1}{2 \bar{d}_2} |\bar{d}_1| \right)^2 \\
= \frac{1}{4 \beta} \ddot{d}_2 \max + \frac{1}{4 \gamma} \bar{d}_1 \max
\] (28)
It can be observed that the size of the boundedness ball \( Br \) mainly depends on \( \gamma \) and \( \beta \). Thus, the high observer gain—low values of \( \bar{c}_1 \) and \( \bar{c}_2 \)—can shrink the size of \( Br \).

**VSPC**
The traditional EHSS exhibits inferior efficiency due to the fact that the amount of energy produced by its power unit is constant, while the power needed in the actuator is actually variable. Therefore, in many cases of EHSS operation, a large amount of energy will be constantly consumed by the supply unit to follow a desired trajectory.

For a certain demand flow, the smallest possible pressure drop will occur if the valve is fully opened. However, in that case, the PDV cannot be adjusted in a wide range to compensate for the tracking error. Thus, in order to save energy, the input signal of the PDV should be kept at a relatively large value.

For the forward motion \( (\dot{x}_3 \geq 0) \), the pressure drops across the valve can be expressed as follows
\[
\Delta p_1 = p_s - p_1 \] (29)
\[
\Delta p_2 = p_2 - p_1 \] (30)
Then, the valve orifice equation gives
\[
A_1 \dot{x}_d = k_4 k_4 \alpha \mu_{\text{dmax}} \sqrt{\frac{2}{p}} \Delta p_1 \] (31)
\[
A_2 \dot{x}_d = k_4 k_4 \alpha \mu_{\text{dmax}} \sqrt{\frac{2}{p}} \Delta p_2 \] (32)
where \( \mu_{\text{dmax}} \) is the absolute maximum input signal of the PDV, and \( \alpha \) is the desired normalized input signal of PDV determined by the users/designers to indicate the importance of energy saving.

\( p_1 \) can be evaluated by the following force equation
\[
p_1 A_1 - p_2 A_2 = m \dot{x}_d + B_c \dot{x}_d + k_4 x_4 - m \dot{d}_1 \] (33)
By taking the pipe pressure loss into consideration, the desired pump pressure can be obtained from equations (29)–(33).
Thus, an initial value of

\[ p_{a1} = \frac{1}{A_1} \left[ \left( \frac{\rho A_1^2 x_d^2}{2k_i^2 k_r^2 \alpha^2 u_{dmax}^2} + p_i \right) A_2 + \right. \]

\[ \left. m x_d + B_c x_d + k x_d - m \dot{x} + \Delta p_{\text{loss}} \right] \]

where \( p_{a1} \) is the desired pump pressure in the forward motion, and \( \Delta p_{\text{loss}} \) is the pipe pressure loss which is set as a constant value. Similar to the case of forward motion, the desired pump pressure when \( \dot{x}_d < 0 \) is expressed as

\[ p_{a2} = \frac{1}{A_2} \left[ \left( \frac{\rho A_1^2 x_d^2}{2k_i^2 k_r^2 \alpha^2 u_{dmax}^2} + p_i \right) A_1 - \right. \]

\[ \left. m x_d - B_c x_d - k x_d + m \dot{x} + \Delta p_{\text{loss}} \right] \]

and

\[ \frac{\rho A_1^2 x_d^2}{2k_i^2 k_r^2 \alpha^2 u_{dmax}^2} + \Delta p_{\text{loss}} \]

It can be observed that the desired pump pressure \( (p_{a1}) \) in the forward motion and \( p_{a2} \) in the backward motion mainly depends on the system parameters. Furthermore, the lumped uncertainties can be compensated by the DOB.

Since the flow rate of the pump is always higher than the flow rate required in actuator, the supply pressure always exceeds the cracking pressure (i.e. the PRV is always open). Thus, the normalized input signal of the PRV is calculated by the following simplified equation\(^6\)

\[ u_r = \begin{cases} \frac{p_{a1}}{k_r u_{rmax}} & \dot{x}_d \geq 0 \\ \frac{p_{a2}}{k_r u_{rmax}} & \dot{x}_d < 0 \end{cases} \]

where \( k_r \) is the gain of PRV, and \( u_{rmax} \) is the absolute maximum input signal of the PRV.

**Remark 2.** A suitable initial value of \( \dot{a}_1 \) can be set in the DOB. Thus, an initial value of \( p_{a1} \) which is large enough to meet the control demand can be achieved according to equations (34) and (35). Once the pump pressure meets the control demand, the estimation error \( \dot{a}_1 \) will be stabilized by the nonlinear position tracking controller. Consequently, the pump pressure can always be kept at a relatively low level by the VSPC.

**Result and discussion**

In order to evaluate the performance of the proposed control algorithms, experiments were conducted on an experimental test rig which is shown in Figure 3. The position of the cylinder was measured by a displacement sensor (LS 628C). The pressures in the two cylinder chambers and supply pressure were measured by pressure sensors (HDP702). A dSpace1104 data acquisition board was used to acquire the feedback signals from sensors and generate control signals to the system. The main parameters of the system and the controller are shown in Table 1.

Figure 4 depicts the tracking errors of the proposed controller with three different values of \( \alpha \), and the tracking error of a proportional–integral (PI) controller is also demonstrated for verification. The parameters of the PI controller are set to the values that give the best tracking performance in the test

\[ u_d = 76 \left[ (x_d - x_1) + \int_0^t (x_d - x_1) dt \right] \]
It can be observed that the tracking errors of the proposed controller when $\alpha$ is selected as 0.75 and 0.8 are almost the same as 1.2 mm. However, a large tracking error occurs when $\alpha$ is selected as 0.85. It is clear that a higher value of $\alpha$ can further improve the system efficiency at the cost of a larger tracking error. Thus, the value of $\alpha$ is tuned as 0.8 by taking both energy saving and tracking performance into consideration. The largest deviations of actuator position always appear in the peaks of the reference signal (Figure 5) due to the fact that the pressure differences across the valve are relatively small in these points and the designed controller always applies high command signals to the PDV (Figure 6). Furthermore, the input signal of PRV (Figure 7) is variable and controlled by the VSPC to achieve an energy-saving performance. Thus, the system is prone to be saturated in these points. Furthermore, the results also demonstrated that the proposed controller exhibits better performance than PI controller in terms of tracking errors.

The observe results of $d_1$ and $d_2$ are shown in Figures 8 and 9, respectively. In order to verify the observer performance of $d_1$, the observer performance is evaluated by means of a comparison between the driving force and estimated load force, which is shown in Figure 10. The driving force means $(p_1 - ap_2)A_1$, and the estimated load force is $m\ddot{x}_d + B_c\dot{x}_d + kx_d - md_1$. It can be seen from Figure 10 that the estimated load force is similar to the driving force. Thus, we can conclude that the unknown disturbance is well estimated, and the parametric uncertainties and uncertain nonlinearities of equations (34) and (35) could also be compensated by the DOB. Although the estimation accuracy of $d_2$ cannot be verified, we can still draw a conclusion that the disturbance $d_1$ is well estimated due to the precise tracking performance.

The pump pressure and chamber pressures are shown in Figure 11. It can be observed that the pressure is boosted when the demand flow is increasing and vice versa. A higher pressure margin always appears at the middle stroke of the cylinder in both forward and backward motions, which can verify the effectiveness of the energy-saving method. For the purpose of comparison, the VSPC part is omitted by setting the PRV input to a constant value. In this case, the system is similar as a fixed displacement system, in which the supply pressure is set as 42 bar. Figure 12 demonstrates the pump pressure and chamber pressures of this case. Figure 13 shows the comparison of energy supply between the fixed displacement system and the proposed system. The energy is calculated as follows

$$W = \int_0^t \frac{p_s Q_L}{600} \, dt \quad (38)$$

where $W$ denotes the energy whose unit is kJ, $p_s$ is the supply pressure whose unit is bar, and $Q_L$ is the flow rate of the pump which is 24 L/min.

It can be seen from Figure 13 that the energy supply of the fixed displacement system is 13.1 kJ, and the energy supply of the proposed system is 7.2 kJ. Therefore, it can be concluded from the
The aforementioned analysis that the proposed method is capable of saving 45% energy during a typical harmonic trajectory tracking test with the maximum tracking error of 1.2 mm.
To further verify the performance of the proposed method in position tracking and energy-saving aspects, other tests which adopt the multi-step profile as the desired trajectory are conducted by setting the value of $\alpha$ as 0.73. The tracking performance and tracking error are depicted in Figures 14 and 15, respectively. The results demonstrate that the position tracking performance is satisfied with a maximum tracking error of 1.2 mm. Furthermore, it can be seen that the dynamic response of the proposed system has not been changed compared with that of a fixed displacement system by comparing Figures 14 and 16. It is due to the fact that the desired pump pressure mainly depends on the system parameters instead of feedback signals. Meanwhile, it can be seen from Figures 17 and 18 that the energy-saving effect is also achieved. The amount of energy provided by proposed system is only 4.2 kJ and about 68% energy can be saved (Figure 19).

**Conclusion**

1. A DOB-based cascade controller was developed in this study for precise tracking control of EHSS. The proposed method exhibits a good tracking capacity with the maximum tracking error of 1.2 mm in two kinds of tests. Compared with Tivay et al.,\textsuperscript{14} the proposed method has a relatively small tracking error.
2. A DOB-based VSPC was established for energy saving. Results show that the amount of saved energy in this study is 45% in the harmonic trajectory tracking test and 68% in the multi-step tracking test.
trajectory tracking test compared with a fixed displacement system. Compared with Baghestan et al., the proposed method has a better energy-saving performance in terms of the harmonic trajectory tracking task.

3. The effectiveness of the proposed nonlinear energy-saving control strategy was validated by both harmonic trajectory test and multi-step trajectory test. The results demonstrate that both precise tracking performance and significant energy-saving effect of EHSS were achieved. Moreover, this method could be combined with the independent metering to achieve even more energy in mobile machinery and other applications.

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