A practical nonlinear robust control approach of electro-hydraulic load simulator

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Abstract This paper studies a nonlinear robust control algorithm of the electro-hydraulic load simulator (EHLS). The tracking performance of the EHLS is mainly limited by the actuator’s motion disturbance, flow nonlinearity, and friction, etc. The developed controller is developed based on the nonlinear motion loading model. The problems of the actuator’s disturbance and flow nonlinearity are considered. To address the friction problem, the friction model of the loading motor is identified experimentally. The friction disturbance is compensated using the obtained friction model. Therefore, this paper considers the main three factors comprehensively. The developed algorithm is easy to apply since the controller can be obtained just with one step back-stepping design. The stability of the developed algorithm is proven via Lyapunov analysis. Both co-simulation and experiments are performed to verify the effectiveness of this method.

1. Introduction

Load simulator is crucial equipment in hardware-in-the-loop (HIL) experiments, which is widely used in aviation and aerospace fields. Its main function is to generate the torque/force to simulate the aero-dynamic load acting on the actuator system, so that the whole flight control system, which includes the performance of the flight control algorithm and the reliability of the actuator system, can be verified under the real flight condition in the laboratory. The designer of the actuator system, by means of the load simulator, can foresee and detect potential problems related to the flight control algorithm and the actuator mechanics. A load simulator offers a much more efficient development platform in terms of time and cost. According to the type of the energy source, a load simulator can be classified into three types: electro-hydraulic load simulator (EHLS), electric load simulator (ELS), and pneumatic load simulator (PLS). Compared to ELS and PLS, EHLS has many advantages such as: durability, high power to weight ratio, controllability, accuracy, and reliability. In view of these advantages, EHLS have found a wide range of applications in aircraft and missile industries, automotive industry, robotics and fault tolerant fields.

Aside from the common problems such as parameter uncertainties and nonlinear characteristics which all hydraulic servo systems possess, the most crucial problem for an EHLS is the external disturbance caused by the actuator’s active operation. Improvement of tracking performance of EHLS has been of
great interest from both academic and industrial perspectives, and extensive research has been done to resolve these problems. A common and natural idea is to carry out a feed-forward compensation using actuator’s velocity signal. Liu, Jacazio and Balossini exploited actuator velocity to improve the tracking performance of an EHLS. Jiao et al. proposed to make use of the actuator valve’s input to decouple the actuator’s motion disturbance. Based on this study, the velocity gap between the actuator and the loading system was extracted to make a further compensation and the dual-loop scheme was developed. Li et al. developed the double-valve method in which a pressure servo valve was paralleled with a flow valve for the EHLS. The pressure valve was mainly responsible for tracking load instruction and the paralleled flow valve was responsible for releasing the disturbance flow caused by the actuator’s exercise. The robustness against actuator disturbance was improved partially because the pressure valve was less sensitive to flow variation than the flow servo valve. Li proposed a control scheme for an EHLS that was composed by a constant compensator, an inner-loop controller, and an outer-loop controller. The function of the constant compensator and the inner controller was to suppress the motion disturbance and the outer-loop controller was designed to improve tracking performance of the loading system. Su et al. developed a novel load structure, of which another set position servo system was introduced to connect in series to an EHLS for releasing the disturbance flow. One deficiency of this method is the mechanical structure was too complicated and the cost was high. In Ref. [3], a hybrid cylinder was investigated as the actuator of an EHLS and the grey predictor–identifier control, the fuzzy technology, neural networks, and the $H_{\infty}$ mixed sensitivity theory, all have been implemented for EHLS. The central issue of these studies is to apply a certain robust algorithm for loading systems. However, the disturbance boundary was hard to determine and the disturbance strength was very serious in some occasions.

For most papers concerning on hydraulic position servo control, it is very common to focus the efforts on the problem of nonlinear and parametric uncertainty. For most studies about load simulator, however, they usually give the center stage to suppress the actuator’s motion disturbance while deemphasizing the nonlinear nature and the parameter uncertainty problem. Although a few researches have taken the nonlinear and parametric uncertainty problem into consideration for force/pressure systems, the loaded objectives in these studies have no active exercise. Alleyne et al. has detailed the reason of limitation when using a simple control for a hydraulic force servo system. The EHLS is actually a motion force/torque hydraulic servo system with serious external disturbance; hence, it is necessary to resort to some advanced control technologies to improve the tracking performance.

Seminal works in the field of adaptive robust control for uncertain nonlinear systems were done in Refs. [35–37] which treated uncertain nonlinearity and parameters in a systematic way for hydraulic position servo systems. In this work, the adaptive robust torque control for an EHLS is studied. The principal contribution of this work is that the EHLS is addressed as a nonlinear motion loading system rather than a linear torque servo system with external disturbance. This result in the developed nonlinear controller is decoupling against the actuator’s motion disturbance. In addition to the actuator’s disturbance, friction and flow nonlinearity are also addressed with the developed controller. The rest of the paper is organized as follows. Section 2 gives a brief introduction about the EHLS. The nonlinear robust controller based on load flow planning is developed in Section 3. Both the design procedure and the stability analysis are presented. Section 4 presents co-simulation and experiment results. Finally, conclusions are drawn in Section 5.

2. System description

In general, an HIL experiment is mainly composed by two sets of servo systems which are the actuator and the EHLS system. The schematic diagram and the oil line principle of the EHLS are described by Fig. 1. As shown in Fig. 1(a), the left part denotes the position actuator system which is equipped with a servo valve, a hydraulic swing motor, and an angular encoder. The signal from the angle encoder is fed back to the actuator controller to achieve servo angle control. The right part is the EHLS which is composed by a valve controlled hydraulic swing motor, an angular encoder, a torque sensor, and an inertia disk to simulate the inertia of the control surface. The EHLS exerts torque to simulate the air dynamic load acting on the control surface of the actuator system in a flight process. Obviously, the actuator’s active exercise will impact the torque tracking performance of the EHLS.

Unlike a common force/torque servo system in which a loaded objective has no active motion, the control objective of the EHLS, i.e., the output of the torque sensor (see Fig. 1), is governed by the angle difference between the ends of the torque sensor. Therefore, torque output can be expressed as

$$T_L = K_S (\varphi_L - \varphi_A)$$  \hspace{1cm} (1)

where $T_L$ is the torque output of the loading system (N·m), $K_S$ is the stiffness of the torque sensor (N·m/rad), $\varphi_L$ and $\varphi_A$ are the angular displacements of the EHLS and the actuator system, respectively (rad).

According to the oil principle shown in Fig. 1(b), the flow continuity equation of the loading motor can be established. For simplification, we assume that the servo valve is matched symmetrically with ideal zero opening and zero lapping and the spool of the valve radial-clearance leakage and the external leakage of the load motor are both negligible, as well as stable oil source pressure, zero return pressure, and constant oil elastic modulus. Based on these assumptions, the load flow equation can be given:

$$Q_L = D_L \dot{\varphi}_L + \frac{V}{4\beta_e} P_L + C_I P_L$$  \hspace{1cm} (2)

where $D_L$ is the displacement of the hydraulic motor ($m^3$/rad), $P_L = P_1 - P_2$ (N/m²) is the pressure difference between the two chambers of the loading motor, $P_1$ and $P_2$ are the pressure in forward and return chamber, respectively. $Q_L$ is the load flow rate ($m^3$/s), $\dot{\varphi}_L$ is the angular position of the loading system (rad/s), $V$ is the total control volume of the EHLS system ($m^3$), $\beta_e$ is the effective bulk modulus (N/m²), and $C_I$ is the coefficient of the total internal leakage of the loading motor due to pressure ($m^3$/N·s)).
Rewriting Eq. (2) yields
\[ \dot{\phi}_l = zQ_L - \theta_1 \dot{P}_L - \theta_2 P_L \]
where \( z = 1/D_L \), \( \theta_1 = V/(4\beta_6 D_L) \), and \( \theta_2 = C_d/D_L \).

The load flow \( Q_L \) governed by the spool displacement can be given as
\[ Q_L = C_d \omega x_V \sqrt{P_A - \text{sgn}(x_V) P_L / \rho} \]
where \( C_d \) is the flow coefficient of the EHLS valve, \( \omega \) is the area gradient of the EHLS valve \((\text{m}^2)\), \( x_V \) is the spool displacement \((\text{m})\), \( P_A \) is the oil source pressure \((\text{N/m}^2)\).

Considering that the dynamic of the servo valve used in this paper is much higher than that of the system, the spool dynamic is ignored
\[ x_V = k_V u \]
where \( k_V \) is the valve gain \((\text{m/V})\), \( u \) is the control output \((\text{V})\), and \( \text{sgn}(\cdot) \) denotes the discontinuous sign function which is defined as
\[ \text{sgn}(\cdot) = \begin{cases} 1 & \text{if } \cdot > 0 \\ 0 & \text{if } \cdot = 0 \\ -1 & \text{if } \cdot < 0 \end{cases} \]

Given the desired torque reference \( T_d \), our aim is to synthesize a control output \( u \) so that the torque output of the loading system \( T_L \) tracks \( T_d \) as closely as possible in spite of the external disturbance caused by the actuator’s active exercise, nonlinear characteristics, and parametric uncertainty. In next section, the nonlinear robust controller will be developed for the EHLS.

Before the design process, the following assumption should be introduced first.

**Assumption 1.** It is assumed that both the torque reference \( T_d \) and the actuator’s angle velocity are known and differentiable; the extent of the parametric uncertainties is known, which satisfies
\[ \theta \in \Omega_\theta \triangleq \{ \theta : \theta_{\text{min}} \leq \theta \leq \theta_{\text{max}} \} \]

where \( \theta = [\theta_1, \theta_2]^T \), and \( \theta_{\text{min}} = [\theta_{1\text{min}}, \theta_{2\text{min}}]^T \) and \( \theta_{\text{max}} = [\theta_{1\text{max}}, \theta_{2\text{max}}]^T \) are known lower and upper bound vectors of \( \theta \).

Define the parameter uncertainty bound vector \( \theta_u \)
\[ \theta_u = \theta_{\text{max}} - \theta_{\text{min}} \]
Define the parameter error vector
\[ \Delta \theta = \theta - \hat{\theta} \]
where \( \theta = [\theta_1, \theta_2]^T \) is the estimated parameter vector of \( \theta \), and \( \hat{\theta}_1 \) and \( \hat{\theta}_2 \) are the estimated parameters of \( \theta_1 \) and \( \theta_2 \), respectively.

### 3. Nonlinear robust controller design for EHLS

#### 3.1. Controller design

Define the torque tracking error \( e_1 \)
\[ e_1 = T_L - T_d \]

Its time derivative along Eq. (10) is given as
\[ \dot{e}_1 = K_\delta (\dot{\phi}_L - \dot{\phi}_A) - T_d \]

Substituting Eq. (3) into Eq. (11) yields
\[ \dot{e}_1 = K_\delta (zQ_L - \theta_1 \dot{P}_L - \theta_2 P_L - \dot{\phi}_A) - T_d \]

Define the Lyapunov function \( V_1 \)
\[ V_1 = \frac{1}{2} \dot{e}_1^2 \]

Its time derivative noting Eq. (12) is given as
\[ \dot{V}_1 = e_1 [K_\delta (zQ_L - \theta_1 \dot{P}_L - \theta_2 P_L - \dot{\phi}_A) - T_d] \]
\[ = e_1 [K_\delta (zQ_L - \theta_1 \dot{P}_L - \theta_2 P_L - \dot{\phi}_A + \dot{\theta}_1 \dot{P}_L + \dot{\theta}_2 P_L) - T_d] \]

Define
\[ \psi = [\dot{P}_L \ P_L]^T \]

For Eq. (14), if we treat \( Q_L \) as the control input, we can synthesize a desired load flow to ensure that \( V_1 \) is semi-negative definite, equivalently, to make the torque tracking error dynamic stable. The planned load flow consists of two parts,
\[ Q_{I} = Q_{Lr} + Q_{La} \]  

(16)

Rewriting Eq. (14) noting Eqs. (15) and (16) yields

\[ \dot{V}_{1} = e_{1}\left[ K_{S}\left( zQ_{La} - \vartheta^{T}\psi - \phi_{A} \right) - \dot{T}_{a} + K_{S}\left( zQ_{La} - \vartheta^{T}\psi \right) \right] \]  

(17)

To ensure that \( V_{1} \) is semi-negative definite, the calculable load flow \( Q_{La} \) can be synthesized as follows

\[
\begin{align*}
Q_{La} &= \frac{1}{2}\left( \dot{\theta}_{d}P_{L} + \dot{\theta}_{d}P_{L} + \dot{T}_{a}/K_{S} + \phi_{A} \right) \\
Q_{Lr} &= -\frac{1}{2}ke_{1}
\end{align*}
\]  

(18)

where \( k > 0 \) is a design parameter which can be arbitrarily big theoretically.

**Remark 1.** \( Q_{La} \) is the calculable load flow to achieve the torque tracking and decouple the external disturbance caused by the actuator’s motion. As shown by Eq. (18), \( Q_{La} \) is composed by the following four parts, in which the first two terms are generated due to the oil compressibility and the internal leakage flow. The third term aims at achieving the torque load and the forth term is responsible for decoupling the external disturbance caused by the actuator’s active exercise. \( Q_{Lr} \) is the linear error feedback term used to address the parameter uncertainty and stabilize the error dynamic system in Eq. (12).

The remaining task, at this stage, is to obtain the real control output based on the planned load flow \( Q_{I} \). With Eq. (4), it is easy to see that the main difficulty to obtain the real control output is how to determine the value for the parameters \( C_{d} \) and \( \phi_{A} \). Fortunately, it is easy to know the rated flow of the servo valve at a certain valve pressure drop in general. For clarity, all the servo valves of Moog Company give the rated flow at a 7 MPa pressure drop through the valve.

\[ C_{d}x_{v_{max}} \max\sqrt{\Delta P_{drop}/p} = Q_{\text{rated}} \]  

(19)

where \( x_{v_{max}} \) is the maximum spool displacement (m), which corresponds to the saturated input \( u_{max} \) and \( Q_{\text{rated}} \) is the rated flow (m\(^3\)/s) when the pressure drop thorough the valve is \( \Delta P_{\text{drop}} \).

Combining Eqs. (19) and (4), the real control output is synthesized as

\[ u = \frac{Q_{I}u_{max}}{Q_{\text{rated}}} \frac{\Delta P_{\text{drop}}}{P_{S} - \text{sgn}(x_{v})P_{L}} \]  

(20)

The developed method can be shown by Fig. 2.

### 3.2. Stability analysis

**Theorem 1.** Let the error feedback control parameter be large enough. Then, the synthesized control output based on the planned load flow Eq. (18) guarantees that the motion loading system is exponentially stable and its torque output tracks error can be determined with the controller parameter.

**Proof of Theorem 1:** Substituting Eq. (18) into Eq. (17) yields

\[
\begin{align*}
\dot{V}_{1} &= e_{1}\left[ K_{S}\left( -ke_{1} - \vartheta^{T}\psi \right) \right] \leq e_{1}\left[ K_{S}\left( -ke_{1} + ||\vartheta^{T}||\psi \right) \right] \\
&\leq -ke_{1}^{2} + K_{S}\left[ ||\vartheta^{T}||\psi \right]e_{1} \\
&\leq -ke_{1}^{2} + K_{S}\left[ ||\vartheta^{T}||\psi \right]e_{1}
\end{align*}
\]  

(21)

Let \( k \) be large enough so that

\[ k > k_{1} + \frac{||\vartheta^{T}||^{2}||\psi||^{2}}{2e} \]  

(22)

holds, where \( k_{1} > 0 \) and \( e > 0 \).

Substituting Eq. (22) into Eq. (21) and by completion of square, it yields that

\[
\begin{align*}
\dot{V}_{1} &\leq -ke_{1}^{2} + K_{S}\left( -\frac{||\vartheta^{T}||^{2}||\psi||^{2}}{2e}e_{1}^{2} + ||\vartheta^{T}||\psi \right) \\
&\leq -k_{1}ke_{1}^{2} + K_{S}\frac{e}{2}
\end{align*}
\]  

(23)

For clarity, let \( k_{e} = k_{1}K_{S} \) and \( \delta = K_{S}e/2 \). Rewriting Eq. (23), we have

\[
\begin{align*}
\dot{V}_{1} &\leq -k_{e}e_{1}^{2} + \delta \Rightarrow V_{1}(t) \\
&\leq \exp(-2k_{e}t)V_{1}(0) + \frac{\delta}{2k_{e}}[1 - \exp(-2k_{e}t)] \Rightarrow e_{1}^{2} \\
&\leq \exp(-2k_{e}t)e_{1}^{2}(0) + \frac{\delta}{k_{e}}[1 - \exp(-2k_{e}t)]
\end{align*}
\]  

(24)

This is to say the converging rate of the error dynamic and the tracking error can be governed by the controller parameter \( k \) in Eq. (18) in the way shown by Eq. (24). Therefore, Theorem 1 is proven.

### 4. Simulation and experiment

#### 4.1. Simulation verification

##### 4.1.1. Simulation modeling

To test the feasibility of the developed controller, co-simulation based on AMESim and MATLAB/Simulink was carried out. The load simulator model was build up in AMESim and the developed controller was achieved in MATLAB/Simulink, as illustrated in Fig. 3. As shown, the left part stands for the loading system and the right part is the actuator system to generate the motion disturbance. The basic mechanism of one co-simulation cycle is that: firstly, the states of the EHLS model are solved by AMESim and fed into the controller developed in Simulink, then the control signal from Simulink is fed back into the EHLS model in AMESim, and finally the states in AMESim are updated. In order to be consistent with the experiment test, the sampling rate was set as 0.0005 s.

In order to improve the accuracy of the co-simulation, the simulation model and their simulation parameters in AMESim were consistent with the test rig as real as possible. The simulation parameters are shown in Table 1. As shown, the
The following characteristics have been taken into consideration: the internal leakage, the static friction, the coulomb friction, and the viscous friction of the loading motor, the stiffness torque sensor (the stiffness of the load shaft), the stiffness of the connecting shaft, the inertia of the loading shaft, the inertia disc, etc. The internal leakage and the friction parameters used in this simulation were obtained through experiments. To obtain the friction parameters, the following experiments were conducted to obtain the data pairs about friction torque versus velocity for the hydraulic swing motor: the actuator system was operated as triangular trajectory and the loading motor was free-state of which two pressure chambers were short-circuit connected. The purpose of setting the triangular reference for the actuator system is to manipulate the actuator system operating as a uniform motion so that the impact of the inertial torque could be avoided. Obviously, the loading motor was driven with the same velocity. With such experiments, the friction torque could be measured directly by the torque sensor. The extensive pairs of data about velocity versus friction torque are shown in Fig. 4. Based on the tested friction torque data, the relative friction parameters such as static friction, viscous friction, and coulomb friction parameters could be determined.

4.1.2. Simulation results

The loading system tracked sinusoidal torque instruction 500 N·m and 6 Hz under external sinusoidal motion disturbance with an amplitude of 2 degrees and a frequency of 6 Hz, which was generated by the actuator system. The following comparative simulations between the PID and the developed NRC (nonlinear robust control) were conducted. As illustrated in Fig. 5, the PID parameters were well tuned until the system produced a slight shock. The control parameters of the PID in the simulation were: P-gain 0.015, I-gain 0.05, and the control parameters of the NRC were selected as: k = 1200, the estimates for \( \hat{\theta}_1 \) and \( \hat{\theta}_2 \) are selected as 4.2054 \( \times 10^{-10} \) and 5.0739 \( \times 10^{-9} \), respectively.

Fig. 6 (a) shows that there is an obvious phase lag just based on the PID control and the tracking error was more than 200 N·m. Both the serious external disturbance and the response lag of the loading system lead to the worse performance. With the developed approach, the tracking performance was improved significantly and the tracking error was reduced to nearly 30 N·m. The feasibility and effectiveness of the developed method were verified by the co-simulation results.

4.2. Experiment verification

4.2.1. Experimental setup configuration

The developed control approach was also performed on an experiment platform which consisted of four parts: the oil source system, the mechanical bed, the loading system, and the simulation actuator system. The photograph of the test rig is depicted as in Fig. 7(a) and its structure is shown in Fig. 7(b). In the following experiments, the motion disturbance was generated by the angle position system, so the real HIL work conditions could be reproduced. The nozzle flapper type servo-valves (D765) manufactured by MOOG company were equipped. The effective angle range of the hydraulic swing motor was ±45 degree. The angle position and the torque feedback were obtained by the angle encoder and the patch type torque sensor, respectively.

<table>
<thead>
<tr>
<th>Table 1 Simulation parameters in AMESim.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameter</td>
</tr>
<tr>
<td>Motor displacement (cm³/rad)</td>
</tr>
<tr>
<td>Angular stroke (rad)</td>
</tr>
<tr>
<td>Coulomb friction (N·m)</td>
</tr>
<tr>
<td>Viscous friction (N·m·rad⁻¹·s⁻¹)</td>
</tr>
<tr>
<td>Connecting stiffness (N·m/rad)</td>
</tr>
<tr>
<td>Control volume (cm³)</td>
</tr>
<tr>
<td>Inertia disc (kg·m²)</td>
</tr>
</tbody>
</table>

As illustrated in Fig. 5, the PID parameters were well tuned until the system produced a slight shock. The control parameters of the PID in the simulation were: P-gain 0.015, I-gain 0.05; and the control parameters of the NRC were selected as: \( k = 1200 \), the estimates for \( \hat{\theta}_1 \) and \( \hat{\theta}_2 \) are selected as 4.2054 \( \times 10^{-10} \) and 5.0739 \( \times 10^{-9} \), respectively.
About the pressure supply system, a variable displacement pump was used and driven by an AC motor so that the pump was capable of supplying pressured oil. The security pressure was limited to 31 MPa by a relief valve. The pressure output can be set to any valve between 0 and 21 MPa by regulating the proportional pressure reducing valve. The EHLS consists of a hydraulic swing motor, a mechanical framework, a servo valve, a torque sensor, an angle encoder, and a computer included PCI-bus multifunction card. A 16-bit A/D converter and a 16-bit D/A converter were used. The specific parameters and brands of the main components of the test bed are listed in Table 2.

4.2.2. Comparative experiment results

Actual system parameters were the same as those in the simulation section. The derivatives of load pressure required in the control law, both in the above co-simulation and in the experiments, were implemented by a fourth-order Butterworth filter with a 100 Hz cut-off frequency.

For the sake of convenience, we rewrite the developed controller as follows:

\[
\begin{align*}
Q_{Lr} &= \frac{\Delta P_{\text{drop}}}{1} \\
Q_{La} &= \frac{1}{2} (\dot{\phi}_A + \dot{T}_d/K_s + \dot{\phi}_l P_L + \dot{\phi}_l P_L) \\
Q_L &= Q_{Lr} + Q_{La} \\
u &= \frac{Q_L}{Q_{\text{rated}}} \sqrt{P_S - \text{sgn}(x_V) P_L} \cdot u_{\text{max}}
\end{align*}
\]  

(25)
To test the performance of the developed controller, the developed algorithm is decomposed into the following five cases:

- **Case 1**: Firstly, only the error feedback term (subscript 1) and the disturbance velocity term (subscript 2) were used (see Eq. (25)). It is worth pointing out that the flow nonlinearity was not considered in this case. Case 1 is actually the proportional control plus disturbance velocity feed-forward method which is a traditional control strategy for load simulators.

- **Case 2**: Secondly, based on Case 1, the flow nonlinearity was taken into consideration.

- **Case 3**: Then, based on Case 1, the term \( \hat{T}_{d}/K_S \) (subscript 3) was added.

- **Case 4**: Moreover, based on Case 3, the term \( \hat{\theta}_T P_L \) (subscript 4) was exploited.

- **Case 5**: Finally, the term \( \hat{\theta}_T P_L \) (subscript 5) was added. Case 5 is the developed approach in this paper.

The five cases experiments were performed and compared under the same external disturbance condition as in the co-simulation section: the actuator tracked sine instruction \( 2\pi \), 6 Hz and the loading system tracked 500 N/m, 6 Hz. The actuator’s motion disturbance is shown in Fig. 8, and the loading experiment data are shown in Fig. 9(a)-(j).

As shown in Fig. 9(a) and (b), the tracking error based on Case 1 is 104 N·m; Fig. 9(c) and (d) shows that, after taking the flow nonlinearity into consideration, the tracking error is slightly reduced to 96 N·m. The performance was not improved significantly with considering the nonlinear factor. It is worth noting that, in order to eliminate the shocking caused by the sign function, the sign function was replaced by a smooth hyperbolic function during the experiments. This leads to the performance not being improved significantly when the nonlinear factor was considered.

The principal factors that limit the performance of the loading system are the actuator’s exercise disturbance and the response lag of the loading system. This can be verified by the experiment data based on Case 3, which is shown in Fig. 9(e) and (f). The phenomenon of phase lag was improved significantly after the term \( T_{a}/K_S \) was exploited and the tracking error was reduced to 40 N·m. This can be interpreted as that the role of this term is like the instruction feed-forward to some extent; therefore, it could improve the response performance of the loading system. Fig. 9(g) and (h) illustrates the results based on Case 4. The tracking error was further reduced to nearly 30 N·m. The reason of the performance improvement can be interpreted as that the internal flow was compensated with the term \( \hat{\theta}_T P_L \). Based on Case 5 as shown in Fig. 9(i) and (j), the tracking was further reduced slightly to less than 30 Nm. Experiment data illustrated that the role of the term \( \hat{\theta}_T P_L \) was not very important. This is easy to understand since the role of this term was to compensate the flow due to the oil compressibility in theory, while the modulus of hydraulic oil was very large so the effect of this term was very weak. Fig. 10 gives the control output, the load pressure and its derivative of Case 5. Fig. 11 shows the experimental results of tracking random torque load spectrum.

5. Conclusions

(1) The developed control law shows that, besides the torque tracking error and the actuator’s velocity, the derivative of the torque instruction, the load pressure, and the load pressure’s derivative of the loading system can also be used to improve the torque tracking performance for the EHLS.

(2) Physically speaking, the role of the actuator’s velocity is to decouple the external disturbance caused by the actuator’s active operation; the load pressure compensates the internal leakage of the loading motor; the load pressure’s derivative occurs because the oil is compressible; the derivative of the torque instruction compensates the load flow of deformation resulted from the load torque, and can be viewed as the instruction feed-forward to some extent.

### Table 2 Main component of test rig.

<table>
<thead>
<tr>
<th>Element</th>
<th>Type</th>
<th>Marks</th>
<th>Quantity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Piston pump</td>
<td>A4VSO40DR/10PR25NOO</td>
<td>REXROTH</td>
<td>1</td>
</tr>
<tr>
<td>AC motor</td>
<td>30 kW, 280 V,4 poles, B35</td>
<td>ABB</td>
<td>1</td>
</tr>
<tr>
<td>Relief valve</td>
<td>DBW10B1-5X/315-6EG24Nqk4</td>
<td>REXROTH</td>
<td>1</td>
</tr>
<tr>
<td>Hydraulic motor</td>
<td>DL (115 cm³/rad), range: ±45 deg</td>
<td>Self-development</td>
<td>2</td>
</tr>
<tr>
<td>Servo valve</td>
<td>D765–1633–5,38 L/min</td>
<td>MOOG</td>
<td>2</td>
</tr>
<tr>
<td>Digital encoder</td>
<td>ECN413</td>
<td>HEIDENHAIN</td>
<td>2</td>
</tr>
<tr>
<td>Torque sensor</td>
<td>Strain gauge(HBM)</td>
<td>Institute 701</td>
<td>1</td>
</tr>
<tr>
<td>Pressure sensor</td>
<td>US175-c00002-20086</td>
<td>MEAS</td>
<td>2</td>
</tr>
<tr>
<td>Computer</td>
<td>IEI Ws-855GS</td>
<td>ADVANTECH</td>
<td>1</td>
</tr>
<tr>
<td>A/D card</td>
<td>PCI-1716, 250 kHz/s</td>
<td>ADVANTECH</td>
<td>1</td>
</tr>
<tr>
<td>D/A card</td>
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Fig. 9  Experimental data based on Case 1–Case 5.
A practical nonlinear robust control approach of electro-hydraulic load simulator

Fig. 10 Control output, load pressure and its derivative based on Case 5.

Fig. 11 Experiment result of tracking random load spectrum.

(3) Moreover, the advantage of the proposed controller is the simplicity. It can be obtained just through one step back-stepping design process.

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