Self-Anti-Disturbance Control of a Hydraulic System Subjected to Variable Static Loads

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Featured Application: An improved Self-Turbulent Flow (STF) algorithm is proposed. A mathematical model of the Electrohydraulic Servo (EHS) control system for Lubricating Oil Static Pressure (LOSP) acquisition is presented, and the STF controller is designed for numerical analysis. The Self-Anti-Disturbance (SAD) control strategy for the LOSP of an EHS system is discussed. The proposed SAD control algorithm is verified by experiments, and the LOSP acquisition followability and monitoring accuracy are greatly improved.

Abstract: A hydraulic system’s lubricating oil is subject to serialized variable static loads with performance. An improved self-turbulent flow algorithm, based on the real-time acquisition and monitoring of lubricating oil static pressure in a hydraulic system to simulate variable static loads, is proposed. A mathematical model of the electrohydraulic servo control system for lubricating oil static pressure acquisition is presented, and the self-turbulent flow controller is designed for numerical analysis. The self-anti-disturbance control strategy for the lubricating oil static pressure of an electrohydraulic servo system is discussed, which is used for quadratic optimization, pole placement, PID, and self-turbulent flow control, and the lubricating oil static pressure simulation model of self-turbulent flow control is constructed by a SIMULINK module. The numerical simulation results indicate that the overshoot is significantly reduced. The proposed self-anti-disturbance control algorithm is verified by experiments, and the lubricating oil static pressure acquisition followability and monitoring accuracy are greatly improved. Variable hydraulic lubricating oil static pressure acquisition and monitoring can be effectively and stably adjusted by a predesigned electrohydraulic servo control system in the field of power hydraulic fluid lubrication.

Keywords: hydraulic system; variable static loads; lubricating oil static pressure; self-turbulent flow algorithm; electrohydraulic servo; self-anti-disturbance control strategy

1. Introduction

A hydraulic system typically transfers a fluidic medium from one hydraulic element to another, capable of delivering serialized variable hydraulic loads to contact lubrication areas [1,2]. The core components of a hydraulic system are an actuator or hydraulic motor, different control valves, accumulators, and a hydraulic pump [3,4]. The variable Lubricating Oil Static Pressure (LOSP) acquisition and Self-Anti-Disturbance (SAD) control strategy of the entire hydraulic system are key links, which are adversely constrained by the serialized Variable Static Loads (VSL) of different hydraulic components [5]. The acquisition efficiency is lower with the increase in the number of hydraulic components, which weakens the effectiveness of the control strategy of the hydraulic system. As a result, a hydraulic system’s
optimal design may maximize the timeliness of the control strategy [6–8]. When it is mandatory for the hydraulic system to have dual functions, the number of hydraulic components is usually higher [9,10]. Hence, the SAD function affecting the overall control strategy of the hydraulic system is relatively larger, and the key corrective measure is to reduce the connection of hydraulic components or seek to optimize the control link design [11,12]. In this investigation, a new hydraulic control system strategy is proposed to verify the simulation results of LOSP acquisition and monitoring. At present, others have studied state stochastic linear quadratic optimal control, proposing that the optimal control strategy is a piecewise affine function of the system state [13,14]. Consideration should be given to the controllable high precision of hydraulic systems that have been widely used in many industries. The hydraulic system can provide a high rigidity LOSP, which is a prominent advantage and necessary for the above description. The bottleneck of the high-precision control of a hydraulic system is its nonlinear characteristics and modeling uncertainty; to be specific, the parametric quantification uncertainty and Self-Anti-Disturbance (SAD) are the main obstacles now. Therefore, an optimal control strategy for parameter uncertainty handling and SAD improvement is realized synergistically. A new method for inverse resonance assignment and regional pole assignment for linear time-invariant vibrational systems was presented [15,16]. Currently, although many scholars have used quadratic optimal control and pole configuration to solve related practical engineering problems, this control method has not been introduced into the optimization research of a hydraulic lubrication system [17,18]. There are many research results on the abovementioned issue, and most of the current controllers are mainly anti-interference, represented by adaptive robust control [19,20]. Researchers have applied a variety of advanced control methods to ensure the transient performance of the hydraulic system and the specified steady-state performance. An Electrohydraulic Servo (EHS) control system for LOSP acquisition has been widely used in many fields of modern industry [21,22]. Considering the uncertainty of the EHS system model, for important equipment such as the hydraulic system of a warship’s power rear transmission system, the traditional control algorithm can no longer meet the requirements of this hydraulic system. In this regard, the improved Self-Turbulent Flow (STF) controller for hydraulic system state estimation based on the EHS mathematical model is an effective solution for LOSP acquisition and monitoring [23,24]. The precise control of an EHS system for LOSP through an SAD strategy has been studied in many respects, but it has not been applied in practice and remains at the theoretical level. An active disturbance rejection control algorithm to optimize the performance of the hydraulic system was adopted, and the specific servo control technical structure and regulation error feedback mechanism was revealed [25]. It was shown that the integrator series structure of a hydraulic servo control system could correspond to a linear or nonlinear system under the condition of feedback. By considering the system function as state feedback, a scalable state controller was proposed, which used servo feedback to control the hydraulic system [26,27]. To be more specific, a LOSP simulation model of STF control algorithm adjusted by a predesigned EHS system was introduced to simultaneously to deal with the parameter uncertainties and time-varying disturbances demonstrated in hydraulic system. The problem is that the time-varying disturbances cause an asymptotic loss of hydraulic servo system stability, although the desired LOSP acquisition and monitoring performance can be guaranteed at low feedback gains [28,29]. For a stochastic hydraulic system, when multiple heterogeneous disturbances exist simultaneously, the closed-loop system is asymptotically bounded on the mean square, and with the disappearance of the additive disturbance, the equilibrium is globally asymptotically stable theoretically. When the uncertainty terms could be linearly parameterized, an adaptive control scheme with the addition of a power integrator technique was proposed, and the mismatch disturbance was estimated by the designed disturbance controller [30,31]. The global asymptotic stability of the closed-loop system was guaranteed only when the perturbation was constant. Fuzzy logic systems were used to approximate unknown nonlinear functions, and nonlinear disturbance controllers were used to estimate unknown external disturbances. In the above
work, all the signals were semi-globally consistently eventually bounded, and the local neighborhood inclusion errors could converge to the prescribed bounds. The hydraulic actuator adopted a high-order sliding mode controller, and the control gain was adaptive online. The proposed controller still could not achieve asymptotic following control performance. In this paper, a disturbance controller for a pump-controlled hydraulic system based on the error function is developed, and asymptotic following control is realized. The desired asymptotic LOSP control can also be achieved by using a jitter-free robust finite-time output feedback control scheme. Failure to consider the parameter uncertainties will result in too high a learning burden for the disturbance controller, and the hydraulic system servo performance deteriorates when the considered control device is subject to severe parameter uncertainty.

2. EHS Control Modeling of Hydraulic System

Hydraulic systems transmit power using a fluidic medium. The medium converts the mechanical power from an engine or an electric motor to hydraulic power by rotating the shaft of a hydraulic pump. The hydraulic pump provides flow to the control valve, which directs the same flow to the hydraulic actuator and converts the hydraulic power back to mechanical power. In this subproject, two different non-closed-loop hydraulic systems are presented, as shown in Figure 1, which are determined to analyze the function of an Adjustable Flow Throttle Valve (AFTV). The AFTV is preferred over the choice of a Pilot Operated Relief Valve (PORV) to perform two different performances simultaneously. Moreover, it is more suitable for providing a constant flow whenever the input flow has a fluctuating nature. This novel idea is preferred in a warship power rear transmission system for consistent output power and torque.

![Figure 1. Schematic diagram of a non-closed-loop hydraulic system with AFTV and PORV.](image)

In Figure 1, a lubricating oil hydraulic pump with serialized VSL rigidity supplies a variable flow to the AFTV. The AFTV is responsible for two different loads, namely a first-order load and a second-order load of the hydraulic system. The AFTV directs the Electric Three-Screw Pump (ETSP) flow towards the primary and secondary loads of the hydraulic system. Depending on the load of the hydraulic system, the ETSP flow is
regulated by controlling the movement of the pump shaft. A PORV is installed between the ETSP and AFTV to maintain normal system pressure and thus acts as a safety valve. In the present study, the dynamic model of the hydraulic system is not considered, as the central issue is to analyze the performance of the AFTV using the hydraulic mechanism. The scheme proposes to obtain constant power and torque from the warship Gear Transmission System (GTS) to the warship Controlled Pitch Propeller (CPP) adopting an AFTV and two PORVs. In this case, a hydraulic ETSP is used to provide variable flow to the AFTV. The Primary Lubricating Oil Circuit (PLOC) of the AFTV is tunable per stable load conditions and assembled at a threshold. Once it is above the given flow from the ETSP needed to maintain a steady load, the additional given flow from the ETSP will be allowed through the auxiliary port of the PORV. The surplus flow is returned to the lubricating oil tank in the Secondary Lubricating Oil Circuit (SLOC) of the hydraulic system and stored in the hydraulic accumulator. In contrast to the above, if the given flow from the pump to the AFTV is too low, the flow is unable to provide the hydraulic motor with the required working flow to maintain the constant speed of the GTS and CPP.

In this case, the EHS control valve receives a signal from the on/off controller. Thereafter, the stored oil fluid from the accumulator compensates the supplied flow to the hydraulic motor, which helps maintain the constant speed and torque of the GTS/CPP when the ETSP output flow is undersupplied. The hydraulic system discussed above can replace hydraulic power transmission in turbine applications to obtain the required stable power elements. The schematic diagram of the EHS control system is shown in Figure 2. The block diagram of the hydraulic control system is shown in Figure 3.

![Figure 2. Schematic diagram of the EHS control system.](image)

![Figure 3. Block diagram of the hydraulic control system.](image)

The hydraulic control valve flow equation, the cylinder flow continuity equation, and the force balance equation between the cylinder and the load are established, respectively.

\[ q_L = K_q x_v - K_c p_L \]  \hspace{1cm} (1)

\[ q_L = A_p \frac{dx_p}{dt} + C_{tp} p_L + \frac{V_t}{4\beta_p} \frac{dp_L}{dt} \]  \hspace{1cm} (2)

\[ q_L A_p = m_i \frac{d^2 x_p}{dt^2} + B_p \frac{dx_p}{dt} + K x_p \]  \hspace{1cm} (3)
where \( q_L \) is the VSL flow (m\(^3\)/s), \( K_f \) denotes the supplied flow gain factor, \( x_v \) represents the displacement of EHS valve spool (mm), \( K_c \) is the pressure–flow gain factor, \( p_L \) is the VSL pressure (Pa), \( A_p \) denotes the effective working area of a hydraulic servo cylinder (m\(^2\)), \( x_p \) is the displacement of the piston rod of the hydraulic cylinder (mm), \( C_{tp} \) represents the total leakage coefficient of the hydraulic cylinder, \( V_i \) is the total compressed volume (m\(^3\)), \( \beta_v \) is the effective bulk elastic modulus, \( B_p \) is the VSL flow damping factor, \( m_i \) is the VSL flow quality (kg), and \( K \) is the VSL flow elastic stiffness.

The above equations can be expressed by Laplace transform as:

\[
\begin{align*}
Q_L &= K_f X_v - K_c P_L \\
Q_L &= A_p s X_p + C_{tp} P_L + \frac{V_i}{\beta_v} s P_L \\
A_p P_L &= m_i s^2 X_p + B_p s X_p + K X_p \\
\end{align*}
\]  

(4)

The control system deviation voltage signal equation and feedback link pressure sensor equation are established, respectively.

\[
U_e = U_r - U_f
\]

(5)

\[
U_f = K_{ff} F_g
\]

(6)

where \( U_r \) is the input voltage signal (V), \( U_f \) is the feedback voltage signal in the system link (V), \( K_{ff} \) is the sensor pressure gain amount (V/N), and \( F_g \) is the hydraulic servo cylinder output force (N).

If only the static performance of the amplifier is considered, its output current is

\[
\Delta I = K_a U_e
\]

(7)

where \( K_a \) denotes the servo amplifier gain amount (A/V). The EHS valve transfer function is expressed as

\[
\frac{X_v}{\Delta I} = K_{sv} G_{sv}(s)
\]

(8)

where \( X_v \) is the EHS valve displacement (mm), and \( K_{sv} \) is the EHS valve hydraulic gain amount (m\(^3\)/s·A); then, \( K_{sv} - G_{sv} = 1 \). Based on the above formulas, the hydraulic system block diagram is shown in Figure 4, where \( K_{sv} = K_c + C_{tp} \) holds.

![Figure 4. Hydraulic system block diagram with EHS control.](image)

Considering the complex dynamic performance model of an EHS system, there are fifth-order, fourth-order, and third-order functions. Moreover, the response speed of the EHS valve is fast; only the static performance is considered here, and it is directly set as the proportional link. Then, the mathematical model of the hydraulic system in this paper is simplified as

\[
G(s) = \frac{K_a K_{sv} K_f}{A_p K_{ff} \left( \frac{s^2}{\omega_n^2} + 1 \right)} \frac{1}{\left( \frac{s}{\omega_d} + 1 \right) \left( \frac{\omega_n}{\omega_d} + \frac{2 \zeta \omega_n}{\omega_d} s + 1 \right)}
\]

(9)
The parameter assignment, simplification of Equation (9), and the transfer function is derived, which can be regarded as

\[
G(s) = \frac{4.18 \times 10^{-3}s^2 + 3.36}{5.98 \times 10^{-3}s^3 + 5.86 \times 10^{-3}s^2 + 3.63s + 1} \tag{10}
\]

In this scheme, the classical PID was optimized through the Self-Turbulent Flow (STF) control algorithm, and the specific structure of the STF controller was designed. The Simulink module was used to simulate the control system. After comparative analysis, the changing laws of the response curves of the four control strategies were proposed.

3. Simulation Analysis of Optimal Control Strategy

The Self-Anti-Disturbance (SAD) control strategy for the LOSP of an EHS system is discussed in detail, which is used for quadratic optimization, pole placement, PID, and STF control, and the LOSP simulation model of STF control is constructed by the SIMULINK module.

3.1. Quadratic Optimal Control Analysis with Linearity

The principle of optimal control is to find a control variable \(u(t)\) that has a small value and can satisfy the minimum system error \(x(t)\), so that the output of the system quickly follows the input, and the energy consumption is low. The minimum value of the index can be obtained from the Pontryagin principle, and the essence of the quadratic optimal control is the approximation of the feedback \(K(t)\) of the original system. In this study, using the feedback of the optimal regulator to approximate the optimization, the simplified optimal control rate can be expressed as

\[
u(t) = -K(t)x(t) = R^{-1}(t)B^T(t)P(t)x(t) \tag{11}
\]

The algebraic equation for the Riccati matrix can be given as

\[
PA + A^TP - PBR^{-1}B^TP + Q = 0 \tag{12}
\]

where \(A\) denotes the EHS system matrix, \(B\) is the control matrix of the EHS system, and \(Q\) and \(R\) are the weighting matrices of the EHS system; herein, \(Q = \text{diag}[275, 000, 1, 1]\), \(R = 0.0001\), and \(P\) is the numerical solution of the equation.

Therefore, the optimized system matrix can be described as

\[
A - BK = \begin{bmatrix}
0 & 1 & 0 \\
0 & 0 & 1 \\
-55,138 & -59,727 & -141
\end{bmatrix} \tag{13}
\]

Using Matlab to simulate the transfer function under quadratic optimal control, the step response curve of the EHS system after quadratic optimal control with linearity is revealed, as shown in Figure 5. It can be seen from Figure 5 that the EHS control system reached stability in 4.8 s. After the control system underwent secondary optimization control, the stabilization time was shortened by 15.3 s, and the control performance was greatly improved; however, unfortunately the time was relatively long.
3.2. Pole Optimized Configuration

We introduce the state feedback gain matrix $K = [K_1 \quad K_2 \quad K_3]$; then, the characteristic polynomial can be expressed as

$$a(\lambda) = \text{det}(\lambda I - A + bK) = \begin{bmatrix} \lambda & 0 & 0 \\ 0 & \lambda & 0 \\ 0 & 0 & \lambda \end{bmatrix} - \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ -17,036 & -59,625 & -98 \end{bmatrix} + \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ k_1 & k_2 & k_3 \end{bmatrix}$$

$$= \lambda^3 + (k_3 + 98)\lambda^2 + (k_2 + 59,625)\lambda + (k_1 + 17,036)$$

The core content of analyzing the performance of the control system is to determine the dominant pole, where the effect of the far pole is ignored. Therefore, the EHS control system is considered to be equivalent to a second-order control system containing a main pair of poles. Determining the location of the expected dominant pole from the representation of the dynamic indicators $\sigma_p\%$ and $t_s$ is as follows

$$\sigma_p\% = e^{-\pi \xi/\sqrt{1-\xi^2}} \times 100\%$$

where $\sigma_p\%$ represents the maximum overshoot, and $\xi$ is the damping ($0 < \xi < 1$). Assuming that the allowable error of the EHS control system is set to 5%, there is

$$t_s = \frac{3}{\xi \omega_n}$$

where $t_s$ is the adjustment time (s), and $\omega_n$ denotes the undamped frequency (rad/s). The expected dominant pole of an EHS system can be expressed as

$$\lambda_{1,2} = -\xi \omega_n \pm j\omega_n \sqrt{1 - \xi^2}$$

where the system expects $\lambda_{1,2}$ to dominate the poles. Considering that the maximum overshoot of the EHS control system is limited to $\sigma_p\% \leq 5\%$, and the adjustment time is set to $t_s \leq 0.5s$, the expected dominant poles ($\lambda_1$ and $\lambda_2$) of the EHS system can be given by the following Equation (18). Analyzing this equation, the following settings are specified

$$\begin{cases} \sigma_p\% = e^{-\pi \xi/\sqrt{1-\xi^2}} \times 100\% \leq 5\% \\ t_s = \frac{3}{\xi \omega_n} \leq 0.5, \\ \text{where,} \xi \geq 0.69, \xi \omega_n \geq 6.0 \end{cases}$$

Herein, the clear value is $\xi = 0.707$, $\xi \omega_n = 6.0 = 6$; then, there is

$$\lambda_{1,2} = -\xi \omega_n \pm j\omega_n \sqrt{1 - \xi^2} = -6 \pm j6$$
Let the third pole $\lambda_3 = 10R_p[A_1] = -60$ be set; so, the following expression is

$$a^*(s) = (s + 60)(s + 6 + j6)(s + 6 + j6) = s^3 + 72s^2 + 792s + 4,320$$  \hspace{1cm} (20)

We establish $a(s) = a^*(s)$, and then obtain the following expression

$$K = \begin{bmatrix} k_1 & k_2 & k_3 \end{bmatrix} = \begin{bmatrix} -27 & -58,836 & -12,732 \end{bmatrix}$$

which shows the step response curve of the pole optimal configured EHS control system obtained by the Simulink module simulation. As can be seen from Figure 6, the EHS system can be stabilized within 1.3 s. After the EHS control system was optimized by the pole configuration, the stabilization time was shortened by 20.3 s, and the control performance was significantly improved; however, the overshoot phenomenon occurred prematurely.

![Step Response](image)

**Figure 6.** Step response curve of the EHS system after pole optimal configured control.

### 3.3. Simulation Analysis of Classical PID Control

The simple structure and convenient parameter calibration of the classical PID control make it widely used in many fields. In this section, we describe how the proportional coefficient $K_p$ was firstly determined; then, the integral constant and differential constant were set to zero. The proportional gain gradually increased from zero until the response curve oscillated; then, the value decreased slowly until the oscillation stopped, 30–70% of the proportional gain can be used as the system proportional coefficient, and 60% was taken in this study. The proportional coefficient was determined, and the larger value was taken as the initial value of the system integral constant. After several adjustments, the system stopped oscillating, and 150–180% of the integral constant was taken as the final value of the integral constant of the system, which was set as 165%. In the case of the EHS system output force, the differential constant $K_d$ is usually chosen to be 0. The parameters of the classical PID controller were selected as $K_p = 11.17$, $K_i = 63.11$, and $K_d = 0$. The step response curve of the classic PID control of an EHS system simulated by Simulink is shown in Figure 7.

![Step Response](image)

**Figure 7.** Step response curve of the classic PID control of the EHS system simulated by Simulink.
As shown in Figure 8, the EHS system had some improved performance through PID control. When a unit step signal was input, the EHS system tended to achieve a stable state within 1.55 s, and its input and output signals had good matching performance; however, the EHS system was also accompanied by a certain amount of overshoot.

![Designed block diagram of the proposed SAD third-order controller.](image)

Figure 8. Designed block diagram of the proposed SAD third-order controller.

3.4. Simulation Analysis of SAD Control

To further improve the control effect of the control system, on the basis of the SAD control algorithm, the classical PID was comprehensively optimized, the optimal scheme of the overall performance of the system was proposed, and the SAD controller was analyzed and designed in combination with the specific structure. The designed block diagram of the proposed SAD third-order controller in this paper is shown in Figure 8.

The SAD controller consists of three modules, the control parameter Following Differentiator (TD), the error feedback Extended State Controller (ESC), and the EHS Control Law (CL). Based on the system state space expression, the controlled object was set to be third-order, where the TD was third-order, and the ESC was fourth-order. Seeking to improve the dynamics of this control system, we used state feedback to configure the desired poles. If the zero-point configuration of the control system can be realized, the dynamic performance of the control system can be improved. However, the zero-point configuration needs to use the differential signal. The previous differential signal cannot filter out noise, nor is it used for zero-point configuration, while the following differential signal can filter out the noise well, and the differential signal is zero-point configuration. The TD follows a given signal and obtains its differential equation. The transition process is arranged by the TD to prevent the overshoot caused by the sudden change in the given signal, so that the controlled objects approach the target smoothly, thereby improving the stability of the control system. By eliminating the poles of the system with the configured zeros, the system can be viewed as a control system close to unity. This sets the string level object to the following expression.

\[
y = \frac{1}{p_1(s)} \frac{1}{p_2(s)} u
\]  
\[
(21)
\]

Assuming that \(p_2(s)\) is known, the virtual control volume is

\[
U(t) = \frac{1}{p_2(s)} u
\]  
\[
(22)
\]

The above expression (21) can be written as

\[
y = \frac{1}{p_1(s)} U(t)
\]  
\[
(23)
\]

Once the virtual control quantity \(U(t)\) is clarified, the actual control quantity acts as

\[
u = p_2(s)U(t)
\]  
\[
(24)
\]

where \(p_1(s)\) and \(p_2(s)\) are polynomials.
The virtual control volume $U(t)$ is determined by applying a improved controller design of Self-Turbulent Flow (STF) algorithm approach to the control subsystem $y = U(t)/P_1(s)$, which allows for a certain range of uncertainties to exist. Assuming that the mathematical model of the controlled object contains zeros and poles, its general form is described as

$$y = \frac{q(s)}{p(s)}u$$  \hspace{1cm} (25)$$

We set the virtual control quantity $U(t) = q(s)u$; then, the system becomes $y = U(t)/P(s)$, and the actual control quantity becomes $u = U(t)/q(s)$. If the controlled object conforms to the minimum phase system, the $q(s)$ belongs to a stable polynomial and is known. The SAD controller can be used to determine the virtual control quantity $U(t)$, and the actual control quantity $u(t)$ can be obtained by solving the system $u = U(t)/q(s)$, with $U(t)$ as the input. Through the above analysis, the zero-pole diagram of the controlled object was derived using Matlab, as shown in Figure 9.

![Pole-Zero Map](image)

**Figure 9.** Zero-pole diagram of the controlled object exported by Matlab.

As can be seen in Figure 9, the system model had no zeros and poles in the positive part of the complex plane and was considered as a minimum phase system. In this case, the virtual control variable $U(t)$ was determined by the active disturbance rejection controller, and $U(t)$ was used as the input to solve the system $u = U(t)/q(s)$ to obtain the actual control variable. The transfer function of the control system is known to satisfy the cubic polynomial $p_3(s) = s_3 + a_2s^2 + a_2s + a_1$. The parameters $a_1$, $a_2$, and $a_3$ are known, and the STF algorithm of the third-order TD differentiator is written as

$$\begin{align*}
fs &= -r(r(v_1 - v(t)) + 3v_2) \\
v_1 &= v_1 + hv_2 \\
v_2 &= v_2 + hv_3 \\
v_3 &= v_3 + hfs \\
U(t) &= v_1(t)
\end{align*}$$  \hspace{1cm} (26)$$

where $r$ is a fast factor, and $h$ denotes the sampling step size.

The error feedback Extended State Controller (ESC) follows the output of the control system, monitors the state of each stage of the research object and the total perturbation of the EHS system in real time, and then compensates the perturbation accordingly. If the system suffers from unknown disturbances, the control object is uncertain. This indeterminate object is expressed as

$$\begin{align*}
x^{(n)} &= f(x, \dot{x}, \cdots, x^{(n-1)}, t) + w(t) + bu \\
y &= x(t)
\end{align*}$$  \hspace{1cm} (27)$$

where $f(x, \dot{x}, \cdots, x^{(n-1)}, t)$ is the system’s unknown function, $w(t)$ is the unknown perturbation, $b$ is a control gain, $u$ is the system input, and $y$ is the system output.
The error feedback ESC in this paper is fourth-order. Under linear conditions, the fourth-order ESC algorithm of the control system is written as

\[
\begin{align*}
\{ & z_1 = z_1 + h(z_2 - \beta_{01}e), \quad z_2 = z_2 + h(z_3 - \beta_{02}e), \quad z_3 = z_3 + h(z_4 - \beta_{03}e + bu), \quad z_4 = z_4 + h(-\beta_{04}e) \\
\text{where, } & e = z_1 - y
\end{align*}
\] (28)

where \(\beta_{01}, \beta_{02}, \beta_{03}, \beta_{04}\) are the ESC parameters, and \(e\) is the feedback error term.

The transfer relationship from the input to output of the error feedback ESC is expressed as

\[
\begin{align*}
\{ & z_1 = w_1(s)y = \frac{\beta_{03}s^3 + \beta_{02}s^2 + \beta_{01}s + \beta_{04}}{s^4 + \beta_{01}s^3 + \beta_{02}s^2 + \beta_{03}s + \beta_{04}}y, \quad z_2 = w_2(s)y = \frac{\beta_{03}s^3 + \beta_{02}s^2 + \beta_{01}s + \beta_{04}}{s^4 + \beta_{01}s^3 + \beta_{02}s^2 + \beta_{03}s + \beta_{04}}y \\
& z_3 = w_3(s)y = \frac{\beta_{03}s^3 + \beta_{02}s^2 + \beta_{01}s + \beta_{04}}{s^4 + \beta_{01}s^3 + \beta_{02}s^2 + \beta_{03}s + \beta_{04}}y, \quad z_4 = w_4(s)y = \frac{\beta_{03}s^3 + \beta_{02}s^2 + \beta_{01}s + \beta_{04}}{s^4 + \beta_{01}s^3 + \beta_{02}s^2 + \beta_{03}s + \beta_{04}}y
\end{align*}
\] (29)

In the control system, the ESC contains four output variables: \(z_1\) follows the system output \(y\), \(z_2\) follows \(y'\), \(z_3\) follows \(y''\), \(z_4\) follows the system integral disturbance, and the disturbance is compensated by the feedforward method. In order to achieve a predetermined estimation accuracy, a larger gain coefficient was selected, that is, a larger value of the gain coefficients \(\beta_{01}, \beta_{02}, \beta_{03}, \beta_{04}\) was required to satisfy the high-gain ESC mode. Based on the four optimization methods and control strategy system models of the classical PID control, quadratic optimization control, pole configuration, and SAD, the EHS system was simulated by Simulink, and the step response curves of the unit step signal under the four control strategies were compared and analyzed, as shown in Figure 10.

![Figure 10. Step response curve of unit step signal under four control strategies.](image)

In Figure 10, the SAD performance was the most stable after the system was optimized. The response speed of PID was 0.6 s slower than that of SAD control, and the response was accompanied by a large amount of overshoot. The response time of the pole configuration was 0.3 s longer than that of the SAD control. Compared with the PID control, although the overshoot was significantly reduced, the overshoot still existed. Both the quadratic optimal control and SAD control did not have overshoot, but the response time of the SAD control was 4.8 s shorter than that of the quadratic optimal control.

4. Experimental Analysis

To verify the effectiveness of the proposed four control strategies, a hydraulic platform was established, which is shown in Figure 11. To test the proposed controllers, the experimental error performance was shown by quantitative acquisition and monitoring, as described in this section. The following-up errors of the four control strategies within 1.0 s are shown in Figure 12. The following-up errors of the four control strategies within 4.0 s are shown in Figure 13, where, Case1 represents the pole optimized configuration, Case2 represents the quadratic optimal control, Case3 represents the classical PID control, and Case4 represents the SAD control.
The performance of the conventional PID controller (Case 1) was the least ideal. Following-up errors within 4.0 s. As shown in Figures 12 and 13, the following-up performance of the Case 4 controller was better than that of the others. It is also noted that the following-up performance of the Case 2 controller was relatively weak, which indicates that the parameter uncertainty had a strong influence on the high frequency band of the controller. Similarly, the following-up performance of the conventional PID controller (Case 1) was the least ideal.

5. Discussion of the Results

From the results above, it is possible to conclude that the following-up of the controller of Case 3 was greatly increased compared with that of Case 1 and Case 2. An improved
STF algorithm was introduced into the EHS system with the LOSP optimized for the SAD control strategy, which is a novel move. Another interesting consideration is the comparison of the gains of the Cases; it is possible to state that the following-up performance of the Case 4 controller was better than that of Case 3. On the other hand, it was verified that the variable hydraulic LOSP acquisition and monitoring proposed in this paper was effectively and stably adjusted by a predesigned EHS control system.

6. Conclusions

The motivation for this study was the serialized Variable Static Loads (VSL) following-up performance challenge presented by hydraulic systems that required developing a Self-Anti-Disturbance (SAD) control optimal strategy for the Lubricating Oil Static Pressure (LOSP) of an Electrohydraulic Servo (EHS) system that was efficient at the performance level and asymptotically stable at the theoretical level. For this purpose, an improved Self-Turbulent Flow (STF) algorithm was introduced, which could simultaneously deal with the parameter uncertainty and time-varying disturbances. A novel strategy for control adaptation in VSL, called SAD, was defined, and its stability analysis was provided, showing that it preserved all the properties of the finite time convergence of the state estimation error of its following-up performance counterpart. Based on the proposed algorithm, the VSL asymptotic stability of the hydraulic system was achieved. Considering that the dynamics of the valves, such as the Adjustable Flow Throttle Valves (AFTV) and Pilot Operated Relief Valves (PORV), are often overlooked in a hydraulic system lubrication design, one of the future studies will investigate EHS valve-dynamics modeling techniques to improve the optimal dynamic following-up performance. In addition, considering the practical application of volume/cost/weight, the full state feedback was not perfectly realized in this study, which will prompt further in-depth discussions on the output feedback control of the hydraulic system.

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