Dual-valve parallel prediction control for an electro-hydraulic servo system

Shi-jie Su1, Yuan-yuan Zhu1, Cun-jun Li2, Wen-xian Tang1, and Hai-rong Wang2
1School of Mechanical Engineering, Jiangsu University of Science and Technology, Zhenjiang, China
2Zhoushan Institute of Calibration and Testing for Quality and Technology Supervision, Zhoushan, China

Abstract

To improve the dynamic response performance of a high-flow electro-hydraulic servo system, scholars have conducted considerable research on the synchronous and time-sharing controls of multiple valves. However, most scholars have used offline optimization to improve control performance. Thus, control performance cannot be dynamically adjusted or optimized. To repeatedly optimize the performance of multiple valves online, this study proposes a method for connecting a high-flow proportional valve in parallel with a low-flow servo valve. Moreover, this study proposes an algorithm in which a proportional–integral–derivative system and multivariable predictive control system are used as an inner loop and outer loop, respectively. The simulation and experimental results revealed that dual-valve parallel control could effectively improve the control accuracy and dynamic response performance of an electro-hydraulic servo system and that the proportional-integral-derivative–multivariable predictive control controller could further dynamically improve the control accuracy.

Keywords

Dual-valve control, proportional-integral-derivative–multivariable predictive control, electro-hydraulic servo system, predictive control

Corresponding author:
Shi-jie Su, School of Mechanical Engineering, Jiangsu University of Science and Technology, No.2 Mengxi Road, Zhenjiang 212003, China.
Email: sushijie@just.edu.cn

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Introduction

Because of the development of electro-hydraulic servo technology,1–3 multiple hydraulic valves have been widely used to drive an actuator in parallel, and scholars have achieved favorable results by using this method in some engineering applications.

Currently, the multivalve parallel control of an electro-hydraulic servo system is implemented using two approaches. In the first approach, multiple heterogeneous parallel valves are operated using a time-sharing approach.4–6 For example, Bai and Quan5 proposed a new hydraulic control circuit with two valves arranged in parallel for jointly controlling the actuator. Ren and Ruan6 used an improved two-dimensional valve as the primary control component, which was connected in parallel with a servo valve to obtain high-frequency vibration. In the second approach, multiple isomorphic parallel valves are operated synchronously. Hodgson et al.7 connected four solenoid valves in parallel for time-sharing control to improve the stability and control accuracy of a closed-loop system.

Although the aforementioned control schemes provide solutions specific to certain applications, they cannot dynamically adjust the performance of multiple valves. Therefore, in this study, a control method that achieved the online and repeated optimization of the control performance was designed. Because predictive control can provide real-time optimal control under the constraint conditions through a predictive model, rolling optimization, and feedback correction, a dual-valve proportional-integral-derivative (PID)–multivariable predictive control (MPC) algorithm based on predictive control is proposed.8–12 The PID–MPC controller uses the PID and MPC systems as the inner and outer loops, respectively.

This article is organized as follows: In section “Mathematical model of the dual-valve parallel electro-hydraulic servo system,” the mathematical model of the electro-hydraulic servo system using proportional and servo valves is provided. In section “PID–MPC composite algorithm for dual-valve parallel control,” the PID–MPC composite control algorithm is outlined. Section “Simulation” discusses the PID–MPC control simulation model established using Siemens AMESim and MathWorks MATLAB/Simulink. Section “Experimental results” presents the experimental verification of the slope tracking accuracy of PID–MPC control. Finally, the conclusions are provided in section “Conclusion.”

Mathematical model of the dual-valve parallel electro-hydraulic servo system

As displayed in Figure 1, the dual-valve parallel electro-hydraulic servo system primarily comprises a servo valve, proportional valve, hydraulic cylinder, displacement sensor, and controller. The controller sends a control command to the servo and proportional valves, which generates movement in the piston rod of the hydraulic cylinder. The displacement sensor continuously detects the position of the piston rod and provides feedback to the controller.

First, single-valve control was considered. According to the operational characteristics of a hydraulic spool valve, the linear flow equation is obtained as follows

\[ q_L = K_q x_v - K_C p_L \]  

(1)
where $q_L$, $K_q$, $K_C$, $x_v$, and $p_L$ are the spool flow, spool flow gain, flow-pressure coefficient, spool displacement, and load pressure, respectively.

On the basis of the flow continuity equation, the following equation can be obtained

$$q_L = A \frac{dx_p}{dt} + c_{tc} p_L + \frac{V_t}{4\beta_e} \frac{dp_L}{dt}$$

(2)

where $A$, $x_p$, $c_{tc}$, $V_t$, and $\beta_e$ are the effective cross-sectional area of the rodless cavity, displacement of the piston rod, equivalent leakage coefficient of the hydraulic cylinder, sum of the hydraulic cylinder and tubing volumes, and effective volume elastic modulus of the hydraulic cylinder, respectively. The balance equation between the hydraulic cylinder and load pressure is given as follows

$$p_L = \frac{1}{A} \left( m_t \frac{d^2 x_p}{dt^2} + B_e \frac{dx_p}{dt} + K_p x_p + F \right)$$

(3)

where $m_t$ is the total mass of the moving components of the system, $B_e$ is the viscous damping coefficient of the piston and load, $K_p$ is the elastic stiffness of the load, and $F$ is an external interference force. By performing the Laplace transform on the aforementioned equations, the following equation is obtained

**Figure 1.** Schematic of the dual-valve parallel electro-hydraulic servo system.

1. Piston pump; 2. three-phase motor; 3. overflow valve; 4. pressure gauge; 5. hydraulic cylinder; 6. servo valve; 7. proportional valve; 8. filter.
\[
\begin{aligned}
q_L &= K_q x_v - K_C p_L \\
p_L &= \frac{1}{A} \left( m_t s^2 x_p + B_c s x_p + K_p x_p + F \right) \\
q_L &= c_c p_L + A s x_p + \frac{V_t}{4 \beta_e} s p_L \\
\end{aligned}
\]  

(4)

On the basis of the aforementioned equations, the following equation can be obtained

\[
x_p = \frac{K_d}{A} x_v - \frac{K_{ce}}{A^2} \left( \frac{V_t}{4 \beta_c K_{ce}} + 1 \right) F \\
\]

(5)

where \( K_{ce} = c_e + K_C \) and \( K_{ce} \) are the total flow-pressure coefficient including the leak. An amplifier is then included, and the external disturbance force \( F \) is neglected. The transfer function \( G_1(s) \) between the displacement of the piston rod \( x_p \) and the control signal \( u \) is as follows

\[
G_1(s) = \frac{K_d K_a K_{sv}}{A} \\
\]

(6)

where \( K_a \) and \( K_{sv} \) are amplifier gain and spool gain, respectively.

By neglecting the nonlinear coupling between the proportional and servo valves, the flow equation representing the parallel connection of the two valves obtained using a linear superposition method is as follows

\[
Q_L = K_{q1} K_{a1} K_{sv1} u_1 + K_{q2} K_{a2} K_{sv2} u_2 - \left( K_{C1} + K_{C2} \right) p_L \\
\]

(7)

where \( K_{q1} \) and \( K_{q2} \) are the flow gains of the proportional and servo valves, respectively; \( K_{C1} \) and \( K_{C2} \) are the flow-pressure coefficients of the proportional and servo valves, respectively; \( K_{a1} \) and \( K_{a2} \) are the gains of the proportional and servo amplifiers, respectively; \( K_{sv1} \) and \( K_{sv2} \) are the gains of the proportional and servo valves, respectively; \( u_1 \) and \( u_2 \) are the control signals of the proportional and servo valves, respectively. \( Q_L \) is the total flow when the two valves are connected in parallel. The state space model is established according to equations (2), (3), and (7) as follows

\[
\begin{bmatrix}
\dot{x}_1 \\
\dot{x}_2 \\
\dot{x}_3 \\
\end{bmatrix} = 
\begin{bmatrix}
0 & 1 & 0 \\
0 & 0 & 1 \\
-A_3 & -A_2 & -A_1 \\
\end{bmatrix}
\begin{bmatrix}
x_1 \\
x_2 \\
x_3 \\
\end{bmatrix} 
+ 
\begin{bmatrix}
0 & 0 \\
0 & 0 \\
B_1 & B_2 \\
\end{bmatrix}
\begin{bmatrix}
u_1(t) \\
u_2(t) \\
\end{bmatrix}
\]

(8)
\[
\begin{bmatrix}
  x_1 \\
  x_2 \\
  x_3
\end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 \end{bmatrix} \begin{bmatrix}
  x_1 \\
  x_2 \\
  x_3
\end{bmatrix}
\]  

(9)

where \(x_1\), \(x_2\), and \(x_3\) are the displacement, speed, and acceleration of the piston rod, respectively. Moreover, \(B_1 = 4\beta_e AK_q1K_{at}K_{sv1}/V_i m_1\), \(B_2 = 4\beta_e AK_q2K_{at}K_{sv2}/V_i m_1\), \(A_1 = (4\beta_e/V_i m_i)(K_{CE} m_i + (V_i B_e / 4\beta_e))\), \(A_2 = (4\beta_e A^2 / V_i m_i)((K_p V_i / 4\beta_e A^2) + (K_{CE} B_e / A^2) + 1)\), \(A_3 = 4\beta_e K_{CE} K_p / V_i m_1\), and \(K_{CE} = C_e + K_{C1} + K_{C2}\).

**PID–MPC composite algorithm for dual-valve parallel control**

According to the mathematical model presented in section “Mathematical model of the dual-valve parallel electro-hydraulic servo system,” the dual-valve parallel electro-hydraulic servo system is a two-input single-output system. Figure 2 illustrates the PID–MPC composite control algorithm. As displayed in the figure, the PID–MPC composite control algorithm comprises the PID control system as the inner loop and MPC system as the outer loop, respectively. \(G_1(S)\) and \(G_2(S)\) are the transfer functions of the proportional and servo control loops, respectively. In Figure 2, the inner-loop control system is represented by the solid line on the right side, where \(u_1\) and \(u_2\) are the inputs of the proportional and servo control loops, respectively.

A dynamic matrix control (DMC) algorithm was used to design the system controller.\(^{13,14}\) First, the prediction model was developed. A step response was sampled for each closed-loop control system in the inner-loop system, and the sample values of input \(a_{11}(t)\) corresponding to \(u_1\) and input \(a_{12}(t)\) corresponding to \(u_2\) were obtained. Subsequently, the modeling time domain \(N\) was determined, and \(a_{11} = [a_{11}(1), \ldots, a_{11}(N)]^T\) and \(a_{12} = [a_{12}(1), \ldots, a_{12}(N)]^T\) were obtained according to the sampled values.

At time \(k\), when the inputs \(u_1\) and \(u_2\) had \(M\) consecutive control increments \(\Delta u_1(k),\ldots,\Delta u_1(k+M-1)\) and \(\Delta u_2(k),\ldots,\Delta u_2(k+M-1)\), respectively, the predicted value \(\hat{y}\) of the output at future \(P\) moments could be obtained. Here, \(P\) and \(M\) are the time domain of control and optimization, respectively.
\[
\hat{y}_{PM}(k) = \hat{y}_{P0}(k) + \sum_{j=1}^{2} A_{ij} \Delta u_{j,M}(k)
\]

where \( \hat{y}_{P0}(k) = \) and \( \hat{y}_{P0}(k) \) is the initial predicted value of the output at future \( P \) moments at time \( k \)

\[
A_{i1} = \begin{bmatrix}
                \quad a_{i1}(1) & \cdots & 0 \\
                \vdots & \ddots & \vdots \\
                a_{i1}(M) & \cdots & a_{i1}(1) \\
                \vdots & \cdots & \vdots \\
                a_{i1}(P) & \cdots & a_{i1}(P-M+1)
        \end{bmatrix}
\]

and

\[
A_{i2} = \begin{bmatrix}
                \quad a_{i2}(1) & \cdots & 0 \\
                \vdots & \ddots & \vdots \\
                a_{i2}(M) & \cdots & a_{i2}(1) \\
                \vdots & \cdots & \vdots \\
                a_{i2}(P) & \cdots & a_{i2}(P-M+1)
        \end{bmatrix}
\]

where \( A_{i1} \) and \( A_{i2} \) are the dynamic matrices comprising \( a_{i1} \) and \( a_{i2} \), respectively.

\[
\Delta u_{1,M}(k) = \begin{bmatrix}
                \quad \Delta u_{1}(k) \\
                \vdots \\
                \Delta u_{1}(k+M-1)
        \end{bmatrix}
\]

, and \( \Delta u_{1,M}(k) \) is the input increment of the proportional control loop.

\[
\Delta u_{2,M}(k) = \begin{bmatrix}
                \quad \Delta u_{2}(k) \\
                \vdots \\
                \Delta u_{2}(k+M-1)
        \end{bmatrix}
\]

, and \( \Delta u_{2,M}(k) \) is the input increment of the servo control loop.

\[
\hat{y}_{PM}(k) = \begin{bmatrix}
                \quad \hat{y}_{M}(k+1|k) \\
                \vdots \\
                \hat{y}_{M}(k+P|k)
        \end{bmatrix}
\]

, and \( \hat{y}_{PM}(k) \) is the predicted value of the output.

The rolling optimization strategy of the system was determined. In rolling optimization, the changes in \( u_1 \) and \( u_2 \) at future \( M \) moments were adjusted to ensure that the
output could track the expected value $w$ accurately at future $P$ moments. To prevent abrupt changes in $u_1$ and $u_2$, a weight matrix was added to suppress their performance indices. The performance index of the optimization of the DMC algorithm at time $k$ can be represented as follows

$$
\min J(k) = \sum_{i=1}^{P} q(i) \left[ w(k+i) - \tilde{y}_M(k+i) \right]^2 + \sum_{j=1}^{M} r_1(j) \Delta u_1^2(k+j-1) + r_2(j) \Delta u_2^2(k+j-1)
$$

(11)

The index can be given in a vector form as follows

$$
\min J(k) = \left\| w(k) - \tilde{y}_{PM}(k) \right\|_Q^2 + \left\| \Delta u_M(k) \right\|_R^2
$$

(12)

where $w(k) = \begin{bmatrix} w(k) \\ \vdots \\ w(k+P) \end{bmatrix}$ is the expected output value at $P$ moments, $Q = \text{diag}[q(1), \ldots, q(P)]$ is the error matrix corresponding to $y$, and the elements in $Q$ correspond to the tracking error of $y$ at different times. $R = \text{diag}(R_1, R_2)$ is the control matrix, whereas $R_1 = \text{diag}[r_1(1), \ldots, r_1(M)]$ and $R_2 = \text{diag}[r_2(1), \ldots, r_2(M)]$ correspond to inputs $u_1$ and $u_2$, respectively. Moreover, the elements of $R_1$ and $R_2$ correspond to the suppression of the increments in $u_1$ and $u_2$, respectively, at different times.

A large weight of the element $q$ in $Q$ is positively associated with the values of $y$ and $w$ being close. A large value of $P$ corresponds to high stability and low rapidity of the system. Similarly, the large weight of element $r_1$ in $R_1$ corresponds to the high increment suppression of the input of the proportional control loop. Moreover, the larger the weight of element $r_2$ in $R_2$, the higher the increment suppression of the input of the servo control loop. Furthermore, a large value of $M$ is associated with improved rapidity and reduced stability and robustness of the system.

The high-flow proportional valve cannot track a target curve, which abruptly varies because of the low response frequency and dead zone of the valve. Therefore, the value of $r_1$ must be increased to suppress the variation in the input speed of the proportional valve. The servo valve can track a target curve, which varies abruptly because of its high frequency response. Therefore, the value of $r_2$ should be reduced to improve system speed.

The constraints of $u$ and $y$ were added to the DMC algorithm as follows

$$
u_{\min} \leq u \leq u_{\max}
$$

(13)

$$
y_{\min} \leq y \leq y_{\max}
$$

(14)
Equations (13) and (14) were incorporated into the rolling optimization equation as follows

\[
\begin{align*}
    u_{j,\min} & \leq u_j(k) = u_j(k-1) + \Delta u_j(k) \leq u_{j,\max} & j = 1, 2 \\
    u_{j,\min} & \leq u_j(k + M - 1) = u_j(k-1) + \Delta u_j(k) + \cdots + \Delta u_j(k + M - 1) \leq u_{j,\max}
\end{align*}
\]

(15)

\[
y_{\min} - \tilde{y}_{P0}(k) \leq A\Delta u_M(k) \leq y_{\max} - \tilde{y}_{P0}(k)
\]

(16)

where \( y_{\min} = [y_{\min}, \ldots, y_{\min}]^T \) and \( y_{\max} = [y_{\max}, \ldots, y_{\max}]^T \) are the minimum and maximum values of the output in the range of \( P \), respectively.

For optimization problems involving quadratic properties with inequality constraints, quadratic programming (QP)\textsuperscript{15} can be used to obtain the optimal solution.

Finally, after developing the prediction model and rolling optimization, feedback correction was added to the algorithm. Thus, the system could be adjusted according to the actual output. The error vector calculated at time \( k + 1 \) is as follows

\[
e(k+1) = y(k+1) - \tilde{y}_1(k+1 | k)
\]

(17)

According to equation (17), the predicted output of the corrected system is as follows

\[
\tilde{y}_{\text{cor}}(k+1) = \tilde{y}_{N1}(k) + He(k+1)
\]

(18)

where \( \tilde{y}_{N1}(k) \) and \( \tilde{y}_{\text{cor}}(k+1) \) are the predicted and corrected values of the output, respectively, and \( H = \begin{bmatrix} \vdots \end{bmatrix} \) is the error correction matrix.

The predicted value \( \tilde{y}_{\text{cor}}(k+1) \) is moved forward to reach time \( k + 1 \) as follows

\[
\tilde{y}_{N0}(k+1) = S\tilde{y}_{\text{cor}}(k+1)
\]

(19)

where \( S_0 = \begin{bmatrix} S & \cdots & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & S \end{bmatrix} \) is the diagonal matrix formed using the matrix block \( S = \begin{bmatrix} 0 & 1 & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & 0 & 1 \end{bmatrix} \) and \( \tilde{y}_{N0}(k+1) \) is the predicted output corrected at time \( k + 1 \).
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Simulation

**Simulink modeling for dual-valve parallel PID–MPC composite control**

The Simulink model of the PID–MPC composite algorithm for dual-valve parallel control was developed (Figure 3(a)) according to the mathematical model and PID–MPC composite algorithm presented in sections “Mathematical model of the dual-valve parallel electro-hydraulic servo system” and “PID–MPC composite algorithm for dual-valve parallel control,” respectively.
Figure 3(b) confirms the effectiveness of the PID–MPC composite algorithm; here, \( w \) is the target curve, \( pv \) is the response curve of the proportional control loop, \( sv \) is the response curve of the servo control loop, and \( mpc \) is the response curve of the dual-valve parallel control. As can be seen from Figure 3(b), the response time of \( mpc \) is significantly less than that of \( pv \) and \( sv \), and the tracking accuracy of \( mpc \) is higher than that of \( pv \) and \( sv \).

In order to avoid too complex mathematical modeling, the nonlinear factors such as dead zone and maximum flow limit of the valves are neglected, resulting in the fact that Figure 3(b) cannot reflect the actual characteristics of proportional and servo valves, for example, a significant deviation between \( sv \) and \( w \), and a dead zone at the peak of \( pv \) should be observed. However, the actual physical characteristics of the valves can be shown in AMESim. Therefore, to better reflect the dynamic response performance of the electro-hydraulic servo system, AMESim–Simulink co-simulation models were developed in the following study.

**AMESim–Simulink co-simulation modeling for single-valve control**

First, the proportional valve was used for establishing the single-valve simulation model.

Figure 4 depicts the AMESim–Simulink co-simulation model of the proportional valve (similar to that of the servo valve). The primary parameters were as follows: The flow rate of the plunger pump was 15.8 L/rev, relief valve opening pressure was 140 bar, signal of the proportional valve was ±40 mA, response time was less than or equal to 60 ms, dead zone was 20% of the valve stroke, and maximum flow rate was 30 L/min at a P–T pressure drop of 70 bar. The input signal of the servo valve was ±40 mA, and the response time was less than or equal to 10 ms for a natural frequency of 62 Hz and maximum flow rate of 8 L/min at a P–T pressure drop of 70 bar. The diameters of the cylinder, piston rod, and stroke were 100, 50, and 200 mm, respectively. The initial position was at 100 mm, and the weight was 100 kg. The parameters of the PID controller of the proportional control loop were as follows: \( K_p = 25.5 \), \( K_i = 5 \), and \( K_d = 0 \). The parameters of the servo control loop were as follows: \( K_p = 100 \), \( K_i = 2 \), and \( K_d = 0 \).

Figure 5 displays the step response curve of the proportional and servo control loops. The amplitudes of the proportional and servo control loops were 1 and 0.2 mm, respectively. In Figure 5, \( w_1 \) is the target curve of the proportional control loop, \( y_1 \) is the response curve of the proportional control loop, \( w_2 \) is the target curve of the servo control loop, and \( y_2 \) is the response curve of the servo control loop. The step response curve and target curve exhibited a constant deviation because of the dead zone of the proportional valve.

The sampling period was set to 0.01 s. The value of the modeling time domain \( N \) was 20, with two inputs and one output. The following equations were obtained after sampling

\[
a_{11} = [0.0000; 0.1046; 0.4077; 0.7361; 0.9508; 1.0025; 1.0027; 1.0025; 1.0025; 1.0027; 1.0025; 1.0025; 1.0027; 1.0025; 1.0027; 1.0026; 1.0025; 1.0026; 1.0027; 1.0026; 1.0027; 1.0026; 1.0027]
\]

\[
a_{12} = [0.0649; 0.1430; 0.1757; 0.1895; 0.1952; 0.1967; 0.1984; 0.1988; 0.1988; 0.1987; 0.1990; 0.1990; 0.1999; 0.1994; 0.1998; 0.1999; 0.1995; 0.2002; 0.1995; 0.2002]\
\]
where \( a_{11} \) and \( a_{12} \) are the model vectors of the proportional and servo control loops, respectively. The value of the optimized time domain \( P \) was 10, and the value of the control time domain \( M \) was 4. On the basis of the obtained model vectors \( a_{11} \) and \( a_{12} \), the dynamic matrix \( A = [a_{11} \ a_{12}] \) was constructed for the constrained QP solution and the coefficient matrix \( Aa = [a_{11} \ a_{12}] \) was used to calculate the predicted value of the output at any time. On the basis of the status of the proportional and servo valves, the constraints of the input signal were set as follows: \( u_{1,\text{min}} = -40, u_{1,\text{max}} = 40, u_{2,\text{min}} = -40 \), and \( u_{2,\text{max}} = 40 \). Moreover, the predicted output constraints were set as follows: \( y_{\text{min}} = -5 \) and \( y_{\text{max}} = 5 \). The error correction matrix was as follows: \( H = [1, 0.8, \ldots, 0.8] \).

Figure 4. AMESim–Simulink co-simulation model for proportional valve: (a) AMESim model and (b) Simulink model.
AMESim–Simulink co-simulation modeling for dual-valve parallel PID–MPC control

Figure 6 displays the simulation model for dual-valve parallel PID–MPC composite control (similar to the model of PID control). The parameters of the simulation were the same as those for single-valve control.

Figure 7 illustrates the slope-tracking inputs of the PID and PID–MPC controls. The parameters in the PID control were adjusted as follows: $K_p = 100$, $K_i = 2$, and $K_d = 0.02$. As displayed in Figure 6(a), the control signals of the servo and proportional control loops in PID control were the same. Therefore, the input could not be optimized according to the valve characteristics. By contrast, Figure 6(b) indicates that the control signals of the two control loops were significantly different. The control signal of the proportional control loop changed gradually, whereas the variation in the signal of the servo control loop was relatively abrupt. Therefore, the proportional and servo control loops were optimized to obtain a high-flow output and maintain a high-frequency response, respectively.

Figure 8 illustrates the slope-tracking output for proportional PID control, dual-valve PID control, and dual-valve PID–MPC control. In the figure, $w$, $mpc$, $pid$, and $pv$ are the target curve, tracking curve for PID–MPC control, tracking curve for PID control, and tracking curve for single proportional valve control, respectively. The highest variation was observed between the outputs for single proportional valve control and the target curve. PID control marginally reduced the tracking deviation; however, it did not effectively track the target curve. PID–MPC control effectively compensated for the tracking deviation by using rolling optimization, and the PID–MPC control curve almost coincided with the tracking curve from 0.1 s.

Figure 9 presents the sinusoidal tracking input of dual-valve PID control and dual-valve PID–MPC control. The frequency and amplitude were 10 Hz and 1 mm, respectively. The
parameters in PID control were adjusted as follows: $K_p = 60$, $K_i = 2$, and $K_d = 0.02$. The parameters of the control and error matrices for PID–MPC control were adjusted as follows: $R = \text{diag}(18, 18, 18, 1, 1, 1, 1, 1, 1, 1, 1)$ and $Q = \text{diag}(10, 10, 10, 1, 1, 1, 1, 1, 1, 1, 1)$. As illustrated in Figure 9(a), the control signals for servo and proportional PID control were identical. Figure 9(b) indicates that the control signal of the servo control loop accurately tracked the target curve, whereas the control signal of the proportional control loop maintained a high-flow output.
Figure 10 illustrates the sinusoidal tracking curves for single proportional valve control, dual-valve PID control, and dual-valve PID–MPC control. The figure indicates that the sinusoidal tracking curve for single proportional valve control had a large phase deviation. The tracking deviation caused by the dead zone is particularly evident near the peak of the curve. Although dual-valve PID control reduced the amplitude deviation, a large phase deviation was observed. Dual-valve PID–MPC control exhibited significant improvement in the control accuracy from the second cycle, which indicated that PID–MPC control can reduce the phase deviation and improve the dynamic performance of the system.

Figure 7. Input of slope tracking: (a) PID control and (b) PID–MPC control.
Experimental results

After simulation, the proposed method was verified using an experimental platform. Figure 11 depicts the electro-hydraulic servo testing machine. The testing machine comprised a testing machine body, control cabinet, industrial personal computer, and hydraulic station. The hydraulic cylinder was driven by the proportional valve ATOS DLHZO-AE–071-L1 and servo valve ATOS DHZO-TE-040-L01. The response time of the proportional valve was less than or equal to 30 ms, and the dead zone was 20% of the valve stroke. The response time of the servo valve was less than or equal to 10 ms. The dynamic responses for the ±100% and ±5% rated strokes were 62 and 180 Hz, respectively.

Figure 12 illustrates the experimental curve of slope tracking for single proportional valve control, dual-valve PID control, and dual-valve PID–MPC control. In the figure, \( w \) is the target curve, whereas \( mpc, pid, \) and \( pv \) are curves for PID–MPC control, PID control, and single proportional valve control, respectively. Figure 13 displays the deviation curve of the slope tracking for the three control methods.

As displayed in Figure 12, the slope tracking curve for single proportional valve control and the target curve exhibited a constant deviation because of the dead zone. Figure 13 indicates that PID control considerably reduced the tracking error; however, its initial tracking error was still 1.2 mm. The deviation of PID–MPC control was very close to zero except initially, when deviation was 0.15 mm.

Figures 14 and 15 illustrate the experimental curves of slope tracking for the three control methods and the deviation curve, respectively. The frequency and amplitude of the target sinusoid were 1 Hz and 5 mm, respectively. Figures 14 and 15 indicate that single proportional valve control could not accurately track the target curve because of the low-frequency response of the proportional valve. Furthermore, the curve for dual-valve PID control was close to the target curve except at the peak position. The deviation of dual-valve PID–MPC control could be maintained in the range of 0–0.25 mm, which...
indicated that the rolling optimization of dual-valve PID–MPC control could effectively suppress the influence of the dead zone on the control accuracy.

The integrated time and absolute error (ITAE)\(^{16}\) was used to quantitatively evaluate the tracking accuracy of the three control methods. The equation is as follows

\[
ITAE = \int_{0}^{T} t |e(t)| \, dt
\]  

(20)
where the upper limit of integral $T$ is the experimental time and $|e(t)|$ is the difference between the actual output and the target values.

Table 1 summarizes the ITAE deviations for the three methods. According to the conducted slope tracking, the accuracy of dual-valve PID control was 39.3% higher
than that of single proportional valve control, whereas the accuracy of dual-valve PID–MPC control was 79.6% higher than that of single proportional valve control. Similarly, according to the conducted sinusoidal tracking, the accuracy of dual-valve PID control was 75.5% higher than that of single proportional valve control, whereas the accuracy of dual-valve PID–MPC control was 90.0% higher than that of single proportional valve control.
Few studies have examined the dynamic optimization of the control performance of a multivalve parallel electro-hydraulic servo system. In this study, a new parallel method and control algorithm were designed. The contributions of this study are as follows:

A technical solution is proposed by connecting a high-flow proportional valve with a low-flow servo valve. For predictive control, a dual-valve parallel PID–MPC composite control algorithm was designed with PID and MPC systems as the inner and outer loops, respectively.

**Figure 14.** Experimental curve of sinusoidal tracking.

**Figure 15.** Deviation curve of sinusoidal tracking.

**Conclusion**

Few studies have examined the dynamic optimization of the control performance of a multivalve parallel electro-hydraulic servo system. In this study, a new parallel method and control algorithm were designed. The contributions of this study are as follows:

A technical solution is proposed by connecting a high-flow proportional valve with a low-flow servo valve. For predictive control, a dual-valve parallel PID–MPC composite control algorithm was designed with PID and MPC systems as the inner and outer loops, respectively.
The simulation and experimental results revealed that dual-valve PID control could effectively improve the control accuracy and dynamic response performance of the electro-hydraulic servo system to a certain extent. Moreover, the PID–MPC control algorithm proposed in this study could efficiently utilize the advantages of the servo and proportional valves by conducting rolling optimization, thereby further dynamically improving the control accuracy of the system.

**Declaration of conflicting interests**

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**ORCID iD**

Shi-jie Su https://orcid.org/0000-0003-2784-9832

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**Author biographies**

Shi-jie Su graduated in Mechanical Design, Manufacturing and Automation (Master Degree) from Jiangsu University of Science and Technology in 2002. He holds a PhD in Mechatronic Engineering from Nanjing University of Aeronautics and Astronautics. Currently, his research line is targeted at the modeling and control of electro-hydraulic servo system.

Yuan-yuan Zhu graduated in Mechanical Design, Manufacturing and Automation from Jiangsu University of Science and Technology in 2017. Currently, she is studying for a master’s degree in mechanical engineering college of Jiangsu University of Science and Technology. Her research is focused on the development and application of modern control theory.

Cun-jun Li graduated from Shanghai Jiaotong University in 1988. He is a professor-level senior engineer, and he is the president of Zhoushan Institute of Calibration and Testing for Quality and Technology Supervision. Currently, he is mainly engaged in the development and application of mechanical testing system for large structural parts of ships.

Wen-xian Tang received his master’s and doctor’s degrees in Jiangsu University of Science and Technology(1994) and Shanghai University (2003). He studied as a postdoctoral researcher at the Postdoctoral Research Station in mechanical engineering of Nanjing University of Science and Technology (2003-2005). Currently, his research is focused on the digital design theory and method of Marine engineering equipment.

Hai-rong Wang graduated from China Jiliang University in 2000. He is a senior engineer, and he assists the president in charge of ship inspection, quality inspection and fire control inspection as the vice president. Currently, his research is focused on the design of hydraulic system and high precision testing technology.