

Impact of Spool Position Feedback on the Tracking Performance of Hydraulic Actuator Systems

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Abstract: This paper investigates the influence of spool position feedback on the tracking behavior of an electro-hydraulic actuator system. Two servo valve configurations, with and without spool position feedback, are modeled and evaluated under identical operating conditions using a nonlinear backstepping position controller. The reference trajectory is generated from an ISO 8608-based random road profile to represent realistic excitation conditions. In the spool-feedback configuration, the valve spool dynamics are explicitly included in the control loop, whereas the configuration without spool feedback is modeled using a simplified valve representation. The performance of both configurations is assessed through time-domain responses and quantitative error metrics, including RMS tracking error, maximum absolute error, ISE, ITAE, and RMS control effort. The results show that incorporating spool dynamics reduces tracking deviations and improves transient behavior while maintaining a comparable level of control effort. These findings indicate that valve-level dynamics should be considered in the design of electro-hydraulic control systems, particularly in applications requiring accurate reference tracking.

Key-Words: Nonlinear control, Backstepping control, Hydraulic actuator, Servo valve, Spool feedback, Position control, Tracking performance

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1 Introduction

Electro-hydraulic servo systems are widely used in high-performance motion control applications due to their high power density and fast dynamic response. However, achieving accurate tracking remains challenging because of nonlinearities, parameter uncertainties, actuator dynamics, and external disturbances, [1], [2].

To address these challenges, various advanced control strategies have been proposed in the literature. Adaptive robust control methods have demonstrated high tracking accuracy and robustness against parameter uncertainties in hydraulic actuator systems, [1], [3]. In addition, nonlinear adaptive backstepping and output-feedback backstepping approaches have been applied to electro-hydraulic systems to improve tracking performance and closed-loop stability, [4], [5].

Beyond controller design, the internal structure and dynamics of servo valves play a critical role in determining overall system performance. In particular, spool position feedback significantly affects the dynamic characteristics and accuracy of electro-hydraulic servo valves, as demonstrated in detailed modeling studies, [6]. Nevertheless, most existing research focuses on actuator-level control strategies, while the effect of spool position feedback on track-

ing behavior is less explicitly addressed.

Recent studies have investigated road simulators and hydraulic test systems designed to reproduce realistic operating conditions in laboratory environments. These studies include motorcycle road simulator design, real-world road profile reproduction on multi-post rigs, road simulator tracking control, electro-hydraulic waveform replication, and acceleration response analysis in electro-hydraulic shaking tables, [7], [8], [9], [10], [11]. Similar experimental and analytical efforts have also been reported for motorcycle testing applications, [12]. Since accurate tracking is essential for high-fidelity road-profile reproduction, control performance remains a key requirement in such systems.

Motivated by these considerations, this study investigates the influence of spool position feedback on the tracking performance of electro-hydraulic actuator systems. Two servo-valve configurations, with and without spool position feedback, are evaluated using identical ISO 8608-based road-profile reference signals, [13]. Both time-domain responses and quantitative performance metrics are considered.

Unlike previous studies that focused on road simulators, waveform replication, shaking-table response improvement, and hydraulic test-system implementa-

tion aspects, [7], [8], [9], [10], [11], [12], the objective of the present work is not to develop a new control algorithm. Instead, a nonlinear backstepping controller is employed as a common framework to isolate the effect of spool position feedback. By using the same controller for both valve configurations, the influence of spool feedback can be assessed independently of controller-related factors, enabling a direct and fair comparison.

The main contribution of this work is to quantify how spool feedback affects the tracking behavior of an electro-hydraulic actuator system. The results show that incorporating spool dynamics into the control loop improves tracking accuracy without increasing the overall control effort. These findings provide further insight into the role of valve-level dynamics and may support the design of high-performance electro-hydraulic control systems.

2 System Modeling

This section presents the physical model and the governing equations of the electro-hydraulic servo system considered in this study. The system consists of a hydraulic actuator driven by a servo valve, where the actuator motion is controlled through hydraulic pressure differences.

The physical model of the test system is illustrated in Fig. 1. In this model, m represents the equivalent mass of the actuator-load system, P_1 and P_2 denote the pressures in the two chambers of the cylinder, and A is the effective piston area. The supply and return pressures are denoted by P_s and P_r , respectively. The control input is represented by u . The load pressure P_L is defined as the pressure difference between the two chambers, i.e., $P_L = P_1 - P_2$.

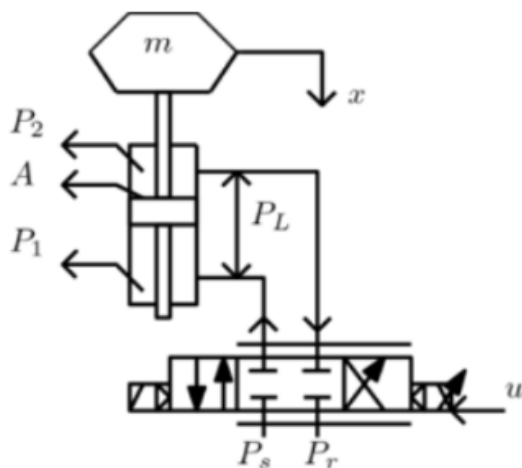


Fig. 1: Physical model of the electro-hydraulic system

Source: created by the authors.

Based on the physical model shown in Fig. 1, the governing equations of motion are derived and presented in Eqs. (1)-(3).

Let x_1 denote the piston position, x_2 represent the piston velocity, and x_3 denote the load pressure-related state variable. The coefficients f_1 , f_2 , and f_3 are system parameters derived from the physical properties of the hydraulic system, including fluid bulk modulus, leakage coefficients, and valve flow characteristics.

$$m\ddot{x}_1 = A \cdot x_3 \quad (1)$$

$$\dot{x}_3 = f_1 x_2 + f_2 x_3 + f_3 \sqrt{P_s - \text{sign}(k_g u) x_3} \cdot u \quad (2)$$

$$f_1 = -\frac{4\beta_e A}{V_t}, f_2 = -\frac{4\beta_e C_{tl}}{V_t}, f_3 = \frac{4\beta_e C_d w k_i}{V_t \sqrt{\rho}} \quad (3)$$

These equations capture the essential nonlinear dynamics of the electro-hydraulic actuator and form the basis for the control design presented in the following sections. The parameters used in the simulations are summarized in Table 1. The parameters used in the simulations are selected based on typical values reported in the literature and are kept consistent for both configurations to ensure a fair comparison.

Table 1: System parameters

Parameter	Description	Unit	Value
m	Equivalent mass	kg	124
A	Effective piston area	m ²	9.45e-4
β_e	Effective bulk modulus	Pa	1.4e9
V_t	Total cylinder volume	m ³	9.45e-5
C_{tl}	Leakage coefficient	m ³ /(s·Pa)	3e-11
C_d	Discharge coefficient	-	0.62
w	Servo valve area gradient	m	0.024
k_i	Electrical gain	m/A	3e-4
ρ	Hydraulic oil density	kg/m ³	850
P_s	Supply pressure	Pa	2.1e7
k_g	Valve flow gain constant	-	1e8

Source: created by the authors.

3 Control Design

This section presents the nonlinear backstepping-based position control design for the electro-hydraulic servo system considering two valve configurations.

First, the design is formulated for the valve configuration without spool position feedback. Then, spool dynamics are incorporated into the model, resulting in an augmented backstepping structure for the feedback configuration.

3.1 Nonlinear Backstepping Position Control for the Valve Without Spool Feedback

In this subsection, a nonlinear backstepping-based position control strategy is developed for the servo valve without spool position feedback. The control objective is to ensure that the actuator position tracks a desired reference trajectory.

Let e denote the tracking error, where x_r represents the reference position. The tracking error is defined as

$$e = x_r - x_1 \quad (4)$$

A Lyapunov candidate function is selected as

$$V_1 = \frac{1}{2}e^2 \quad (5)$$

Taking the time derivative of V_1 yields

$$\dot{V}_1 = e\dot{e} = e(\dot{x}_r - \dot{x}_2) \quad (6)$$

Since the control input does not explicitly appear in this expression, a virtual control law is introduced.

Let x_{2d} denote the desired (virtual) velocity. Let z_1 denote the first error variable, defined as the difference between the actual piston velocity and its desired value:

$$z_1 = x_2 - x_{2d} \quad (7)$$

Substituting x_2 into $\dot{V}_1$ gives

$$\dot{V}_1 = e(\dot{x}_r - z_1 - \dot{x}_{2d}) \quad (8)$$

To render the Lyapunov derivative partially negative, the virtual control input is chosen as

$$x_{2d} = k_1 e + \dot{x}_r \quad (9)$$

which leads to

$$\dot{V}_1 = -k_1 e^2 - z_1 e \quad (10)$$

To eliminate the remaining cross-term, an augmented Lyapunov function is defined as

$$V_2 = V_1 + \frac{1}{2}z_1^2 \quad (11)$$

Its time derivative becomes

$$\dot{V}_2 = \dot{V}_1 + z_1(\dot{x}_2 - \dot{x}_{2d}) \quad (12)$$

Using the actuator dynamics given in Eq. (1), we obtain

$$\dot{V}_2 = \dot{V}_1 + z_1 \left(\frac{A}{m} x_3 - \dot{x}_{2d} \right) \quad (13)$$

The recursive design is continued by introducing a second virtual control variable associated with the

load pressure dynamics. Let x_{3d} denote the desired (virtual) control input associated with the load pressure dynamics. Let z_2 denote the second error variable, defined as the difference between the actual load pressure state and its desired value:

$$z_2 = x_3 - x_{3d} \quad (14)$$

Substituting x_3 results in

$$\dot{V}_2 = \dot{V}_1 + z_1 \left(\frac{A}{m} (z_2 + x_{3d}) - \dot{x}_{2d} \right) \quad (15)$$

The desired virtual control x_{3d} is selected as

$$x_{3d} = \frac{m}{A} (e + \dot{x}_{2d} - k_2 z_1) \quad (16)$$

which leads to

$$\dot{V}_2 = -k_1 e^2 - k_2 z_1^2 + \frac{m}{A} z_1 z_2 \quad (17)$$

To complete the recursive stabilization procedure, the Lyapunov function is further extended as

$$V_3 = V_2 + \frac{1}{2}z_2^2 \quad (18)$$

Taking its derivative gives

$$\dot{V}_3 = \dot{V}_2 + z_2 \dot{z}_2 \quad (19)$$

Using the system dynamics, the final expression becomes

$$\dot{V}_3 = \dot{V}_2 + z_2 \left(f_1 x_2 + f_2 x_3 + f_3 \sqrt{P_s - \text{sign}(k_g u) x_3} u - \dot{x}_{3d} \right) \quad (20)$$

Finally, the control input is designed as

$$u = \frac{1}{f_3 \sqrt{P_s - \text{sign}(k_g u) x_3}} v \quad (21)$$

where $v = -f_1 x_2 - f_2 x_3 + \dot{x}_{3d} - k_3 z_2 - \frac{m}{A} z_1$.

Substituting the final control law into the Lyapunov derivative yields

$$\dot{V}_3 = -k_1 e^2 - k_2 z_1^2 - k_3 z_2^2 \leq 0 \quad (22)$$

Thus, for positive gains k_1 , k_2 , and k_3 , the Lyapunov derivative is negative semi-definite. This ensures closed-loop stability and asymptotic convergence of the tracking errors.

3.2 Nonlinear Backstepping Position Control for the Valve With Spool Feedback

In this configuration, the spool dynamics are explicitly included in the system model by introducing an additional state for the spool displacement. In the spool feedback configuration, the discontinuous sign function used in the previous model is replaced by a smooth hyperbolic tangent function to improve numerical stability.

Let x_4 denote the spool displacement, T_{sv} the servo valve time constant, and K_{sv} the valve gain. The control input corresponding to the spool feedback configuration is denoted by u_{sf} . The spool dynamics are then given by

$$\dot{x}_4 = -\frac{1}{T_{sv}}x_4 + \frac{K_{sv}}{T_{sv}}u_{sf} \quad (23)$$

With the inclusion of spool dynamics, the hydraulic pressure dynamics are modified accordingly. The updated nonlinear relationship can be expressed as

$$\dot{x}_3 = f_1x_2 + f_2x_3 + f_3\sqrt{P_s - \tanh(k_gx_4)x_3x_4} \quad (24)$$

Unlike the previous configuration, the valve flow is not directly driven by the control input. Instead, it is represented through the spool displacement x_4 . Therefore, the input gain k_i is not included in f_3 in this formulation.

First, the tracking error is defined as the difference between the desired reference position and the actual piston position:

$$e_1 = x_r - x_1 \quad (25)$$

An augmented Lyapunov function is selected as

$$V_4 = \frac{1}{2}e_1^2 \quad (26)$$

Taking the time derivative of V_4 gives

$$\dot{V}_4 = e_1(\dot{x}_r - x_2) \quad (27)$$

Since the actual piston velocity x_2 is not the direct control input, a desired virtual velocity ϕ_1 is introduced. The first error variable z_3 is defined as

$$z_3 = \phi_1 - x_2 \quad (28)$$

After substituting this expression, the derivative becomes

$$\dot{V}_4 = e_1(\dot{x}_r - \phi_1 + z_3) \quad (29)$$

The first virtual control input is selected as

$$\phi_1 = \dot{x}_r + k_1^f e_1 \quad (30)$$

where k_1^f is a positive control gain. Substituting ϕ_1 into $\dot{V}_4$ results in

$$\dot{V}_4 = -k_1^f e_1^2 + z_3 e_1 \quad (31)$$

To eliminate the remaining cross-term $z_3 e_1$, the Lyapunov function is further augmented as

$$V_5 = \frac{1}{2}e_1^2 + \frac{1}{2}z_3^2 \quad (32)$$

By differentiating V_5 , we obtain

$$\dot{V}_5 = \dot{V}_4 + z_3 \dot{z}_3 \quad (33)$$

Substituting $\dot{z}_3 = \dot{\phi}_1 - \dot{x}_2$ into $\dot{V}_5$ yields

$$\dot{V}_5 = -k_1^f e_1^2 + z_3 e_1 + z_3 (\dot{\phi}_1 - \dot{x}_2) \quad (34)$$

Using the actuator dynamics given in Eq. (1), the above expression becomes

$$\dot{V}_5 = -k_1^f e_1^2 + z_3 e_1 + z_3 \left(\dot{\phi}_1 - \frac{A}{m} x_3 \right) \quad (35)$$

The next backstepping step is formulated by defining the second virtual control input ϕ_2 for the load pressure state. The corresponding error variable z_4 is introduced as

$$z_4 = \phi_2 - x_3 \quad (36)$$

Using this relation, $\dot{V}_5$ can be rewritten as

$$\dot{V}_5 = -k_1^f e_1^2 + z_3 e_1 + z_3 \left(\dot{\phi}_1 - \frac{A}{m} \phi_2 + \frac{A}{m} z_4 \right) \quad (37)$$

The second virtual control input is selected as

$$\phi_2 = \frac{m}{A} \left(\dot{\phi}_1 + k_2^f z_3 + e_1 \right) \quad (38)$$

where k_2^f is a positive control gain. Substituting ϕ_2 into $\dot{V}_5$ yields

$$\dot{V}_5 = -k_1^f e_1^2 - k_2^f z_3^2 + \frac{A}{m} z_3 z_4 \quad (39)$$

To continue the recursive stabilization procedure, the Lyapunov function is extended as

$$V_6 = V_5 + \frac{1}{2}z_4^2 \quad (40)$$

The time derivative of V_6 is obtained as

$$\dot{V}_6 = \dot{V}_5 + z_4 \dot{z}_4 \quad (41)$$

Since $\dot{z}_4 = \dot{\phi}_2 - \dot{x}_3$, and using the modified pressure dynamics for the spool feedback configuration, $\dot{V}_6$ becomes

$$\dot{V}_6 = \dot{V}_5 + z_4 \left[\dot{\phi}_2 - (f_1 x_2 + f_2 x_3 + f_3 \sqrt{P_s - \tanh(k_g x_4) x_3 x_4}) \right] \quad (42)$$

The recursive design is then extended to the spool displacement dynamics by introducing the third virtual control input ϕ_3 . Accordingly, the error variable z_5 is written as

$$z_5 = \phi_3 - x_4 \quad (43)$$

By using $x_4 = \phi_3 - z_5$, the following expression is obtained:

$$\dot{V}_6 = \dot{V}_5 + z_4 \left[\dot{\phi}_2 - f_1 x_2 - f_2 x_3 - f_3 \sqrt{P_s - \tanh(k_g x_4) x_3 (\phi_3 - z_5)} \right] \quad (44)$$

The third virtual control input is selected as

$$\phi_3 = \frac{1}{f_3 \sqrt{P_s - \tanh(k_g x_4) x_3}} \left(-f_1 x_2 - f_2 x_3 + \dot{\phi}_2 + k_3^f z_4 + \frac{A}{m} z_3 \right) \quad (45)$$

where k_3^f is a positive control gain. Substituting ϕ_3 into $\dot{V}_6$ results in

$$\dot{V}_6 = -k_1^f e_1^2 - k_2^f z_3^2 - k_3^f z_4^2 + f_3 \sqrt{P_s - \tanh(k_g x_4) x_3 z_4 z_5} \quad (46)$$

To handle the remaining coupling term $z_4 z_5$, the Lyapunov function is augmented as

$$V_7 = V_6 + \frac{1}{2} z_5^2 \quad (47)$$

Differentiating V_7 with respect to time gives

$$\dot{V}_7 = \dot{V}_6 + z_5 \dot{z}_5 \quad (48)$$

Since $\dot{z}_5 = \dot{\phi}_3 - \dot{x}_4$, and using the spool dynamics, $\dot{V}_7$ becomes

$$\dot{V}_7 = \dot{V}_6 + z_5 \left[\dot{\phi}_3 - \left(-\frac{1}{T_{sv}} x_4 + \frac{K_{sv}}{T_{sv}} u_{sf} \right) \right] \quad (49)$$

The final control input for the spool feedback configuration is selected as

$$u_{sf} = \frac{T_{sv}}{K_{sv}} \left(\frac{x_4}{T_{sv}} + \dot{\phi}_3 + k_4^f z_5 + f_3 \sqrt{P_s - \tanh(k_g x_4) x_3 z_4} \right) \quad (50)$$

where k_4^f is a positive control gain. Substituting u_{sf} into $\dot{V}_7$ yields

$$\dot{V}_7 = -k_1^f e_1^2 - k_2^f z_3^2 - k_3^f z_4^2 - k_4^f z_5^2 \leq 0 \quad (51)$$

Similarly, for positive gains k_1^f, k_2^f, k_3^f , and k_4^f , the extended Lyapunov derivative satisfies $\dot{V}_7 \leq 0$. Since the Lyapunov function is positive definite and its time derivative is negative semi-definite, all closed-loop signals remain bounded. Furthermore, according to Lyapunov stability theory, the spool-feedback closed-loop system remains stable, and the tracking errors converge asymptotically to zero.

Compared to the valve model without spool feedback, the inclusion of spool dynamics increases the system order and introduces an additional control design step. This extended model provides a more realistic representation of valve behavior and enables improved tracking performance. The control parameters used for both configurations are summarized in Table 2.

Table 2: Control parameters used in the simulations

Parameter	Description	Unit	Value
k_1	Gain (Valve1 - No FB)	-	1
k_2	Gain (Valve1 - No FB)	-	50
k_3	Gain (Valve1 - No FB)	-	1500
k_1^f	Gain (Valve2 - FB)	-	1e2
k_2^f	Gain (Valve2 - FB)	-	1e2
k_3^f	Gain (Valve2 - FB)	-	1e-3
k_4^f	Gain (Valve2 - FB)	-	1e-5
K_{sv}	Servo valve gain	m/A	2e17
T_{sv}	Servo valve time constant	s	1.2e-2

Source: created by the authors.

The controller gains were selected through iterative simulations to balance tracking accuracy, transient response, control effort, and closed-loop stability. Since the state variables in the backstepping design have different magnitudes, the gains span several orders of magnitude and should be interpreted as tuning parameters associated with the scaled states, rather than directly comparable physical quantities. The servo valve gain K_{sv} is introduced as an equivalent model parameter relating the electrical control

signal to the spool displacement dynamics. Its numerical value is affected by the scaling introduced through the pressure–flow relationships, hydraulic parameters, and unit conversions in the state-space formulation. Therefore, it does not represent a directly measurable physical valve gain, but an effective gain used to preserve the actuator dynamics in the simplified simulation model. Dimensional consistency is maintained through the adopted formulation, and the closed-loop responses were verified to remain physically meaningful.

4 Simulation Setup and Results

This section presents the simulation setup and the comparative performance results of the two valve configurations. Both control structures were evaluated under identical reference inputs and operating conditions to ensure a fair comparison.

4.1 Reference Signal and Simulation Conditions

The reference signal was generated as a synthetic random road profile based on the ISO 8608 spatial PSD representation, [13]. The profile was constructed using a power spectral density formulation defined by a reference spatial frequency and a specified road roughness coefficient. For a fair comparison, both valve configurations were evaluated under identical simulation conditions and subjected to the same reference input signal. The simulations were performed in the time domain with a fixed sampling time.

The performance of the control systems was evaluated using both qualitative and quantitative criteria. Time-domain responses were analyzed for tracking behavior, while root mean square (RMS) tracking error, maximum absolute tracking error, and control effort were used for quantitative comparison. The generated reference road profile is shown in Fig. 2.

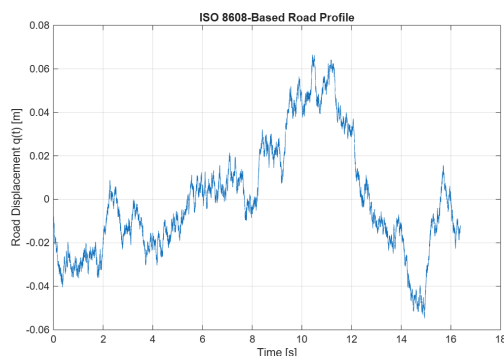


Fig. 2: Generated ISO 8608-based road profile used as the reference input signal

Source: created by the authors.

4.2 Tracking Performance Comparison

This subsection compares the tracking behavior of the two valve configurations under the same reference input using time-domain responses and quantitative performance metrics.

The position responses are presented in Fig. 3. Both configurations are able to follow the reference trajectory, indicating that the backstepping controller provides stable tracking for both valve models. However, Valve2 remains closer to the reference signal, particularly during intervals where the road profile exhibits rapid variations. This suggests that the inclusion of spool position feedback improves the dynamic response of the actuator and reduces the deviation from the desired trajectory. The improvement is more clearly observed in the zoomed-in response shown in Fig. 4 for the interval $t = 2\text{--}3$ s.

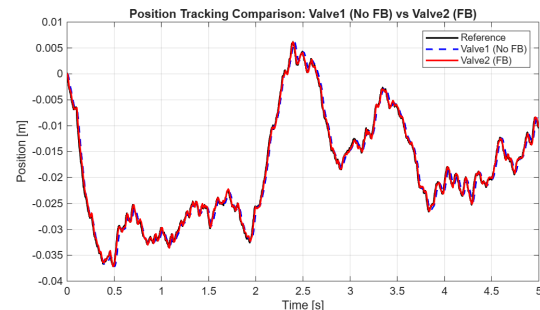


Fig. 3: Position tracking comparison of Valve1 (without spool feedback) and Valve2 (with spool feedback)

Source: created by the authors.

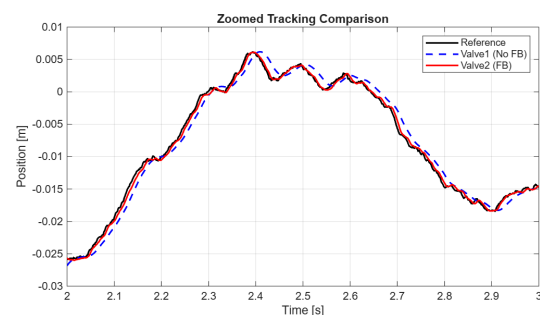


Fig. 4: Zoomed tracking comparison in the interval of $t = 2\text{--}3$ s

Source: created by the authors.

The zoomed response indicates that Valve2 captures small variations in the reference signal more accurately than Valve1.

The tracking error responses of both configurations are presented in Fig. 5.

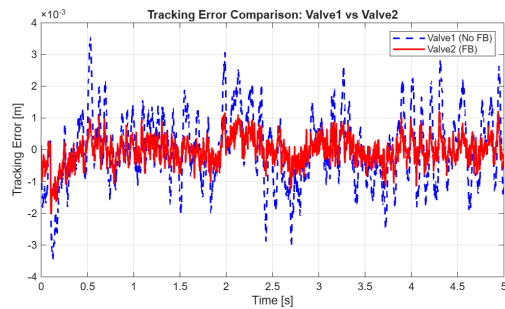


Fig. 5: Tracking error comparison of Valve1 and Valve2

Source: created by the authors.

As shown in Fig. 5, Valve2 produces lower tracking errors than Valve1. The error response also exhibits smaller oscillations, indicating improved dynamic behavior. A zoomed-in view of the tracking error in the interval of $t = 2-3$ s is provided in Fig. 6.

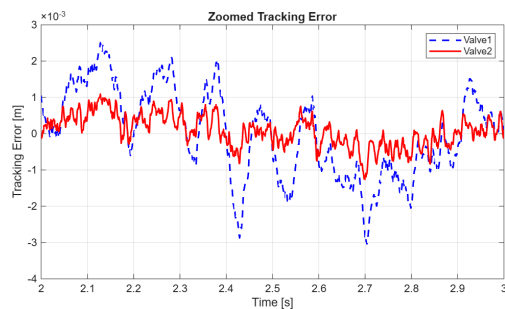


Fig. 6: Zoomed tracking error comparison in the interval of $t = 2-3$ s

Source: created by the authors.

The zoomed error response in Fig. 6 confirms that Valve2 maintains smaller deviations in the selected interval. The moving RMS comparison of the control input signals is shown in Fig. 7.

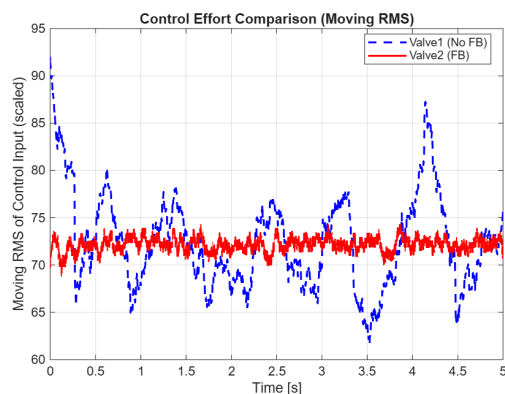


Fig. 7: Moving RMS comparison of the control input signals for Valve1 and Valve2

Source: created by the authors.

In addition to tracking performance, the control effort of the two configurations is evaluated using the moving RMS of the control input signals, as shown in Fig. 7. The signals are scaled to allow a clearer visual comparison of their behavior, independent of their absolute magnitudes. This metric provides a time-varying representation of the effective control magnitude.

It is observed that both control strategies operate within comparable control effort levels. However, the spool-feedback configuration shows a more consistent control behavior, whereas the case without feedback exhibits larger fluctuations and occasional peaks.

This suggests that the improved response is mainly related to the inclusion of spool dynamics, which provides a more accurate representation of internal valve behavior. Therefore, the performance gain can be attributed to improved utilization of the control input rather than an increase in control demand. A quantitative comparison of the performance metrics is summarized in Table 3.

Table 3: Performance comparison of Valves

Metric	Valve1 (No FB)	Valve2 (FB)
RMS Tracking Error [m]	1.125e-3	4.402e-4
Max Absolute Error [m]	3.534e-3	2.017e-3
ISE	6.327e-6	9.688e-7
ITAE	1.127e-2	3.950e-3
RMS Control Effort	72.66	72.14

Source: created by the authors.

The quantitative results indicate that the spool-feedback configuration provides lower tracking errors across all error-based metrics. The RMS tracking error decreases from 1.125×10^{-3} m to 4.402×10^{-4} m when spool position feedback is included, corresponding to an approximate reduction of 60.9%. Similarly, the maximum absolute error is reduced from 3.534×10^{-3} m to 2.017×10^{-3} m.

The integral error metrics also show a consistent reduction. The ISE value decreases by approximately 84.7%, indicating a lower accumulated squared tracking error over the simulation interval. In addition, the ITAE value is reduced by about 65.0%, suggesting an improved transient error response.

It is also observed that the RMS control effort remains at a comparable level for both configurations. This suggests that the error reduction is mainly associated with the inclusion of spool dynamics rather than an increase in control demand. Therefore, the quantitative results support the time-domain observations and show that spool position feedback improves both steady-state tracking accuracy and transient response while maintaining a similar control effort.

4.3 Robustness and Practical Implementation

Although the simulations were conducted using nominal system parameters, practical electro-hydraulic systems are inherently subject to parameter variations, actuator nonlinearities, and external disturbances. The proposed nonlinear backstepping controller is designed based on Lyapunov stability theory, which provides a degree of robustness against moderate modeling uncertainties while preserving closed-loop stability. In addition, the controller was evaluated using ISO 8608-based randomly varying road profiles, representing disturbance-like operating conditions. From an implementation perspective, the discontinuous sign function was replaced by a smooth hyperbolic tangent function in the spool-feedback configuration to improve numerical stability and reduce potential implementation issues. Nevertheless, practical applications may still be influenced by sensor noise, actuator saturation, and unmodeled dynamics. Therefore, experimental validation and a broader robustness assessment under uncertain operating conditions remain necessary steps for future work.

5 Conclusion

This paper investigated the influence of spool position feedback on the tracking behavior of an electro-hydraulic servo system. Two servo valve configurations, with and without spool position feedback, were modeled and controlled using the same nonlinear backstepping framework to provide a consistent basis for comparison.

The simulation results showed that including spool dynamics in the control loop reduces tracking errors and improves the transient response. These gains were obtained without a noticeable increase in control effort, suggesting that the spool-feedback model uses the available control action more efficiently. The results also suggest that an explicit representation of spool dynamics helps the controller capture internal valve behavior and compensate for nonlinear hydraulic effects.

The study demonstrates that valve-level dynamics can have a significant influence on closed-loop performance and should not be neglected in high-precision electro-hydraulic control applications. In this context, spool feedback can be considered a practical way to improve reference tracking in high-precision hydraulic applications.

The observed improvement is mainly associated with the inclusion of spool dynamics rather than a specific feature of the selected controller. Therefore, similar benefits may also be expected when spool position feedback is incorporated into other nonlinear control architectures. Nevertheless, the degree of im-

provement may depend on the controller structure and design methodology.

Although the presented results demonstrate the effectiveness of the proposed approach, experimental validation and a more comprehensive robustness assessment under uncertain operating conditions remain important topics for future investigation. In addition, future studies may extend the analysis by comparing the proposed approach with other advanced nonlinear control strategies available in the literature. Such comparisons may provide further insight into the role of spool position feedback under different control architectures and operating conditions.

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Contribution of individual authors to the creation of a scientific article (ghostwriting policy)

Talha Oguz carried out the modeling, simulations, and implementation of the study. He also performed the analysis of the results and wrote the manuscript.

Muzaffer Metin contributed through supervision, guidance, and overall project management. He also provided critical revisions and contributed to the conceptual development of the study.

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