

Article

High-Frequency Position Servo Control of Hydraulic Actuator with Valve Dynamic Compensation

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Abstract: Hydraulic actuators play an important role in various industries. In the last decades, to improve system performance, some advanced control methods have been developed. Backstepping control, which can deal with the system nonlinearities, is widely used in hydraulic system motion control. This paper focuses on the high-frequency position servo control of hydraulic systems with proportional valves. In backstepping controllers, valve dynamics are usually ignored due to difficulty of controller implementation. In this paper, valve dynamics of the proportional valve were decoupled into phase delay and amplitude delay. The valve dynamics are compensated without increasing the system order. The phase delay is compensated by desired engine valve lifts transformation. For amplitude delay, the paper proposes a compensation strategy based on the integral flow error. By introducing the feedback of the integral flow error to the backstepping controller, the system has faster dynamic responses. Besides, the controller also synthesized proportional valve dead-zone and system uncertainties. The comparative experiment results show that the controller with integral flow compensation can improve engine valve lift tracking precision both in steady and transient conditions.

Keywords: valve dynamics; backstepping control; nonlinear systems; electro-hydraulic; variable valve technology



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1. Introduction

Electro-hydraulic servo systems have been widely used in various industries over the past years. With the development of industrial applications, there are strong demands to improve control performance of hydraulic actuators especially in high-frequency position servo systems [1–3]. However, traditional linear control strategies can no longer meet the needs of high-performance control due to ignoring the nonlinear characteristics in the systems.

The paper focuses on the high-performance position servo control of hydraulic systems with proportional valves. The systems have complicated nonlinear characteristics, including valve dead zone [4–6], valve dynamics [7], system uncertainties [8–10], et al. The reference [4] shows that the dead-zone has significant influence on a state observer. In [5], Riccati equation is used to develop a state observer, which is able to fully consider dead-zone hard nonlinearity. In [6], a proportional valve was used to drive a hydraulic manipulator. By compensating valve dead-zone, the system is capable of accurate piston motion tracking, despite the use of a slow-response pressure compensated directional valve with a bandwidth of only 4 Hz. The reference [7] shows that the valve time delay characteristic will delay shifting process and increase the sliding friction time of a friction plate. Many advanced control strategies have been proposed to compensate system dead zone, valve delay and system uncertainties [4,7,8,11–13]. In [11], a system symmetric control method was brought out, and the state space mathematical model was derived to solve the asymmetry, inconsistent dynamic characteristics for positive/negative directions and poor stability for asymmetric control system. In [12], a sliding model controller which

incorporated the derivatives of the control input was developed to reduce the position tracking errors. In [13], A PID controller together with a feedforward (FF) controller is applied to the valve. To further improve the response of the servo valve, an input shaping filter (ISF) is incorporated into the valve control system.

Backstepping control, which can deal with system nonlinear characteristics through a recursive design procedure, is one of the most popular methods [14–16]. In [14], an indirect adaptive backstepping controller was developed to compensate system nonlinear characteristics such as time-varying friction, leakage, et al. In [15], a nonlinear sliding model controller based on backstepping method was presented for electro-hydraulic single-rod actuators in a practical projectile transfer arm. In [16], a backstepping controller was designed to derive a nonlinear control law for force control of an hydraulic actuator. However, valve dynamics are usually ignored in backstepping controllers, which makes the system only work in low-frequency conditions [17–19]. One of the most important reasons is that the high-order valve model is not accepted by backstepping controllers, although valve dynamics can be described by second-order or higher-order models, because it is possible to perform high-order difference of displacement signals if high-order models are used [17,18]. The displacement signals acquired from sensors are noisy, and high-order difference will amplify the noise, which makes the real signal merge with noises and become difficult to be distinguished [19]. Besides, “explosion of complexity” is inevitable in recursive procedures [20,21]. In [22,23], to implement the controllers, valve models were simplified as a linear gain of control input, although first-order or second-order models were used in the controller designment process. When system frequency is low, the valve dynamics have little influence on system control performance. In [24–26], backstepping controllers were developed to track desired trajectories or rotate speed in systems controlled by proportional valves or servo valves, and the control performance was apparently improved. The system frequency in [24–26] is 2, 0.5 and 0.05, respectively. When system frequency is high, it is unreasonable to ignore valve dynamics.

The step response of a proportional valve is shown in Figure 1. Proportional valve dynamics can be decoupled into valve phase delay and amplitude delay.

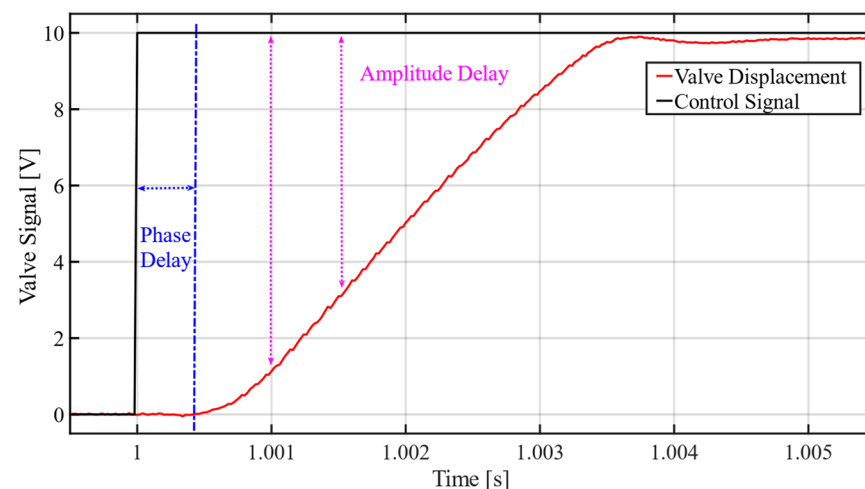


Figure 1. Step response of proportional valve.

For system phase delay, some compensation strategies based on the delay model or velocity, and acceleration signals have been developed. In [27,28], feedforward controllers were designed to compensate system delay. In [29], Ding et al. developed a delay observer to compensate the valve phase delay. There are also some compensation strategies which do not rely on accurate delay models. In [30], Cao et al. proposed an adaptive neural network controller designed to compensate system input delay. However, complicated calculations of RBF neural network parameters must be performed in [30]. For hydraulic systems without a velocity sensor and acceleration sensor, these compensation strategies are difficult

to implement. For valve amplitude delay, there have been almost no reports on amplitude delay compensation. For high-frequency systems, valve amplitude delay will cause large control errors. By introducing a high gain feedback of output, the system can achieve quick dynamic responses, but it may destabilize the closed-loop system [31–34]. The paper is dedicated to compensating the proportional valve dynamics in backstepping controllers without using high-order proportional valve models in high-frequency hydraulic systems.

The paper is organized as follows. The valve dynamics compensation strategy is shown in Section 2. Problem statement and system model are given in Section 3. Section 4 develops backstepping controllers with proportional valve dynamics compensation. Section 5 reveals experiment comparisons with controllers without compensation. Finally, some conclusions are given in Section 6.

2. Valve Dynamics Compensation Strategy

In this paper, we proposed a desired trajectory transformation strategy to compensate for the valve phase delay. The system delay is calculated through the position signal. Besides, a feedback of integral flow error is added to the backstepping controller to compensate for the proportional valve amplitude delay. The actuator position is determined directly by oil flow into the hydraulic cylinder. By calculating the integral flow error instead of the instantaneous flow error and applying its linear feedback directly to the control input, the position servo error caused by the amplitude delay can be compensated faster, so the position tracking performance can be improved.

The proposed controller is applied to a kind of fully variable valve hydraulic actuator (FVVHA) of medium and high-speed diesel engines. Traditional engine valve lift is determined by engine camshaft. There is only one engine valve lift in all engine conditions, which makes the combustion unable to maintain an optimal state in all conditions (e.g., engine speed, engine load) [35,36]. For an engine configured with a FVVHA, engine valve lift can be changed easily by adjusting control signals of the hydraulic actuator. Flexible valve lifts can make the engine work in optimal states in all work conditions, so the engine performance can be improved [37,38]. In [39], the optimal valve lifts are obtained using the multi-objective optimization method. A typical engine valve lift is shown in Figure 2.

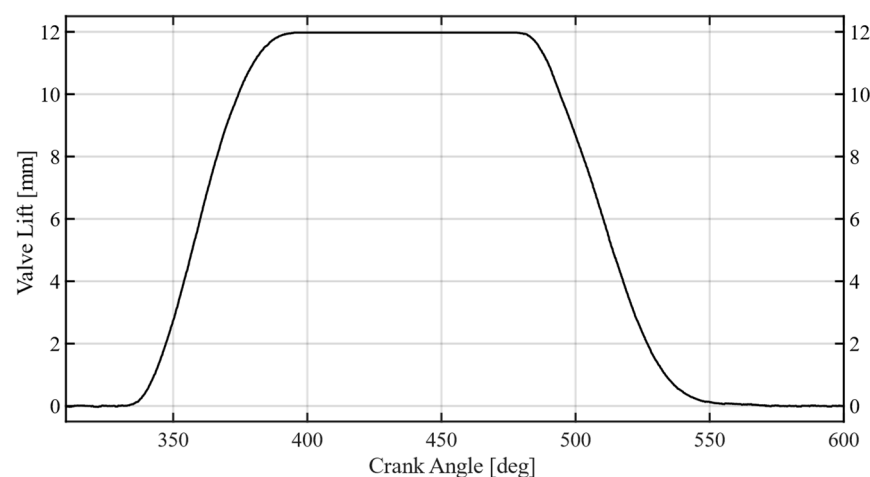


Figure 2. A typical engine valve lift.

The FVVHA schematic and prototype are shown in Figure 3. The engine valve is driven by a hydraulic actuator. The movement of the engine valve is controlled by the proportional valve. So, the most important part of a FVVHA system is the tracking controller. For a four-stroke medium and high-speed diesel engine, the maximum engine speed is 1200 r/min in this paper. The maximum valve opening duration is 300 °CA (41.67 ms), and the minimum valve opening duration is 220 °CA (30.55 ms) [40]. After optimization, the minimum time of valve movement is 80 °CA (11.1 ms) in an engine cycle. Therefore, the engine valve

lift tracking control is a high-frequency position servo event, and the proportional valve dynamics have to be taken into account.

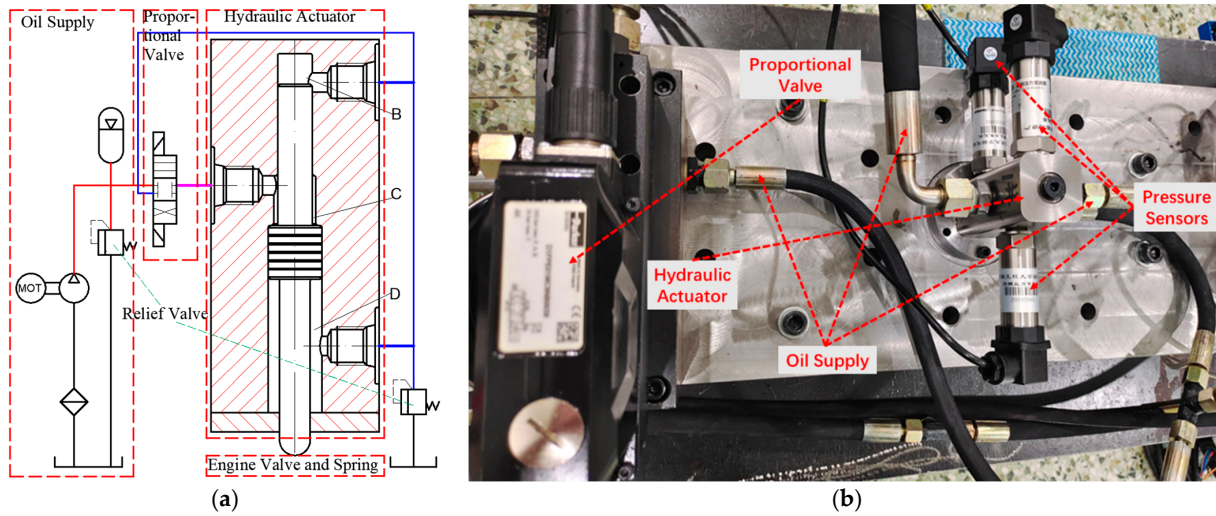


Figure 3. FVVHA schematic and prototype. (a) FVVHA schematic; (b) FVVHA prototype.

For engine valve lift tracking control, this paper has the following technical novelties:

- (1) The proportional valve dynamics are innovatively decoupled into phase delay and amplitude delay. Therefore, proportional valve dynamics can be compensated by phase delay compensation and amplitude delay compensation, respectively.
- (2) Proportional valve dynamics are compensated in a backstepping controller without increasing system order. Valve phase delay can be compensated by trajectory transformation strategy because the desired engine lifts are known in advance. Valve amplitude delay can be compensated by feedback of integral flow. The paper innovatively proposed feedback of integral flow instead of instantaneous flow. Compared with gain of instantaneous flow error, integral flow error gain is smaller, which can achieve smaller tracking errors. The experiment results in Section 5 can verify this conclusion.

3. Problem Statement and System Modeling

As shown in Figure 3a, the engine valve is driven by a hydraulic cylinder. When Chamber B is connected with high-pressure oil, the engine valve spring will be compressed, and the engine valve will open. When Chamber B is connected with low-pressure oil, engine valves will close under action of the valve spring. Engine valve lift is determined by forces acted on the engine valve, including hydraulic force, spring force, friction force and unknown force. Engine valve lift can be calculated by:

$$ma = P_B A_B + P_C A_C - P_D A_D + mg - F_f - F_s + d \quad (1)$$

where m is the mass of all movement parts; a is the acceleration of the valve; P_B , P_C , P_D are pressures of Chamber B, C and D; A_B , A_C , A_D are effective action areas of Chamber B, C and D; F_f is the friction force of the system and the F_s is the spring force.

P_B , P_C and P_D can be described by:

$$\dot{P}_B = \frac{\beta_e}{V_B} (Q_B + Q_{CB} - A_B v) \quad (2)$$

$$\dot{P}_C = \frac{\beta_e}{V_C} (Q_C - Q_{CB} - Q_{CD} - A_C v) \quad (3)$$

$$\dot{P}_D = \frac{\beta_e}{V_D} (Q_D + Q_{CD} + A_D v) \quad (4)$$

where β_e is the elasticity modulus of oil; V_B , V_C , V_D are effective volumes of Chamber B, Chamber C, Chamber D and their auxiliary pipeline, respectively; Q_B , Q_C , Q_D are flow rates into Chamber B, C and D. Q_{CB} and Q_{CD} are the flow rates leaked from Chamber C to Chamber B and Chamber D, respectively. V_B , V_C and V_D change with the movement of the valve, but their variations are much smaller than the total volume. Therefore, the variations are ignored in this paper.

Q_B , Q_C , Q_D , Q_{CB} and Q_{CD} can be described by:

$$Q_B = C_{dB} A_{vB} \sqrt{\frac{2}{\rho} |P_l - P_B| \text{sgn}(P_l - P_B)} \quad (5)$$

$$Q_C = C_{dC} \omega x_v \left[s(x_v) \sqrt{\frac{2}{\rho} (P_h - P_C)} + s(-x_v) \sqrt{\frac{2}{\rho} (P_C - P_l)} \right] \quad (6)$$

$$Q_D = C_{dD} A_{vD} \sqrt{\frac{2}{\rho} |P_l - P_D| \text{sgn}(P_l - P_D)} \quad (7)$$

$$Q_{CB} = C_{lB} (P_C - P_B) \quad (8)$$

$$Q_{CD} = C_{lD} (P_C - P_D) \quad (9)$$

where C_{dB} , C_{dD} are equivalent flow coefficients of oil port in Chamber B and D; C_{dC} is equivalent flow coefficient of proportional valve; A_{vB} , A_{vD} are effective flow areas of oil port; ω is area gain coefficient of proportional valve; x_v is the real valve openings of the proportional valve instead of the valve command signal; C_{lB} and C_{lD} are leakage coefficients leaked from Chamber C to B and D, respectively.

$$\text{sgn}(x) = \begin{cases} 1, & \text{if } x \geq 0 \\ -1, & \text{if } x < 0 \end{cases} \quad (10)$$

$$s(x_v) = \begin{cases} 1, & \text{if } x_v \geq 0 \\ 0, & \text{if } x_v < 0 \end{cases} \quad (11)$$

The proportional valve can be described by:

$$x_v' = u \quad (12)$$

Note that x_v' is the valve displacement instead of valve openings because of the proportional valve's dead zone. The experiment results show that the positive dead zone value is 25.5%, and the negative dead zone value is -25%. So, the real valve opening is:

$$x_v = \begin{cases} x_v' - 2.55, & \text{if } x_v' \geq 0 \\ x_v' + 2.5, & \text{if } x_v' < 0 \end{cases} \quad (13)$$

In this paper, proportional valve displacement is standardized to $[-10, 10]$.

The friction force can be described by an improved Lurers model. The improved LuGre model introduces bristle direction coefficient and deformation coefficient of seals to describe the transient friction force in systems with one-way seals. The FVVHA friction is different from other hydraulic actuators due to its sealing designment. The improved friction model is given by:

$$F_f = \mu_d \delta (\sigma_0 z + \sigma_1 \dot{z} + \sigma_2 \dot{x}) \quad (14)$$

$$\mu_d = f(\dot{x}) = \begin{cases} \mu_{d+}, & \text{if } \dot{x} \geq 0 \\ \mu_{d-}, & \text{if } \dot{x} < 0 \end{cases} \quad (15)$$

$$\delta = f(x) = e^{\frac{x}{4x_{max}}} \quad (16)$$

where σ_0 is the bristle stiffness; σ_1 is the damping coefficient of bristle; σ_2 is the viscous damping coefficient of FVVHA; z is the deformation length of bristle; $\dot{z}$ is the deformation velocity; $\dot{x}$ is the engine valve velocity; μ_d is the bristle direction coefficient; δ is the deformation coefficient of seals; x is the engine valve lift; x_{max} is the maximum engine valve lift. In this paper, μ_{d+} is 1.04 and μ_{d-} is 0.45.

The engine valve spring force can be described by a quadratic polynomial:

$$F_s = \beta_1(x + 0.0085)^2 + \beta_2(x + 0.0085) + \beta_3 \quad (17)$$

In this paper, β_1 is 248, β_2 is 23,660 and β_3 is 26.69. Note that the pre-compression length of the engine spring is 8.5 mm.

The model parameters are shown in Table 1. A_B , A_C and A_D were calculated by actuator structure parameters. m was tested by the balance. β_e was set as the default value of the oil. σ_0 , σ_1 and σ_2 were obtained by the identification of the friction model. The rest parameters in Table 1 can be identified by the engine valve response experiments.

Table 1. Model parameters.

Parameters (Units)	Value	Parameters (Units)	Value
m (kg)	0.17	C_{dC}	1×10^{-7}
β_e (Pa)	1.7×10^9	C_{dD}	2×10^{-7}
A_B (m ²)	6.362×10^{-5}	ω	12.489
A_C (m ²)	9.0321×10^{-5}	C_{lB}	5×10^{-13}
A_D (m ²)	1.0367×10^{-4}	C_{lD}	5×10^{-13}
V_B (m ³)	1.2×10^{-5}	σ_0 (N/m)	400,000
V_C (m ³)	1.3×10^{-4}	σ_1 (N/m/s)	300
V_D (m ³)	1.8×10^{-5}	σ_2 (N/m/s)	10
C_{dB}	2×10^{-7}		

4. Controller Design

4.1. Desired Engine Valve Lifts Transformation

In the FVVHA system shown in Figure 3a, engine valve motion and proportional valve spool motion delay behind the control signal. The engine valve motion delay is bigger than the proportional valve spool motion delay, because the maximum static friction of the system is also an important factor causing the delay in addition to the proportional valve phase delay. However, the maximum static friction is difficult to model. Therefore, model-based feedback compensation is not feasible. Fortunately, for engine valve lift tracking control, the desired lift is stable if engine work conditions remain unchanged. Therefore, system delay of FVVHA can be calculated with the lift date of the last engine cycle. The engine valve motion delay can be compensated by tuning the desired valve lift, and the desired valve lift after tuning can be described by:

$$x_{1d'}(k) = x_{1d}(k + k_\tau) \quad (18)$$

$$k_\tau = k_r \tau \quad (19)$$

$$k_r = \frac{sr}{RPM \times 360/60} = \frac{sr}{6 \times RPM} \quad (20)$$

where $x_{1d'}$ is valve lift after tuning; x_{1d} is valve lift before tuning; τ is valve motion delay (crank angle); k_r is linear gain from crank angle to sampling count corresponding to engine speed and sampling rate. In this paper, sr is 50,000.

4.2. Backstepping Controller Design with Integral Flow Error Feedback

Define state variables below:

$$x = [x_1, x_2, x_3, x_4, x_5, x_6]^T = [y, \dot{y}, P_B, P_C, P_D, z]^T \quad (21)$$

The system model can be expressed by:

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = \frac{1}{m}(x_3 A_B + x_4 A_C - x_5 A_D - F_f - F_s) + d_2(t) \\ \dot{x}_3 = \frac{\beta_e}{V_B}(Q_B(x_3) + C_{IB}(x_4 - x_3) - A_B x_2) \\ \dot{x}_4 = \frac{\beta_e}{V_{OC}}(Q_C(x_4) - C_{IB}(x_4 - x_3) - C_{ID}(x_4 - x_5) - A_C x_2) + d_4(t) \\ \dot{x}_5 = \frac{\beta_e}{V_D}(Q_B(x_5) + C_{ID}(x_4 - x_5) + A_D x_2) \\ \dot{x}_6 = x_2 - \frac{|x_2|}{\alpha_0 + \alpha_1 e^{-(\frac{x_2}{0.2})^2}} x_6 \end{cases} \quad (22)$$

where $d_2(t)$ is unmatched uncertainty and $d_4(t)$ is matched uncertainty.

There is no additional velocity and acceleration sensor at the test bench; also, bristle deformation length is also unmeasurable. Besides, matched uncertainty can be estimated. So, a state observer is necessary. Extend $d_4(t)$ as a new state variable x_7 and the extend observer is given by:

$$\begin{cases} \dot{\hat{x}}_1 = \hat{x}_2 + \frac{\sigma_1}{\varepsilon}(x_1 - \hat{x}_1) \\ \dot{\hat{x}}_2 = \frac{1}{m}(x_3 A_B + x_4 A_C - x_5 A_D - \hat{F}_f - F_s) + \frac{\sigma_2}{\varepsilon^2}(x_1 - \hat{x}_1) \\ \dot{\hat{x}}_6 = x_2 - \frac{|x_2|}{\alpha_0 + \alpha_1 e^{-(\frac{x_2}{0.2})^2}} \hat{x}_6 + \frac{\sigma_6}{\varepsilon^4}(x_1 - \hat{x}_1) \\ \dot{\hat{x}}_7 = x_7 + \frac{\sigma_7}{\varepsilon^4}(x_1 - \hat{x}_1) \end{cases} \quad (23)$$

where $\sigma_1, \sigma_2, \sigma_6, \sigma_7$ are observer gains and $\frac{1}{\varepsilon}$ is a constant can be adjusted to ensure the stability of observer. References [41,42] show the detailed explanation about extend state observer (ESO).

Based on the ESO, a backstepping controller can be designed:

Step1: Define the valve lift error and valve velocity error:

$$z_1 = x_1 - x_{1d} \quad (24)$$

$$z_2 = x_2 - x_{2eq} \quad (25)$$

where x_1 is engine valve before tuning; x_{2eq} is virtual desired valve velocity.

$$x_{2eq} = \dot{x}_{1d'} - k_1 z_1 \quad (26)$$

where k_1 is positive feedback gain of engine valve lift tracking error.

The time derivative of z_2 is calculated with:

$$\dot{z}_2 = \dot{x}_2 - \dot{x}_{2eq} = \frac{1}{m}(x_3 A_B + x_4 A_C - x_5 A_D - F_f - F_s + d_2) - (\dot{x}_{1d'} - k_1 \dot{z}_1) \quad (27)$$

If x_4 is the input of (27), design the virtual control law α_2 of x_4 . Define z_3 :

$$z_3 = x_4 - \alpha_2 \quad (28)$$

α_2 can be calculated by:

$$\alpha_2 = \alpha_{2a} + \alpha_{2s1} + \alpha_{2s2} \quad (29)$$

where α_{2a} is feedforward controller based on system model and desired input; α_{2s1} is linear feedback to z_2 and α_{2s2} is used to compensate the unmatched uncertainty.

α_{2a} can be calculated by:

$$\alpha_{2a} = \frac{m(\ddot{x}_{1d'} - k_1 \dot{z}_1) - (x_3 A_B - x_5 A_D - F_f(\hat{x}_1, x_2) - F_s(x_1))}{A_C} \quad (30)$$

α_{2s1} can be calculated by:

$$\alpha_{2s1} = -k_{2s1}z_2 \quad (31)$$

where k_{2s1} is positive feedback gain;

Apply (28)–(30) to (27):

$$\dot{z}_2 = \frac{1}{m} \left(A_C z_3 - A_C k_{2s1} z_2 + A_C \alpha_{2s2} - \tilde{F}_f + d_2 \right) \quad (32)$$

To ensure the controller stability, α_{2s2} should meet

$$z_2 \left(\alpha_{2s2} - \tilde{F}_f + d_2 \right) \leq \varepsilon_2 \quad (33)$$

$$z_2 \alpha_{2s2} \leq 0 \quad (34)$$

According to [43], α_{2s2} can be calculated by:

$$\alpha_{2s2} - k_{2s2} z_2 = -\frac{h_2}{2\varepsilon_2} z_2 \quad (35)$$

$$h_2 \geq F_{fm}^2 + d_{2m}^2 \quad (36)$$

where F_{fm} is the maximum estimation error of friction force; d_{2m} is the maximum unmatched uncertainty.

Step2: The time derivative of z_3 is calculated with:

$$\begin{aligned} \dot{z}_3 = \dot{x}_4 - \dot{\alpha}_2 = & \frac{\beta_e}{V_{OC}} \left(C_d \omega x_v \left(s(x_v) \sqrt{\frac{2}{\rho} (P_h - x_4)} + s(-x_v) \sqrt{\frac{2}{\rho} (x_4 - P_l)} \right) \right. \\ & \left. - C_{IB}(x_4 - x_3) - C_{ID}(x_4 - x_5) - A_C x_2 \right) + \dot{d}_4(t) - \dot{\alpha}_2 \end{aligned} \quad (37)$$

where $\dot{\alpha}_2$ is the time derivative of α_2 :

$$\dot{\alpha}_2 = \frac{\partial \alpha_2}{\partial t} + \frac{\partial \alpha_2}{\partial x_1} \dot{x}_1 + \frac{\partial \alpha_2}{\partial x_2} \dot{x}_2 + \frac{\partial \alpha_2}{\partial x_3} \dot{x}_3 + \frac{\partial \alpha_2}{\partial x_5} \dot{x}_5 = \dot{\alpha}_{2c} + \dot{\alpha}_{2u} \quad (38)$$

$$\dot{\alpha}_{2c} = \frac{\partial \alpha_2}{\partial t} + \frac{\partial \alpha_2}{\partial x_1} \dot{x}_1 + \frac{\partial \alpha_2}{\partial x_2} \hat{x}_2 + \frac{\partial \alpha_2}{\partial x_3} \dot{x}_3 + \frac{\partial \alpha_2}{\partial x_5} \dot{x}_5 \quad (39)$$

$$\dot{\alpha}_{2u} = \frac{\partial \alpha_2}{\partial x_2} \tilde{x}_2 \quad (40)$$

where $\dot{\alpha}_{2c}$ is term can be calculated and $\dot{\alpha}_{2u}$ is term can't be calculated.

x_v is the input of (37). x_v can be calculated by:

$$x_v = x_{va} + x_{vc} + x_{vs1} + x_{vs2} \quad (41)$$

where x_{va} is feedforward controller based on system model and desired input; x_{vs1} is linear feedback to z_3 and x_{vs2} is used to compensate the estimation error of matched uncertainty; x_{vc} is the linear feedback of integral flow error e_f to compensate the amplitude delay of proportional valve.

x_{va} can be calculated by:

$$x_{va} = \frac{\frac{V_C}{\beta_e} (\dot{\alpha}_{2c} - \hat{x}_7) + C_{IB}(x_4 - x_3) + C_{ID}(x_4 - x_5) + A_C \hat{x}_2}{C_d C \omega \left(s(x_v) \sqrt{\frac{2}{\rho} (P_h - x_4)} + s(-x_v) \sqrt{\frac{2}{\rho} (x_4 - P_l)} \right)} \quad (42)$$

x_{vs1} can be calculated by:

$$x_{vs1} = -k_{vs1} z_3 \quad (43)$$

where k_{vs1} is positive feedback gain;

x_{vc} can be calculated by:

$$x_{vc} = -k_{vc}e_f \quad (44)$$

where k_{vc} is positive feedback gain and e_f can be calculated by:

$$e_f = \int C_{dC}\omega z_4 \left(\sqrt{\frac{2}{\rho}(P_h - \hat{x}_4)} + \sqrt{\frac{2}{\rho}(\hat{x}_4 - P_l)} \right) dt \quad (45)$$

z_4 is the error of desired proportional valve opening and actual valve opening.

$$z_4 = x_v - x_{vd} \quad (46)$$

where x_v is actual opening and x_{vd} is desired valve opening. x_{vd} can be calculated by (12) and (13). Note that when the engine cycle is at its end, e_f will be reset to 0, so that the steady estimation error of e_f can be removed.

Apply (38)–(46) to (37):

$$\dot{z}_3 = \dot{\alpha}_{2u} + \tilde{x}_7 - \dot{\alpha}_2 + \frac{\beta_e}{V_{OC}} \left(-A_C \tilde{x}_2 - C_{dC} k_{vc} e_f(z_4) - C_{dC} k_{vs1} z_3 + C_{dC} x_{vs2} \right) \quad (47)$$

To ensure the controller stability, α_{2s2} should meet

$$z_3 \left(-C_{dC} k_{vc} e_f(z_4) + C_{dC} x_{vs2} - \dot{\alpha}_{2u} + \tilde{x}_8 - \frac{\beta_e}{V_{OC}} A_C \tilde{x}_2 \right) \leq \varepsilon_3 \quad (48)$$

$$z_3 x_{vs2} \leq 0 \quad (49)$$

x_{vs2} can be calculated by:

$$x_{vs2} - k_{vs2} z_3 = -\frac{h_3}{2\varepsilon_3} z_3 \quad (50)$$

$$h_3 \geq (\dot{\alpha}_{2u})^2 + (\tilde{x}_{7m})^2 + \frac{\beta_e^2}{V_{OC}^2} A_C^2 \tilde{x}_{2m}^2 \quad (51)$$

where x_{2m} and x_{7m} are the maximum estimation error of x_2 and x_7 .

Step3: From step2, the proportional valve opening can be calculated. To calculate the control signal u , the dead zone of proportional valve has to be considered. Reference [44] shows a smooth dead zone inverse, which can circumvent chattering phenomena of control signal at zero.

The smooth dead zone inverse is given by:

$$u(x_v) = \frac{x_v + m_r b_r}{m_r} \Phi_r(x_v) + \frac{x_v + m_l b_l}{m_l} \Phi_l(x_v) \quad (52)$$

$$\Phi_r(x_v) = \frac{e^{x_v/\epsilon}}{e^{x_v/\epsilon} + e^{-x_v/\epsilon}} \quad (53)$$

$$\Phi_l(x_v) = \frac{e^{-x_v/\epsilon}}{e^{x_v/\epsilon} + e^{-x_v/\epsilon}} \quad (54)$$

The proportional valve dead zone has been obtained from experiment and the parameters of dead zone inverse are shown in Table 2.

Table 2. Parameters of dead-zone inverse.

Parameters (Units)	Value	Parameters (Units)	Value
m_r	1	m_l	1
b_r	2.55	b_l	−2.5
ϵ	0.2		

4.3. Stability Analysis

Define a positive semi-definite V_2 function as

$$V_2 = \frac{1}{2}k_1^2 z_1^2 + \frac{1}{2}m z_2^2 \quad (55)$$

where k_1 is a positive gain. Its time derivative can be calculated by:

$$\dot{V}_2 = k_1^2 z_1 \dot{z}_1 + m z_2 \dot{z}_2 = -k_1^3 z_1^2 - A_C k_{2s1} z_2^2 + k_1^2 z_1 z_2 + A_C z_2 z_3 + z_2 (\alpha_{2s2} - \tilde{F}_f) \quad (56)$$

Apply (33) to (56):

$$\dot{V}_2 \leq -k_1^3 z_1^2 - A_C k_{2s1} z_2^2 + k_1^2 z_1 z_2 + A_C z_2 z_3 + \varepsilon_2 \quad (57)$$

Define a positive semi-definite V_3 function as

$$V_3 = V_2 + \frac{1}{2}z_3^2 \quad (58)$$

The time derivative of V_3 can be calculated by:

$$\begin{aligned} \dot{V}_3 &\leq -k_1^3 z_1^2 - A_C k_{2s1} z_2^2 + k_1^2 z_1 z_2 + A_C z_2 z_3 + \varepsilon_2 \\ &+ z_3 (\alpha_{2u} + \tilde{x}_7 - \dot{\alpha}_2 + \frac{\beta_e}{V_{OC}} (-A_C \tilde{x}_2 - C_{dC} k_{vc} e_f(z_4) - C_{dC} k_{vs1} z_3 \\ &+ C_{dC} x_{vs2})) \end{aligned} \quad (59)$$

Apply (48) to (59):

$$\dot{V}_3 \leq -k_1^3 z_1^2 - A_C k_{2s1} z_2^2 + k_1^2 z_1 z_2 + A_C z_2 z_3 - \frac{\beta_e}{V_{OC}} k_{vs1} z_3^2 + \varepsilon_2 + \varepsilon_3 \quad (60)$$

Define controller gain matrix:

$$\Lambda_3 = \begin{bmatrix} k_1^3 & -\frac{1}{2}k_1^2 & 0 \\ -\frac{1}{2}k_1^2 & A_C k_{2s1} & -\frac{1}{2}A_C \\ 0 & -\frac{1}{2}A_C & \frac{\beta_e}{V_{OC}} k_{vs1} \end{bmatrix} \quad (61)$$

Apply (61) to (60):

$$\dot{V}_3 \leq z^T \Lambda_3 z + \varepsilon \leq \varepsilon - \mu V_3 \quad (62)$$

$$V_3 \leq e^{-\mu t} V_3(0) + \frac{\varepsilon}{\mu} [1 - e^{-\mu t}] \quad (63)$$

$$\mu = 2\lambda_{\min}(\Lambda_3) \left\{ \frac{1}{k_1^2}, \frac{1}{m}, 1 \right\} \quad (64)$$

Thus, all controller parameters are bound.

5. Experiment Results and Discussion

To verify the tracking performance of engine valve lifts, some comparative experiments have been carried out on the prototype shown in Figure 3. The parameters of controller, which includes x_{vc} in (41) can be found in Table 3. The parameters of controller which do not include x_{vc} in (41) can be found in Table 4.

Table 3. Controller parameters (with integrating flow term).

Parameters (Units)	Value	Parameters (Units)	Value
k_1	100	k_{vs1}	10
k_{2s1}	50	k_{vs2}	10
k_{2s2}	50	k_{vc}	1

Table 4. Controller parameters (without integrating flow term).

Parameters (Units)	Value	Parameters (Units)	Value
k_1	300	k_{vs1}	20
k_{2s1}	100	k_{vs2}	20
k_{2s2}	100		

Case1: Steady experiment: engine speed is 1200 r/min.

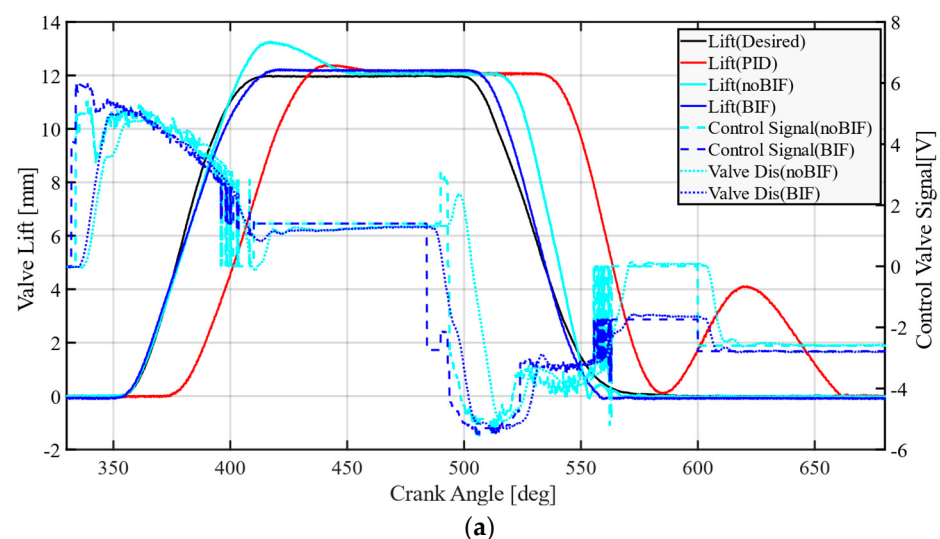
Figure 4 shows the comparison of backstepping controller with integral flow term x_{vc} (BIF) in (41), backstepping controller without x_{vc} (noBIF) and PID controller (PID). $K_P = 500$, $K_I = 0.5$, and $K_D = 0.1$. In figures below, “Lift” represents engine valve lift; “Control Signal” represents proportional valve’s control signal and “Valve Dis” represents proportional valve’s spool displacement.

Figure 4 shows the comparative tracking experiment when maximum valve lift is 12 mm. The tracking error of PID controller is bigger than BIF and noBIF. The PID controller cannot suppress the system uncertainties, which make inconformity obvious in each engine cycle, as shown in Figure 4b.

For BIF and noBIF, the error of BIF is smaller than noBIF as shown in Figure 4c. noBIF controller causes a bigger overshoot at maximum valve lift. The main reason is that feedback gain of noBIF is bigger than BIF, as shown in Tables 3 and 4, which is driven by transient error. The integral flow term in BIF is driven by integral error, which can reduce feedback parameters except for k_{vc} .

Figure 5 shows comparative tracking experiments when maximum valve lift is 10 mm, 8 mm and 5 mm, respectively. For a PID controller, the valve time delay cannot be compensated because the controller is driven by tracking errors. Therefore, the valve motion with PID controllers has significant delays compared to desired valve lifts. Unlike PID controllers, delays of noBIF and BIF controllers are smaller, which verify the effectiveness of engine valve lifts transformation in Section 4.1. Moreover, the experiments show that the valve dynamics cannot be ignored in high-frequency position tracking.

Figures 4 and 5 verify the effectiveness of a BIF controller. Compared with noBIF and PID controllers, the tracking errors of a BIF controller are the smallest, especially when the valve reaches its maximum lift and the valve is seated, which helps to reduce the impact of valve seating. As shown in Figure 4c, the maximum valve lift with a BIF controller is 1 mm, while the maximum valve lift error with noBIF controller is greater than 3 mm.

**Figure 4.** Cont.

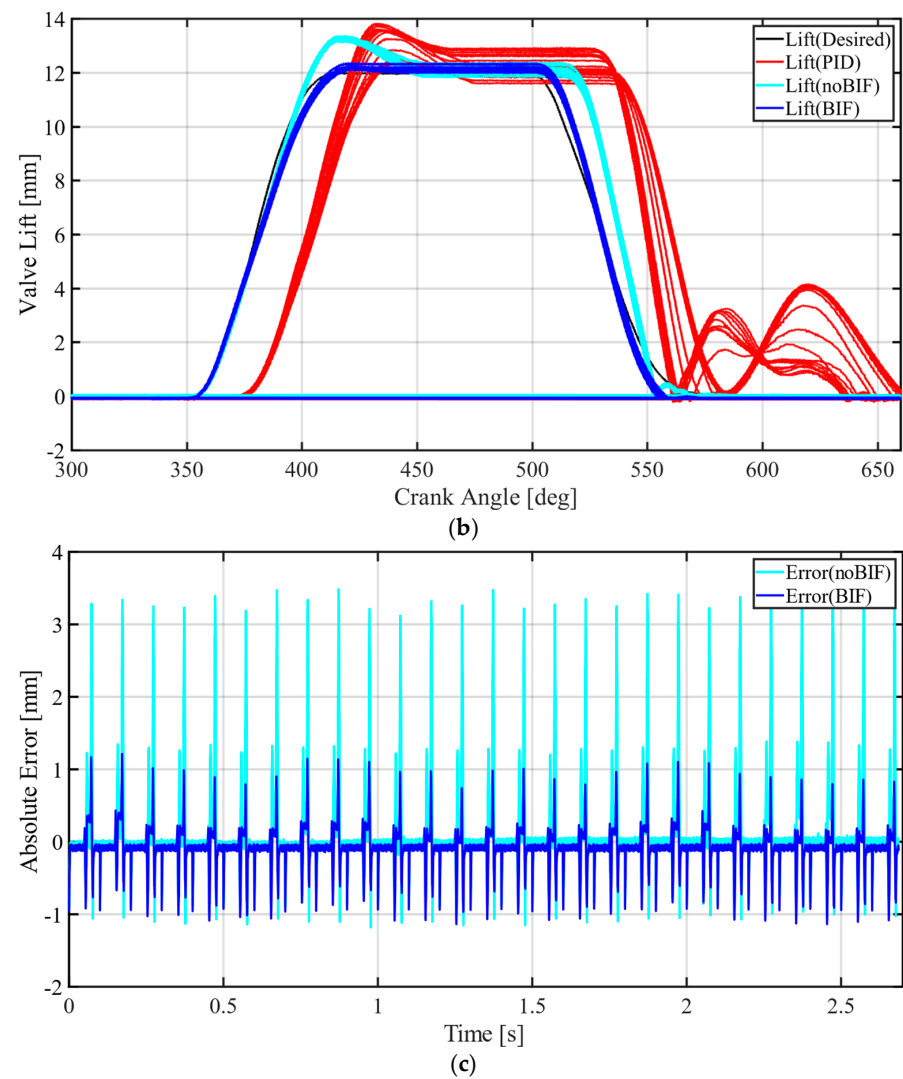


Figure 4. Engine valve lift tracking control comparison (12 mm). (a) Single-cycle tracking experiments; (b) Multi-cycle tracking experiments; (c) Tracking errors.

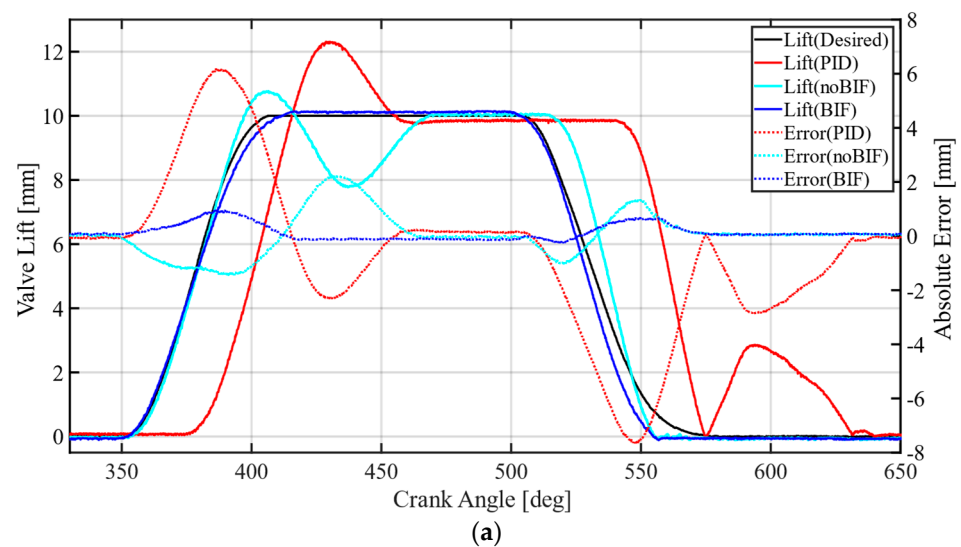


Figure 5. Cont.

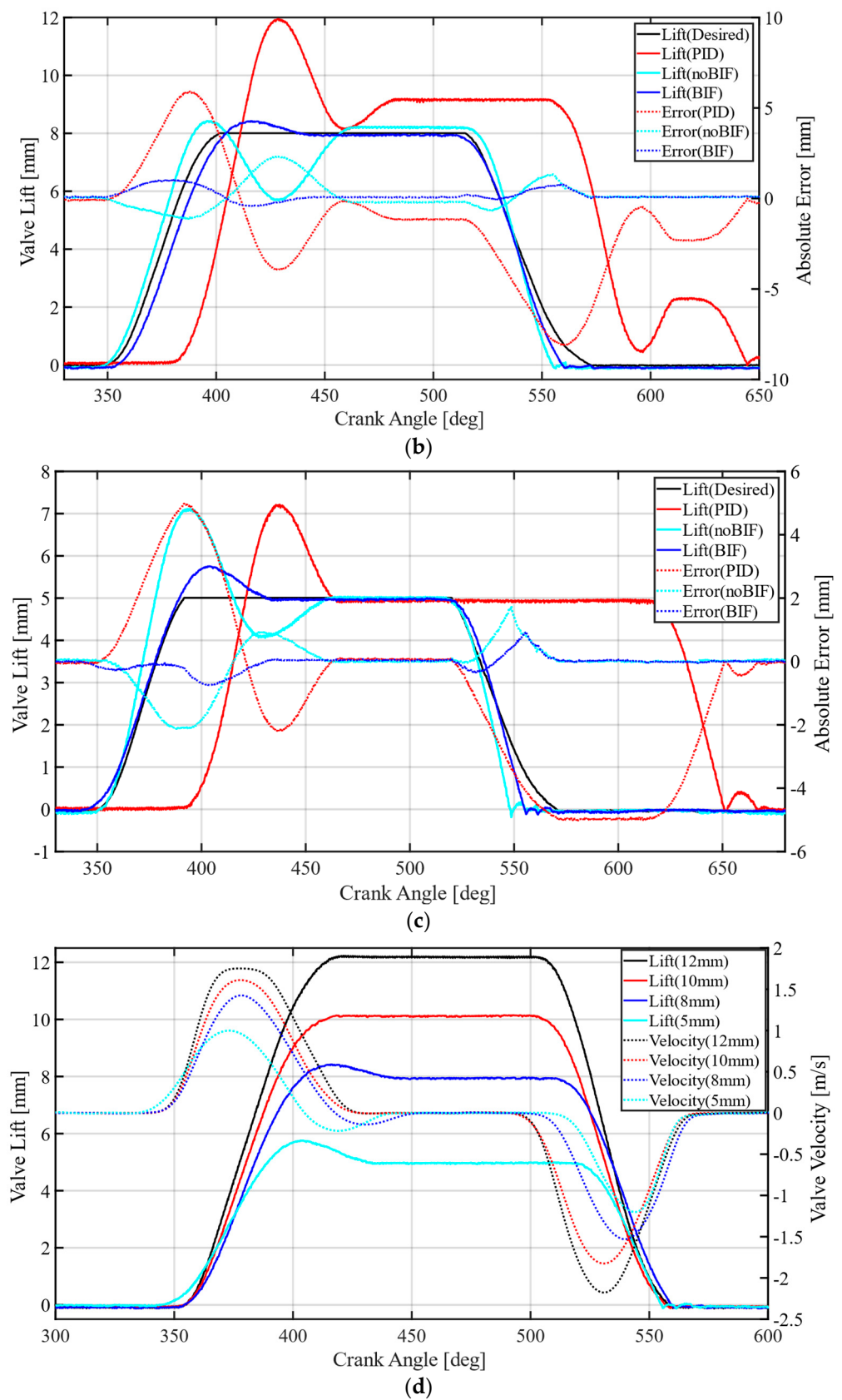


Figure 5. Engine valve lift tracking control comparison (1200 r/min). (a) The maximum valve lift is 10 mm; (b) The maximum valve lift is 8 mm; (c) The maximum valve lift is 5 mm; (d) The engine valve velocities.

Case2: Steady experiment: engine speed is 120 r/min.

Figures 4 and 5 show that the noBIF controller and PID controller cannot meet control performance demands. In case2, 3 and 4, the BIF controller is implemented to track desired valve lifts.

Figure 6 shows engine valve lift tracking experiments with integral flow compensation when engine speed is 120 r/min. Similar to case1, the smaller the maximum valve lift, the greater the overshoot at the maximum lift. Note that the desired valve lifts in 120 r/min are different with lifts in 1200 r/min. It can be seen from Figures 5 and 6 that the overshoots are unavoidable with the control parameters in Table 4 when maximum valve lift is less than 12 mm, no matter what the engine speed is.

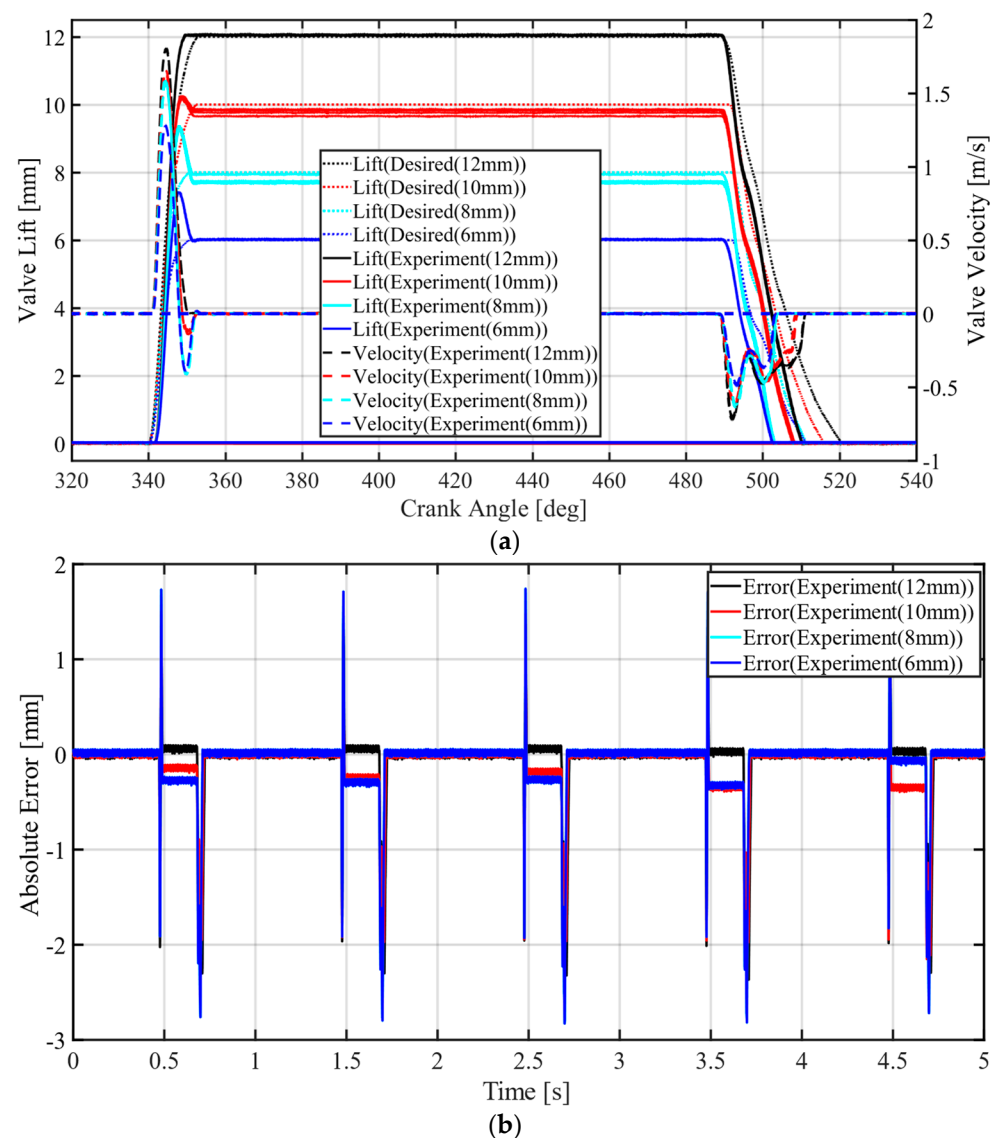


Figure 6. Engine valve lift tracking control comparison (120 r/min). (a) Valve lift tracking experiments; (b) Tracking errors.

With a BIF controller, the maximum valve lift errors are less than 3 mm, which is bigger than errors when engine speed is 1200 r/min. One of the most important reasons is that the engine valve velocities are too small. As shown in Figure 6a, the maximum valve velocity is less than 1 m/s when the engine valve is closing. However, the maximum valve velocity is about 2 m/s. To make the air valve close at a slower speed, the opening of the proportional valve must be also be small. Unfortunately, the flow rate coefficient of the proportional valve when its opening is small is strongly nonlinear, which will cause a big

error of estimated flow rate through the valve. Therefore, there are big engine valve lift tracking errors in Figure 6. For an engine event, the desired valve lift should be optimized to bring the engine valve velocity within a reasonable range.

Case3: Transient experiment: with different maximum engine valve lift.

Figure 7 shows the transient tracking control performance of an engine valve based on the BIF controller with different maximum valve lift when engine speed is 1200 r/min. The tracking errors of BIF are slightly larger than steady conditions in Figure 5. BIF tracking errors in transient conditions are smaller than noBIF and PID controllers in steady conditions. As shown in Figure 7, although the tracking errors are greater than the errors in a steady state condition, the errors keep getting smaller as time goes on, which verifies stability of the system.

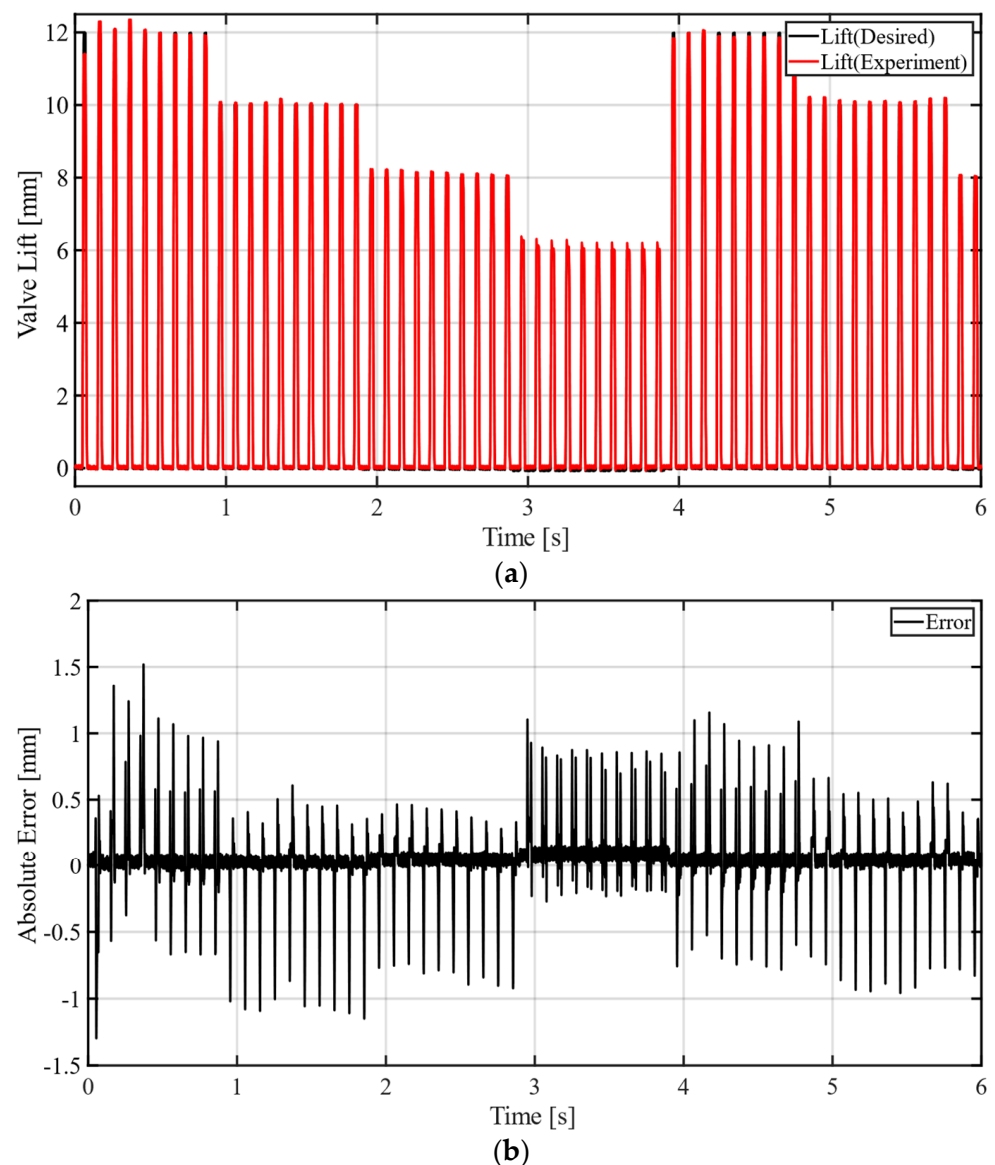


Figure 7. Transient maximum engine valve lift tracking experiment. (a) Valve lift tracking experiments; (b) Tracking errors.

Case4: Transient experiment: with different engine speed.

Figure 8 shows the transient tracking control performance when engine speed and maximum valve lift are variable. Compared with case3, the tracking error is bigger, espe-

cially when engine speed is 120 r/min. The maximum tracking error is less than 2 mm when engine speed is 1200 r/min, which is 3 mm when the engine speed is 120 r/min.

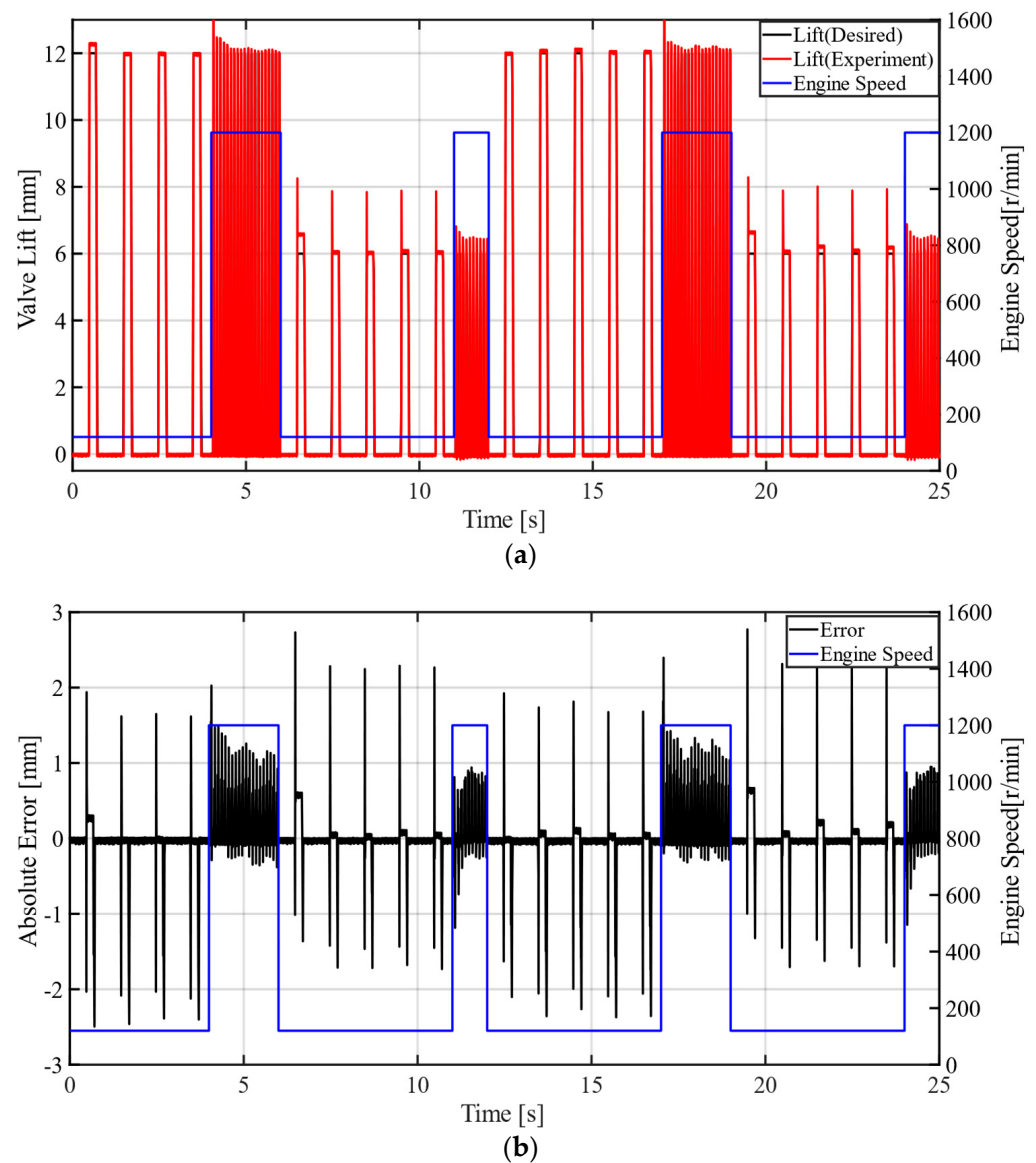


Figure 8. Transient engine speed tracking experiment. (a) Valve lift tracking experiments; (b) Tracking errors.

As shown in Figure 8b, when the engine speed is changing, there will be a large valve lift error, and as time goes on, the valve lift errors will be steady. The most important reason for this is that the opening of the relief valve with low-pressure (the right one shown in Figure 3a) is different at varying engine speeds. The pressure of the relief valve is smaller when the engine speed is 120 r/min, which will cause a bigger overshoot when engine speed changes from 120 r/min to 1200 r/min.

6. Conclusions

In this paper, a tracking controller with proportional valve dynamic compensation is proposed for a high-frequency fully variable valve hydraulic actuator (FVVHA). The valve phase delay can be compensated by feedforward transformation of desired engine valve lifts. The valve amplitude delay can be compensated by integral flow feedback, which is related to the piston in a position servo system. The compensation strategy is implemented in a backstepping controller of an FVVHA. Comparative experiments show

that the tracking errors of PID controller are bigger than BIF and no BIF. The PID controller cannot suppress system uncertainty, which causes great cycle inconsistency. Comparing BIF and noBIF controller, tracking errors with the BIF controller are smaller, which verifies the effectiveness of integral flow compensation. With the BIF controller, the maximum valve lift errors at 1200 r/min are less than 1 mm, while the lift error with noBIF controller is greater than 3 mm. The comparative test shows that in the high-frequency servo control system, the dynamic of the proportional valve cannot be ignored. The engine valve lift tracking experiments verify the effectiveness of the compensation strategy proposed in this paper. The compensation strategy in this paper provides a new method to apply the proportional valve to high frequency systems without increasing the system order.

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Abbreviations

m	mass of all movement parts
a	acceleration of valve
P_B	pressures of Chamber B
P_C	pressures of Chamber C
P_D	pressures of Chamber D
A_B	effective action areas of Chamber B
A_C	effective action areas of Chamber C
A_D	effective action areas of Chamber D
F_f	friction force of the system
F_s	spring force of the engine valve
Q_B	flow rate into Chamber B
Q_C	flow rate into Chamber C
Q_{CB}	flow rate leaked from Chamber C to Chamber B
Q_{CD}	flow rate leaked from Chamber C to Chamber D
Q_D	flow rate into Chamber D
β_e	elasticity modulus of oil
v	engine valve velocity
C_{dB}	equivalent flow coefficient of oil port in Chamber B
A_{vB}	effective flow areas of oil port in Chamber B
C_{dC}	equivalent flow coefficient of proportional valve
ω	area gain coefficient of proportional valve
x_v	real valve openings of proportional valve
x_v'	proportional valve displacement
P_h	oil supply pressure
C_{dD}	equivalent flow coefficient of oil port in Chamber D
A_{vD}	effective flow areas of oil port in Chamber D

C_{IB}	leakage coefficient leaked from Chamber C to B
C_{ID}	leakage coefficient leaked from Chamber C to D
μ_d	bristle direction coefficient
δ	deformation coefficient of seals
σ_0	bristle stiffness
σ_1	damping coefficient of bristle
σ_2	viscous damping coefficient
z	bristle displacement
$\dot{z}$	bristle velocity
$\dot{x}$	engine valve velocity
x_{1d}	desired valve lift before tuning
$x_{1d'}$	desired valve lift after tuning
z_1	valve lift error
z_2	valve velocity error
z_3	error of α_2 and P_C
x_{2eq}	virtual desired valve velocity
α_2	virtual control law of P_C
α_{2a}	feedforward controller in step 1
α_{2s1}	linear feedback to z_2
α_{2s2}	unmatched uncertainty compensation term in step 1
x_{va}	feedforward controller in step 2
x_{vs1}	linear feedback to z_3
x_{vs2}	unmatched uncertainty compensation term in step 2
x_{vc}	linear feedback of e_f
e_f	integral flow error

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