Dynamics Modelling and Control of a Novel Fuel Metering Valve Actuated by Two Binary-Coded Digital Valve Arrays

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Abstract: A fuel metering valve actuated by two binary-coded digital valve arrays (BDVAs) is proposed to improve the reliability of conventional fuel metering valves piloted by a servo valve. The design concept of this configuration is obtained from the structural characteristics of the dual nozzle-flapper and the flow regulation method of the digital hydraulic technology. The structure and working principle of the fuel metering valve are presented. Then, a mathematical model of the entire valve is developed for dynamic analysis. Subsequently, the mechanism of the transient flow uncertainty of the BDVA is revealed through simulation to determine the fluctuation in the velocity of the fuel metering valve. Furthermore, step response indicates that the delay time of the fuel metering valve is within 4.1 ms. Finally, to improve the position tracking accuracy of the fuel metering valve, a velocity feedforward proportional-integral controller with pulse code modulation is proposed. A series of comparative analyses indicate that compared with those of the velocity feedforward controller, the average and standard deviation of the position error for the proposed controller are reduced by 78 and 72.7%, respectively. The results prove the feasibility of the proposed valve and the effectiveness of the proposed control strategy.

Keywords: fuel metering valve; binary-coded digital valve array; dynamic characteristics; velocity feedforward proportional-integral controller; pulse code modulation

1. Introduction

Full authority digital engine control (FADEC) has become a conventional control technology for aero-engines [1–3]. The fuel metering system controlled by the electronic control unit in FADEC regulates the fuel flow rate into the engine. The fuel metering system comprises a fuel pump, an electro-hydraulic servo valve, a fuel metering valve, and a pressure differential valve [4]. The electro-hydraulic servo valve used as the pilot stage controls the position of the fuel metering valve. The stability and reliability of the electro-hydraulic servo valve directly determine the safety of the fuel metering system as it is the most vulnerable component. An anomaly in the fuel metering valve may lead to flight safety hazards such as flight shut-down or loss of thrust control [5]. Therefore, it is crucial to improve the reliability of the fuel metering valve, particularly in the pilot stage.

There are two stages for the fuel metering valve of an aero-engine. A dual nozzle-flapper actuated by an electromagnetic torque motor [6] or an electric-hydraulic servo valve [7] are typically used in the pilot stage because of their high precision and high response speed [8]. The main stage represents a metering spool in which the position of the spool is measured using a linear variable differential transformer. The electric-hydraulic servo valve is severely affected because aero-engines operate under conditions such as long-span temperature variation, strong vibration, and high centrifugal acceleration. Several scholars have conducted meaningful research in this field. For example, Zhang...
et al. [9] studied the effects of temperature on the flow force of a servo valve. The results indicated that the flow force initially increases and then decreases with an increase in the temperature under a fixed opening. Yan et al. [10,11] proposed a comprehensive temperature affecting model to describe the relationship between the temperature variation and the performance of the torque motor. The research results indicated that the control error of the servo valve was increased by 15% when the temperature increased from 20 to 270 °C. In addition, under a random vibration environment, servo valves are prone to fatigue damage and a null shift. To calculate the reliability sensitivities considering independent random parameters, a Monte Carlo method was used [12]. Therefore, while facing the extreme working environment of aero-engines, the performance and reliability of the servo valve cannot fulfill requirements such as strong sensitivity to oil contamination caused by micron-level fitting tolerances and strong temperature sensitivity of the structural parameters [13,14].

Compared with that of electro-hydraulic servo valves based on continuous flow regulation, digital valves based on the discrete flow regulation have two states, namely on and off, which have the advantages of high reliability and efficiency, and low cost [15]. Selecting digital valves as the pilot stage of the spool valve has become a research hotspot in the field of digital hydraulic technology. Wang et al. [16] proposed a three-way main stage valve that was piloted by two two-position and three-way high speed digital valves. The results indicated that the position control accuracy of the main stage was enhanced with a high switching frequency of the digital valves. Xiong and Huang [17] proposed a two-stage proportional flow control valve in which several switching valves were used as the pilot stage. In this study, the pilot switching valves output a discrete flow because they utilize a pulse width modulation (PWM) signal, which can result in the fluctuation of the displacement and outlet flow rate of the valve. In reference [18], to overcome the high complexity and internal leakage of the pilot stage in conventional servo valves, two small two-position, three-way digital valves actuated by piezoelectric ring benders were used as the pilot stage of the servo valve. Gao et al. [19] proposed a two-stage proportional valve considering conventional dual nozzle-flapper servo valves in which two small two-position, three-way high speed on/off valves were used to replace the nozzle-flapper structure. However, owing to the hysteresis of the electromagnetic component, the switching frequency of the digital valves only reached 100 Hz, which resulted in a low position tracking accuracy of the main valve.

According to the above research, selecting digital valves as the pilot stage of the proportional/servo valve can improve the reliability to a certain extent. The service life of the digital valve can be further improved by incorporating intelligent fault diagnosis methods [20]. However, the utilization of the PWM signal limits the position control accuracy of the main valve core owing to the switching frequency. Moreover, the pilot digital valves output a high-frequency discrete flow, resulting in the displacement fluctuation of the main valve.

Therefore, if the digital valve is controlled using a PWM control signal, the displacement of the fuel metering valve fluctuates, which may severely affect the performance and safety of the aero-engine. Compared with that of a digital valve system using the PWM method, the digital valve array system controlled using pulse number modulation signals or pulse code modulation (PCM) signals can improve the reliability because the system exhibits strong redundancy and does not require high frequency switching [21,22]. An equal coded digital valve array (EDVA) system consisting of several parallel digital valves with the same structure only outputs the same number of discrete flows as that of the digital valves. Therefore, the accuracy of the EDVA is the worst, and should be combined with the PWM signal to improve the flow control accuracy [23,24]. Conversely, the binary-coded digital valve array (BDVA) system can output a discrete flow in an exponential number, wherein the flow rate of each digital valve is expressed in binary code. Therefore, the accuracy of BDVA is higher than that of EDVA. Moreover, the control accuracy of BDVA exponentially increases with an increase in the number of digital valves. Sun et al. [25]
successfully applied BDVA technology to the aircraft antiskid brake system to improve the control accuracy of the antiskid brake pressure. Matti et al. [26] performed multiple studies on parallel digital valve technology, and proposed a digital flow control unit and digital hydraulic power management system. A binary-coded parallel digital valve technology is typically used in a digital hydraulic system to control the displacement or force of the hydraulic actuator [27,28]. Lantela and Pietola [29] proposed a high flow rate miniature digital valve system based on BDVA technology, which has been experimentally proved to be a substitute for conventional proportional valves.

According to the above analysis, BDVA technology has been applied in engineering machinery, industries, and aerospace fields owing to its advantages of high reliability and efficiency. To improve the reliability of conventional fuel metering valves, this study proposes a fuel metering valve actuated by two BDVAs from the viewpoint of the dual nozzle-flapper structure. The structure and working principle of the valve and mathematical model of the entire valve system are presented, including the flow model of the pilot BDVA and the dynamic model of the fuel metering valve. Based on the simulation model, the transient flow uncertainty and step response of the fuel metering valve are analyzed. To improve the position tracking accuracy of the fuel metering valve, a velocity feedforward proportional-integral control strategy with PCM coding is proposed. Finally, comparative research results prove the feasibility of the principle and the effectiveness of the position control strategy of the proposed valve.

The remainder of this study is organized as follows. The structure and working principle of the proposed fuel metering valve are presented in Section 2. The mathematical model of the fuel metering valve is established in Section 3. Section 4 analyses the dynamic characteristics of the fuel metering valve. A position controller is designed and verified in Section 5. Finally, Section 6 concludes the research.

2. Structure and Working Principle

A dual nozzle-flapper servo valve is used as the pilot stage of conventional fuel metering valves. However, the disadvantages of the servo valve, such as low efficiency and reliability due to their sensitivity to oil containments, lead to erroneous operation. To solve this problem, a fuel metering valve actuated by two BDVAs is presented in this study, as shown in Figure 1. Each BDVA is composed of five parallelly connected digital valves in which the flow of each valve is expressed in binary code.

![Figure 1. Schematic diagram and structure of the fuel metering valve: (a) Schematic diagram; (b) Structure.](image-url)
hydraulic force of chambers A and B; (2) when only BDVA-A is open, the pressure in chamber A decreases and the resultant force of the metering valve is to the left, which causes the metering valve to move to the left; (3) when only BDVA-B is open, the pressure in chamber B decreases and the resultant force of the metering valve is to the right, which causes the valve to move to the right.

In the closed-loop mode, an LDVT is used to detect the position of the fuel metering valve in real time. The position controller outputs the digital signal to control the opening combinations of BDVA-A and BDVA-B according to the amplitude and symbol of the position error.

3. Mathematical Model of the Fuel Metering Valve

3.1. Binary-Coded Digital Valve Array

(1) Static flow model of the BDVA

The schematic and normalized flow of the BDVA are shown in Figure 2. The flow rates of the five parallel digital valves were $1q$, $2q$, $4q$, $8q$, and $16q$ ($q$ denotes the flow rate of the smallest digital valve). The control accuracy of the metering valve is dependent on the static flow characteristic and dynamic performance of the BDVA. The static flow characteristics of the BDVA describe the relationships between the input combinations and output flow rate.

![Figure 2](image)

The opening combination of the BDVA is defined using a $2^N \times N$ binary matrix $u_{\text{int}}$

$$u_{\text{int}} = \begin{bmatrix}
0 & 0 & 0 & 0 & 0 \\
1 & 0 & 0 & 0 & 0 \\
. & . & . & . & . \\
. & . & . & . & . \\
. & . & . & . & . \\
0 & 1 & 1 & 1 & 1 \\
1 & 1 & 1 & 1 & 1
\end{bmatrix} \rightarrow \begin{bmatrix}
0 \\
1 \\
. \\
. \\
. \\
2^N - 2 \\
2^N - 1
\end{bmatrix}$$  \hspace{1cm} (1)

where “0” and “1” denote the open and closed states of the digital valves, respectively. $N$ denotes the number of the digital valves in one BDVA.

It can be observed from Equation (1) that the output flow combination number of the BDVA is $2^N$. Assuming that the flow coefficient of each valve is the same, the static flow rate of the BDVA is shown in Figure 2b. Since binary coding represents a step speed regulation method, the flow accuracy is dependent on the flow rate of the smallest valve. In addition, the number of output flow combinations exponentially increases with an increase in the number of digital valves. Therefore, the BDVAs have two advantages: high flow resolution and strong scalability.

(2) Dynamic model of the BDVA

The dynamic performance of the BDVA is mainly determined by the opening and closing times of the digital valve. The relationship between the command signal and
displacement of the valve is used to assess the dynamic performance of the digital valves shown in Figure 3. “\( t_{od} \)”, “\( t_{om} \)”, “\( t_{cd} \)”, and “\( t_{cm} \)” denote the opening delay time, opening movement time, closing delay time, and closing movement time, respectively. “\( t_{ton} \)” and “\( t_{toff} \)” denote the total opening time and total closing time, respectively. “\( T \)” and “\( t_{off} \)” denote the period time and closing command time, respectively. “\( x_{d_{max}} \)” denotes the displacement of the digital valve.

Figure 3. Dynamic characteristics of the digital valve.

Figure 3 indicates that the displacement lags behind the command signal owing to the effect of the coil inductance and mechanical hysteresis of the digital valve actuated by the solenoid [30,31]. Moreover, to facilitate the modelling of the dynamic characteristics of the BDVA, a similar mode is used from reference [19], in which a “transport delay module” and a “rate limiter module” were used to simulate the delay time (\( t_{od} \) and \( t_{cd} \)) and movement time (\( t_{om} \) and \( t_{cm} \)), respectively. The dynamic times are listed in Table 1.

Table 1. Dynamic time of the digital valves.

<table>
<thead>
<tr>
<th>Dynamic Time</th>
<th>Valve 1</th>
<th>Valve 2</th>
<th>Valve 3</th>
<th>Valve 4</th>
<th>Valve 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Opening delay time</td>
<td>( t_{od1} )</td>
<td>( t_{od2} )</td>
<td>( t_{od3} )</td>
<td>( t_{od4} )</td>
<td>( t_{od5} )</td>
</tr>
<tr>
<td>Opening movement time</td>
<td>( t_{om1} )</td>
<td>( t_{om2} )</td>
<td>( t_{om3} )</td>
<td>( t_{om4} )</td>
<td>( t_{om5} )</td>
</tr>
<tr>
<td>Closing delay time</td>
<td>( t_{cd1} )</td>
<td>( t_{cd2} )</td>
<td>( t_{cd3} )</td>
<td>( t_{cd4} )</td>
<td>( t_{cd5} )</td>
</tr>
<tr>
<td>Closing movement time</td>
<td>( t_{cm1} )</td>
<td>( t_{cm2} )</td>
<td>( t_{cm3} )</td>
<td>( t_{cm4} )</td>
<td>( t_{cm5} )</td>
</tr>
</tbody>
</table>

3.2. Flow Model of the Pilot Stage

In the pilot stage, two orifices were used to produce the control pressure in the two chambers (chambers A and B) of the fuel metering valve. The flow rates of the two orifices are defined as

\[
Q_{o1} = C_o \frac{\pi d_o^2}{4} \sqrt{\frac{2(p_s - p_a)}{\rho}}
\]

\[
Q_{o2} = C_o \frac{\pi d_o^2}{4} \sqrt{\frac{2(p_s - p_b)}{\rho}}
\]

where \( Q_{o1} \) and \( Q_{o2} \) are the flow rates of orifices 1 and 2, respectively. \( P_s \), \( p_a \), and \( p_b \) are the supply pressure, and the pressures in chamber A and B, respectively. \( C_o \) and \( d_o \) denote the flow coefficient and the diameter of the orifice, respectively. \( P \) is the oil density.

Since BDVA-A and BDVA-B have the same structure, the output flow rates of BDVA-A and BDVA-B are

\[
Q_{ba} = u_a C_d A_{d_{max}} \sqrt{\frac{2(p_a - p_t)}{\rho} u_a} \in [0, 2^N - 1]
\]
\[ Q_{bb} = u_b C_d A_{d\text{max}} \sqrt{\frac{2(p_b - p_t)}{\rho}} \quad u_b \in [0, 2^N - 1] \] (5)

where \( Q_{ba} \) and \( Q_{bb} \) are the output flow rates of BDVA-A and BDVA-B, respectively. \( p_t \) is the tank pressure, and \( C_d \) is the flow coefficient of the digital valve. \( U_a \) and \( u_b \) are the opening combinations (integer) of BDVA-A and BDVA-B, respectively. \( A_{d\text{max}} \) is the maximum opening area of the smallest valve and \( N \) is the number of valves in one BDVA.

The flow rates of chambers A and B of the fuel metering valve are defined as

\[ Q_a = Q_{ot1} - Q_{ba} \] (6)
\[ Q_b = Q_{ot2} - Q_{bb} \] (7)

Subsequently, the flow continuities of chamber A and B are given by

\[ \frac{dp_a}{dt} = \frac{\beta_e}{V_f + A_f x_f} (Q_a - A_f \frac{dx_f}{dt}) \] (8)
\[ \frac{dp_b}{dt} = \frac{\beta_e}{V_f - A_f x_f} (A_f \frac{dx_f}{dt} - Q_b) \] (9)

where \( V_f \) is the initial volume of chambers A and B, respectively. \( A_f \) is the cross-sectional area of the fuel metering valve, \( \beta_e \) is the fluid bulk modulus, and \( x_f \) is the displacement of the fuel metering valve.

3.3. Dynamic Model of the Fuel Metering Valve

The dynamic model of the fuel metering valve is defined as

\[ m_f \ddot{x}_f = (p_a - p_b) A_f - B_f \dot{x}_f - F_c - F_s \] (10)

where \( m_f \) denotes the mass of the fuel metering valve, \( B_f \) denotes the viscous friction coefficient, and \( F_c \) and \( F_s \) denote the Coulomb friction force and the steady flow force of the fuel metering valve, respectively.

The steady flow force \( F_s \) is defined as

\[ F_s = 2 C_v C_d A_f \Delta p_i \cos \theta \] (11)

where \( A_f \) denotes the opening area of the fuel metering valve, \( \Delta p_i \) denotes the pressure difference across the variable orifices, \( C_v \) and \( C_d \) denote the fluid velocity coefficient and flow coefficient, and \( \theta \) denotes the fluid jet angle.

3.4. Simulation Model and Parameters

Before analyzing the dynamic characteristics of the fuel metering valve, a simulation model of the entire valve system was developed using MATLAB/Simulink according to Equations (1)–(11), as shown in Figure 4.
Figure 4. Simulation model of the fuel metering valve system.

The main parameters are given in Table 2.

Table 2. Main parameters of the entire valve system.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Variables</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Opening delay time</td>
<td>$t_{od}$</td>
<td>2 ms</td>
</tr>
<tr>
<td>Opening movement time</td>
<td>$t_{om}$</td>
<td>1.5 ms</td>
</tr>
<tr>
<td>Closing delay time</td>
<td>$t_{cd}$</td>
<td>2 ms</td>
</tr>
<tr>
<td>Closing movement time</td>
<td>$t_{cm}$</td>
<td>1.5 ms</td>
</tr>
<tr>
<td>Digital valve’s flow coefficient</td>
<td>$C_d$</td>
<td>0.65</td>
</tr>
<tr>
<td>Maximum opening area of the smallest valve</td>
<td>$A_{d_{max}}$</td>
<td>0.128 mm$^2$</td>
</tr>
<tr>
<td>Main valve’s mass</td>
<td>$m$</td>
<td>205 g</td>
</tr>
<tr>
<td>Viscous friction coefficient</td>
<td>$B_f$</td>
<td>17.6 N/(m/s)</td>
</tr>
<tr>
<td>Initial volume of chamber</td>
<td>$V_f$</td>
<td>3750 mm$^3$</td>
</tr>
<tr>
<td>Coulomb friction force</td>
<td>$F_c$</td>
<td>6 N</td>
</tr>
<tr>
<td>Fixed orifice’s flow coefficient</td>
<td>$C_f$</td>
<td>0.85</td>
</tr>
<tr>
<td>Fluid jet angle</td>
<td>$\theta$</td>
<td>69°</td>
</tr>
<tr>
<td>Cross-sectional area</td>
<td>$A_f$</td>
<td>314 mm$^2$</td>
</tr>
</tbody>
</table>

4. Dynamic Characteristics of the Fuel Metering Valve

It is necessary to study the dynamic characteristics of the fuel metering valve because it directly affects the combustion performance of the aero-engine. The dynamic characteristics of the fuel metering valve actuated by two BDVAs consists of two parts, namely the transient flow uncertainty and the step response.

4.1. Transient Flow Uncertainty of the BDVA

First, the causes of the transient flow uncertainty of the BDVA should be determined to analyze the effects of the transient flow uncertainty of the BDVA on the moving characteristics of the fuel metering valve. In reference [24], the transient flow uncertainty of the BDVAs was preliminarily analyzed and it was observed that the timing of switching of the digital valves was not exact. Conversely, the transient flow uncertainty also indicates that the digital valves cannot switch synchronously or the opening and closing are asymmetrical.

In the simulation, to study the effects of the asymmetry of the delay time on the transient flow uncertainty of BDVA, the opening delay times $t_{od1}$–$t_{od5}$ were set as 1, 2, and 3 ms, respectively, and the closing delay times $t_{cd1}$–$t_{cd5}$ were set to 2 ms. To study the effects of the asymmetry of the movement time on the transient flow uncertainty of BDVA, the opening movement times $t_{om1}$–$t_{om5}$ were set as 0.5, 1.5, and 2.5 ms, respectively, and the closing movement times $t_{cm1}$–$t_{cm5}$ were set as 1.5 ms. The transient flow uncertainty of BDVA was obtained shown in Figure 5.
As shown in Figure 5, when the opening delay time was equal to the closing delay time or the opening movement time was equal to the closing movement time, the output flow rate of BDVA was stable and smooth. Conversely, when the opening delay time was shorter than the closing delay time or the opening movement time was shorter than the closing movement time, it indicated that a digital valve was not fully closed and the other valve was open, which resulted in a sudden increase in flow rate. When the opening delay time was longer than the closing delay time or the opening movement time was longer than the closing movement time, it indicates that a digital valve was not fully open and the other valve was closed, which resulted in a sudden decrease in the flow rate. In addition, the maximum flow fluctuation occurred when the state combination changed from [1 1 1 1 0] to [0 0 0 0 1] because the states of the digital valve changed.

In practical applications, it is impossible to ensure that the opening and closing characteristics of each digital valve are the same. Therefore, the transient flow uncertainty of BDVA caused by the asymmetry of the delay time and the asymmetry of movement time certainly exists.

4.2. Effects of the Transient Flow Uncertainty of BDVA on the Movement Fuel Metering Valve

To study the open combinations of the BDVA on the moving characteristics of the fuel metering valve, BDVA-A was closed and BDVA-B was controlled with the command signal, as shown in Figure 6. To obtain the discrete command signal, the slope signal was rounded in the simulation model.

As shown in Figure 6, the velocity and displacement curves were smooth, which was mainly caused by two factors. The first is that BDVA had a high flow resolution due to $2^N$ combinations, and the second is that in the simulation model, the opening/closing delay time and opening/closing movement time of each digital valve were the same. However, this setting does not match the actual situation.
It can be observed from Figure 5 that the transient flow uncertainty of BDVA can be divided into two parts: asymmetry of the delay time of the BDVA and the asymmetry of the movement time of BDVA. To analyze the effects of the asymmetry of the delay time of BDVA on the moving characteristics of the fuel metering valve, the opening delay times $t_{od1}$-$t_{od5}$ were set as 1, 2, and 3 ms, respectively, and the closing delay times $t_{cd1}$-$t_{cd5}$ were set as 2 ms. The velocity and displacement of the fuel metering valve are shown in Figure 7a.

[Figure 7a: Effects of the asymmetry of the delay time of the BDVA on the moving characteristics of the fuel metering valve: (a) Asymmetry of delay time; (b) Asymmetry of movement time; (c) Asymmetry of delay time and movement time. Left: velocity, Right: displacement.]

It can be observed from Figure 7a that when the opening delay times $t_{od1}$-$t_{od5}$ were different from the closing delay times $t_{cd1}$-$t_{cd5}$, the fluctuations in the velocity of the fuel metering valve occurred under the condition of varying state combination. Moreover, the velocity of the metering valve increased with a short opening delay time of the BDVA, and the dynamic characteristics of the metering valve were enhanced.

To analyze the effects of the asymmetry of the movement time of the BDVA on the moving characteristics of the fuel metering valve, the opening movement times $t_{om1}$-$t_{om5}$ were set as 0.5, 1.5, and 2.5 ms respectively, and the closing movement times $t_{cm1}$-$t_{cm5}$ were set as 1.5 ms. The moving characteristics of the fuel metering valve are shown in Figure 7b. The velocity fluctuations caused by the asymmetry of the movement time was smaller than that caused by the asymmetry of the delay time. This is because the movement
time contributed only half of the original value to the output flow rate as the delay time. Therefore, the change in the displacement rising curve of the fuel metering valve caused by the movement time was smaller than that caused by the delay time.

Figure 7c indicates that while simultaneously considering the asymmetries of the delay time and movement time, the velocity fluctuation and displacement changing range significantly increased.

4.3. Step Response of the Fuel Metering Valve

Simulation results of the displacement step response of the fuel metering valve are shown in Figure 8. In the figure, the reference signals include positive and negative movements (±2, ±3, ±4, and ±5 mm).

Figure 8 shows that, the delay time of the fuel metering valve under different reference step signals was within 4.1 ms, which was mainly caused by the opening time of the digital valve, the hysteresis induced by oil compression, and the friction of the valve. The rising times of different reference signals were 19.8, 27.3, 35.1, and 42.8 ms, respectively. Moreover, the maximum overshoot and maximum steady error were 1.10 and 0.074 mm, respectively. The performance indexes for the step response are listed in Table 3. The performance indexes of the positive and negative movements were consistent because the parameters of BDVA-A and BDVA-B were the same in the simulation. The rising time, overshoot, and steady error increased with an increase in the reference step signal.

<table>
<thead>
<tr>
<th>Performance Indexes</th>
<th>Reference Step Signal (Positive Movement)</th>
<th>Reference Step Signal (Negative Movement)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Delay time (ms)</td>
<td>2 mm 4 mm 3 mm 4 mm 5 mm –2 mm –3 mm –4 mm –5 mm</td>
<td>4.1 4.1 4.1 4.1 4.1 4.1 4.1 4.1</td>
</tr>
<tr>
<td>Rising time (ms)</td>
<td>19.8 27.3 35.1 42.8 19.8 27.3 35.1 42.8</td>
<td>4.1 4.1 4.1 4.1 4.1 4.1 4.1 4.1</td>
</tr>
<tr>
<td>Overshoot (mm)</td>
<td>0.61 0.74 0.91 1.10 0.61 0.74 0.91 1.10</td>
<td>0.61 0.74 0.91 1.10 0.61 0.74 0.91 1.10</td>
</tr>
<tr>
<td>Steady error (mm)</td>
<td>0 0.027 0.035 0.074 0 0.027 0.035 0.075</td>
<td>0 0.027 0.035 0.075</td>
</tr>
</tbody>
</table>

5. Position Controller of the Fuel Metering Valve

5.1. Design of the Position Controller

To improve the position tracking accuracy of the fuel metering valve, a velocity feedforward proportional-integral control strategy with PCM coding was proposed and its schematic diagram is shown in Figure 9.
As shown in Figure 9, the position controller consists of a velocity feedforward (VF) controller, a position proportional-integral (PI) controller, and a PCM controller. The velocity feedforward controller was used to calculate the flow rate required to match the velocity of the fuel metering valve in the open-loop model. The position proportional-integral controller was used to compensate the remaining position error of the fuel metering valve. The PCM controller was used to determine the best combination in which the error between the flow reference \( Q_{\text{refc}} \) and estimated flow combination \( Q_{\text{es}(i)} \) was minimum.

The closed-loop reference flow rate \( Q_{\text{refc}} \) was calculated by the velocity feedforward proportional-integral [32]:

\[
Q_{\text{refc}} = k_v A_v v_{\text{ref}} + k_p (x_{\text{ref}} - x_i) + k_i \int (x_{\text{ref}} - x_i) \, dt \tag{12}
\]

where \( v_{\text{ref}} \) and \( x_{\text{ref}} \) are the reference velocity and reference position, respectively. \( k_v, k_p, \) and \( k_i \) are the feedforward gain, proportional gain, and integral gain, respectively.

The estimated flow combination of BDVA-A can be defined as

\[
Q_{\text{es}} = n_{\text{int}} k_{\text{ave}} \sqrt{\Delta p} \tag{13}
\]

\[
n_{\text{int}} = \left[0, 1 \ldots 2^N - 2, 2^N - 1 \right]^T \tag{14}
\]

where \( Q_{\text{es}} \) denotes the estimated flow combination of BDVA-A, \( k_{\text{ave}} \) denotes the average flow coefficient of BDVA-A (assuming that the flow coefficients of the digital valves are the same), \( \Delta p \) denotes the pressure difference of BDVA-A, \( n_{\text{int}} \) denotes the opening combinations of BDVA-A, and \( N \) denotes the number of digital valves in BDVA-A.

The purpose of the PCM coding controller is to determine the best combination in which the error between \( Q_{\text{refc}} \) and \( Q_{\text{es}(i)} \) is minimum. A cost function \( J \) is used, which can be defined as

\[
J = \min \left| Q_{\text{es}(i)} - Q_{\text{refc}} \right| \tag{15}
\]

where \( Q_{\text{es}(i)} \) denotes the value of column \( I \) of \( Q_{\text{es}} \) \( (I = 2^N) \).

5.2. Validation of the Position Controller

To verify the effectiveness of the proposed controller, the following three controllers were compared, and the corresponding parameters are as follows: VF controller: This is the velocity feedforward controller. The velocity gain \( k_v \) was determined as 2 via a trial-and-error method. PI controller: This is the position proportional-integral controller. The proportional and integral gain were carefully tuned via a trial-and-error method and were 0.1 and 0.5, respectively. Velocity feedforward and proportional-integral (VFPI) controller: This is the controller composed of the combination of the velocity feedforward and the position proportional-integral. The velocity was the same as that of the VF controller, and the proportional-integral gains were the same as those of the PI controller.
A comparison between the tracking performances of the different controllers while tracking the sinusoidal trajectory with different frequencies is shown in Figure 10.

Figure 10. Comparison between the tracking performances while tracking the sinusoidal trajectory with different frequencies (amplitude is 4.5 mm): (a) 1 Hz; (b) 2 Hz; (c) 3 Hz.

Figure 10 shows that while tracking the sinusoidal signal with an amplitude of 4.5 mm and frequency of 1 Hz, the average error and standard deviation of the error of the VF controller were 0.244 and 0.274 mm, respectively. Compared with those of the VF controller, the average error and the standard deviation of the error of the PI controller were reduced by 60.7 and 60.6%, respectively. However, while using the proposed VFPI controller, the average error and standard deviation of the error were 0.021 and 0.027 mm, respectively. Additionally, a position tracking lag was not observed, which reflected good tracking performance of the proposed VFPI controller. Furthermore, the tracking errors of the three controllers increased with an increase in the tracking frequency. However, the tracking performance of the VFPI controller had a significant advantage.

Although the VF controller can eliminate the deviation caused by disturbance, it is an open-loop control. Therefore, the error cannot be corrected and its tracking accuracy is the worst. The PI controller can correct the error and improve the control accuracy to a certain extent. However, the controller cannot adapt to different tracking conditions owing to the fixed controller gain, and it is difficult to enhance the tracking performance. The proposed
VFPI controller effectively combines the advantages of the VF and PI controller, and the control accuracy is significantly improved.

The tracking amplitude was increased to 6 mm, and a comparison between the tracking performances of the different controllers while tracking the sinusoidal trajectory with different frequencies is shown in Figure 11.

As shown in Figure 11, while tracking the sinusoidal signal with an amplitude of 6 mm and frequency of 1 Hz, the average error and standard deviation of the error of the VF controller were 0.157 and 0.175 mm, respectively. The tracking accuracy of the PI controller was improved owing to the correction of the position error. Compared with those of the VF and PI controller, the average error and standard deviation of the error of the VFPI controller were 0.022 mm and 0.03 mm, respectively.

Moreover, the tracking errors of the three controllers increased with an increase in the tracking frequency. However, the tracking performance of the VFPI controller had a significant advantage.

A comparison between the tracking errors of the three controllers is presented in Table 4. The average and standard deviation of the tracking error are obtained from ref. [33].
The comparisons shown in Table 4 indicate that varying the tracking frequency and amplitude have a significant effect on the tracking performance of the three controllers owing to the utilization of the fixed controller gain. However, compared with those of the VF controller and PI controller, the average error and standard deviation of the error of the proposed VFPI controller were reduced by 78 and 72.7%, respectively. Therefore, the proposed VFPI controller exhibited the best tracking performance.

### 6. Conclusions

In this study, a fuel metering valve actuated by two BDVAs was proposed. The concept of this unit configuration was derived from the principle of the dual nozzle-flapper structure. Compared with that of the conventional fuel metering valve actuated by a servo valve, the proposed fuel metering valve has the advantages of high reliability, efficiency, and low cost. The main conclusions are as follows:

1. The structure and working principle of the proposed fuel metering valve were discussed. In addition, the mathematical model was established, including the pilot flow model of the BDVA and dynamic model of the metering spool. Subsequently, a simulation model of the entire valve system was developed using MATLAB/Simulink.

2. The mechanism of the transient flow uncertainty of the pilot BDVA was revealed through simulation. The BDVA produces transient flow with uncertainty due to the unsynchronized opening and closing of the on–off valves, but when used in the pilot stage, it has little effect on the movement of the main stage.

3. Simulation results of the position step response indicated that the delay time of the fuel metering valve under different reference step signals was within 4.1 ms. In addition, the maximum overshoot and maximum steady error were 1.10 and 0.074 mm, respectively.

4. To improve the position tracking accuracy of the fuel metering valve, a velocity feedforward proportional-integral control strategy (VFPI) with PCM coding was proposed. A comparison between the controllers demonstrated that the average and standard deviation of the position tracking error under the proposed VFPI controller were reduced by 78 and 72.7%, respectively.

5. The proposed pilot structure consisting of two binary-coded digital valve arrays can be used not only for the pilot stage of fuel metering valves in this research, but also for the pilot stage of proportional/servo valves with large flow rate to improve reliability and digitization.

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