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Development of a piezoelectric high speed on/off valve and its application to pneumatic closed-loop position control system†

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Abstract

The use of smart materials-based high speed on/off valve has exhibited the potential to replace traditional servo and proportional valves in fluid power systems. In this paper, a novel pneumatic high speed on/off valve driven by a piezoelectric stack actuator is developed and its dynamics are studied. For an upstream gauge pressure of 0.5 MPa, the valve exhibited a large flow rate of up to 86 L/min. Furthermore, to study the ability of this novel high speed on/off valve to be applied in closed-loop control systems, a real-time position control system is realized using a PID controller. In order to prove its feasibility and effectiveness, a number of closed-loop trajectory tracking experiments are conducted on a pneumatic cylinder. The system is proved feasible and the results demonstrate good tracking performance.

Keywords: Closed-loop; High-speed on/off valve; Piezoelectric stack; PWM

1. Introduction

High-speed on/off valves (also known as digital valves) are one of the fundamental components in the applications of digital fluid power. They have been researched over the years and demonstrated good characteristics, such as high-precision control, high reliability, pollution-proof and low cost [1, 2]. A number of researchers have highlighted the potential of on/off valves to replace servo/proportional valves [3, 4]. In order to use discrete on/off valves instead of continuously acting traditional valves and obtain similar proportional characteristics, pulse width modulation (PWM) techniques are used. Applying a PWM signal as the control input to a switching valve makes it on and off successively; as a result, discrete packets of fluid mass are delivered to the actuator as the valve is either completely open or completely closed. To achieve continuous-like flow, the valve is switched at high frequency. Typical examples of the applications of PWM driven on/off valves can be found in adaptive braking systems (ABS) [5], fuel injectors [6], and motion control of cylinders [7-9]. Originally developed high speed on/off valves used solenoid-based actuators; however, the use of solenoids has some drawbacks, such as the delays in the response time owing to the delay of the electrical signal in the coil, and the reduction of the response speed due to the high flow forces in the valve [10]. To overcome aforementioned drawbacks, different approaches such as the improvement of control electronics and structure of the solenoids [11-14]; or the use of smart materials as the driving actuators were proposed [15-18]. The results for the latter approach are promising; however, most studies focused on developing hydraulic high speed on/off valve; relatively few studies have been conducted on pneumatic ones. Matti Linjama (one of the pioneers of digital fluid power) expressed a similar observation that, one surprising fact is that digital principles are not studied a lot in pneumatics where they should offer similar advantages [19].

Most of the researchers who studied the possibility of using high speed on/off valves instead of traditional valves to control servo-pneumatic systems employed multiple high speed on/off valves. For instance, Ahn et al. proposed a novel modified pulse-width modulation valve pulsing algorithm that employs eight high speed on/off valves to control a rod-less cylinder and the position control was successfully implemented [9]. Laib et al. presented a control system that uses an averaged state model and sliding mode observer to control a pneumatic actuator using four solenoid on/off valves. This observer was validated experimentally in a closed loop position tracking system [20]. Also, Najjari et al. presented a position controller for a double-acting cylinder. The system employs four on/off valves and a classical PID controller optimized by genetic algorithm [21]. Fawaza et al. investigated

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the ability of a fuzzy PI controller to cope with the non-linearity introduced by the four on/off valves operating in a highly non-linear pneumatic system [22]. The controller performance was tested with sinusoidal, triangular and saw tooth reference wave inputs at a frequency of 0.1 Hz and results were satisfactory.

In this study, a novel structure of a pneumatic high speed on/off valve directly driven by a piezoelectric stack actuator is constructed. Its dynamics are mathematically and experimentally studied to determine its flow characteristics.

Unlike most of previous researches, which employed multiple high speed/off valves to replace traditional servo and proportional valves, this new control system employs a single high speed on/off valve in combination with a directional control valve to control the position of a pneumatic cylinder.

Taking into consideration the working principle of piezoelectric materials and the properties of compressed air, a new structure of a high speed on/off valve is designed. Its mathematical model is established based on the designed structure; and Matlab/Simulink is used to build a simulation model. Furthermore, experimental studies are conducted on the manufactured prototype of the high speed on/off valve to investigate its flow characteristics. After evaluating the flow characteristics, a closed-loop system is proposed to realize real-time position control of a pneumatic actuator using a combination of PWM signal and a PID controller. The tracking performance of the proposed real-time position control system is demonstrated and interpreted.

2. Structure and working principle of the valve

2.1 Structure of the valve

The structure and photograph of the designed high speed on/off valve driven by a piezoelectric stack actuator are shown in Fig. 1. It is a normally-closed (NC) type of valve, which mainly consists of a piezoelectric stack (size: 14 mm× 14 mm×40 mm), a mobile plate, and the valve body. The piezoelectric stack actuator drives the mobile plate, which restricts or allows the air flow through the valve. Since the piezoelectric stack actuator can only provide pushing force, when the actuator is de-energized the mobile plate is returned to the initial position by a set of disc springs acting against it.

The typical expansion of a piezoelectric stack is within 0.075 % to 0.150 %. In some applications, displacement amplification mechanisms are used to amplify the stroke of the piezoelectric material; however, displacement amplification is a trade-off between large displacement and force [23]; thus, in this work, a direct drive method was selected in order to preserve high dynamics of the piezoelectric stack. At the bottom end of the actuator, an adjusting bolt was used to ensure that the output rod is in contact with the mobile plate. Another challenge of the piezoelectric stacks is that, they are highly vulnerable to tensile and lateral forces. The common practice to avoid such damage is to mechanically preload the piezoelectric stack to compensate for excessive tensile load [24]; therefore, in this design the piezoelectric stack was preloaded using another set of disc springs. In addition to that, slots were cut in the actuator's shell to facilitate the dissipation of the heat that could be generated by the piezoelectric stack when driven at high frequency.

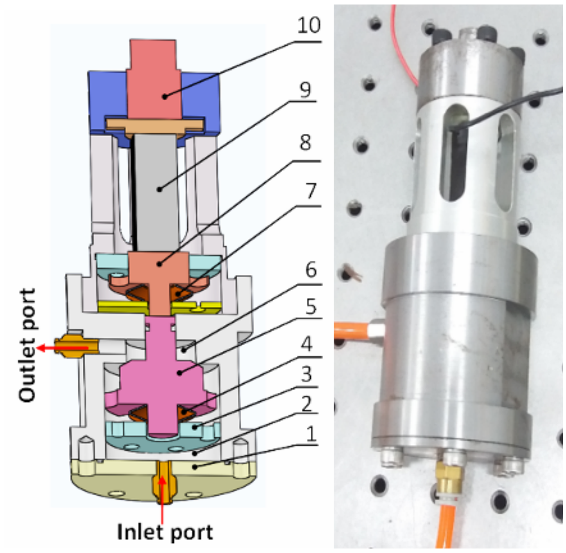


Fig. 1. The designed high speed on/off valve (1. Bottom end cap, 2. Inlet chamber, 3. Spring preloading plate, 4. Disc spring1, 5. Mobile plate, 6. Outlet chamber, 7. Disc spring 2, 8. Output rod, 9. Piezoelectric stack, 10. Adjusting bolt).

2.2 Working principle of the valve

Initially, the mobile plate is pushed against the valve seat by disc spring1; that contact prevents the compressed air to flow from the inlet chamber to the outlet chamber; thus, the valve is "normally closed". When a voltage signal is applied to the piezoelectric stack actuator, the expansion of the piezoelectric stack leads to the motion of the output rod, which in return moves the mobile plate. Consequently, the air flows from the inlet chamber to the outlet. Once the applied voltage signal is switched off, the piezoelectric stack retracts and the mobile plate is returned to the initial position by disc spring1. When a PWM signal is used as the input, the valve switches between two states (on and off) at high frequency. By modulating the duty cycle of the signal, the air flow rate through the valve changes accordingly, which results in a controllable air flow.

3. Mathematical model of the valve

Based on the structure of the valve and its working principle, the mathematical model is divided into two parts: An electromechanical model and a fluidic model as illustrated in the block diagram in Fig. 2. The electromechanical domain represents the conversion of the electrical energy into mechanical energy by the piezoelectric stack actuator. Whereas, the fluidic domain represents the conversion of the valve opening into the

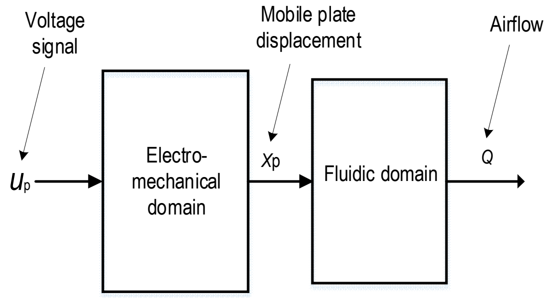


Fig. 2. Block diagram of the mathematical model.

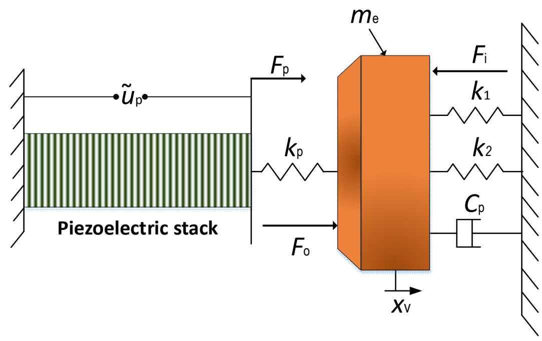


Fig. 3. Electromechanical model representation.

corresponding air flow.

3.1 Electromechanical model

As shown in Fig. 3, the motion of the driving part of the high speed on/off valve can be modeled as a single degree of freedom system [25].

Mathematically, it is expressed as:

$$m_e\ddot{x}_v + c_p\dot{x}_v + (k_p+k_1+k_2)x_v = F_p - (F_i - F_o) \tag{1}$$

where m_e denotes the equivalent mass, c_p denotes the damping constant of the piezoelectric stack; k_p, k_1 and k_2 denote the stiffness of the piezoelectric stack, the disk spring1 and the disc spring 2; respectively. F_i, F_o denote the force exerted by the compressed air to the inlet side of the mobile plate and to the outlet side, respectively. F_p denotes linear output force of the piezoelectric stack. Eq. (1) can be further derived as follows [26].

$$\begin{aligned}&(\frac{1}{3}m_p + m_r + m_m)\ddot{x}_v + c_p\dot{x}_v + (k_p + k_1 + k_2)x_v \\ &= \frac{k_u}{\tau}e^{\frac{-t}{\tau}}u_p(t) - (p_iA_i - p_oA_o)\end{aligned} \tag{2}$$

where m_p, m_r, m_m denote the mass of the piezoelectric stack, the output rod and the mobile plate, respectively. k_u denotes the scale factor between the linear output force and the input voltage. τ denotes the time constant and u_p denotes the input voltage signal. p_i and p_o denote the absolute upstream stagnation pressure and downstream stagnation pressure, respectively. A_i and A_o denote the area of the inlet side and the outlet side of the mobile plate, respectively.

As PWM signal can only have two states (high level and low level), for one period, the driving signal can be expressed as shown in Eq. (4).

$$D = \frac{T_{on}}{T_p} \tag{3}$$

$$u_p(t) = \begin{cases} V & t \le DT_p \\ 0 & t > DT_p \end{cases} \tag{4}$$

where T_{on} denotes the "on-time", T_p denotes the period of the signal, D denotes the duty cycle, and V denotes the amplitude of the PWM signal.

3.2 Fluidic model

The standard equation of the air mass flow rate through the valve is given in Eq. (5). This equation is based on ISO 6358 standard [27, 28].

$$\dot{m} = \begin{cases} \dfrac{C_dA_vp_i}{\sqrt{RT}}\gamma^{\frac{1}{2}}\left(\dfrac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} & \dfrac{p_o}{p_i} \le b_c \\ \dfrac{C_dA_vp_i}{\sqrt{RT}}\left(\dfrac{2\gamma}{(\gamma-1)}\left(\left(\dfrac{p_o}{p_i}\right)^{\frac{2}{\gamma}} - \left(\dfrac{p_o}{p_i}\right)^{\frac{\gamma+1}{\gamma}}\right)\right)^{\frac{1}{2}} & \dfrac{p_o}{p_i} > b_c \end{cases} \tag{5}$$

where for a sharp edged orifice the value of

$$b_c = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}. \tag{6}$$

C_d denotes the discharge coefficient of the valve, A_v denotes the effective cross sectional area of fluid flow path, p_i denotes the absolute upstream stagnation pressure of the valve, p_o denotes the absolute downstream stagnation pressure of the valve, T denotes the upstream stagnation temperature, R denotes gas constant and γ denotes the heat capacity ratio.

The critical pressure ratio b_c of a pneumatic valve is the ratio between the downstream absolute pressure p_o and the upstream absolute pressure p_i at which the air velocity achieves sonic speed. When the ratio between the downstream absolute pressure p_o and the upstream absolute pressure p_i is less than the critical pressure ratio b_c of a valve, further reduction of the downstream pressure p_o does not increase the air mass flow as Eq. (5) indicates.

4. Simulation model of the high speed on/off valve

To theoretically analyze the characteristics of the valve, the equations mentioned in the mathematical modelling section

Table 1. Parameters of the electro-mechanical model.

Name	Unit	Symbol	Value
Mass of the piezoelectric stack	kg	m_p	0.062
Mass of the piston rod	kg	m_r	0.014
Mass of the mobile plate	kg	m_m	0.033
Damping constant of the piezoelectric stack	$N{\cdot}s{\cdot}m^{-1}$	c_p	1500
Stiffness of the piezoelectric stack	$N{\cdot}m^{-1}$	k_p	0.70×10^{8}
Stiffness of the disk spring 1	$N{\cdot}m^{-1}$	k_1	1.5×10^{6}
Stiffness of the disk spring 2	$N{\cdot}m^{-1}$	k_2	1.5×10^{6}
Cross section area of the inlet chamber	m^2	A_i	0.96×10^{-3}
Cross section area of the outlet chamber	m^2	A_o	0.42×10^{-3}
Scale factor between the linear output force and the input voltage	N/V	k_u	57.966
The time constant	s	τ	5×10^{-5}

The value of the damping constant of the piezoelectric stack [26]

Table 2. Parameters of the fluidic model.

Name	Unit	Symbol	Value
Discharge coefficient	-	C_d	0.5
Gas constant	J/kg·K	R	288
Mass density at standard reference atmosphere	kg/m^3	ρ	1,185
Heat capacity ratio	-	γ	1.4
The absolute downstream pressure	MPa	p_o	0.1

were solved numerically using MATLAB/Simulink.

The complete simulation model permits the estimation of the valve's flowrate by specifying the driving voltage signal. Parameters of the complete model are shown in Tables 1 and 2. Using Simulink model, flow characteristics of the valve were simulated under various working conditions and driving signals

First, the effect of the frequency to the air flow was studied. The amplitude of the PWM signal was set to 80 V, the duty cycle to 60 % and the upstream gauge pressure (p_s) to 0.4 MPa. In all simulations in this work, the absolute downstream pressure was assumed to be equal to the atmospheric pressure (0.1 MPa). The flow accumulation at frequencies of 5 Hz, 10 Hz, 15 Hz and 20 Hz, respectively, were compared and the results are shown in Fig. 4.

As it can be seen in Fig. 4, in addition to leading to a more quasi-continuous flow, increasing the frequency increases the resolution of the valve's flow regulation.

Secondly, PWM signals of different duty cycles were applied to the valve's simulation model to study how the duty cycle affects the flow. Fig. 5 shows the comparison of three different duty cycles for upstream gauge pressures of 0.2 MPa and 0.4 MPa, respectively. The frequency of the PWM signals was set to 20 Hz and the amplitude to 80 V.

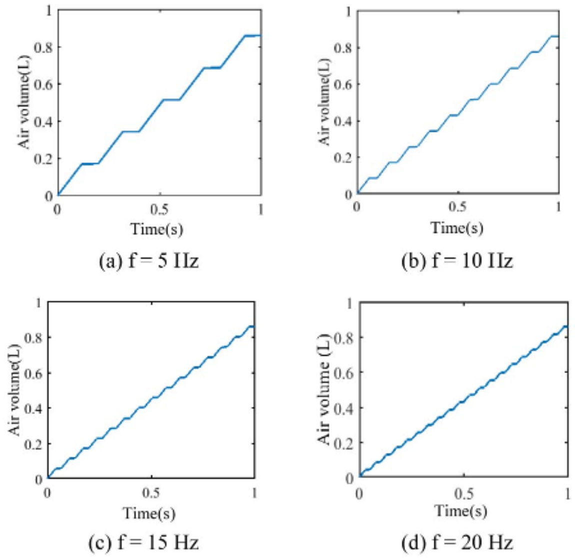


Fig. 4. Air flow accumulation for various frequencies.

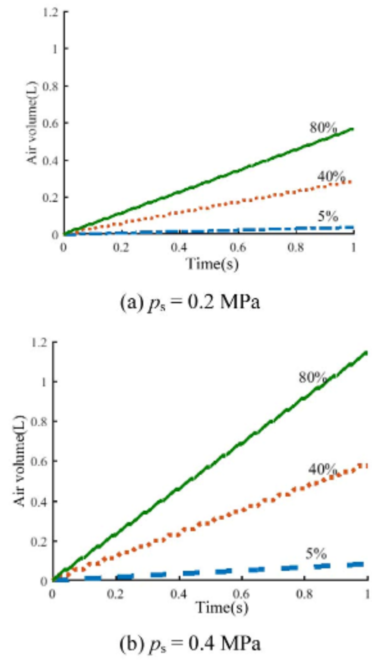


Fig. 5. The effect of the duty cycle to the air flow.

As predicted, the results demonstrated that the flow rate increases with the increase of the duty cycle. These results affirmed that the air flow rate through the valve could be controlled by modulating the duty cycle.

5. Simulation model validation

5.1 Electromechanical model validation

To experimentally validate the electromechanical model, the test bench shown in Fig. 6 was set-up and tests were conducted.

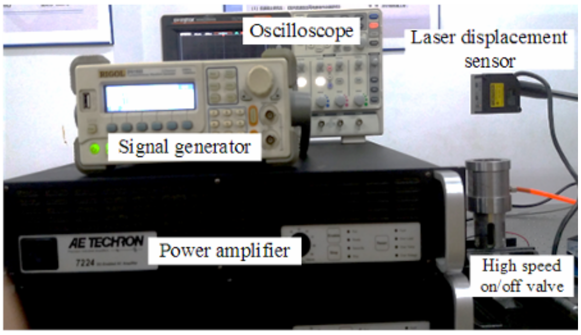
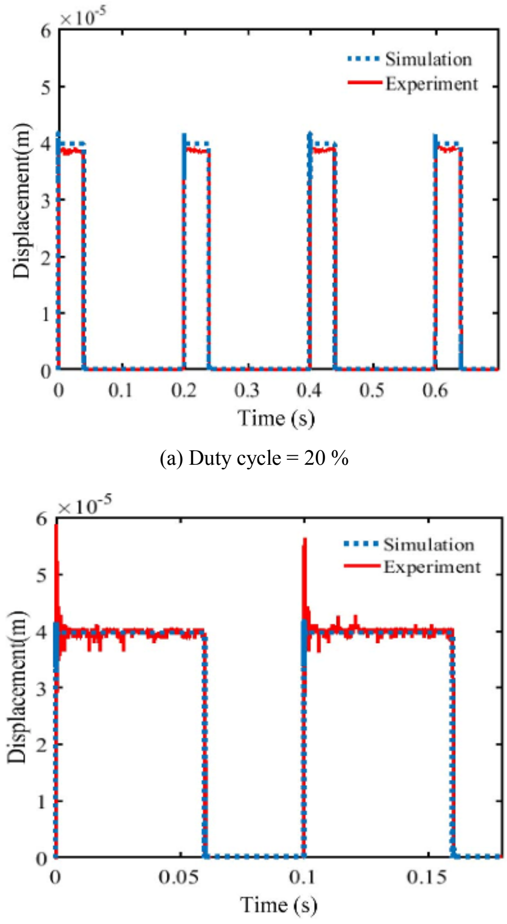


Fig. 6. Electromechanical model validation test bench.

The aim was to capture the displacement of the mobile plate for different input signals. A signal generator (RIGOL DG1022) was used to generate a PWM signal; the signal was amplified by a power amplifier (AE Techron 7224); then, applied to the valve. A laser displacement sensor (Shanghai Sixin CD5-30(A), accuracy linearity ±0.08 %) captured the displacement of the valve's mobile plate and data were stored by the oscilloscope (GW Instek GDS-1104B). Due to the fact that the displacement of the mobile plate had to be captured using the laser displacement sensor, the bottom end cap of the valve was removed, and evidently the inlet port of the valve was not connected to the air supply; therefore, to have the same conditions in the simulation, the absolute upstream pressure (p_i) was set to 0.1 MPa (the atmospheric pressure).

Comparison of the simulation and experimental results are shown in Fig. 7. As it can be seen in the figure, the simulation model can be used to estimate the switching of the mobile plate for different duty cycles.



(a) Duty cycle = 20 %

(b) Duty cycle = 60 %

Fig. 7. Mobile plate displacement (amplitude 100 V).

5.2 Complete model validation

After validating the electromechanical model, another test bench was set-up to validate the complete model of the valve (Fig. 8). In this experiment, since complete simulation model relates the input voltage signal to the output air flow, various voltage input signals were applied to the designed high speed on/off valve and the resulting air flow rates were measured using a flow meter (Shuanghuan LZB-10WB-F). A PWM signal was generated using the signal generator; the signal was sent to the power amplifier; the power amplifier amplified the PWM signal and sent it to the valve. Simultaneously, the air compressor supplied the compressed air and a pressure regulator (Festo FRC-1/4-D-MIDI) was used to set the desired upstream gauge pressure. A flowmeter captured the air flow rate downstream for each tested working conditions.

The valve was experimentally tested under various working conditions and the results were compared with the simulation results to validate it. The variation of the flow rate in respect to the duty cycle at a constant upstream pressure is shown in Fig. 9.

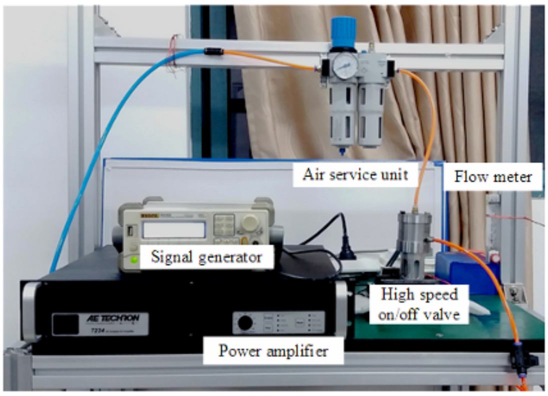
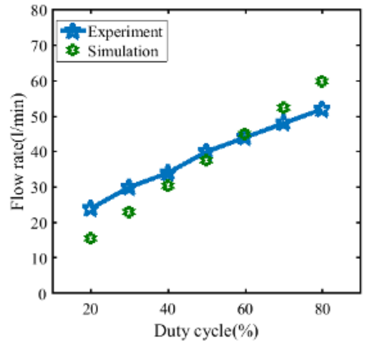
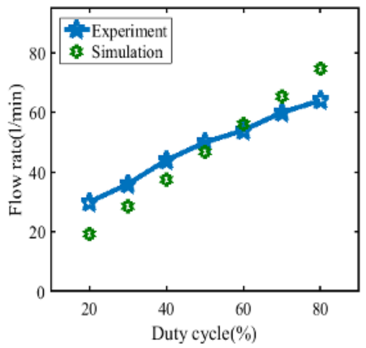


Fig. 8. Photograph of the test bench of the valve's flow characteristics.

The discrepancies between the simulation and the experimental results can be explained by the fact that the simulation model takes several assumptions into account. Some of the assumptions are that, the air in the system is an ideal gas; gas density in the valve and pipelines is uniform; the air flow process is adiabatic; no leak occurs anyway in the system; the temperature change inside the valve is negligible with respect to the temperature of the air supply; and that the supply pressures are constant, which are not the actual conditions during the experiment. Despite that, the developed model can be



(a) Amplitude = 80 V



(b) Amplitude = 100 V

Fig. 9. Variation of the flowrate in respect to the duty cycle (0.4 MPa).

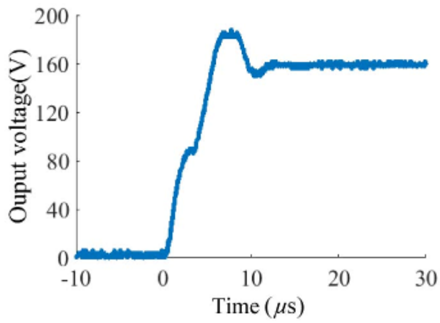
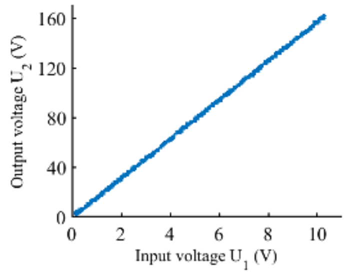
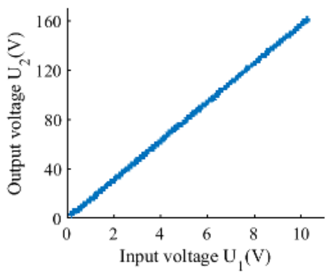


Fig. 10. Step-response of the power amplifier.



(a) f = 10 HZ



(b) f = 100 Hz

Fig. 11. Power amplifier performance testing results.

reliable as far as the analysis of the tendency of the flow rate under specific conditions is concerned. It can be seen that the flow rate increases with the increase of the duty cycle for both results.

6. Experimental analysis

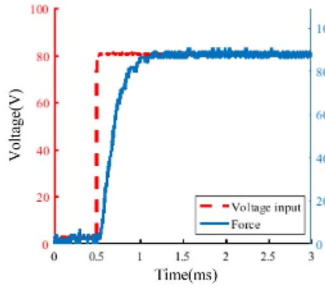
6.1 Power amplifier testing

To drive the piezoelectric stack actuator, a power amplifier is used to amplify the voltage signal before it is sent to the piezoelectric stack; therefore, the output characteristics of the power amplifier have direct impact on the performance of the piezoelectric stack actuator. Experiments were carried out to analyze the voltage amplification process. In this experiment and in all other experiments in this work, the gain of the power amplifier was fixed to 16, which means that for any input voltage, the power amplifier had to amplify it 16 times. First, the step response of the power amplifier to a step input of 10 V is examined. The findings are shown in Fig. 10. It can be seen that the rise time of the power amplifier is approximately 7 µs and the settling time 10 µs, which proves that the power amplifier has an extremely quick response time.

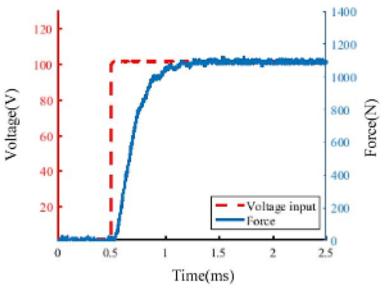
Secondly, the hysteresis between the voltage signal input to the power amplifier and the voltage signal output by the power amplifier are tested at the frequency 10 Hz and 100 Hz, respectively. To test it, the signal generator input sinusoidal voltage signals of a peak to peak amplitude of 10 V. In Fig. 11, U_1 represents the voltage signal input to the power amplifier, while U_2 represents the voltage signal output by the power amplifier. The results showed that there was no lag between the input voltage and the output (Fig. 11).

6.2 Force characteristics

During the valve structure design, the force generated by the piezoelectric stack actuator had to be measured to ensure that its force dynamics are suitable for high speed actuation applications. To measure the force, the preloading disc springs were not installed in the actuator; instead, the preloading plate was tightened against the output rod to ensure that the piezoelectric stack is not damaged during the experiment. A force sensor was installed on the bottom end of the piezoelectric stack to capture the generated force. Two step signals of 80 V and 100 V were input to the piezoelectric stack actuator, respectively. Given that the response time of the piezoelectric stack actuator is in the range of a few milliseconds (ms), the record length of the oscilloscope had to be increased; by doing so, the overshoot shown in Fig. 10, which lasts for few micro-



(a) Amplitude = 80 V



(b) Amplitude = 100 V

Fig. 12. Force step response of the piezoelectric stack actuator.

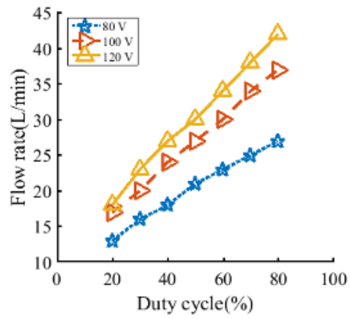
seconds, could not be captured in this specific experiment due to the sample rate of the oscilloscope. The results of the experiment are shown in Fig. 12.

The piezoelectric stack actuator generates 900 N for a voltage input of 80 V and 1100 N for a voltage input of 100 V. In both cases, the rise time of the force generated by the piezoelectric stack actuator is approximately 0.5 ms, which is quick enough to be applied in high speed actuation applications.

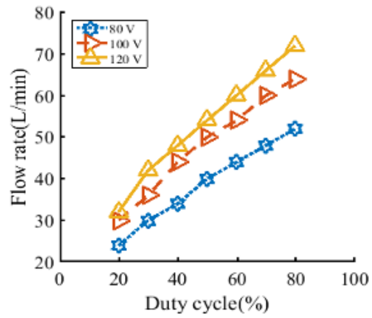
6.3 Flow characteristics

To assess and confirm the viability of the designed high speed on/off valve to be applied in motion control applications. The test bench shown in Fig. 8 was used to further test the valve. In this set of experiments, the effect of the amplitude and the duty cycle of the PWM voltage signal was studied. PWM signals with amplitudes of 80 V, 100 V and 120 V, respectively, were applied to the high speed on/off valve. The duty cycle was varied from 20 % to 80 % for each signal and the flow rate at the outlet port of the valve was measured using the flow meter. The frequency of the driving signal was set to 20 Hz. These experiments were carried out under three different upstream gauge pressures (0.2 MPa, 0.4 MPa and 0.5 MPa). The results in Fig. 13 showed that, as it had been found during the simulation-based analysis, increasing the amplitude of the PWM signal increased the flow rate. In the same way, when the duty cycle increases, the flow rate increases as well. As shown in Fig. 13(c), the valve exhibited a large flow capacity of up to 86 L/min for an upstream gauge pressure of 0.5 MPa.

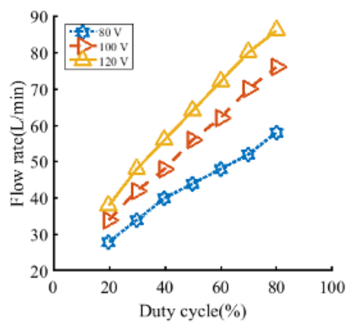
These findings affirmed that the flow through the valve could be regulated by modulating the duty cycle as predicted during the simulation-based analysis.



(a) $p_s = 0.2$ MPa



(b) $p_s = 0.4$ MPa



(c) $p_s = 0.5$ MPa

Fig. 13. The relationship between the duty cycle and the flow rate for different upstream gauge pressures.

7. Dynamics of a pneumatic cylinder controlled by the developed valve in an open-loop system

Before designing a closed-loop system, the dynamics of an open loop system consisting of the designed valve and a double-acting pneumatic cylinder (Chaozhong pneumatic SCJ 32X125-50) were investigated. The schematic of the open-loop control system is shown in Fig. 14. The signal generator was employed to generate a PWM signal. The signal was sent to the power amplifier and the power amplifier fed the amplified signal to the designed high speed on/off valve. Simultaneously, the pressure regulator, which is part of the air service unit, was used to set the supply pressure. The designed valve was directly connected to one of the chambers of the cylinder while the other chamber was open to the atmosphere. The

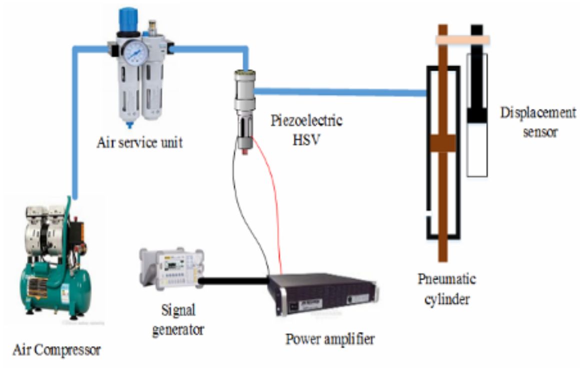


Fig. 14. Schematic diagram of the open-loop control system.

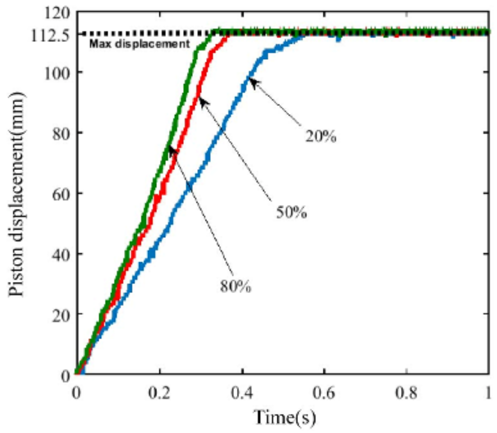


Fig. 15. Effect of the duty cycle on the piston displacement.

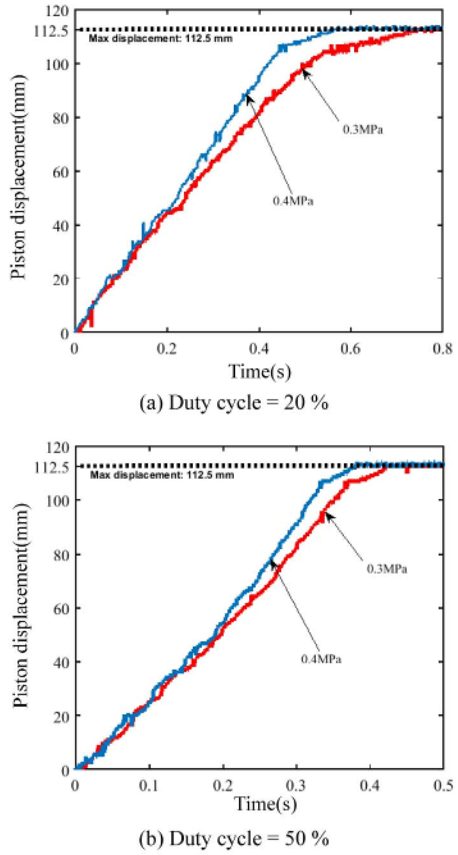


Fig. 16. Effect of the supply pressure to the piston displacement.

movement of the piston was captured by a rod type displacement sensor (GEERT HLC-175 mm, accuracy: ±0.05 %).

First, the effect of the signal's duty cycle to the displacement of the piston was examined. The supply pressure was set to 0.4 MPa and the frequency of the PWM signal to 30 Hz. It can be seen from Fig. 15 (the dotted line indicates the maximum displacement) that, when the duty cycle increases, the speed of the piston increases as well. There is a larger increase in the speed of the piston when the duty cycle is changed from 20 % to 50 % compared to when it is changed from 50 % to 80 %. This is due to the fact that when the duty cycle is low, the flowrate increment step-size is small; thus the speed is relatively low.

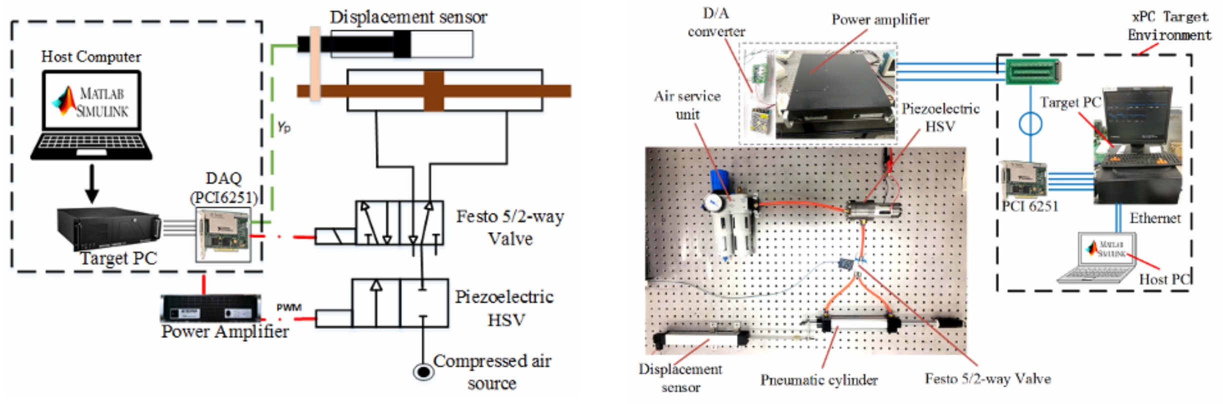
Second, the effect of the supply pressure to the displacement of the piston was also examined. The results are shown in Fig. 16. The speed of the piston increases with the increase of the supply pressure.

8. Closed-loop position control

After the analysis of the flow characteristics of the developed high speed on/off valve and the dynamics of the pneumatic cylinder controlled by the valve in an open-loop system, a closed loop control system was also designed as shown in Fig. 17. The closed loop system consists of a double acting cylinder (Chaozhong pneumatic SCJ 32X125-50, bore diameter: 32 mm, maximum stroke 150 mm), the developed high speed on/off valve, which is used to control the air flow to the cylinder, a directional control valve (Festo MHE2-MSIH-5/2-M7-K) used to change the direction of the air flow, a displacement sensor (GEERT HLC-175, max stroke: 175 mm, accuracy: ±0.05 %) to transmit real-time position data to the data acquisition card (PCI 6251) and a power amplifier(AE Techron 7224) to amplify the signal before it is sent to the high speed on/off valve. Data acquired from the sensors are processed via a controller running on the host computer of an xPC environment to calculate the control signals. A PWM signal whose duty cycle is determined by the controller is sent to the high speed on/off valve to control the air flow rate and depending on the sign of the error between the desired and the actual position, a signal is sent to the directional control valve to direct the air to the proper chamber.

8.1 Controller design

The motion control strategy adopted in this work is mainly based on a combination of PWM signal and PID feedback controller. For over decades, PID controllers have been used in industrial control applications. Despite being around for a longtime, they are still applied in the majority of closed-loop



(a) Schematic diagram (b) Photograph

Fig. 17. Experimental set-up of the proposed closed-loop control system.

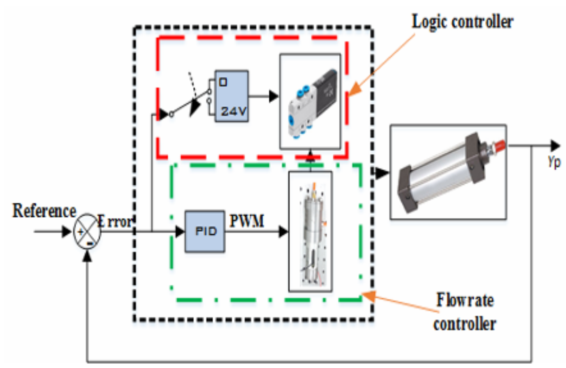


Fig. 18. Schematic diagram of the control system.

applications. PID controller is often combined with logic, sequential functions, selectors, and simple function blocks to build the complicated automation systems [29]. In digital fluid power technology, a flow control strategy such as PWM or PFM is combined with one or multiple feedback control strategies to build a closed-loop control system. For example, Mazare M. et al. used a combination of PWM signal and a PID controller to control a pneumatic actuator using a fast switching valve [30]. Thananchai built a closed-loop system to control both the position and the force of a pneumatic artificial muscle using a combination of PWM signal and fuzzy logic controller [31].

In this work, a combination of PWM signal and a PID controller method is used. The proposed control system consists of two layers of controllers (Fig. 18). A logic controller and a flowrate controller. The former controls the operation of the directional control valve according to the sign of the error between the position command signal and the measured position signal, and the latter controls the switching of the high speed on/off valve by calculating the value of the duty cycle needed to eliminate the error on real-time.

Generally, the input/output relation for an ideal PID controller with error feedback is expressed as:

$$u(t) = K_p e(t) + K_i \int_0^t e(t)dt + K_d \frac{de(t)}{dt} \tag{7}$$

where $e(t)$ is the error between the desired and the actual position, $u(t)$ is the control signal, K_p, K_i and K_d are the proportional ,integral and derivative coefficients.

In relation to the proposed function of the PID controller in the designed closed-loop control system, Eq. (7) can be rewritten as follows:

$$D(t) = K_p |e(t)| + K_i \int_0^t |e(t)| dt + K_d \frac{d|e(t)|}{dt} \tag{8}$$

where $|e(t)|$ is the absolute value of the error between the desired and the actual position; $D(t)$ is the duty cycle of the control PWM signal; and $K_p = 0.04$, $K_i = 0.01$ and $K_d = 0.01$. These values were obtained after manually tuning the PID controller through multiple experiments that aimed at identifying the best parameters.

8.2 Position control results

To test the performance of the proposed control system, a number of reference trajectories were used. First, a sinusoidal trajectory with frequencies of 0.1 Hz, 0.15 Hz, 0.2 Hz, 0.5 Hz, respectively, were input to the system as the reference signal. The frequency of the control PWM signal was set to 30 Hz and the supply pressure to 0.4 MPa. Since the value of voltage output by digital pins of the xPC are 5 V (for high) and 0 V (for low), the gain of the amplifier was preset to 16 in order to obtain a PWM signal of an amplitude of 80 V.

From Figs. 19-21, it can be seen that the proposed control system possess good tracking performance. Fig. 21 shows the results for a higher frequency of 0.5 Hz. A slight increase in the error was noticed compared to lower frequencies (0.1 Hz and 0.2 Hz). Nonetheless, the tracking performance at

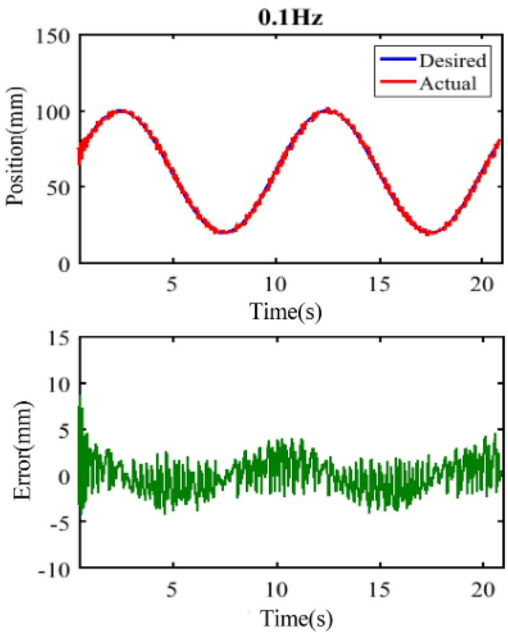


Fig. 19. Tracking performance for a reference sinusoidal signal of 0.1 Hz.

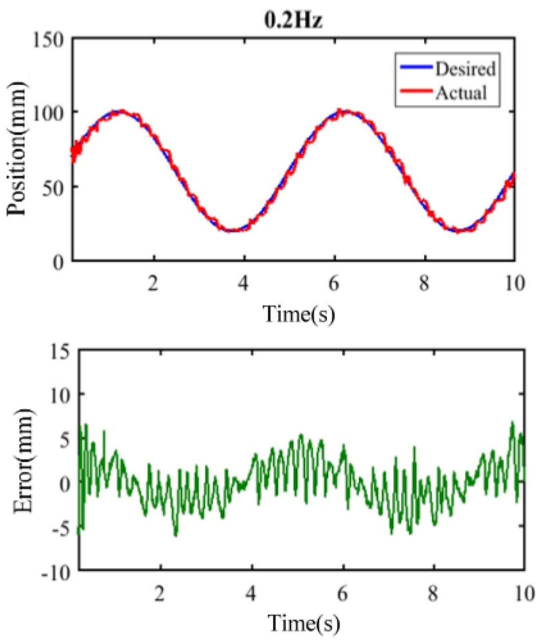


Fig. 20. Tracking performance for a reference sinusoidal signal of 0.2 Hz.

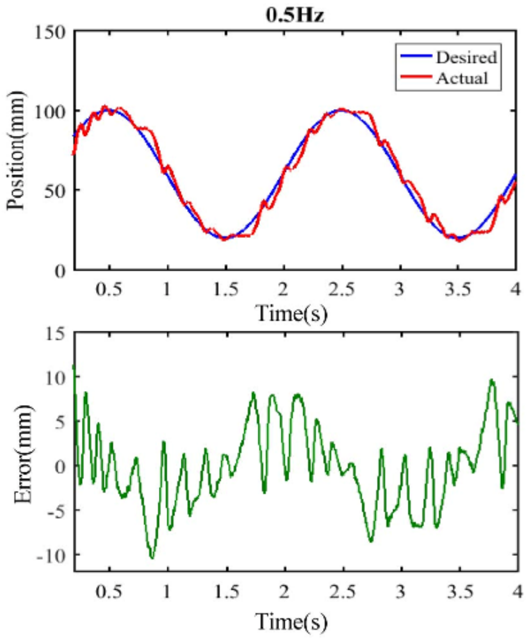


Fig. 21. Tracking performance for a reference sinusoidal signal of 0.5 Hz.

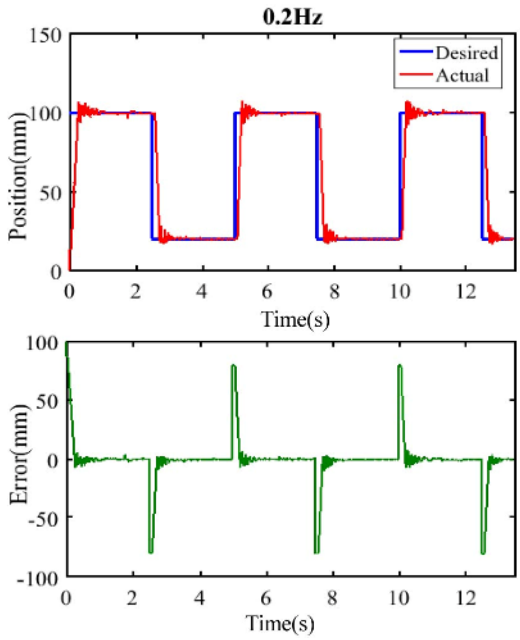


Fig. 22. Tracking performance for a square wave reference signal of 0.2 Hz.

that frequency is still within an acceptable range. No phase lag or amplitude weakening occurred for any tested frequency.

After the performance of the control system to track sinusoidal trajectories was analyzed, the command signal was changed to square waves. Fig. 22 shows the results for a square wave signal of 0.2 Hz and a duty cycle of 50 %.

From the results summarized in Table 3, it can be seen that the designed controller can track the reference signal with a small steady state error, minor overshoots and fast settling time. At the beginning of the motion of the piston, there is a slight delay due to the pressure build-up time during which the force of the air pressure has to overcome the dry friction force. Some oscillations are noticed before the piston settles and this

Table 3. Performance parameters for the tracking of square waves.

Parameter	Value
Delay time	0.08 s
Rise time tr (10 %-90 %)	0.1 s
Maximum overshoot %	9.4 %
Settling time (within 5 %)	0.6 s
Average steady-state error	0.5 mm

behavior is thought to result from the dynamics of the directional control valve. In subsequent studies, this issue will be addressed by thoroughly investigating the dynamics of the directional control valve to further optimize the overall performance.

9. Conclusion

This paper presented a novel pneumatic high speed on/off valve driven by a piezoelectric stack actuator. Its structure and working principle were illustrated.

(1) Mathematical models of the designed valve were established and simulations were carried out using MATLAB/Simulink. To validate the models, a prototype was fabricated and experiments were conducted.

(2) The valve exhibited a large flow rate capability of 86 L/min for an upstream gauge pressure of 0.5 MPa.

(3) To verify its ability to be used in position control tasks, open-loop system was built to analyze how the duty cycle of the driving PWM signal affects the velocity of the pneumatic cylinder's piston. As predicted, the higher the duty cycle, the higher is the velocity of cylinder's piston. This behavior confirmed that it was feasible to control the position of the piston by modulating the duty cycle.

(4) A closed-loop control system based on a combination of PWM signal and a PID controller was designed in order to use the designed high speed on/off valve in the control of a pneumatic cylinder. The results showed that the high speed on/off valve controlled pneumatic cylinder has good position tracking performance. The findings of this study affirmed that, as observed in hydraulic systems, high speed on/off valves are potential candidates to replace traditional servo and proportional valves in pneumatic systems as well.

In subsequent studies, an adaptive robust PID controller will be implemented to improve the tracking performance of the proposed closed-loop control system. In addition to that, the influence of the directional control valve to the overall performance will be thoroughly investigated.

Acknowledgments

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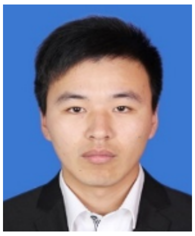

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