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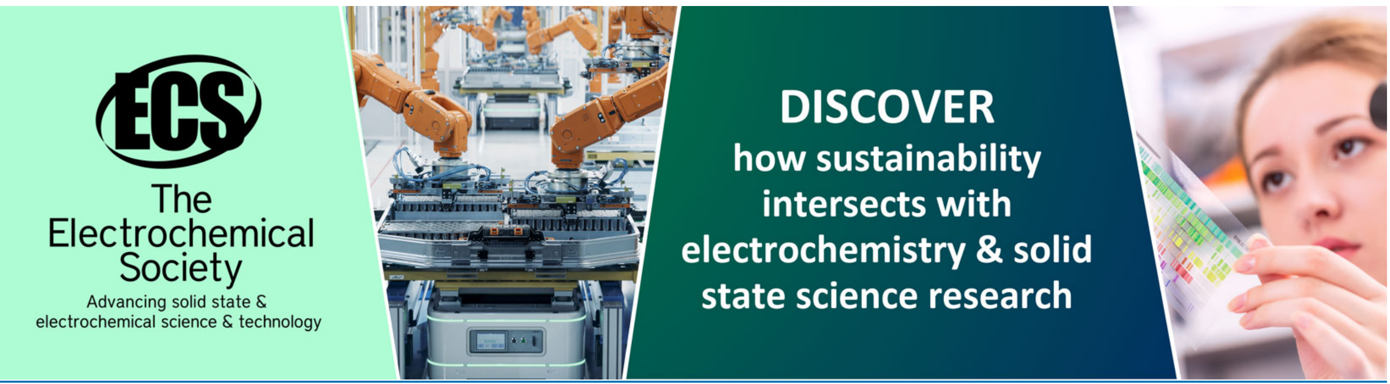

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Fast Switching Valve Utilization to Control Pneumatic Cylinder Speed

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Abstract. The position and speed of the piston of pneumatic cylinder are very important parameters in evaluating the pneumatic system performance. Utilization of Pulse Width Modulation (PWM) technique to control the Fast Switching Valve (FSV) usually causes fluctuation in the piston motion and accordingly, the system performance deteriorates. The objective of this work is to introduce a novel PWM control technique considering the suitable duty cycles according to the switching frequency in order to improve the FSV characteristics and reduce piston speed fluctuation. A test rig for a simple pneumatic circuit with an FSV as a control element has been designed and realized. The displacement of the piston is measured using a Linear Wire Potentiometer Transducer (LWPT), while its derivative is used to estimate the speed considering a low pass filter to reduce the effect of piston position signal noise. A nonlinear mathematical model is introduced and experimentally validated. The results of both computer simulation and experimental measurements show that the integration between fast switching valve and PWM control signal leads to a quasi-linear relationship between duty cycle and piston speed. Moreover; results proved the successful control of piston speed based on the correct duty cycle and valve switching frequency.

Keywords: Fast Switching Valve, Pulse Width Modulation, Pneumatic Cylinder, PWM Scheme.

1. Introduction

Pneumatic systems are widely used in robotics and industry applications. They are clean, cheap, simple in design and control, self-cooling, fast acting and low maintenance cost. This makes pneumatic systems a preferable choice for many cases compared to electro-mechanical and hydraulic systems [1]. The non-linear behavior of pneumatic systems mainly caused by air compressibility, is a major drawback of pneumatic systems. Therefore, proper control of pneumatic actuators becomes difficult to be achieved. Continuous acting Servo Valves (SV) are used to realize servo-pneumatic systems, but those valves are complex in structure, expensive (about $1000 US), and need special controllers. Many researchers developed the servo-pneumatic systems by using digital valves with fast switching electronic circuit (about $ 90 US) and driven by pulse width modulation technique instead of servo valves. This is in order to decrease system's complexity and cost [2, 3].

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PWM-Driven electro-pneumatic systems motivate many researchers to investigate and develop its characteristics in order to improve its dynamic performance. Many studies investigated and developed the modeling and control of servo-pneumatic system including force element (actuator) and command device (valve) [4, 5].

The improvement of dynamic performance of pneumatic system mainly associated with the enhancement of the pneumatic valve characteristics. M. Pipan et al. [6] studied a PWM control algorithm for a closed-loop volumetric flow of a fast switching valve on the basis of experimental results. Topcu et al. [7] investigated the characteristics of 2/2-way fast switching valves and presented a method to reduce closing time. Taghizadeh et al. [8] developed a non-linear dynamic model of 3/2-way fast switching valve (FSV) and validated it experimentally. Shiee et al. [9] used different schemes to investigate the performance of the servo-pneumatic system. Taghizadeh et al. [10] investigated the dynamics performance of PWM-Driven servo-pneumatic system and presents an open loop behavior between the input voltage and the speed of piston rod.

In this paper, a new simple method is investigated to control the speed of cylinder by using the fast switching valve. The effect of control signal pulse width on the speed of cylinder is investigated theoretically and experimentally.

2. Experimental Setup

Referring to Fig. 1, a double acting cylinder actuator (DNSU-20-100-PPV-A) from FESTO including stroke length of 100 mm, and 20 mm bore diameter is used to be under control. FESTO fast switching valve 3/2-way is used as a command device to control the air flow. A manual flow control valve is used to blank the exhaust port or connect it to the atmosphere. An NI-myRIO controller is used to control the Fast Switching Valve (FSV). The control signal of the FSV is transferred through PWM output pin of the I/O port toward the MOSFET amplification circuit. HONTKO Linear Wire Potentiometer Transducer (LWPT) is attached to the end of piston rod to measure its displacement. This signal is transferred through the analog input port of the NI-myRIO to be recorded and monitored. A reciprocating air compressor with enough air tank capacity is used to supply pressurized air of 6 bars. A service unit from FESTO (FRC-M7-D-7-5M) is used. This unit contains filtration, regulation, and lubrication stages in a single unit with pressure gauge. The air pressure is regulated through the service unit to be 5 bars for all experiments.

3. The Proposed Pneumatic Circuit

The proposed pneumatic circuit is shown in Fig. 2. As can be seen from the figure, the full diameter side chamber is connected to the output port of the valve, and the annulus side chamber is connected directly to the service unit. The input port of the valve is connected to the regulated pressure while the exhaust port is connected to the flow control valve. Energizing of the solenoid allows the pressurized air to pass through the valve toward the full side chamber which builds up the pressure inside it. The self-return spring of the valve returns the spool and maintains its normal position when the solenoid is de-energized. The higher switching operating frequency of the valve result in smoother displacement of the piston rod and bigger pulse duty cycles result in bigger packets of air. The air packets delivered to cylinder have a high rate of delivery which is faster than the dynamics of the cylinder, so that, the system filters this discreteness of air packets and piston rod responds to the average of this packets.

The extension of the piston requires to blank or to throttle the exhaust port of the fast switching valve by adjusting the flow control valve. The flow control valve is completely closed during the experiments carried out in the present work and it is only opened to retract the piston when the fast switching valve is not energized. The extension speed of the piston rod is controlled by the FSV control signal.

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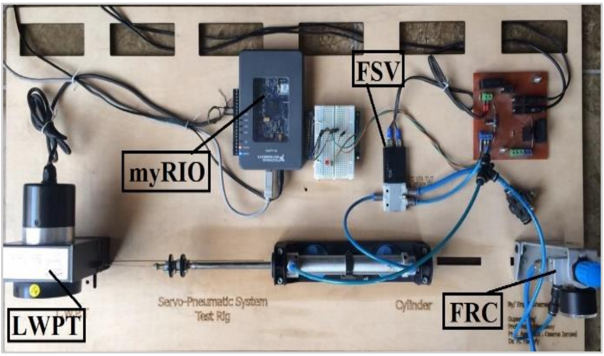

Figure 1. Photograph of Test Rig

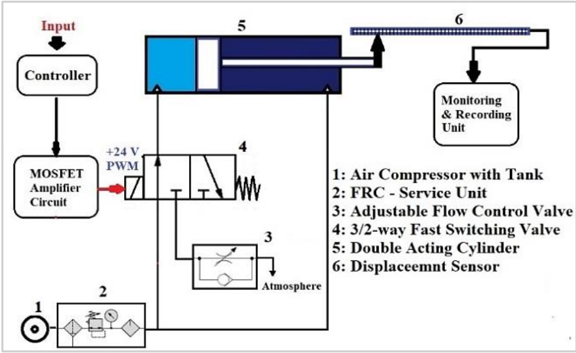

Figure 2. The Proposed electro-pneumatic circuit

4. Modeling and Simulation

In this section, the mathematical equations governing the proposed system are introduced. Moreover, the validation is carried out using the experimental setup described in section 2.

4.1 Mathematical equations

A non-linear dynamic model of the proposed pneumatic circuit is presented including the force element and the command devices. The system can be represented by the following set of differential equations:

- The equation of motion for the pneumatic cylinder:

Applying Newton's second law, the following equation can be obtained:

$$\frac{d^2x}{dt^2} = \frac{1}{m_L} \left[P_1 A_1 - P_s A_2 - b\frac{dx}{dt} \right] \tag{1}$$

where, x is the displacement of piston rod, m_L is the mass of piston and load, P_1 is the pressure inside the full side chamber, P_s is the supply pressure connected to annulus side chamber. A_1 is the full side piston area, A_2 is the annulus side piston area, and b is the cylinder viscous coefficient.

- The equation of rate of change for pressure inside full side chamber:

$$\dot{P}_1 = \frac{RT}{V} (\propto_{in} \dot{m}_{in} - \propto_{out} \dot{m}_{out}) - \propto \frac{P_1}{V} \dot{V} \tag{2}$$

where, $\propto_{in}$ *, and* $\propto_{out}$ are taking values between 1 and 1.4 depending on the actual heat transfer during the process. V is the control volume, T is the temperature of air and R is the ideal gas constant. $\dot{m}_{in}$ and $\dot{m}_{out}$ are the mass flow rate entering and leaving the chamber.

- The mass flow rate through valve orifice of area A_v can be expressed by:

$$\dot{m} = \begin{cases} C_f C_1 A_v \frac{P_u}{\sqrt{T}}, & \frac{P_d}{P_u} \leq 0.528 \\ C_f A_v C_2 \frac{P_u}{\sqrt{T}} (\frac{P_d}{P_u})^{1/k} \sqrt{1 - [\frac{Pd}{Pu}]^{\frac{(k-1)}{k}}}, & \frac{P_d}{P_u} > 0.528 \end{cases} \tag{3}$$

where, C_f is the discharge coefficient, P_u is the supply pressure, P_d is the downstream pressure, T is the temperature of supplied air (293 °K), and k is the specific heat ratio. C_1 = 0.04042, and C_2 = 0.15617 are constants of a given fluid [11]. MATLAB/SIMULINK environment is used to simulate the above dynamical equations of the system. It should be noted that the effect of cushioning at the

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annulus side of the piston has been considered by variable spring and damper coefficients, while it is neglected at the full side.

4.2 Model validation

The validation of the mathematical model for the proposed system is carried out by comparing the results of the system simulation to the corresponding measured ones. Fig. 3 shows the piston displacement obtained from both simulation output and experimental measurements considering 50% duty cycle & 60 Hz PWM valve control signal. Inspecting this figure, it can be seen that there is a very good quantitative and qualitative agreement of both simulation and experimental results (maximum peak to peak error = 4.85% of full displacement stroke).

In this work, the piston speed is estimated from its displacement, it could not be obtained from the direct differentiation of the displacement signal as it was too noisy. Therefore, Digital low pass filter is used to alleviate this problem. Fig.4 shows the piston speed obtained from both simulation output and experimental measurements at the same duty cycle and frequency (50% duty cycle & 60 Hz PWM valve control signal).

It is clear from the figure that there is a good qualitative agreement between both simulation and experimental results. However; there is still a noticeable but acceptable oscillation in the measurement results which, as mentioned before, is due to the fact that the speed is estimated from the measured displacement. Accordingly, to get more accurate results, it is recommended to measure the speed directly using a speed sensor such as a linear encoder.

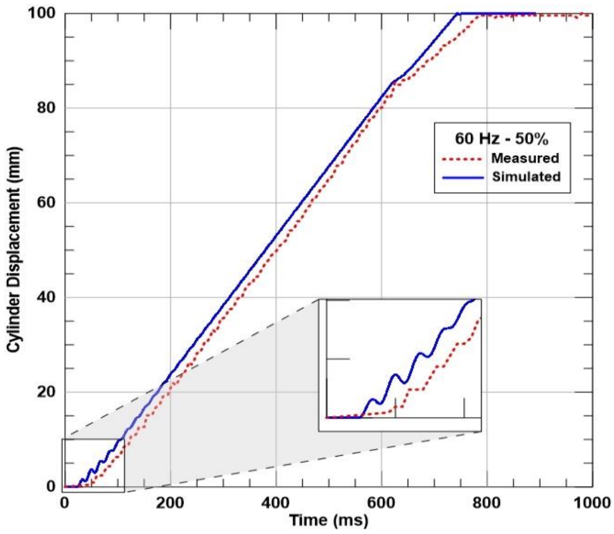

Figure 3. Cylinder response to 60 Hz plain PWM valve control signal.

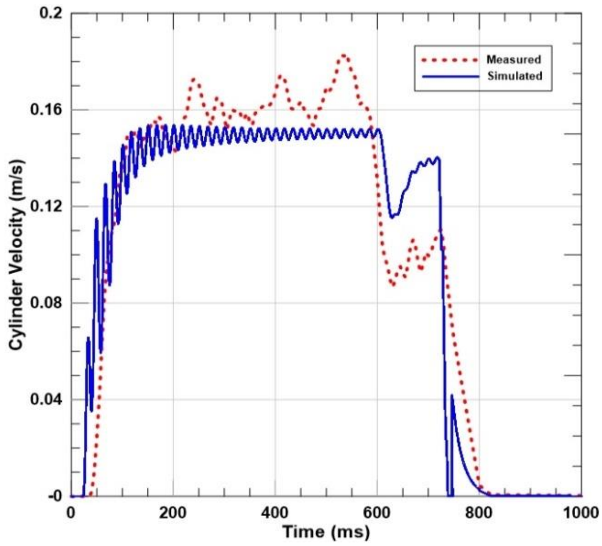

Figure 4. Cylinder velocity under the effect of 60 Hz PWM control signal with 50% duty cycle

5. PWM Scheme

In general, the fast switching valve has a delay opening time (TON) and a delay closing time (TOF). The values of TON & TOF depend on the manufacturing tolerance and the electromagnetic characteristics of the valve solenoid. The existence of those delay times causes a dead band and a saturation band. Accordingly; the valve cannot completely open with duty cycles less than a minimum (DC_{Min}) which corresponds to pulse ON time enough to compensate for the ON delay time. Also, the valve is saturated with duty cycles bigger than a maximum (DC_{Max}) which corresponds to pulse OFF time enough to pass the OFF-delay time. The values of the DC_{Min}, and the DC_{Max} depend on the control signal frequency and the delay times of the valve. Accordingly, the minimum and maximum duty cycle can be given as follows:

$$\mathrm{DC_{Min}} = \frac{\mathrm{TON}}{\mathrm{T_{PWM}}} \quad , \quad \mathrm{DC_{Max}} = \frac{\mathrm{T_{PWM}} - \mathrm{TOF}}{\mathrm{T_{PWM}}} \tag{4}$$

Where, TPWM, TON, & TOF are the valve switching period time, delay opening time & delay closing time; respectively. According to the manufacturer, the maximum switching frequency for this FSV is 330 Hz but the ambient temperature must be limited with switching frequencies in excess of 125 Hz. Also, this valve has a delay opening time, TON=1.7 ms, and a delay closing time, TOF=2 ms. Considering equation 4, DC_{Min} & DC_{Max} are calculated and Table 1 shows the maximum and the minimum duty cycles for the FSV operating frequencies used through this study.

6. Open Loop System Behavior

The open loop behavior of the proposed system is investigated by simulating the model with several duty cycles at different frequencies considering constant pressure supply of 5 bars. Using the developed validated simulation model, the following are to be discussed:

- The pressure in the piston full diameter side chamber and the valve spool position.
- The piston speed time response.

6.1 The pressure in the piston full diameter side and the valve spool position

Fig. 5 shows the relationship between the pressure inside the full side chamber, and the displacement of valves' spool under the effect of 50% duty cycle & 60 Hz PWM control signal.

The PWM valve control signal is applied to start switching of the valve at t = 0 which causes pressure to build up inside the full piston diameter side chamber. After the pressure reaches certain value which overcomes the opposing forces, the piston starts to move causing pressure drop. The new air packet compensates for the pressure drop and it builds up again causing the piston to continue motion.

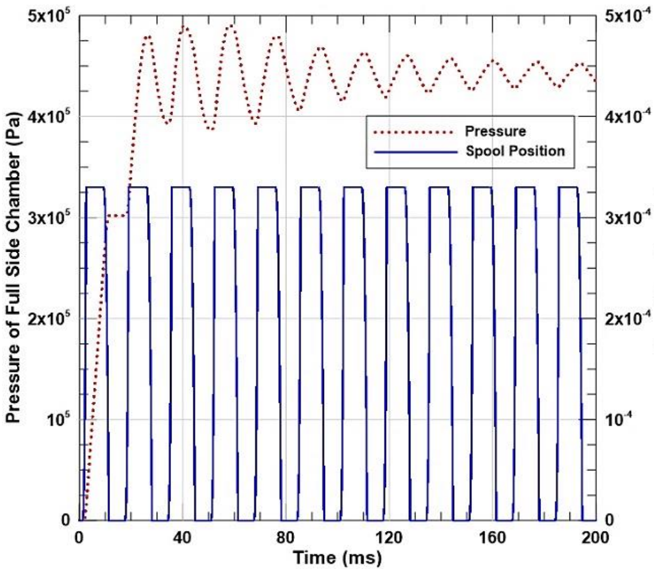

Figure 5. The relation between the pressure inside the full side chamber and the position of spool.

Table 1. limits of duty cycles

Frequency	Min Duty Cycle	Max Duty Cycle
60 Hz	0.1	0.88
80 Hz	0.136	0.84
100 Hz	0.17	0.8

6.2 The piston speed time response

Examples for piston speed time response are shown in Fig. 6 for valve switching frequencies of 60 Hz. The results shown in Fig. 6 indicate the following:

- The stroke of cylinder can be described by three sequential stages, namely: dead zone, fluctuation zone, and smooth motion zone. Both of experimental and simulated results indicated that the larger duty cycles result in smaller dead zone while the higher PWM frequencies and cushioning result in smaller fluctuation zone and larger smooth zone.

- The speed of cylinder can be controlled by the pulse width of the valve control signal.
- The switching frequency of FSV limits the maximum and minimum duty cycles in order to avoid both valve saturation condition (the valve remains completely open) and the unpredicted long dead band.
- In the conditions considered in Fig. 6; as duty cycle increases, the cylinder speed increases till DC_{Max}. For duty cycles above the DC_{Max}, the speed can be considered the same (90 % & 95% duty cycles).
- The use of higher switching operating frequency for the FSV results in a smoother motion of the piston rod.

The cylinder speed versus the control signal duty cycle for three different valve operating frequencies (60 Hz, 80 Hz, and 100 Hz) is plotted in Fig. 7 from which the following can be noticed:

- The input-output relationship is a quasi-linear one.
- The use of higher operating frequency decreases the range between the maximum and the minimum duty cycle (increases DC_{MIN} & decreases DC_{MAX}) which is in agreement with equation 4.
- The piston speed is reduced with the use of the 10% duty cycle of both 80 Hz & and 100 PWM control signal compared to the same duty cycle of 60 Hz PWM control signal as it lies below the minimum limits of duty cycle for both frequencies.

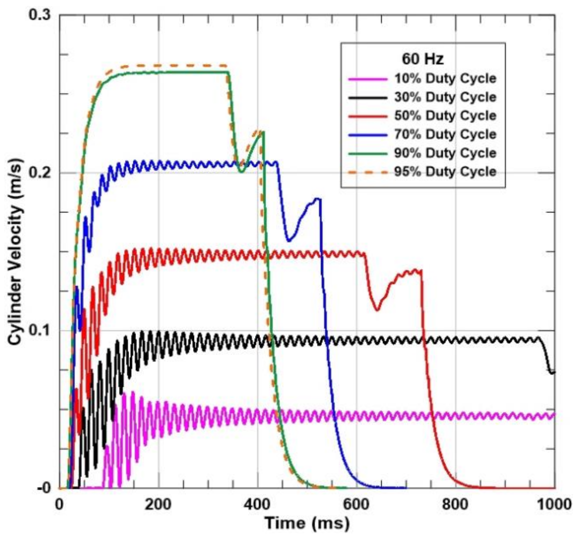

Figure 6. The velocity of simulated cylinder under the effect of 60 Hz PWM control signal.

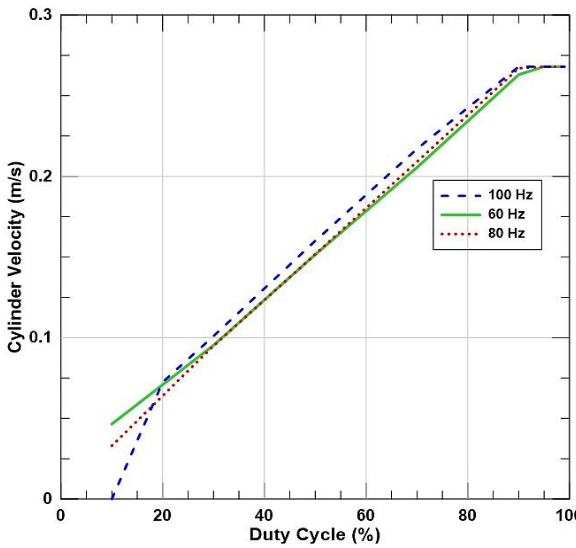

Figure 7. The quasi-linear input-output behavior

7. Concluding Remarks

In this work, a reliable test rig for simple pneumatic circuit including an FSV as a command element has been designed and built. Also, a mathematical model has been introduced and validated using this test rig. This is to investigate the usage of PWM in controlling a fast switching valve as a part of an electro- pneumatic system. From both experimental and simulation results obtained; it can be concluded that a) utilization of PWM technique to control the fast switching valve reduces the non-linearity behavior of the valve, b) the fast switching valve can be used effectively to control the speed of pneumatic actuators, c) direct differentiation of displacement signal to obtain speed signal may not give correct results due to the displacement signal noise and low pass filter may be implemented to

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alleviate this problem, d) the bigger duty cycles result in higher speed of piston and smaller dead zone, and e) a simple algorithm can be used to avoid both dead band and saturation.

8. Future Works

Investigate, theoretically and experimentally, the optimal operating frequency of the 3/2-way fast switching valve for controlling a servo-pneumatic system.

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