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Modelling and Controller Design of Electro-Pneumatic Actuator Based on PWM

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ABSTRACT

In this paper, a nonlinear model associated to the fast switching on-off solenoid valve and pneumatic cylinder is dynamically presented. Furthermore, the electrical, magnetic, mechanical, and fluid sub-systems are studied. Two common control policies to track valve position, a proportional integrator (PI) based on pulse width modulation (PWM) and hysteresis controllers, are investigated. The control cylinder position is simulated usinga programmable logic controller (PLC), andan experimental setup regulated with AVR microcontroller isaccomplished. Simulationand experimental results verified proper performance of the proposed controller.

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1. INTRODUCTION

Electro-pneumatic control valves used to control and convect the air flow, are categorized upon two types. First type is servo valves, with high control accuracy and linear behavior yet expensive and with complex structure. Second one is the fast switching on-off solenoid valve with simple structure and reasonable cost, but presenting the intrinsic nonlinear performance. Thus, PWM is availed to linearize this valve type to behave as similar as a servo valve. Another motivation to PWM is the air compressibility. Once the pneumatic excitation is used, the system is sluggish with too much delay [1], [2]. Hence, in this study, instead of pure pneumatic system, electro-pneumatic is presented to improve the performance. PWM signal as an input, causes the valve to fluctuate between open and closed states to pass the air through the valve, and if needed, be transmitted into the cylinder. Valves energized by PWM, have comprehensive industrial applications, such as electro-pneumatic brake and robotics [1]. To obtain the dynamic characteristic and tracking the desired output, an input-output coordination is analyzed [2]. To control cylinder position, PLC has high reliability, because it is able to simulate the system as many as needed and hence, saving the time and reducing the risk of mistake. In addation, it is an adaptive and robust system[4]. However, for the system under study, PLC is not a cost-effective controller, hence, an AVR microcontroller is employedinto experimental setup.

This paper is organized as follows: In Section 2, the dynamic model of system is presented, considering electrical-magnetic, fluid and mechanical subsystems along with cylinder model. Control valve strategies based on PI controller with PWM and hysteresis method, are performed and compared, in Section 3. Section 4 is dedicated to simulation of system including PLC controller. Finally, an AVR embedded on experimental setup unit are implemented and validated in Section 5. Modelled system and experimental setup contain two 3-2 valves and double-acting cylinder with a rod.

2. MODELING SOLENOID VALVE

Valve model consists of four subsystems (Fig. 1). Electrical and magnetic blocks institute voltage-force together. In mechanical modelplunger movement within valve and in fluid model air flow passing through the valve are considered.

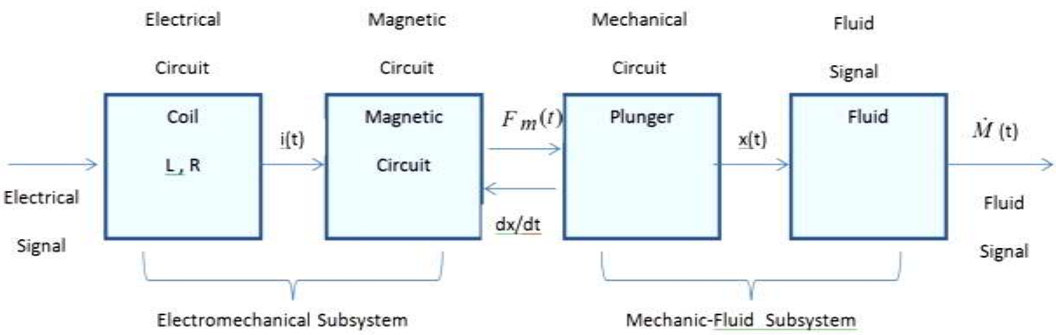

Figure 1. Overview of solenoid valve block diagram.

2.1. Electrical model

This model represents the current passing through the coil within valve *(i)*, in terms of the applied voltage (U_{PWM}) and the plunger position (X_P) and plunger speed ($\dot{X}_p$). Considering the series topology of resistance and inductance, corresponding equations are obtained as follows [1], [2]:

$$x(t) = X_p, \tag{1}$$

$$U_{PWM} = R_c i(t) + V_L(t) = R_c i(t) + N\frac{d\varphi(t)}{dt} = R_c i(t) + \frac{d}{dt}(L(t)i(t)), \tag{2}$$

$$\mu_c = \mu_0 \mu_r, \tag{3}$$

$$\frac{d}{dt}i(t) = \frac{(U_{PWM} - R_c i(t))(2\mu_r(x_t - x(t)) + l_c)}{N^2 \mu_c Ae} - \frac{2\mu_r i(t)\frac{d}{dt}x(t)}{2\mu_r(x_t - x(t)) + l_c} \tag{4}$$

where, U_{PWM} applied voltage, $\varphi(t)$ magnetic flux, *L(t)* inductance of the magnetic circuit, μ_c, μ_0 core permeability and air's, μ_r relative permeability, *N* coil turns, l_c effective length of magnetic circuit inside the core, *Ae* cross-section flux andX_p the overall air gap including constant and variable values. U_{PWM} voltage, in periodic PWM signal in terms of duty cycle (D) per a period (T) is developed as follows:

$$U_{PWM} = \begin{cases} high & t \le DT \\ low & DT < t < T \end{cases} \tag{5}$$

2.2. Magnetic model

This model considers magnetic force applied into the plunger (F_m) to open the air path in terms of the current *(i)* and the plunger position (X_P). Magnetic circuit comprises of the fixed core with a coil and plunger, which is moving by magnetic force (Fig. 2)[1]. For this circuit magnetic force is obtained using coenergy theory as below:

$$W = \frac{1}{2}L(t)\,i(t)^2 \tag{6}$$

$$F_m = \frac{\partial W}{\partial x(t)} = \frac{N^2 \mu_0 \mu_r^2 Ae}{(2\mu_r(x_t - x(t)) + l_c)^2}\,i(t)^2 \tag{7}$$

where$W$ is the work done in magnetic circuit.

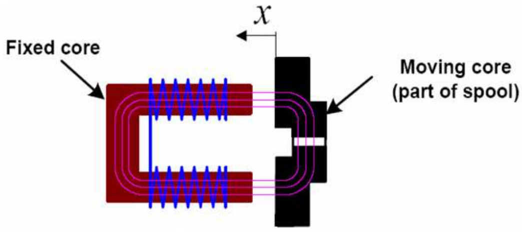

Figure 2. Magnetic circuit[1]

2.3. Mechanical model

This representation establishes the plunger position (X_P) in terms of the magnetic force (F_m) and the air pressure force (F_{prs}). Dynamic equations are achieved as:

$$m_p \ddot{x}(t) = F_{prs} + F_m - Kx(t) - F_{pld} - b\dot{x}(t) \tag{8}$$

$$F_{prs} = (a_1 - a_2) P_{sup} \tag{9}$$

where, m_p plunger mass within valve, k valve spring constant, F_{pld}spring preload, b damping coefficient, a_1, a_2 plunger cross sections andP_{sup} pressure into the valve.

2.4. Fluid model

This model presents mass flow rate passing through the valve ($\dot{m}$) in terms of the plunger position (X_P) and the air pressure force ($Fprs$). Plunger movement provides the required mass flow rate to control the cylinder.

Electromagnetic and mechanical components of the valve are used to control the fluid flow through the valve orifice using the plunger position. Turbulent air flow passing the orifice is modelled by Orifice equation (10) (Fig. 3)[5]. This relation is composed of two different equations for subsonic and sonic flow regimes [4], [7]. As the fluid inside the pneumatic system is compressible, once the ratioof pressure $P_r$is more than critical ratioof pressure P_{cr} , the regime is infrasonic, where the mass flow is nonlinear in terms of upstream and downstream pressure. Otherwise, it is in sonic regime, i.e., mass flow is linear as a function of the upper pressure. In both cases, mass flow behaves linearly in terms of the plunger position.

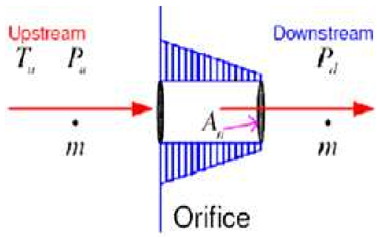

Figure 3.Flow through the orifice[8]

Standard equations to the mass flow passing the orifice are described as:

$$\frac{dm}{dt} = \dot{m} = \frac{c_d c_A A_v P_u}{\sqrt{R T_{air}}} f(P_r) \tag{10}$$

$$P_r = \frac{P_{down}}{P_{up}} \tag{11}$$

$$f(P_r) = \begin{cases} \sqrt{\gamma (\frac{2}{\gamma+1})^{\frac{\gamma+1}{\gamma-1}}} & P_r \le P_{cr} \\ \sqrt{\frac{2\gamma}{\gamma-1}((P_r)^{\frac{2}{\gamma}} - (P_r)^{\frac{\gamma+1}{\gamma}})} & P_r > P_{cr} \end{cases} \tag{12}$$

$$P_{cr} = (\frac{2}{\gamma+1})^{\frac{\gamma}{\gamma-1}} = 0.528 \tag{13}$$

where, cd discharge coefficient, P_{up}, P_{down} downstream and upstream pressure of the valve (Input and output), γ specific heat ratio for air, equals to 1.4, R , general gas constant 287 $\frac{m^2}{s^{2\circ}k}$, T_{air} system temperature, c_A constant coefficient designated to orifice cross sectional area, A_V orifice cross sectional area, P_{cr} critical pressure inside the orifice as the speed equals to the sonic speed and Mach number to unity.

2.5. Cylinder model

Equations governed by the pressure dynamics in the cylinder chambers, can be obtained through the thermodynamic analysis based upon energy conservation and continuity law. Using the ideal gas law:

$$P_i V_i = m_i R T_i, \tag{14}$$

where, i=1,2 denotes indices to two chests through the cylinder piston, P_i, V_i, T_i and m_i are pressure, volume, temperature and the mass on both piston sides. Deriving and incorporating (14) into the energy equation and the heat convection term, differential equation (15) yields. This is also obtainable by deriving $PV^{\gamma} = cte$ and inserting into (14) [2], [2], [9].

$$\frac{dP_i}{dt} = \frac{\gamma RT}{V_i}\frac{dm_i}{dt} - \frac{\gamma P_i}{V_i}\frac{dV_i}{dt} \tag{15}$$

Choosing the displacement coordinates origin at the stroke beginning, volume of each chamber can be written as:

$$V_1 = V_{01} + AV_1 = V_{01} + A_1 y, \tag{16}$$

$$V_2 = V_{02} + A_1(L - y), \tag{17}$$

where, L cylinder course, y displacement of the cylinder piston, V_{01}, V_{02} dead volume in the first and second chest and A_1 and $A_2$are cross sectional areas of the first and second chest.
Inserting (16) into (15), equations describing the pressure dynamics is developed as[2], [10], [11] :

$$\frac{dP_1}{dt} = \frac{\gamma RT}{V_{01} + A_1 y}\frac{dm_1}{dt} - \frac{\gamma P_1 A_1}{V_{01} + A_1 y}\frac{dy}{dt}, \tag{18}$$

$$\frac{dP_2}{dt} = \frac{\gamma RT}{V_{02} + A_2(L-y)}\frac{dm_2}{dt} + \frac{\gamma P_2 A_2}{V_{02} + A_2(L-y)}\frac{dy}{dt}, . \tag{19}$$

According to second Newton's law for the cylinder piston, movement equation turns out to be:

$$M\ddot{y} = P_1 A_1 - P_2 A_2 - B\dot{y} - F_f - P_{atm} A_{rod}, \tag{20}$$

where, M, piston mass, B, damping coefficient, F_f, friction force, P_{atm} atmosphere pressure applied into the cylinder rod and A_{rod} denotes cross section of the rod connected to the piston.

3. CONTROLLER DESIGN

System requirements are indicated as follows [1], [2]. Energizing the valve with input voltage at t=0 sec, the current is rising exponentially due to R-L nature until it meets the desired value, the magnetic force overcome the spring force and the valve plunger staring to move. It takes some time to increase within this limit resulted from the system delay. After the valve is completely opened, it requires the current to keep the valve opened, being less the required current to move the plunger namely as holding current while prime current is called switching current. To close the valve, it is necessary to fetch the electrical current off the coil. To reduce time to close the valve, at the beginning of energizing process, negative voltage must be fed. Valve control circuit is comprised of a regulator to maintain the holding current and a proposed system called Sequence Generator (*SG*) to accomplish fetching the current off solenoid and a switch to fluctuate between these two modes, as shown in Fig. 3. *SG* is composed of a second order low pass filter as (22).

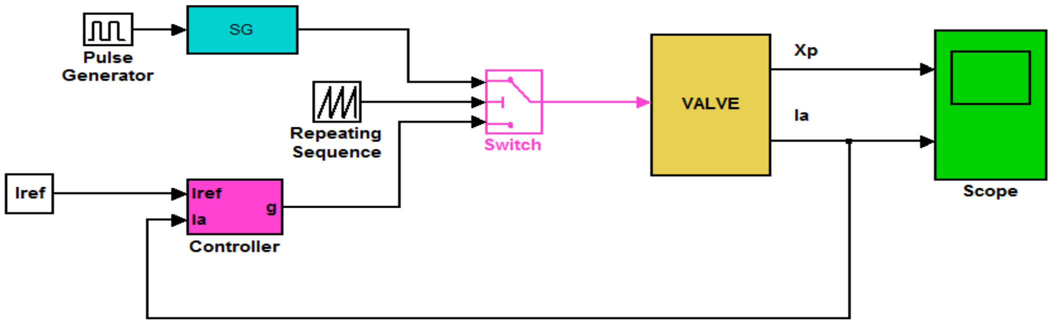

Figure 3. Control valve blockset.

$$H(s) = \frac{1}{\tau s + 1} \tag{22}$$

To implement this filter, an integral loop with its time constant is employed. To have this sequential signal periodically, the mechanism performsin each period. This task is done through reset signal to the filter integrator with the frequency as equal as valve's. Altogether, to achieve aforesaid signal, it is adequate to apply a step signal as an input. Note that the time constant must reasonably be chosen low so that before one period ends, this signal is completely settled down. Fig. 4 demonstrates low pass filter designed to simulate extracting the current off coil.

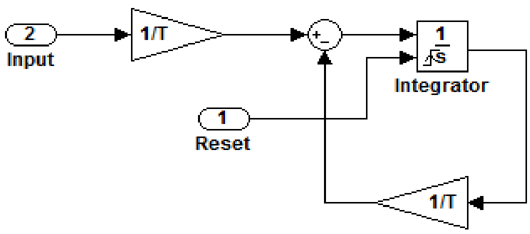

Figure 4. Implemented low pass filter.

3.1. Holding Current Regulator

To track the holding current, two commonly used regulators are hysteresis modulation and PI controller based on PWM controller **Error! Reference source not found.**.

3.1.1. Hysteresis controller

In this method, reference signal is compared to the measured signal. Once measured value is more than the reference, it must be switched on to arise the current; in other hand, when the current is less the desired value, like the previous state, command off is applied to drop off the current. Hence, the current is fluctuating around the reference current persistently. To prevent the valve from big fluctuating, resulted from significant sensitivity to the exact value, a lower and upper bounds are defined, where no signal change is made as:

$$g = \begin{cases} 1 & i_a \le i_{ref} + \frac{h}{2} \\ 0 & i_a \ge i_{ref} - \frac{h}{2} \\ no\ change & |i_a| < \frac{h}{2} \end{cases} \tag{23}$$

where, i_a, i_{ref} denote measured and desired current, h, hysteresis bounds and g output signal into solenoid. This is implementable using a relay with given upper and lower bounds (Fig. 5). Simulation results regarding to hysteresis controller is efficiently obtained as Fig. 6.

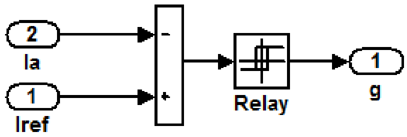

Figure 5. Hysteresis block diagram.

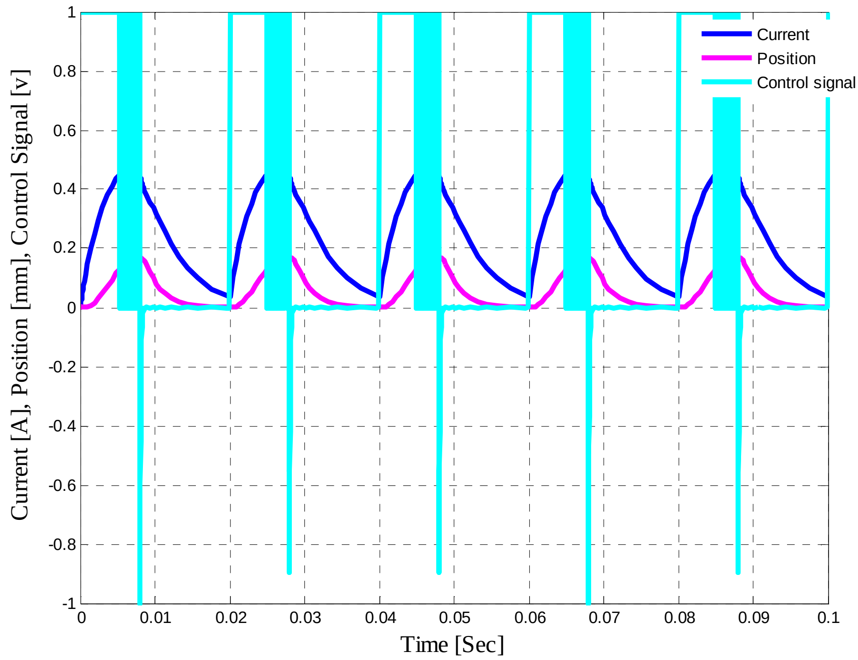

Figure 6.Hysteresis simulation results.

3.1.2. PI controller based on PWM

This approach is compatible with PWM where a control signal is compared to a saw-tooth signal with the valve frequency (Fig. 7).

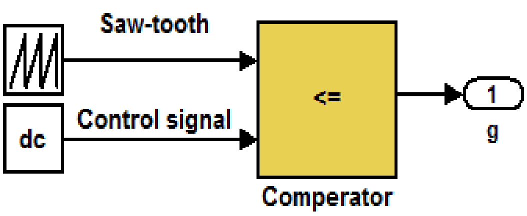

Figure 6. PWM block diagram.

PI controller has been availed in pneumatic systems [1], where it has merit to reduce the error between the desire and measured value resulted from the integrator nature [12]. On the other hand, the sensitivity to noise is eradicated in the better way in comparison to other principles [12]. The most challenging problem with PI design is how to tune the controller coefficients to present as efficient performance as possible. Couple of tuning methods are introduced in [13]. To present comparison different tunning methods, a performance index is considered to show the efficient performance. Integral absolute error (*IAE*) has the reasonable view rather than other indices [12] as follows:

$$J = \int_0^{\infty} |E| dt \tag{23}$$

But these approaches suffer from poor performance when the system nonlinearity is significantly extreme,same as the plant under study. Thus there must be a methodology to guarantee the proper performance irrespective from the policies based on purely mathematical analysis. Evolutionary optimization algorithms are preferred to other optimization techniques due to robust performance compatible with non-explicit functions, while with analytical ones, difficult to implement this type of optimization. Genetic algorithm (*GA*) is employed to establish optimal parameters. A simulation is ran with a step constant reference, as showninFig. 7. The step input contains the broad frequency spectrum [12]. Therefore, if there is a proper system response, the system has acceptable reponse for any input, such as PWM signal, regardless of its form thud. To implement PI over the system, a saturation unit is used to maintain the duty cycle (*dc*) on the interval [0 1]. Parameters of GA simulation is given in Table 1 . Simulation results based upon *GA*, verify the effectiveness of the proposed strategy (Fig. 8).

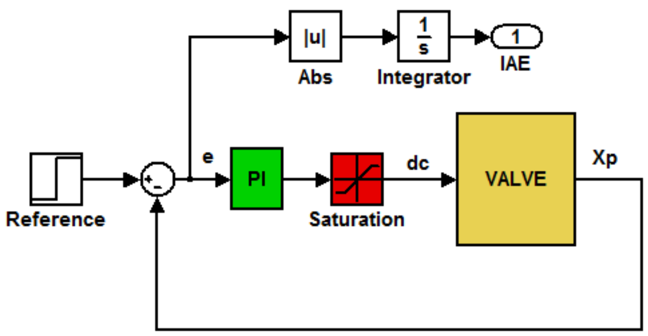

Figure 7. PI design based on GA.

Table 1. GA Parameters

Population	20
Stop criteria	150
Fitness scaling	Rank
Selection function	Roulette
Crossover function	Heuristic
Crossover rate	0.6
Mutation function	Uniform
Mutation rate	0.03

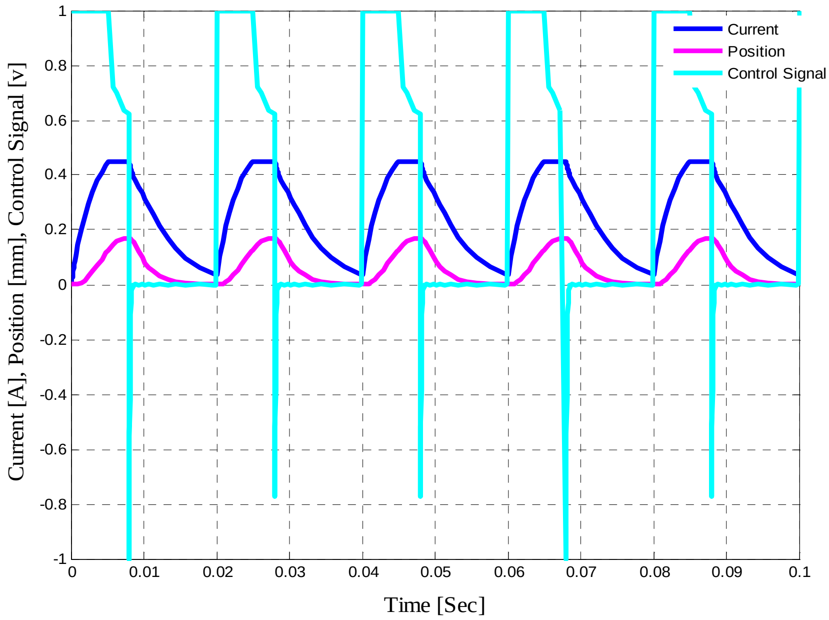

Figure 8. Simulation results from PI controller based upon GA.

3.2. Comparison between PI and hysteresis

From dynamic response point if view, there is no considerable difference between *PI* and hysteresis controller. *PI* Regarding to switching frequency have applied into the valve effecting significantly on its deterioration level, less frequency, longer life time. Hence, *PI* controller with constant switching frequency of *PWM* rather than hysteresis controller, with variable and high frequency, causes valve life time to be longer. Therefore, *PI* controller is preferred to hysteresis to feed valve.

4. PLC CONTROL

Automation systems applicable in electro-pneumatic technology, composed of main three components: actuators or motors, sensors or buttons, and control devices [4]. A double acting cylinder with two 3-2, valve is simulated and controlled via *PLC* (Fig. 9) [15], [16]. Note that, length tubes is high enough to result in high pressure dip [4].Sensors and buttons as the control input and air control valves as the controller output was designed (Fig. 10) using *LADDER* programming [17]. Using a_0 and a_1 proximity cylinder sensors, a command is sent to a_{10} and a_{20} as set and reset buttons, respectively to open the air path.

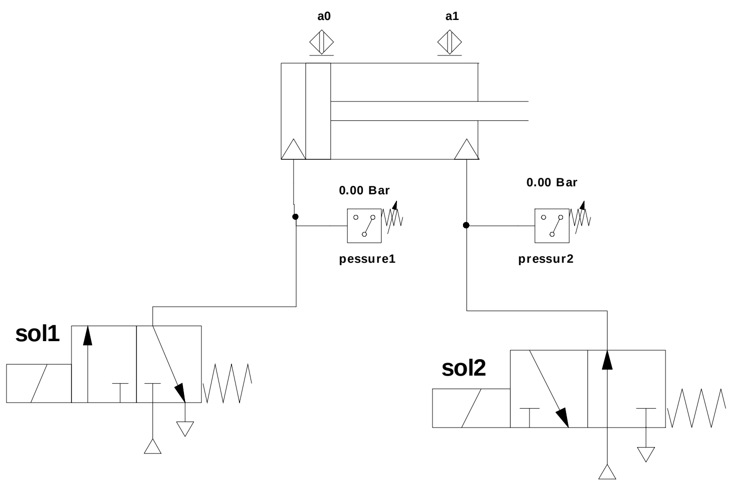

Figure 9.Overall schematic of the system.

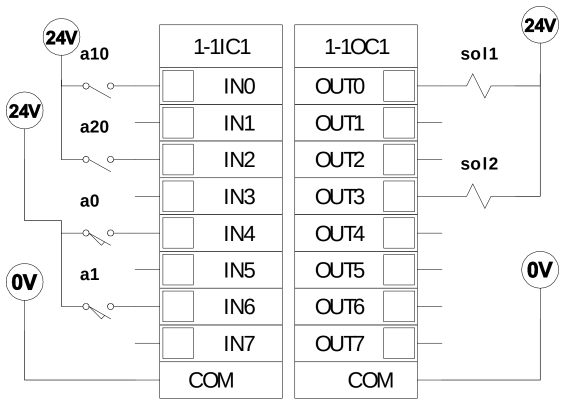

Figure10. Input-output card of PLC.

Appling *PWM* signal into the solenoid of both valves by the timer, the cylinder piston has been controlled. Furthermore, it provides various signals to each valve to change forward-backward mode for a cycle. Control procedures are organized as follows. Position sensor reads the cylinder position and sends it to controller input. Then, microprocessor performs necessary analyses and commands solenoids to open the air path and move the plunger to track the desire position. Initially, a PWMsignal with dc= %50 was applied, then timers change this signal to control the position for 2^{sec} with keep opening by sol1 and closing by sol2 (Fig. 11).

Pressure sensors connected to two cylinder input ports, sense pressures behind them and logged. Once the coil is energized, magnetic force causes the plunger movement and open the air stream. Afterwards, in a bit time, cylinder pressure is increased that results in piston sliding (Fig. 12). Less pressure variations cannot move the piston in light of the air compressibility and the friction force. Later on, plunger moves back, exhaust valve path is opened and the pressure behind cylinder is exhausted to the atmosphere [17]. This procedure follows PWM signal periodically. Parameters of the simulated system are given in Table 2.

Table 2. PLC simulation parameters

Friction force [N]	damping coefficient [Ns/m]	Exhaust pressure [bar]	Supply pressure [bar]	Piston mass [kg]	Stroke [cm]	Piston diameter [cm]
7	0.01	1	6	0.5	12.5	5

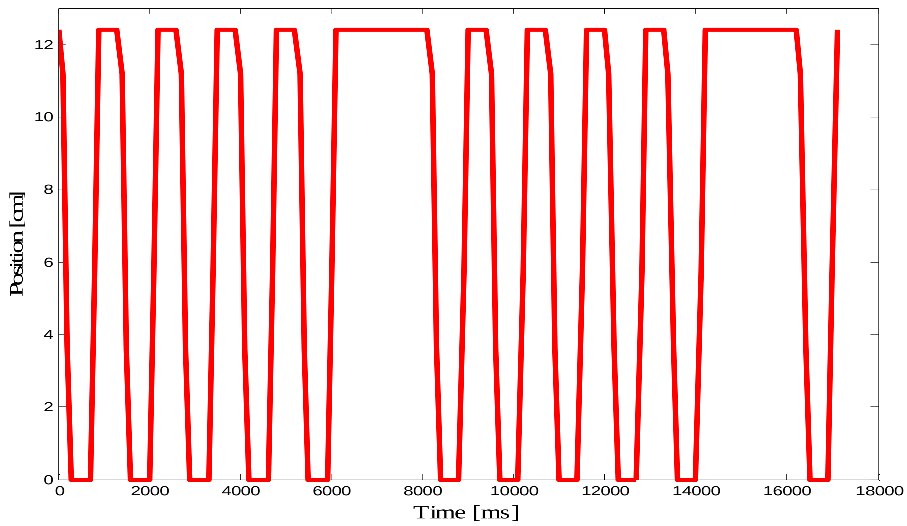

Figure 11.position piston cylinder for dc=50% with 2 seconds delay.

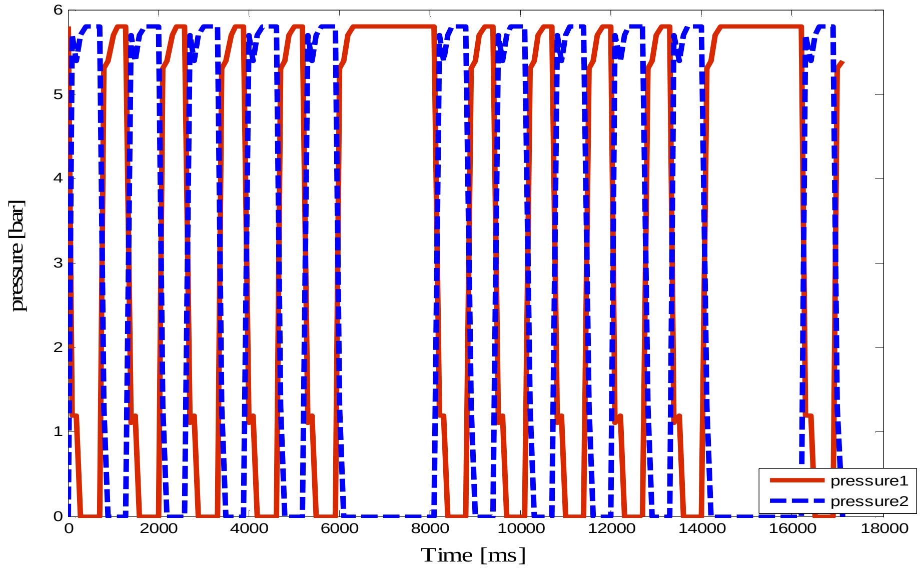

Figure 12. Pressures P_1 , P_2 for dc=50% with 2 seconds delay.

5. EXPERIMENTAL SETUP

The system under control is a double-acting single-rod cylinder with 125 [mm]. Two fast switching 3-2 way valve is used to control the air flow. The air compressor gears up the compressed air to drive the system. Pressure regulator adjusts the pressure behind the valve. An AVR controller is used to control the valve position. A drive circuit of the valve is used to boost the required current for the valve. A position sensor is employed to measure the displacement of the cylinder piston. Experiments are conducted under a pressure of 6 bars. A photograph of the experimental setup is depicted in Fig. 13.

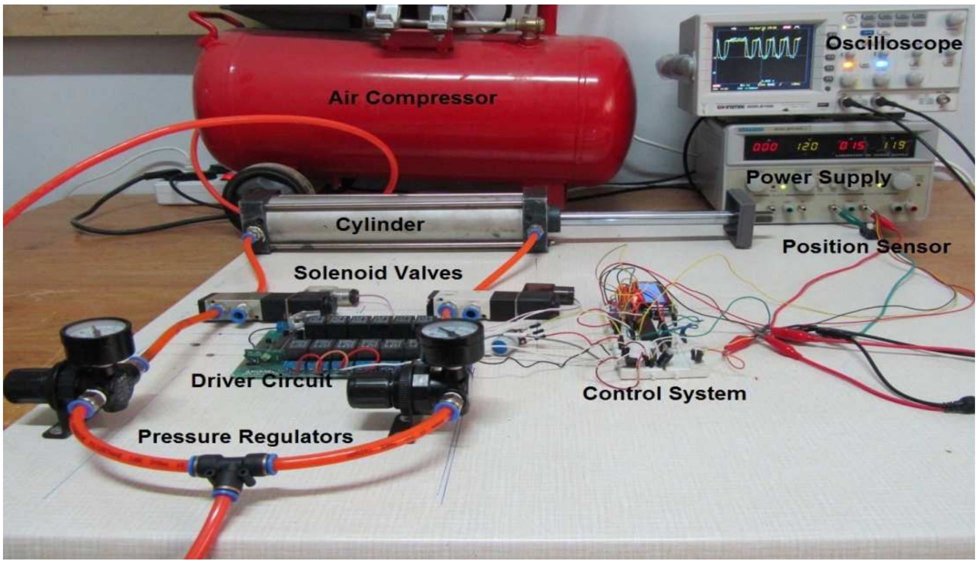

Figure 13. Experimental setup

As PLC implementation is not cost-effective, analogous to the control policy done in PLC, an AVR microcontroller is used in the experimental setup. In AVR system, applying four PWM signals(with dc=50% with 2 seconds delay) to the valve driver, the cylinder piston positionis appropriately controlled. Simulation results of PLC controller are adaptively validated to the experimental results using AVR, as shown in Fig. 14.It is pointed out that cylinder stroke has set to 12.5 ^{cm} by the position sensor.

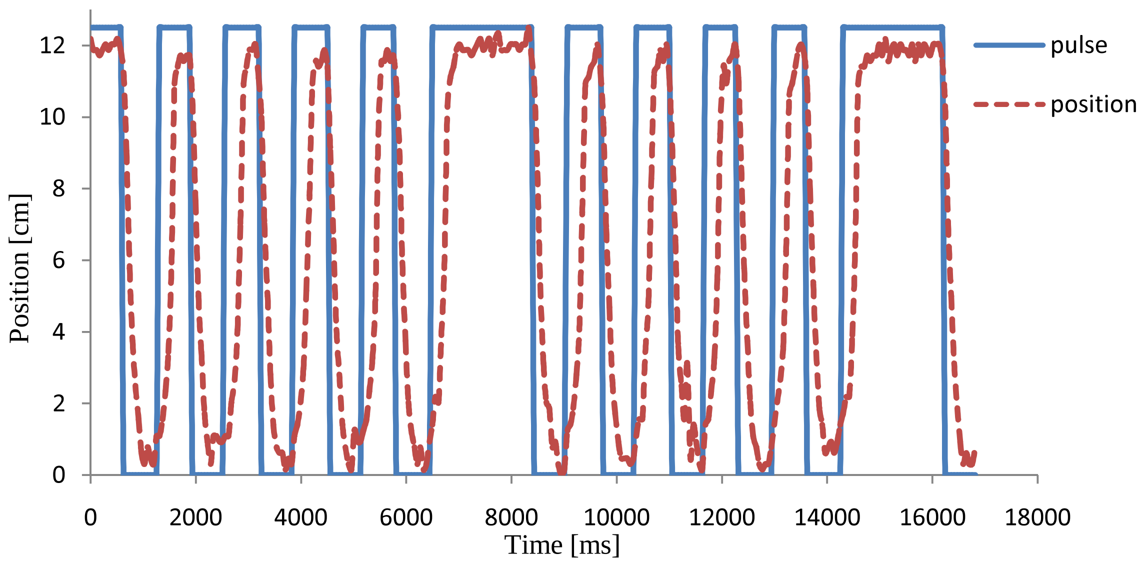

Figure 14. Position piston cylinder and PWM command in experimental setup for dc=50% with 2 seconds delay.

6. CONCLUSION

In this study, dynamic model of an electrical-pneumatic system including electrical, magnetic, fluid, and mechanical subsystems is presented. To control valve plunger position, methodologies of PI controller and hysteresis modulation are implemented. PI controller with constant switching frequency is preferred to hysteresis containing variable high frequency, because it results in longer life time of the valve. Finally, to regulate the cylinder position, a PLC controller based on the simulated plant and an AVR microcontroller incorporated into the experimental setup are conducted. PLC simulation result is adaptively validated to the experimental setup using AVR.

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