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OPTIMAL POSITION CONTROL OF A SERVO-PNEUMATIC SYSTEM USING GENETIC ALGORITHM AND PULSE WIDTH MODULATION WITH SLIDING MODE CONTROL

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Abstract. Speed, low production cost and clarity are of main importance.in developing mechanical equipment which deal with humans such as rehabilitation systems. Therefore, pneumatic systems are suitable systems in this field. However, controlling these systems can be a challenging issue, due to the nonlinearities that arose in these systems: For instance, friction, fluid compression and delays of valves performances. various methods are available to design a controller for pneumatic systems. Varseyeld and Bone [1] designed a discrete PID controller using a model which had been obtained by identification methods. They used their proposed controller to control the position of a servo-pneumatic operator, then, they examined the controller performance for different inputs. the obtained results of their work lacked accuracy because of two reasons: the inaccuracy of their mathematical model and using a PID controller. In [1] solenoid on/off switches have been used. One of the methods of transferring the control signals to command for such valves is to use a PWD algorithm which Ahn and Yokota used in their work [2]. They controlled the position of a pneumatic operator using a smart switching controller via on/off solenoid switches. In this method and ones used later on, inexactness in modeling and ignoring some parts of the model causes inaccuracy in trajectory following. Moreover, the proposed controllers did not have a satisfactory precision as well. Since exact modelling of pneumatic system are complicated and approximations are always within such systems, it makes sense to use robust control methods. For all the aforementioned reasons, in this paper, a sliding mode controller is used to follow a trajectory. After applying an optimization on the controller, its performance is compared with a PID controller. Using a pulse width modulation algorithm, differential equations of a double sided single action operator and solenoid valves, the pneumatic mathematical equations are simulated in MATLAB/Simulink computer software. Afterwards, a PID controller and a sliding mode controller are designed for the trajectory following task. In order to get the best performance of the designed controllers they are then optimized by genetic algorithm. Finally, the simulated results are compared and advantages/disadvantages of each controller are discussed.

Keywords: Sliding mode control, position control, pulse width modulation.
, genetic algorithm, servoneumatic systems

1. Modelling the pneumatic system

As shown in Figure 1, the pneumatic system in this paper, has a pneumatic operator and two on/off solenoid valves. The pneumatic operator is a double sided one and the valves are electrically stimulated. A pulse width modulation has been used to transfer the input control signal based on the valves task and the time they are performing. The governing

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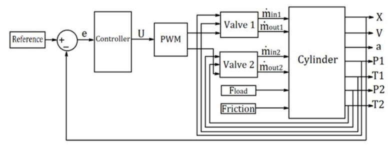

Figure 1. Pneumatic system modeled in MATLAB Simulink

equations of the operator and valves are presented in this paper. Dynamic equations of the operative piston are derived regarding to the pressure difference in the double sided cylinder and viscose-coulomb friction considering cohesion effect as well. The fluid in the cylinder has been considered to be ideal gas. Thermodynamic processes are considered as adiabatic process because it happens rapidly and there and little energy is transferred with the environment. Figure 2. Represents the free body diagram of the servo-pneumatic operator. As shown in Figure 2. The dynamic equations of the pneumatic system are as

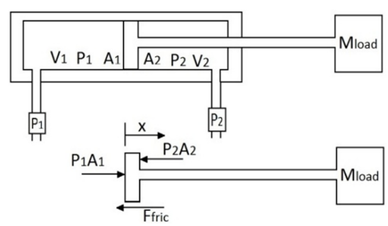

Figure 2. Free body diagram of the servo-pneumatic operator

following:

$$\dot{x} = v \tag{1.1}$$

$$\dot{v} = \frac{F_p - F_{fric} - F_{load}}{m} \tag{1.2}$$

$$\dot{P}_i(t) = \frac{P_i K}{V_i}(\dot{m}_{in_i}\frac{T_{in}R}{P_i} - \dot{m}_{out_i}\frac{T_i R}{P_i} - \dot{V}_i), \quad i = 1,2 \tag{1.3}$$

$$\dot{T}_i(t) = (KT_{in} - T_i)\frac{R\dot{m}_{in_i}}{P_i V_i}T_i - (K-1)(\frac{R\dot{m}_{out_i}}{P_i V_i}T_i^2) - (K-1)(\frac{\dot{V}_i}{V_i}T_i), \quad i = 1,2 \tag{1.4}$$

Where$\dot{x}$ is the piston velocity, $\dot{x}$ piston acceleration and F_p is the pressure force in two sides of the cylinder, F_{fric} refers to the coulomb friction, F_{load} is the external force and m stands for the external loaded mass P_i, V_i and T_i are respectively pressure, volume and temperature of the inside of the cylinder respectively. K is Specific heat capacity, R is the gas constant, $\dot{m}_{in}$ and $\dot{m}_{out}$ are the entering and outgoing mass flow rate of the cylinder. In order to apply the control commands to the intended operator, two on/off solenoid valves are used. By applying the on/off pulse width, compressed air is driven into the operator cases. The mass flow rate to the cylinder is calculated by the following equation [8]:

$$\dot{m} = C_d A_v \begin{cases} \frac{P_u}{\sqrt{RT_0}}, & 0 < r \le b \\ \frac{P_u}{\sqrt{RT_0}}\sqrt{1 - (\frac{r-b}{1-b})^2}, & b < r \le 1 \\ 0, & Otherwise \end{cases} \tag{1.5}$$

Where P_u is the upstream pressure, C_d is discharge coefficient of the valve, A_v is the cross section of the valves orifice at each moment, T_0 is the stagnation temperature and r is the downstream to upstream pressure ratio. Value of b is dependent on the specific heat capacity of the fluid i.e. ideal gas and can be obtained from Eq. (6) as following:

$$b = \sqrt{\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}} \tag{1.6}$$

2. Pulse width modulation algorithm

Pulse width modulation algorithm has been applied to transform the control signals to the task and time of valve operations. Due to the fact that one should constantly transfer different mass flows from valves to the operators, it is quite normal to use servo-or-proportional valves in servo applications. However, solenoid valves can be utilized to lower the cost. These valves work only in the maximum a minimum rate of the mass flow. Pulse width modulation algorithm opens and closes these valves with a high frequency which the so that the optimal mass rate flow from valves to the operator is achieved. In this algorithm, the control signal i.e. u is first sampled, then the opening task is evaluated accordingly and sent to the valves as on/off pulses. Various methods are available to transform the control signals to on/off signals [5-7]. In this study, the cut-off input signals with a saw-tooth shape and the frequency equal to the frequency of the modeled valves is used. As shown in Figure 4,

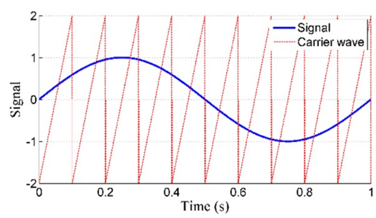

FIGURE 3. generating a PWM pulse

the PWM algorithm calculates the task ratio and the time of the operation of both valves comparing the input and carrier signals together and then generates the PWM signals.

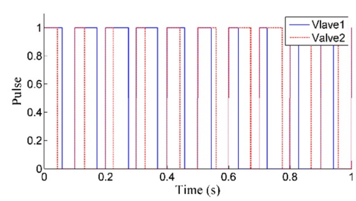

FIGURE 4. applied pulses to the solenoid valves

For higher accuracy in modeling the applied pulses, open/close latencies are modeled, as shown in Figure 5.

3. Designing the PID controller

A PID controller is designed to compare the performance of the proposed controllers for the trajectory following task. General form of a PID controller is as following:

$$u(t) = K_p e(t) + K_i \int_0^t e(t)dt + K_d \frac{de(t)}{dt} \tag{3.1}$$

In which, u is the control command and e is the error. The PID controller has three different components; proportional term i.e. P which amplifies with the control command regarding

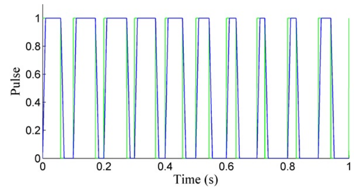

FIGURE 5. applied pulses to the solenoid valves

TABLE 1. Design variables for PID controller

No.	Design variable	Symbol
1	proportional gain	K_p
2	Integral gain	K_i
3	Derivative gain	K_d

the error scale e and has the gain of K_p. The integrator component which integrates the error over a time history from the beginning time of the performance till the moment t and has the control gain of K_i. And the derivative component which accounts for possible future values of the error, based on its current rate of change and has the control gain of K_d. The ranges of the control gains K_d, K_d and K_d are obtained with trial and error method.

The exact values of them are obtained using optimization analysis having considered them as designing variables of the function which calculates the overall errors which is shown in equation 8:

$$RSSE = \sqrt{\int_0^t e^2 dt} \tag{3.2}$$

4. DESIGNING THE SLIDING MODE CONTROLLER

Since the servo-pneumatic operator has rehabilitation applications and as the mathematical model of the system is available, a sliding mode controller is proposed for the presented operator. Moreover, these controllers are unaffected to the ambiguities of systems. To attain the control input, dynamic model of the system is considered as following:

$$\dot{x} = v$$

$$\dot{v} = f(X) + b(X)u \tag{4.1}$$

Where, $f(X) = (-F_{fric} - F_{load})/m$, $b(X) = 1/m$. Regardless to the ineffectiveness of the sliding mode controller to the ambiguities, $f(X)$ and $b(X)$ can be identified with a high accuracy. Design of the sliding mode controller requires system equations to be derived in the form of (9), which leads to state-space model:

$$f(X) = \frac{-F_{fric} - F_{load}}{m} + \frac{1}{m}u \tag{4.2}$$

General sliding surface is assumed as [8]:

$$s(x,t) = (\frac{d}{dt} + \lambda)^{n-1}\tilde{x} \tag{4.3}$$

Where n is the order of the state-space equation and λ is a positive constant related to reaching time. $\tilde{x}$ is defined as:

$$\tilde{x} = x - x_d \tag{4.4}$$

TABLE 2. Numerical results for the example 2.

Parameter	Value	Parameter	Value
Pneumatic cylinder stroke	$160mm$	Valve frequency	$60Hz$
Cylinder inner diameter	$16mm$	Orifice cross section	$10mm^2$
Rod diameter	$6mm$	Compressor pressure	$600kPa$
Valve ON time	$1.7msec$	Initial pressure	$100kPa$
Valve OFF time	$2msec$	Chamber initial temperature	$293K$

Where x and x_d are actual and desired positions, respectively. It is obvious that $n = 2$; therefore sliding surface can be written as:

$$s = (\frac{d}{dt} + \lambda)\tilde{x} \tag{4.5}$$

Where $\tilde{x} = x - x_d$ and x_d is the desired position. Filippov's Construction and $\dot{s} = 0$ yields an equation for equivalent control [8]:

$$\dot{s} = \ddot{\tilde{x}} + \lambda\dot{\tilde{x}} \tag{4.6}$$

$$\ddot{x} - \ddot{x}_d + \lambda\dot{\tilde{x}} = 0 \tag{4.7}$$

Substituting $\ddot{x}$ into (15):

$$f - bu - \ddot{x}_d + \lambda\dot{\tilde{x}} = 0 \tag{4.8}$$

Solving this equation for control signal, yields equivalent control as:

$$u_{eq} = \frac{1}{b}[-f + \ddot{x}_d - \lambda\dot{\tilde{x}}] \tag{4.9}$$

Finally, adding a sign function leads to control signal as follows [8]:

$$u = \frac{1}{b}[-f + \ddot{x}_d - \lambda\dot{\tilde{x}} - K_{SM}fgn(s)] \tag{4.10}$$

Where $sgn(s)$ is defined as:

$$sgn(s) = \begin{cases} -1, & \text{s¡0;} \\ 0, & \text{s=0;} \\ 1, & \text{s¿0.} \end{cases} \tag{4.11}$$

The optimization process has been completed based on RSSE error as the cost function:

$$RSSE = \sqrt{\int_0^t e^2 dt} \tag{4.12}$$

5. SIMULATION AND RESULTS

Simulations and optimizations are done using a combined model in MATLAB/Simulink. Constant parameters are shown in Table 2.

Simulation results for harmonic input with amplitude 0.04m and frequency 2π are shown in Figure.6 - Figure.9:

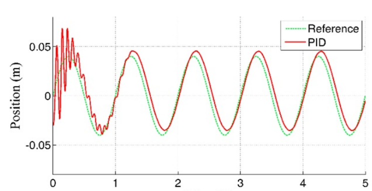

FIGURE 6. Position control with harmonic input for PID controller

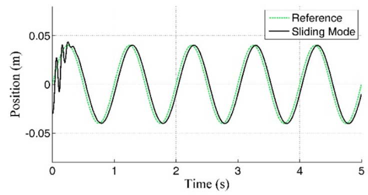

FIGURE 7. Position control with harmonic input for Sliding mode controller

TABLE 3. RSSE error for PID and Sliding mode controllers, under harmonic and step inputs

Input	Controller	RSSE(m)
Harmonic	PID	0.02326
Harmonic	Sliding mode	0.01750
Step	PID	0.01150
Step	Sliding mode	0.00807

Figure.6 and Figure.7 shows position control for sinusoidal desired position of the actuator, for both controllers. Results for step input with 0.04 m amplitude are shown in Figure.8 and Figure.9. Results show that sliding mode controller has better performance, compared

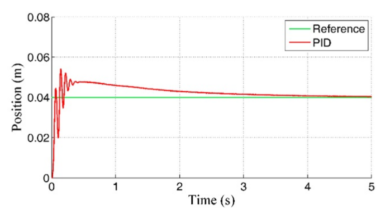

FIGURE 8. Position control with step input for PID

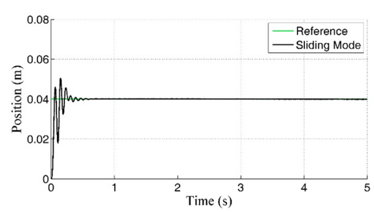

FIGURE 9. Position control with step input for PID

to PID controller. Using RSSE criterion (10), final error of both controllers are shown in Table.3: These results show that sliding mode controller has better performances in position tracking for both harmonic and step inputs. Moreover, sliding mode controller has lower maximum error amplitude compared to PID controller.

6. EXPERIMENTAL RESULTS

In this section, the experimental setup for the analyzed pneumatic system is introduced and the communication method between the software and the installed hardware is explained. Then the pneumatic valves and PWM algorithm, which are used to apply the

Table 4. characteristics of the actuator used in experiments.

Characteristic	Value
Stroke	160 mm
Piston Diameter	16 mm
Cushioning	PPV: Pneumatic cushioning adjustable at both ends
Conforms to standard	ISO 6432
Working pressure	1 ... 10 bar
Mode of operation	double acting
Note on operating and pilot medium	Lubricated operation possible
Ambient temperature	-20 ... 120 ^{o}C
Impact energy in end positions	0.3 J
Theoretical force at 6 bar, advance stroke	247 ... 295 N
Mounting type	With accessories
Pneumatic connection	G1/8
Materials note	conforms to RoHS
Materials information for cover	Wrought Aluminium alloy
Materials information for seals	TPE-U(PU)
Materials information for piston rod	High alloy steel, non-corrosive
Materials information for cylinder barrel	High alloy steel, non-corrosive

control input to the pneumatic setup is demonstrated. Finally, experimental results are shown and compared to simulations. These results confirm the controller design which has been investigated in previous chapters. The experimental pneumatic setup used to apply the designed controllers is shown in Figure.10. This pneumatic system consists of several parts, which are explained briefly later on. Technical properties of the investigated pneumatic

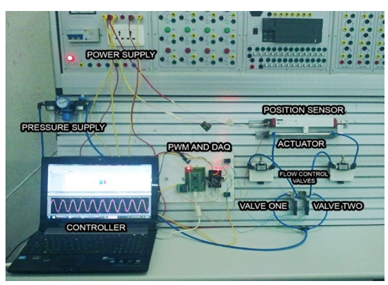

Figure 10. Position control with step input for PID

actuator is shown in Table.4: A linear potentiometer is used to measure actuator's position and actuator's chamber pressures are measured using two pressure 10bar sensors. Figure.11 shows harmonic tracking error for position controlled system, under sliding mode control. During the experiment, the error amplitude stays under 0.001m which is highly acceptable. This is due to optimization of sliding mode controller design variables.

Figure 13 shows position tracking results for step input with amplitude 0.04 m. Initial position is -0.03m. Experimental results show more error in comparison to simulations, but are highly acceptable and sliding mode controller is chosen to be a better controller compared to PID controllers for step inputs. Figure 14 shows position tracking error for step input with amplitude 0.04 m. Initial position is 0.03m. during the experimental test, error amplitude stays under 0.004m which is highly acceptable. For the calculated value for ?, optimization process delivers a small reaching time, which forces the actuator to reach

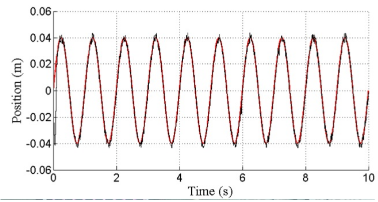

FIGURE 11. Position control with step input for PID

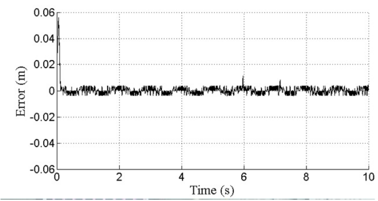

FIGURE 12. Position control with step input for PID

the desired position in a short time. This results confirms the validation of the optimization process for design variables.

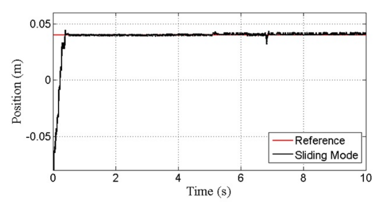

FIGURE 13. Position control with step input for PID

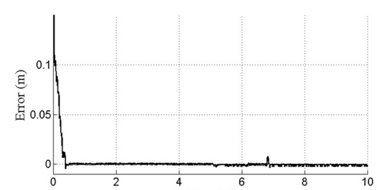

FIGURE 14. Position control with step input for PID

7. Conclusion

In this paper, position control of a pneumatic actuator was investigated, using two in/off solenoid valves, PWM algorithm and a sliding mode controller. A single objective GA algorithm was used to optimize the design variables. The sliding mode controller's performance was compared to a PID controller under harmonic and step inputs. Finally an experimental setup was used to validate the results. Results showed that the sliding mode controller was able to control pneumatic systems, which are highly nonlinear due to the compressibility of air. Moreover it showed better performances compared to a PID controller.

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