Improvement of Output Displacement of Servo Pneumatic System using Fuzzy PI Controller

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Abstract

Pneumatic systems are widely used in industrial automation for their advantages such as speed of motion, low cost of maintenance, safe operation and high power/ weight ratio, cleanliness, simplicity of operation as compared with other systems such as hydraulic and electro - mechanical technologies.

In this work, a fuzzy PI controller is designed for improving the output displacement of the servo pneumatic system, in order to improve the position transient response. The aim of tuning is to reach minimum steady state error in the output displacement and also to compensate the effect of applying different types of external load forces. The parameters of the controller are tuned using a trial and error method.

A comparison between the results of using Fuzzy PI controller and the PID controller showed that the Fuzzy PI controller has improved the maximum error ratio in position up to 45 % by adding a fixed external load force and up to 19 % by adding variable external load force with high compensation for the effects of any external load force.

Keywords: Servo Pneumatic Model, Fuzzy Logic, PID Controller.

تصميم مسيطر مضبب متناسب _ متكامل للمنظومات الهوائية المؤازرة

الخلاصة

لقد أصبحت أنظمة الهواء و دوائرها أكثر انتشارا واستعمالا في الأجهزة الصناعية لعدة فوائد : سرعة الحركة و قلة تكلفة الصيانة و أمان و علو نسبة القدرة إلى الوزن ونظافة الاستعمال وسهولة في عملية المقارنة مع أنظمة أخرى مثل أنظمة الهيدروليك و أنظمة الكهروميكانيك . ان استخدامات الأنظمة الهوائية و تطبيقاتها في عدة مجالات منها : في بعض الاجهزة الطبية و الروبوت , كذلك تدخل في المعامل المصنعة للحديد والخشب و البلاستك ... الخ . بالرغم من ان استخدام الانظمة الهوائية في تطبيقات السيطرة على الموقع تواجه صعوبة وتوعز هذه الصعوبة إلى التأثيرات اللاخطية في أنظمة الهواء . وقد تم في هذا البحث تصميم وتنغيم المسيطر المضبب المتناسب – المتكامل (Fuzzy PI) لأجل تحسين دقة المواقع اللحظية والوصول إلى اقل نسبة خطأ للاستقرار في الموقع وكذلك لتعويض تأثير تسليط أي حمل خارجي , مع العلم إن عوامل كلا من المسيطرين قد نغمت باستخدام طريقة المحاولة والخطأ . ولغرض المقارنة تم تصميم وتنغيم نوع اخر من المسيطرات كالمسيطر المتناسب – المتكامل – المتفاضل (PID) , و تمت المقارنة بين نتائج استعمال كلا من المسيطرين و قد بينت المقارنة أن المسيطر (Fuzzy PI) قد حسّن نسبة الخطأ للدائرة

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بإضافة حمل خارجي ثابت إلى 45 % و بإضافة حمل خـــارجي متغيـــر مع تعويض واضح لتأثير إضافة الأحمال الخارجية إلى 19 % .

Introduction

Pneumatic actuators are well suited for a number of industrially relevant tasks ranging from point - to - point positioning to high accuracy servo positioning and force control because of their high force output to weight ratios, cleanliness and low cost.

In spite of these advantages, pneumatic systems have some undesirable characteristics. These are from the high compressibility of the air and from the nonlinearities of the pneumatic systems, which limit their use in applications that require fast and accurate response [1].

Pneumatic actuators are very common in industrial applications for their easy and simple maintenance, relatively low cost, self cooling properties, good power density (power/dimension rate), fast acting with high accelerations and installation flexibility. Also, compressed air is available in almost all industry plants. The pneumatic actuators have a large band of applications like in motion control to materials and parts handling, packing machines, machine tools, robotics, food processing and process industry.

Pneumatic servo system has many disadvantages that have to be overcome by its control system. It has very low stiffness (caused by air compressibility), inherently non-linear behavior and low damping of the actuators circuits, that cause control difficulties. The main non-linearties in pneumatic servo systems are the air flow-pressure relationship through valve orifice, the air compressibility and friction effects between contact surfaces in actuator seals [1].

In order to meet desired requirements of servo pneumatic system such as fast displacement response with minimum overshoot and good positioning accuracy, a controller should be able to continuously correct the circuit output response. The early control programs of servo pneumatic systems were depended on the linearization of the mathematical model of the servo pneumatic system and using a classical controllers.

As a result, a number of researches had been concerned with the development of various control strategies for pneumatic drives for motion or force control [2]. **Perondi, and Guenther**, addressed in 2000 the problem of the pneumatic positioning system in the presence of friction [3]. Furthermore **Andrighetto, et al**. pointed in 2004 to the control problem of the pneumatic servo actuator in positioning, and they used P, PI and PID controllers and compared them with commercially available Festo SPC- 100 controller [4]. **Situm, et al**. compared in 2004 between the classical PID controller and the fuzzy PID controller, by experimental results for adjustment of the conventional controller parameters [2]. **Šitum, et al**. built in 2005 [5], a mathematical model. The model is based on an analysis of solenoid valve dynamic, thermodynamic changes in pneumatic system and equation of motion including significant nonlinear effects for the dual action rod less pneumatic actuator. Experimental results indicate good agreement between the measured and the corresponding simulation data. Finally **Andrighetto, et al**, presented in 2006 a comparative study among friction behavior of several double-acting pneumatic actuators available to industrial use [1].

However, the aim of this paper is to tune a Fuzzy PI controller to improve the transient response of the model. A comparison between the Fuzzy PI controller and PID controller is made to find which controller has a better influence on the position transient response of servo pneumatic model .

Mathematical Modeling of Servo Pneumatic System:

The study of modeling servo pneumatic system is difficult because of the nonlinearity in the pneumatic compressed air and other reasons. The servo pneumatic system can be basically divided into three components; the compressor, the proportional valve and the actuator. The main equations that represent the pneumatic position servo circuit are the piston – load dynamic equation, the continuity equations and the pneumatic mass flow equation of the system. In the following, the details of these equations are presented:

1 <u>Piston load dynamic equation</u>:

The equation of motion for the piston – load assembly can be expressed by applying Newton's second law as follows [6]:

$$\ddot{y} = \frac{1}{M}(P_1.A - P_2.A - F_f - F_{ext}) \quad ...(1)$$

where y is the actuator position in m, M is the mass of the piston – load in kg, F_{ext} is the external force in Newton, F_f is the friction force in Newton, A is the cylinder area in m^2, P_1 and P_2 are the pressure in Pascal in chamber 1 and in chamber 2 respectively, $\dot{m}_1$ and $\dot{m}_2$ are the mass flow rate in kg/s in chamber 1 and chamber 2 respectively. However, Figure 1 shows the forces applied on the piston of the cylinder:

The most effective non – linearity in pneumatic position servo circuit is the actuator friction force. So, for representing the friction force, the following equations are used [1, 2, 5,6]:

$$F_f = \begin{cases} F_c + B\dot{y} + (F_s - F_c)e^{-(\frac{\dot{y}}{\dot{y}_s})^{\delta}} & \text{if } 0.01 < F_s \\ \begin{cases} F_{ext} & \text{if } |F_{ext}| < F_s \quad if(\frac{P_d}{P_u}) \geq P_{cr} \\ sign(F_{ext}).F_s & \text{if } |F_{ext}| > F_s \quad if(\frac{P_d}{P_u}) < P_{cr} \end{cases} & \text{if } 0.01 > F_s \end{cases} \quad ....(2)$$

where (δ) is an arbitrary exponent =2 , Fs is the static friction, Fc is the coulomb friction, B is viscous damping coefficient in N.s/m, $\dot{y}_s$ is the stribeck velocity in m/s, $\dot{y}$ is the velocity in m/s and F_{ext} is an external force in Newton. However, Figure 2 shows the friction – velocity relationship.

2 <u>Continuity equations</u>:

Continuity equations show the relationship between the air mass flow rate and the pressure changes in the chambers, which is obtained using energy conservation laws [5 - 8]:

$$\dot{P}_1 = -\frac{A.K}{A(\frac{L}{2}+y)}P_1.\dot{y} + \frac{R.K.T}{A(\frac{L}{2}+y)}\dot{m}_1 \quad ....(3)$$

where R is the gas constant in kg.J/K, T is the temperature in K, $\dot{m}$ is the mass flow rate in kg/s, K is the specific heat ratio and L is the stroke length in m. Similarly, the rate of pressure for chamber 2 of the cylinder is obtained from [3, 7]:

$$\dot{P}_2 = \frac{A.K}{A(\frac{L}{2}-y)}P_2.\dot{y} + \frac{R.K.T}{A(\frac{L}{2}-y)}\dot{m}_2 \quad ....(4)$$

3 Mass flow equations:

The air mass flow rate of a valve orifice through a pipe varies according to the control voltage that applied to the valve. However, The mass flow rate for chamber 1 is [5, 7,8]:

$$\dot{m}_1 = \dot{m}_{1\_in} - \dot{m}_{1\_out} \quad ...(5)$$

where

$$\dot{m}_{1\_in} = \begin{cases} C_1.C_f.A_v.P_u.\frac{1}{\sqrt{T}} \\ C_2.C_f.A_v.P_u.\frac{1}{\sqrt{T}}.(\frac{P_d}{P_u})^{\frac{1}{K}}.\sqrt{1-(\frac{P_d}{P_u})^{(\frac{K-1}{K})}} \end{cases} \quad ....(6)$$

$$\dot{m}_{1\_out} = \begin{cases} C_1.C_f.A_v.P_u.\frac{1}{\sqrt{T}} \\ C_2.C_f.A_v.P_u.\frac{1}{\sqrt{T}}.(\frac{P_d}{P_u})^{\frac{1}{K}}.\sqrt{1-(\frac{P_d}{P_u})^{(\frac{K-1}{K})}} \end{cases} \quad ...(7)$$

where $\left(\sqrt{1-(\frac{P_d}{P_u})^{(\frac{K-1}{K})}} > 0\right)$.

The same equations for the mass flow rate can be used to calculate the mass flow rate of chamber 2, where C_f is a non-dimensional discharge coefficient; P_u is the upstream pressure in Pascal, P_d is the downstream pressure, P_s is the supply pressure, P_{cr} is a critical pressure ratio, $P_{cr} = 0.528$, which delineates between the sonic, chocked, and subsonic flow reign, The constants C_1 and C_2 are assumed to be 0.0404 and 0.153 respectively [3, 5, 9].

However, the schematic diagram for the servo pneumatic actuator circuit is shown in Figure 3.

Figure 4 shows the simulated model of the nonlinear mathematical model of Servo pneumatic system using the Matlab/ Simulink [14].

Design of fuzzy PI controller:

Fuzzy logic control is similar to the mankind thinking method. This controller has become one of the most successful today's technologies for developing sophisticated control system. It has been used in all the technical fields including control, modeling, image and signal processing and expert systems [10].

In order to implement the Fuzzy PI controller, a proportional plus integral control equation is [11]:

$$\mathbf{u(t) = Kpf.e(t) + Kif \int e(t).dt} \quad ......(8)$$

It is clear from Equation (8) that there is an integration of the error. There is a difficulty in formulating rules depending on an integral error, because it may have very wide range of universe of discourse. To overcome this problem the integration should be removed by taking the derivative of u(t) in Equation (8) with respect to time. Thus, Equation (8) becomes:

$$\mathrm{D}\mathbf{u(t)} = \mathbf{Kpf}.\mathrm{D}\mathbf{e(t)} + \mathbf{Kif}.\mathbf{e(t)} \quad ....(9)$$

where

$$\mathrm{D}\mathbf{e(t)} = \mathbf{e(t)} - \mathbf{e(t-1)} \quad ...(10)$$

and

$$\mathrm{D}\mathbf{u(t)} = \mathbf{u(t)} - \mathbf{u(t-1)} \quad ...(11)$$

where Kpf is the gain of the change – of – error and Kif is the gain of the error. However, the Simulink model representation of the Fuzzy PI controller is shown in Figure 5.

Fuzzy logic controller has two inputs and one output, each one of the inputs and the output has its own membership function. Each membership function is defined on the common normalized domain [-1, 1], as shown in Figure 6.

According to the number of memberships in each input, the Fuzzy PI controller will have a 7 * 7=49 rules.

The selection of these rules is based on the knowledge of the behavior of the error equation and change of error equation. After several trials, the rules are selected to be as shown in Table (1).

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Simulation results:

To simulate the Simulink model of servo pneumatic circuit, the cylinder used in this modeling is selected to be a linear drive of a part model number DGPL from Festo company, It is a double acting rodless cylinder with internal piston diameter of 18 mm. Moreover, the valve used in this system is selected to be a proportional directional control valve of model number MPYE-5 1/8LF-010-B, in this valve the relation ship between the mass flow rate and the input voltage, where one of the output ports are connected to the supply pressure when the input voltage is from (0 to 5) volt while the other output port is connected when the voltage is from (5 to 10) volt as shown in Figure 7. It is a control application valve, where the acceleration and velocity are controlled by valves switching techniques. The specifications of servo pneumatic system are listed in Table (2) [6, 12, 13]:

By adding the Fuzzy PI controller to the servo pneumatic system and tuning, using trial and error, in order to achieve the performance characteristic; that is minimum position steady state error and compensation for the effect of applying the external load force. The controller gains are selected to be Kpf = 1 and Kif = 10. By applying a fixed external load force of $\mathbf{F_{ext}} = \mathbf{64N}$, shown in Figure 8, this load will affect the output displacement after 2 seconds which makes the maximum error in position to be 0.0333. The ratio in error will be 11 %, while the error has increased between the seconds of 2 - 3 to be 0.0203.

Now, if a variable external load force is applied to the controlled system, as shown in Figure 9, the output displacement will be affected with about 0.063 as a maximum error in position which equals to 21 % as an error ratio, while the error has increased within the seconds (2-3) up to 0.0203.

1 A PID controller design:

The PID controller consists of the Proportional plus Integral plus Derivative PID control. The PID controller is widely used across the industry, it is easy to implement and relatively easy to tune, because of the controller simplicity puts limitations on its capabilities in dealing with complex control problems, such as the hysterias problem. However, the Simulink model representation of the PID controller is shown in Figure 10.

The continuous form of the PID controller Equation is described by the following expression:

$$u(t) = Kp.e(t) + Ki.\int_0^t e(t)dt + Kd.\frac{de(t)}{dt} \quad ....(12)$$

By tuning the controller to achieve cylinder position with minimum steady state error, it was found after several trials that the parameters of the controller are selected to be Kp=86, Ki=69 and Kd=1.

By applying a fixed external load force of F_{ext} = 64 N, the output displacement of the cylinder is shown in Figure 11. The system reduces the maximum error in position to 0.0586 which represents an error ratio of 20 %. By applying a variable external load force, the error has increased

compared with the situation of applying a fixed external load force to 0.079. So, the error ratio will be 26 %. Whilst the output displacement rises faster than that with applying a fixed external load force as shown in Figure 12.

It is observed, from the results above that the PID controller could not track the desired output after applying an external load force. The output did not reach the zero error because the PID controller did not track the model's desired output displacement.

2 Comparison between the Fuzzy PI and PID controller:

In order to prove that the Fuzzy PI controller has improved the output displacement response, a comparison between using the fuzzy PI controller and using the PID controller shows that the fuzzy PI controller enhances the transient response of the piston position than the PID controller. Table (3) shows the results of the maximum error in piston position using the two controllers by applying different types of external load forces.

It can be concluded from the above results, that using the Fuzzy PI controller has reduced the error ratio by 45 % with applying a fixed external load force. Also, after apply a variable external load force the error ratio will reduce by 19 %.

The output displacement has reached the zero error point faster by using the fuzzy PI controller than by sing the PID controller. The output does not reach the zero error because of the nonlinearity of the servo pneumatic model. It can be noticed from the comparison between the results that the fuzzy PI controller compensates the effect of applying external load forces faster with minimum error in position as compared with the same situation by using PID controller. This is because the fuzzy controller generates difficult gains according to the selection of rules compared with a fixed value of gains in PID controller.

However, compared to the industrially dominant PID control that has only three design parameters, the number of design parameters for a fuzzy controller can become overwhelmingly large [11].

Conclusions:

In order to reduce the maximum error in position and to compensate the effect of applying external load forces, an intelligent controller of Fuzzy PI type was designed and tuned using trial and error. The Fuzzy PI controller had improved the output displacement, even with applying an external load force, to reach the zero error.

A comparison between the results obtained from the two controllers shows that the Fuzzy PI controller improves the output displacement response better than using PID type. The Fuzzy PI controller has improved the error ratio of the circuit up to 45 % with adding a fixed external load force and up to 19 % with a variable external load force with a high compensation for adding an external load force. Although the fuzzy controller enhances the transient response, there are several parameters to be tuned, compared with three parameters; the proportional, integral and derivative gains.

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Table (1): Rule base of fuzzy logic PI controller.

e \ Δe	N_B	N_M	N_S	Z	P_S	P_M	P_B
N_B	N_B	N_B	N_B	N_B	Z	N_B	N_B
N_M	N_B	N_S	N_B	Z	N_B	P_M	N_B
N_S	N_B	N_B	N_B	N_B	P_M	P_B	P_B
Z	P_B	P_B	P_M	P_S	P_B	N_M	N_B
P_S	P_M	P_M	P_M	P_B	P_B	Z	P_B
P_M	P_B	P_M	P_B	P_B	P_B	P_M	N_B
P_B	N_B	P_B	P_M	P_M	P_B	P_M	Z

Table (2): Specifications of servo pneumatic circuit.

Parameter	Value
Temperature (T)	294.5 K
Area (A)	2.543410^(-4) m^2
Mass (M)	2 kg
Length of piston (L)	0.3 m
Supply pressure (P_s)	6e5 pa
Atmosphere pressure (P_{atm})	1e5 pa
Columbic friction (F_c)	32.9 N
Static friction (F_s)	38.5 N
Viscous damping coefficient (B)	65 N.s/m^2
Stribeck velocity ($\dot{y}_s$)	4e-3 m/s

Table (3): Results of the maximum position error.

Load / Controller	Fixed Load	Variable Load
PID controller	20 %	26 %
Fuzzy PI controller	11 %	21 %

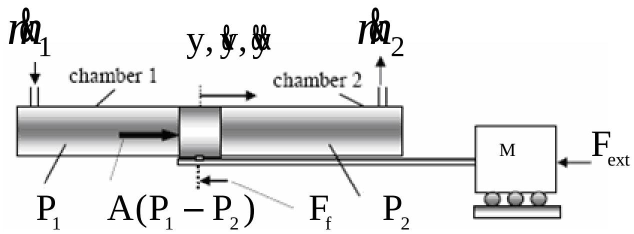

Figure 1: Force equilibrium at the piston.

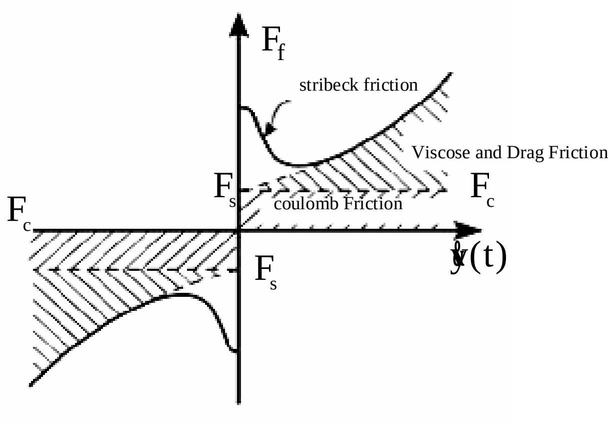

Figure 2: Friction – Velocity map [1].

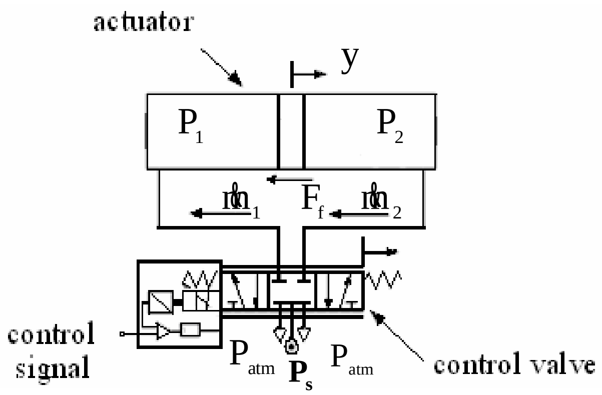

Figure 3: Schematic diagram of pneumatic actuator.

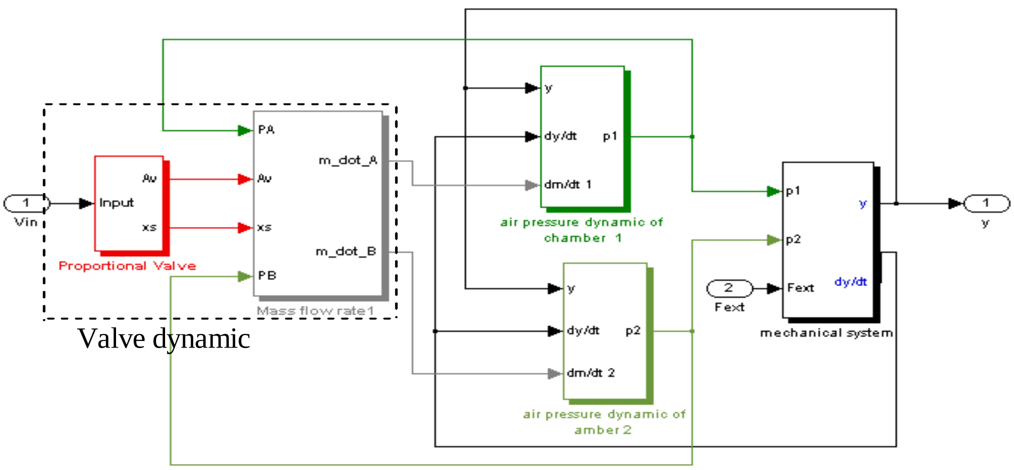

Figure 4: Simulink block of nonlinear model for pneumatic actuator system.

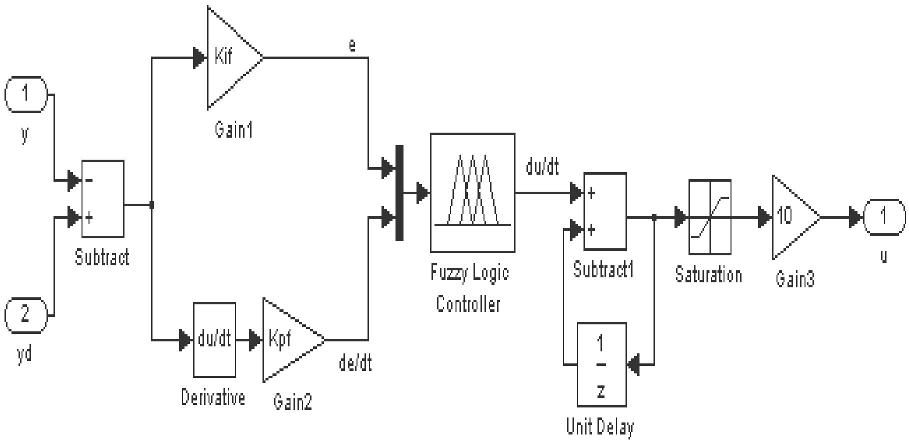

Figure 5: Simulink model of Fuzzy PI controller.

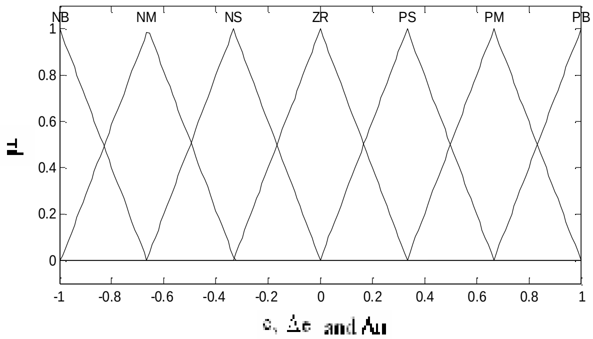

Figure 6: Membership functions of the inputs (e and Δe) and the controller output (Δu).

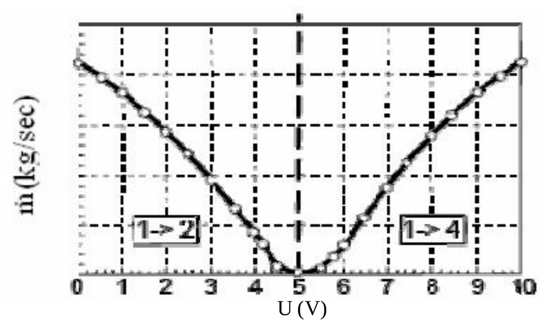

Figure 7: The relationship between the mass flow rate and the input voltage.

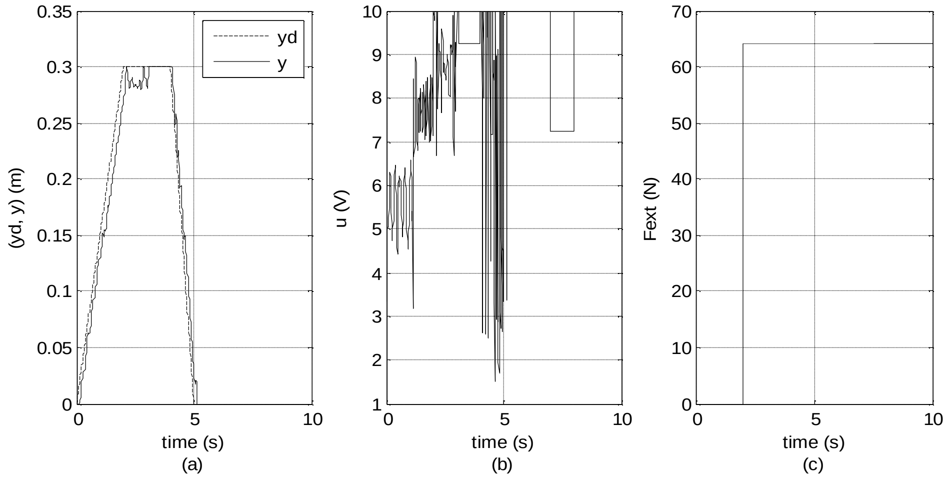

Figure 8: Simulation of closed loop model with a fixed external load force:
a) Displacement of the model and the desired displacement.
b) Output of the controller.
c) External load force.

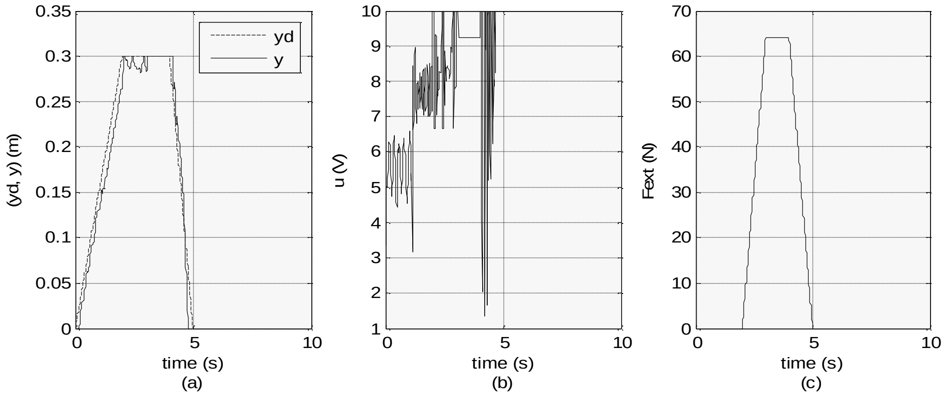

Figure 9: Simulation of closed loop model with applying a variable external load force:
a)Displacement of the model and the desired displacement.
b) Output of the controller.
c) Variable external load force.

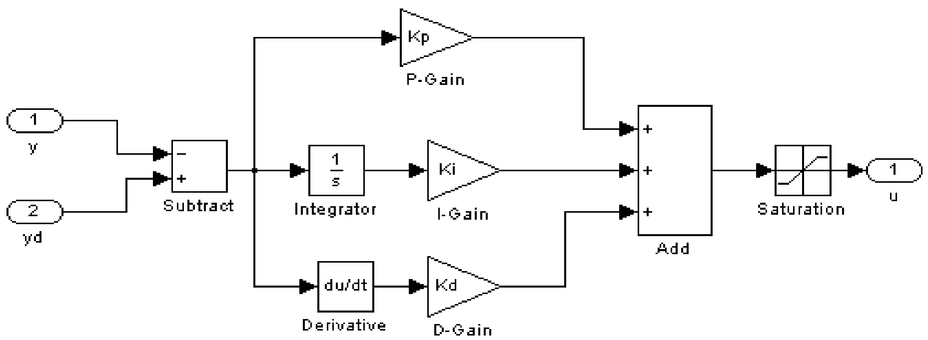

Figure 10: Simulink model of PID controller.

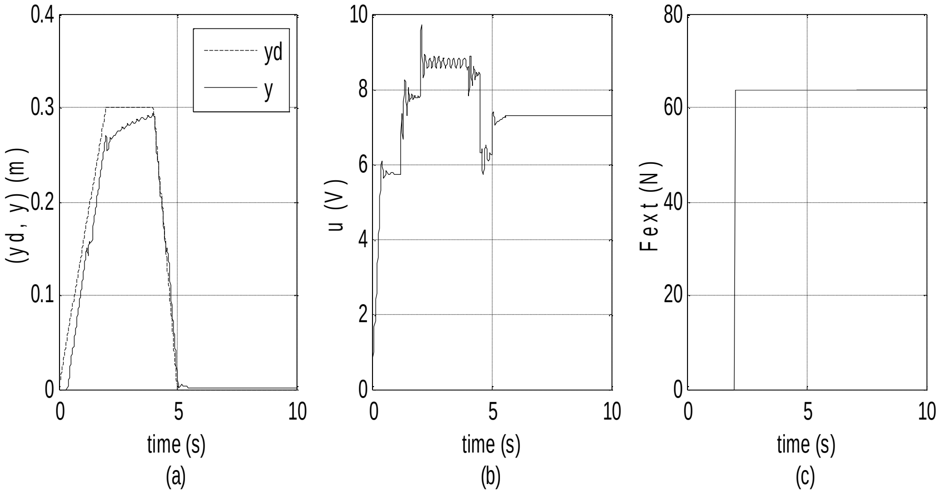

Figure 11: Simulation of PID controlled servo Pneumatic model with applying a fixed external load force.
a) Displacement of the model and the desired displacement.
b) Output of the controller.
c) External load force.

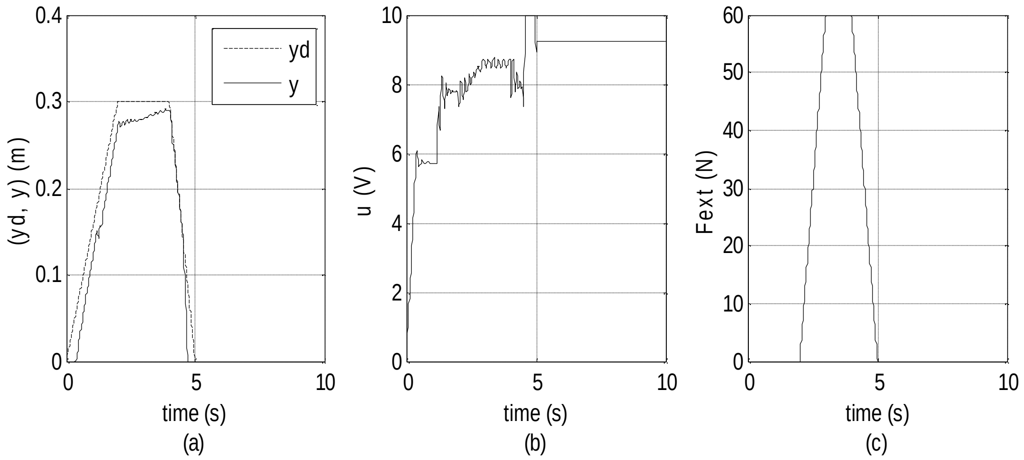

Figure 12: Simulation of PID controlled servo pneumatic model with applying a variable external load force.
a) Displacement of the model and the desired displacement.
b) Output of the controller.
c) Variable external load force.