Multiple Model Controller Design For Electropneumatic Servo Actuator

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Abstract

An application of the multiple model control to a pneumatic servo system will be presented. This approach has been considered in order to assure high control performance in the presence of large load variation on the system. The advantage of the multiple model control system with respect to the classical PID control will be illustrated. In the multiple model control approach a database of models is first determined in a system identification phase. A supervisor is then used to determine the model, which best represents the process at a particular instant. The controller associated with this model is then selected.

الخـــلاصة

يتضمن هذا ألبحث تطبيق المسيطر ذي النماذج المتعددة على منظومة المؤازر الكهروهوائي. واستخدام هذا النوع من المسيطرات لتحقيق سيطرة عالية الاداء مع حدوث تغيرات كبيرة في حمل المنظومة ويوضح هذا البحث امكانية هذا النوع من المسيطرات في تحسين اداء المنظومة مقارنة بالمسيطر الاعتيادي الـــ (PID). ولتصميم هذا النوع من المسيطرات هنالك مرحلتان الاولى، مرحلة تعريف النظام، حيث يتم انشاء قاعدة بيانات تضم مجموعة من النماذج للمنظومة المراد السيطرة عليها ولمختلف الاحمال وفي المرحلة الثانية، مرحلة تصميم المشرف، حيث يقوم المشرف باختيار المسيطر المناسب لنموذج المنظومة في تلك اللحظة.

1. Introduction

The servo actuator unit represents the main force control operator in the control system, where it's static and dynamic characteristics play an important role in the overall behavior of the control system. Therefore, improving the dynamic behavior of the servo actuator is of prime interest to control system designers. Most of the earlier pneumatic control systems were used in the process control industries where low-pressure air of the order of 7 - bar was easily obtainable and give sufficiently fast response. Pneumatic positions servos systems are used in numerous applications because of their ability to position loads with high dynamic response and to augment the force required to move the loads and are very reliable. There is a wide spectrum of applications of pneumatic servos such as milling machines, robots and advanced train suspension [1]. Many real industrial plants are nonlinear in nature or have characteristics, which change over time due to load variation, changes in operating

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conditions.etc. The plants in general, and the electropneumatic servo in our case, subject to abrupt and large parameter variations generally very difficult to control. A classical adaptive controller encounters the difficulties to solve this problem. An adaptive controller is not enough last to follow the parameter variations and unacceptable transients occur. Whereas a fixed robust controller normally leads to poor performance because of large uncertainties [2], [3]. It has been shown by Karemi in [4] that the multiple model control lei-strategy can cope, effectively, with abrupt and large changes in the load on a flexible transmission system, as compared with a fixed robust controller. In the other hand; Gregrcic in [5] compared the performance of self - tuning pole - placement algorithm and multiple model controller, and it has shown that the multiple model controller provides significant improved transient performance for large, abrupt set point changes for a nonlinear Continuous Stirred Tank Reactor (CSTR) system.

2. Electropneumatic Servo Actuator

a. System Description [l]

The purpose of the servo actuator unit is to move the load by displacement (x) in compliance with command signals from the control section. Figure (1) shows the basic construction and operation of the electropnumatic servo actuator.

The source of power used in this type of actuator is compressed air supplied to the jet pipe. An electromagnetic force generated by the flowing electric current rotates the jet pipe. Reacting to the pressure differential in the cylinder cavities, the piston together with the rod moves with speed dependent on the airflow, air pressure, and load. The diameter and stroke of the piston are 80mm and 50mm, respectively. Output piston feedback is provided by a linear potentiometer, the slider of which is driven by the piston. The servo unit consists of a control surface actuator, a feedback transmitter, a polarized jet relay, and a power amplifier. The minor loop of the servo shown in Figure (2) is used to ensure a proportional movement with respect to input commands.

b. System Model and Dynamics

The analysis of pneumatic actuators requires a combination of thermodynamics, fluid dynamics and the dynamics of motion. For constructing a mathematical model, three major considerations must be, involved.

1) The determination of the mass flow rates through the valve.

2) The determination of the pressure, volume and temperature of the air, in cylinder.

3) The determination of the dynamics of the load. The value is a four port pneumatic jet pipe valve. This valve is treated as equivalent to two three-port valves, one for each side of the cylinder. Considering the left hand side of the cylinder, figure (1), and the thermodynamic system is enclosed in the box, or control volume. After applying the continuity equation, the conversation of energy law and considering the cylinder is perfectly insulated (adiabatic conditions) the mass flow rates equations can be rewritten:-

$$\dot{M}_a = \frac{1}{RT_a}(p_a \frac{dv_a}{dt} + \frac{v_a}{\gamma}\frac{dp_a}{dt}) \qquad (1)$$

$$-\dot{M}_b = \frac{1}{RT_b}(p_b \frac{dv_b}{dt} + \frac{v_b}{\gamma}\frac{dp_b}{dt}) \qquad (2)$$

Where:-

M_a, M_b : Mass flow rate in chamber a and b.

R: Gas constant.

P_a, P_b: Pressure in chamber a and b.

T_a, T_b : Temperature in chamber a and b.

V_a, V_b: Volume in chamber a and b.

γ : Specific heat ratio

The volume inside the actuator is a function of "dead volume", or the volume of air when the actuator piston is in the mid point (Vo) and "swept volume" or the area times the stroke:-

$$V_a = v_o + Ay \quad (3)$$

$$V_b = v_o + Ay \quad (4)$$

Where: A: Piston area.

y: Piston displacement.

v_o: Equilibrium piston volume.

Differentiating equation (3) and (4) and substitute them, in equations (1) and (2), we obtain:

$$\dot{M}_a RT_a = P_a A y + (vo + Ay) \frac{p_a}{\gamma}) \quad (5)$$

$$-\dot{M}_b RT_b = P_b A y + (vo + Ay) \frac{p_b}{\gamma}) \quad (6)$$

The load dynamics will influence the overall performance of the piston motion since the accelerating force is proportional to the piston pressure differential. The load equation is:

$$(P_a - P_b)A = m\ddot{y} + f\ddot{y} + F_f + F \quad (7)$$

By simplifying equations (5) to (7) we obtain:

$$\ddot{y} = \frac{A}{M} P_a - \frac{A}{M} P_b - \frac{f}{M}\dot{y} - \frac{F_f}{M} - \frac{F}{M} \quad (8)$$

$$\dot{P}_a = \frac{-A\gamma}{V_o - Ay} P_a \dot{y} + \frac{\gamma RT_s}{V_o - Ay} \dot{M}_a \quad (9)$$

$$\dot{P}_b = \frac{-A\gamma}{V_o - Ay} P_b \dot{y} + \frac{\gamma RT_s}{V_o - Ay} \dot{M}_b \quad (10)$$

In the system considered, a four port valve is used to control a double acting through rod cylinder, so that $A_a = A_b = A$ in Figure (1). There are three nonlinearities in the pneumatic servo system. The first one is the nonlinear characteristic of the valve, and the other two nonlinearities are the volume and bulk modulus when they are used as coefficients in the equations. Some model parameters were determined experimentally [1]. The main parameters required for simulating system equations are given in table (1). Figure (3) shows a linear analysis of the system given in appendix [B]. The open loop Bode plots are plotted for different load values (no load 0.1 Kg, half load 50 Kg, and full load 100 Kg). From the Bode plots it is clear that when the load increases, the bandwidth decreases, tills make the system slower. The phase margin, in the other hand, decreases with the increasing of the load and this makes the system tends to oscillate in closed loop [6].

Table (1): The main parameters required for simulating the system

Parameter	Value
A	0.005 m^2
ρ	1.185 Kg/m^3
R	287.0 $J/Kg.k^0$
K_p	400 v/m
K_a	0.000145 Kg/V.S
V_o	0.000251 m^3
P_i	300000 N/m^2
L_a	-0.000082 S/m

In this section two experiments have been done to compare the performance of the Multiple Model Controller (MMC) with the conventional PID controllers. In experiment 1 three different PID controllers have been used (controller 1, controller 2, and controller 3) each one of them was designed for a specific load where as experiment 2 uses the same controllers in a MMC structure to get the best from each one of the three controllers.

Experiment 1 (A PID controller)

A square wave signal with amplitude of 1 V and a period of 10 seconds is applied on the reference input. The experiment is started without load and at the instants 9s, 19s, 29s, and 39s, 25% of the total load (100kg) is added on the system. Therefore the system without load becomes full loaded in four stages, see Figure (6). From Figure (7), it is clear that controller1 is the best, in terms of time domain specifications, for the period $t \geq 0_{sec}$ to $t < 30_{sec.}$ This result was expected because controller1 had been designed for no load. Similarly, controller2 is the best for the period ($t \geq 10$ sec. to $t < 30$ sec.) Which is corresponds to a load between 25% to 50%. And finally conlroller3 is the best for the last period, $t \geq 30_{sec.}$ for load between 50% and 100%. Each one of the three controllers can cope with load variations within a certain limit, but non of them can cope with the whole rang of load variations.

Experiment 2 (MMC controller)

In this experiment the MMC is used and in order to compare with experiment1, the same reference input signal and load variations have been used. Where e_i (K) is the output error for the ith model. The parameters α and β are the weighting factors on the instaneous measures and long - term accuracy respectively. The forgetting factor λ determines the memory of the index, and will have a direct effect on the decision speed of the supervisor. The forgetting factor λ also assures the boundedness of the criterion for bounded e_i (k). The model with the lowest cost is selected, and its associated controller is switched in.

3. Multiple Model Controllers

a. The principles of the MMC

When a single identification model is used, it will have to adapt itself to the operating condition before appropriate control can be taken. After the operating condition of a system changes abruptly the original model (and hence controller) is no longer valid. Slowness of adaptation may result in a large transient error [5]. If models are available for different operating conditions, controllers corresponding to them can be found in advance. The main idea of this method, the MC is to choose the best model for the plant from an a priori known set of models at every instant and apply the output of the corresponding controller to the plant. This control structure can be represented as shown in Figure (4). The input and output of the plant are u (t) and y (t), respectively. The control system contains *n* models model 1, model 2, ..., model *n* which are either linear or nonlinear, either fixed or adaptive models. The difference between the i_{th} output $\hat{y}_i$, of the model and the plant output

$$e_{i(t)} = y(t) - \hat{y}_i(t) \tag{11}$$

For each model *i*, there is a controller, controller *i* that satisfies the control objective for model *i*. this means that a controller is designed for each model using pole - placement algorithm, robust PID controller or any other suitable controller. In fact strategies different control can be used for each operating point. A supervisor then compares the output errors for the *n* models. A discrete equivalent of the performance index proposed in [7] is given in [5] and can be written as follows:

$$J_i(k) = \alpha e_i^2(k) + \beta \sum_{j=1}^{m} exp(-j\lambda_i) e_i^2(k-j) \quad (12)$$

for experiment 2. The control structure is as shown in Figure (5). The role of the supervisor is to select one of three controllers (controller1, 2, 3) which achieve the best performance, or in other words, to achieve minimum value for the cost function given by equation (12). The parameters α, β and λ are usually chosen empirically and can be different for each model. Choosing a different set of parameters for each model can be a difficult task, particularly when the number of models is large. However, in [9] some general guide lines on how to choose α, β and λ are given. For our system, the parameters α, β where fixed and λ was chosen according to the bandwidth of each particular model. The following set of arameters for J was chosen: m = 8, $\alpha = 0.5$, $\beta = 0.7$, $\lambda_1 = 0.4$, $\lambda_2 = 1$, and $\lambda_3 = 1.6$. As seen from Figure (8) the MMC approach using just three models, gives a relatively good response for the whole range of load variations. Figure (9) shows the corresponding switching diagram. As seen from this figure (9), the supervisor may take repeated switches of the controller between a number of possible candidates, The performance can be improved by introduction of a hysteretic based switching algorithm as show in figure (10). Figure (11), shows the system response after adding the backlash blocks and figure (12) shows the corresponding diagram. It is clear that the backlash is very effective to reduce repeated switching between controllers. The transient response can be improved if more than three models were used. A criterion for addition of new models and the management of the models can be studied as a future work. It is important to note that the supervisor have chosen the best controller for each load value, for light load, controller 1 was the best (has the minim cost function), and hence controller 1 is switched in. thus same thing is true for middle load, where controller 2 is the best. And finally for very high load controller 3 is the best one among the three controllers and the supervisor switch in controller 3.

b.Multiple Model Control of the Pneumatic Servo System

The aim of this section, is to apply the multiple model control to the nonlinear system described by equations (8) to (10), for a wide range of load variation (from no load m = 0.1 Kg to full load m = 100 Kg) and see how it can cope with this wide range of dynamics variations described in Figure (3) and compare the results with the fixed PID controller. Three models for no-load (m = 0.1 Kg), half- load (m = 50 Kg) and full load (m = 100 Kg) are considered in the set of models. The block diagram of the control architecture is illustrated in Figure (5). M_1, M_2, and M_3, represent unloaded, half-loaded, and full-loaded models, respectively. At every instant, the supervisor chooses the best model M^* for the current operating condition according to the performance index of eq. (12) and activate the corresponding controller, a PID controller in our case, for each operating region. In this case we have three models and hence three corresponding controllers. The controllers are similar, but with different parameters (P, I, D). The parameters have been determined using the nonlinear control design (NCD) block set in MATLAB [8]. The NCD block set provides a graphical user interface (GUI) to assist in time domain based control design. With this block set, we can tune parameters within nonlinear Simulink model to meet time domain performance requirements by graphically placing constraints within a time domain window. Uncertainty bounds can be placed on the load mass for robust PID control design. Table (2), shows the parameters for each controller to achieve rise time of 1 sec, max over shoot ≤ 10%, and settling time ≤ 2 sec.

Table (2): Parameters for each controller

The controller	For Model	Parameters (P, I, D)
Controller 1	Model 1 no load	100,10,20
Controller 2	Model 2 half load	110,8,30
Controller 3	Model 3 full load	110,0.1835, 55.718

Note: The PID transfer function is: P+I/s+Ds where s is the laplace operator

Conclusions

An application of the multiple model control to a premutic servo system has been presented. This paper compared the performance of the multiple model controller with the classical PID controller. For large load variations on the system, it was found that the performance of the classical PID controller was poor. A multiple model controller approach, based on only three models, with associated controllers was found to provide significantly improved transient performance. This paper discussed a number of issues: The choice of model selection criterion, the need to have a different cost for each model, and the problem of repeated switching between controllers.

Appendix (A): List of Symbols

A: Piston area (m^2).

J: The cost function

K_a: Slop for the curve of the relationship between mass flow rate and input voltage (kg/v.s).

K_p: Potentiometer constant (V/m)

L_a: Slop for the curve of the relationship between mass flow rate and load pressure (s/m).

m: Mass of piston (kg).

M_a: Mass flow rate in chamber a (kg/s).

M_b: Mass flow rate in chamber b (kg/s).

P_a: Pressure in chamber a (bar).

P_b: Pressure in chamber b (bar).

P_i: Pressure at operating point (bar).

P_s: Supply pressure (bar).

γ: Specific heat ration.

R: Gas constant (J/kg.k^o).

t: Time (sec).

T_a: Temperature in chamber a (k^o).

T_b: Temperature in chamber b (k^o).

Ts: Supply Temperature (k^o).

V_a: Volume in chamber a (m^3).

V_b: Volume in chamber b (m^3).

V_{in}: Input voltage (volt).

V_o: Equilibrium piston volume (m^3).

y_o: Piston displacement from mid point of piston rod

Appendix (B): Linear model of the electro pneumatic system

$$\frac{y(s)}{V_{in}(s)} =$$

$$\frac{K_1}{\frac{K_3 m}{A} s^3 + (\frac{K_3 f}{A} - \frac{K_2 m}{A}) s^2 + (A - \frac{K_2 f}{A})}$$

Where:

$K_1 = RT_sK_a/P_i$
$K_2 = RT_sL_a/2P_i$
$K_3 = V_i/(2\gamma P_i)$
and Ff=f=0

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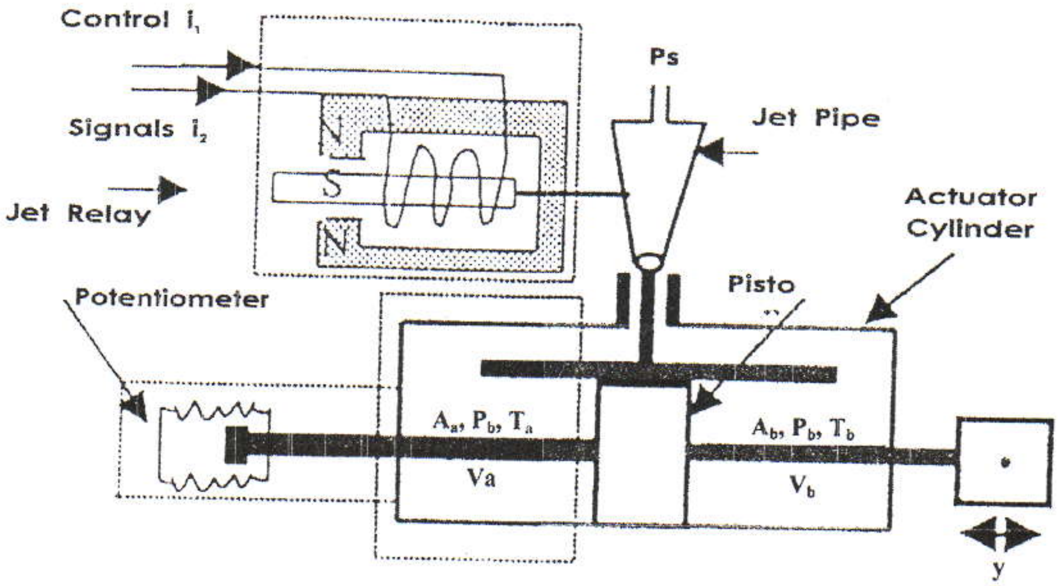

Fig. (1): Construction diagram of electro pneumatic servo actuator system

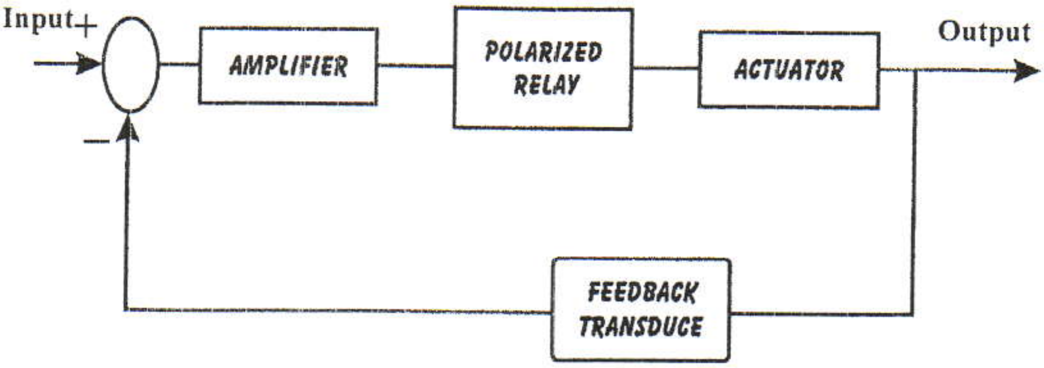

Fig. (2): Block diagram of servo actuator system

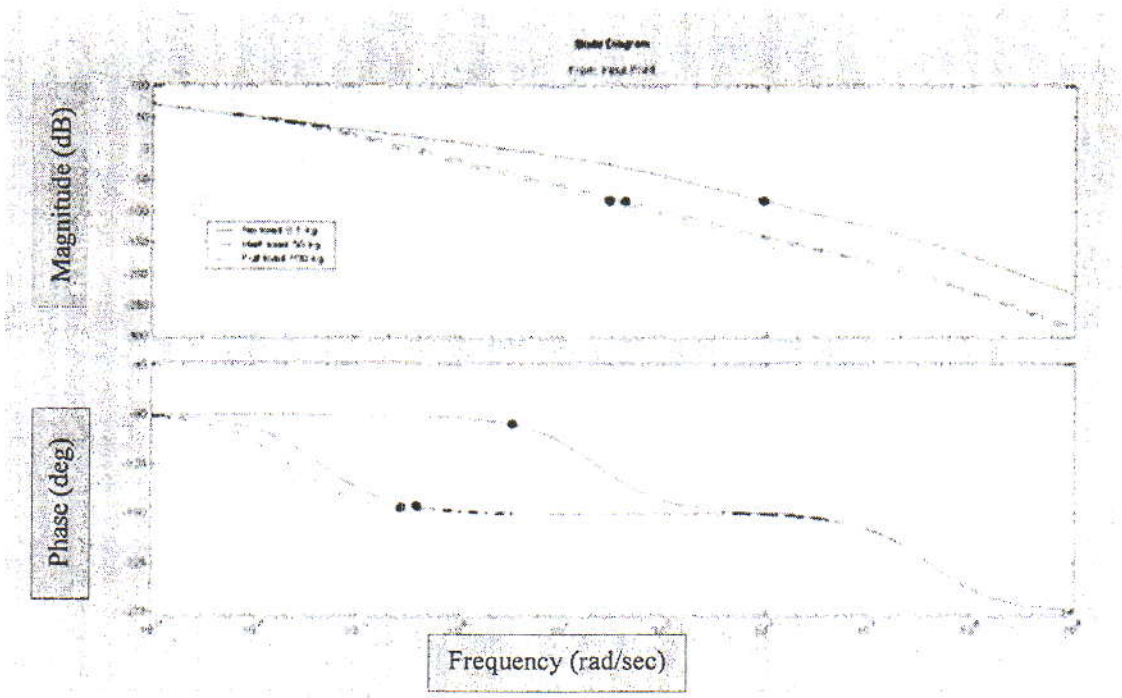

Fig. (3): The frequency characteristics of the pneumatic servo for various loads.

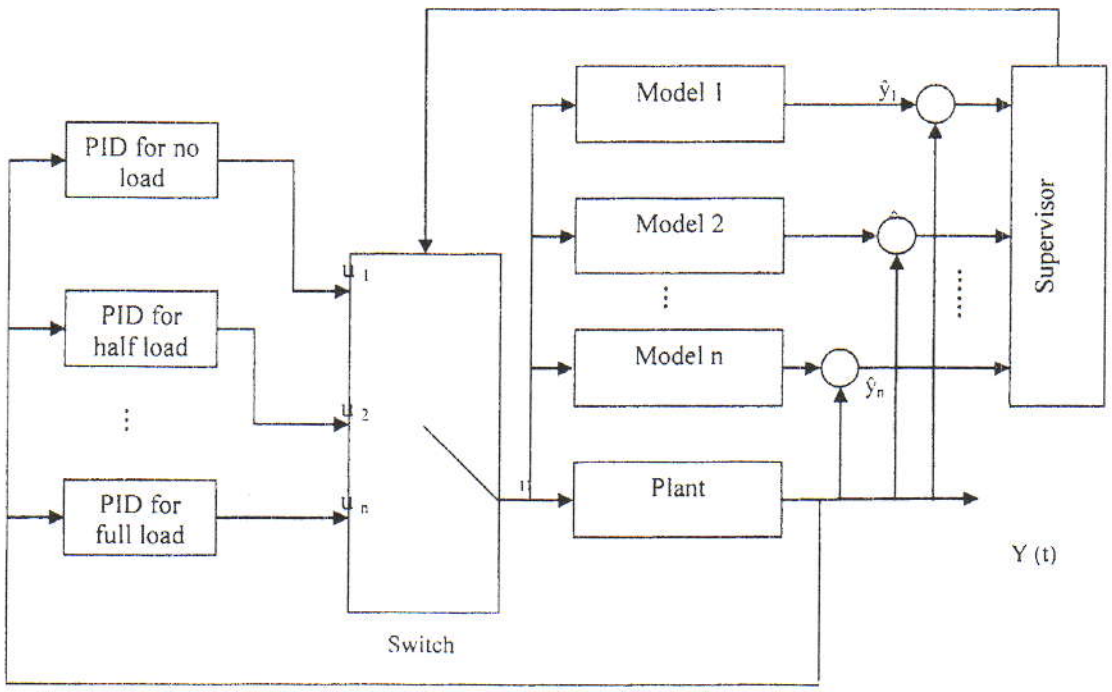

Fig. (4): Schematic diagram of the multiple model approach.

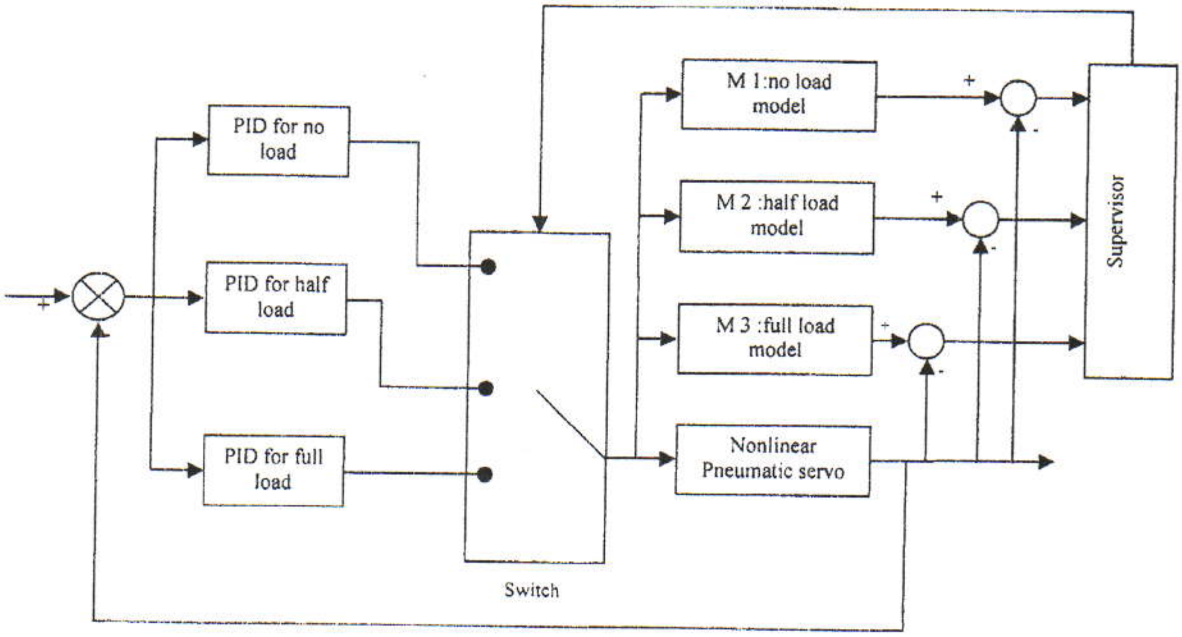

Fig. (5): Schematic diagram of the MMC for the pneumatic servo with three models and three corresponding controllers.

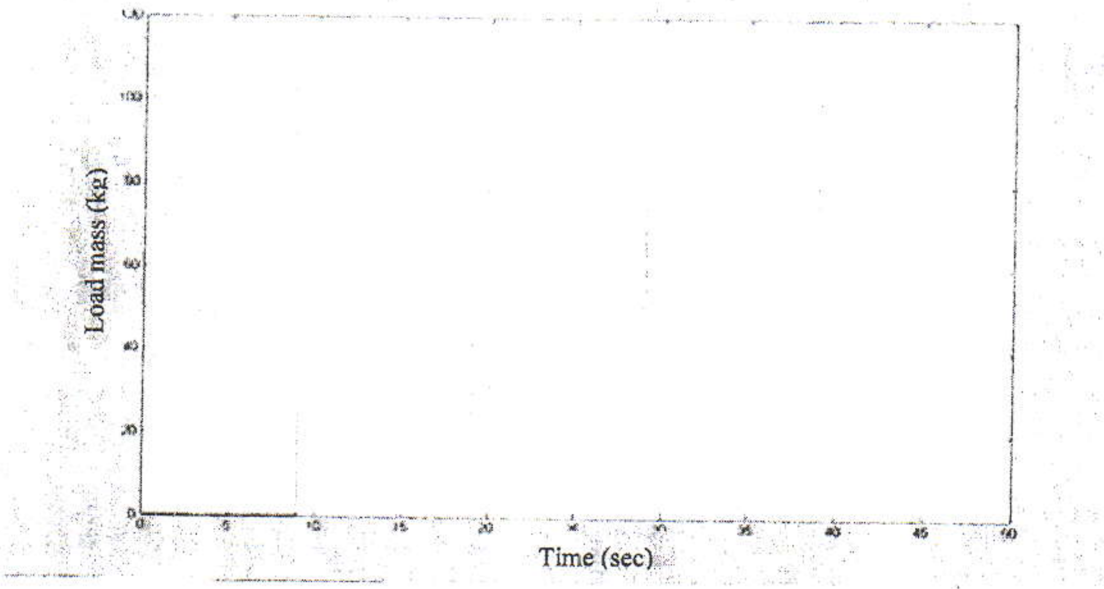

Fig. (6): Load variation with time.

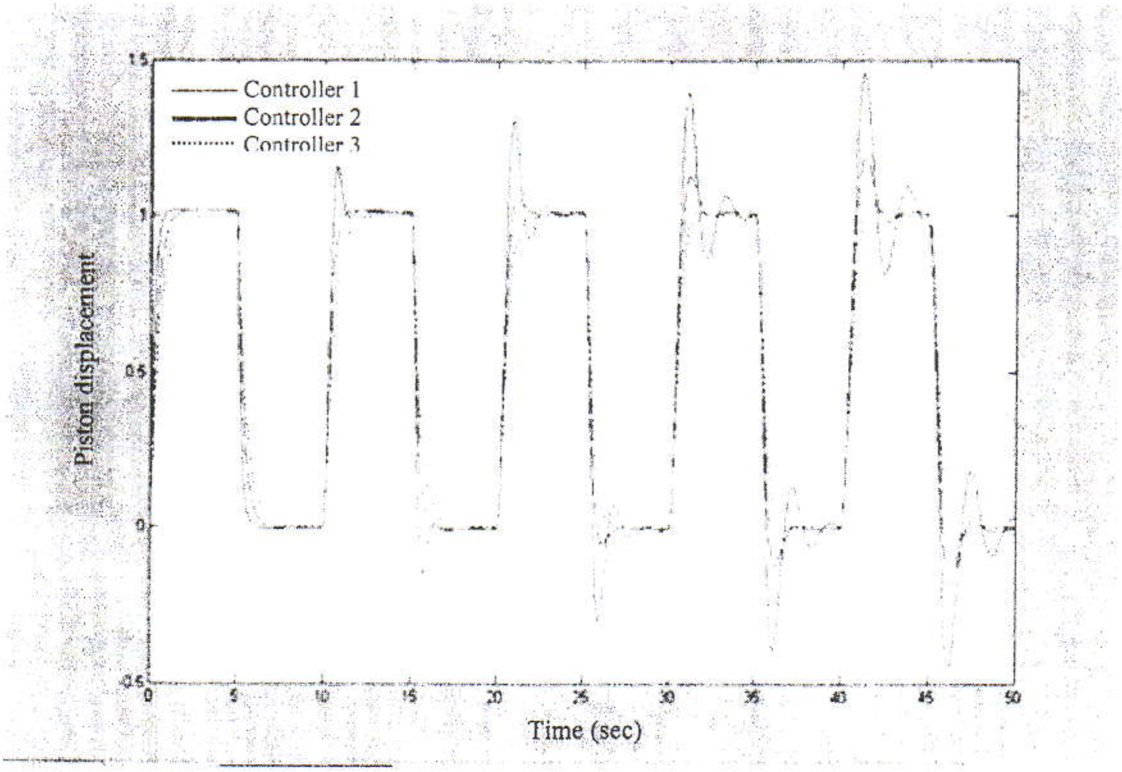

Fig. (7): Comparison of three fixed PID controllers: controller 1 designed for no load, controller 2 designed for half load, and controiler 3 for full load.

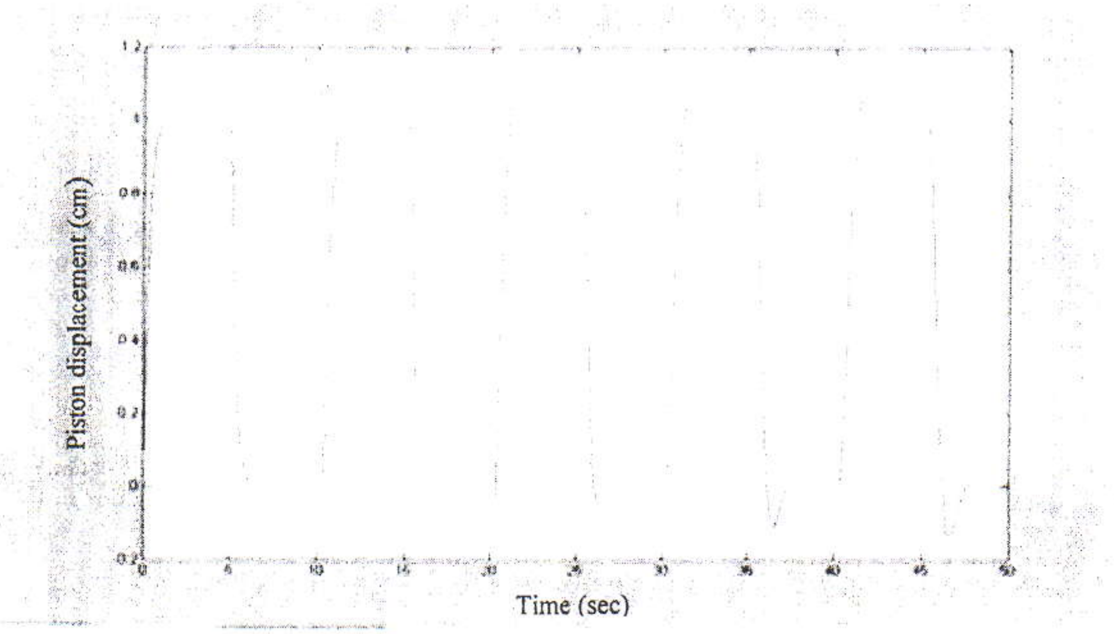

Fig. (8): The plant output for the MMC without hysteretic.

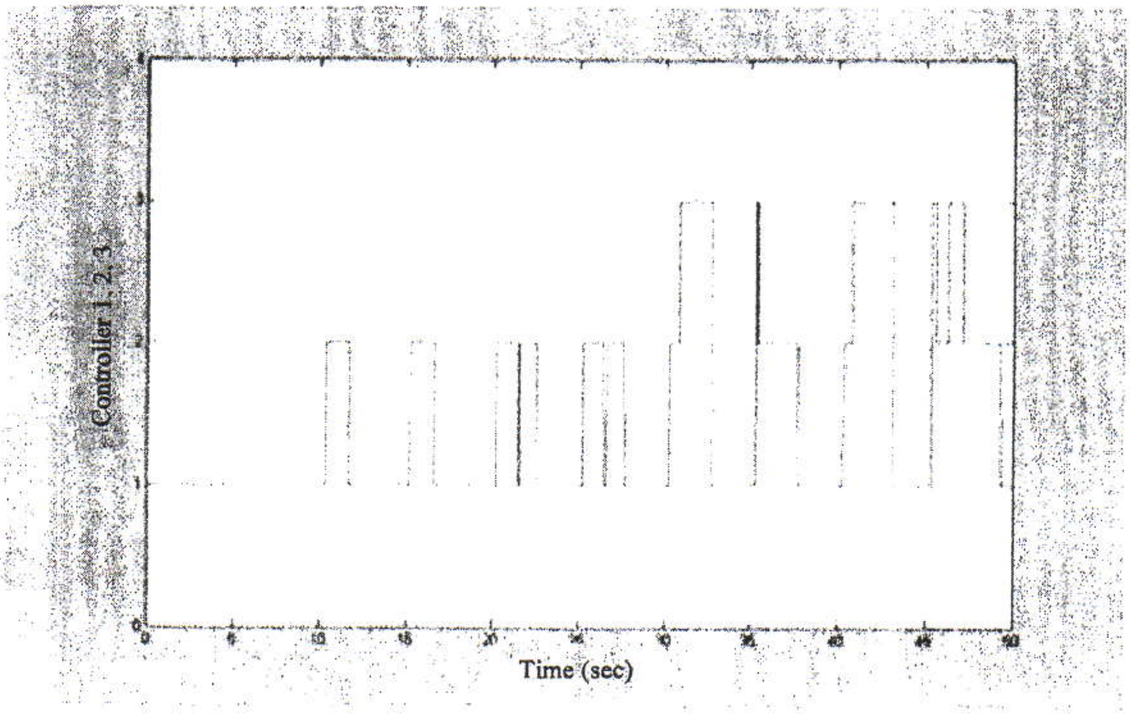

Fig. (9): Controller switching without hysteretic.

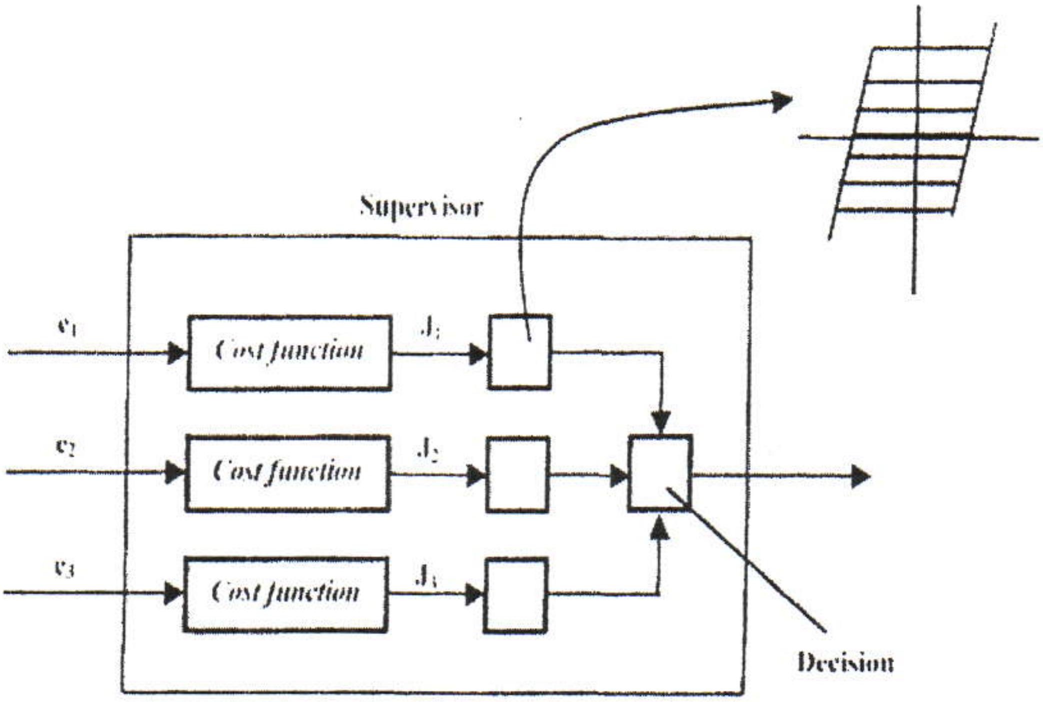

Fig. (10): Reduction of repeated switches using backlash.

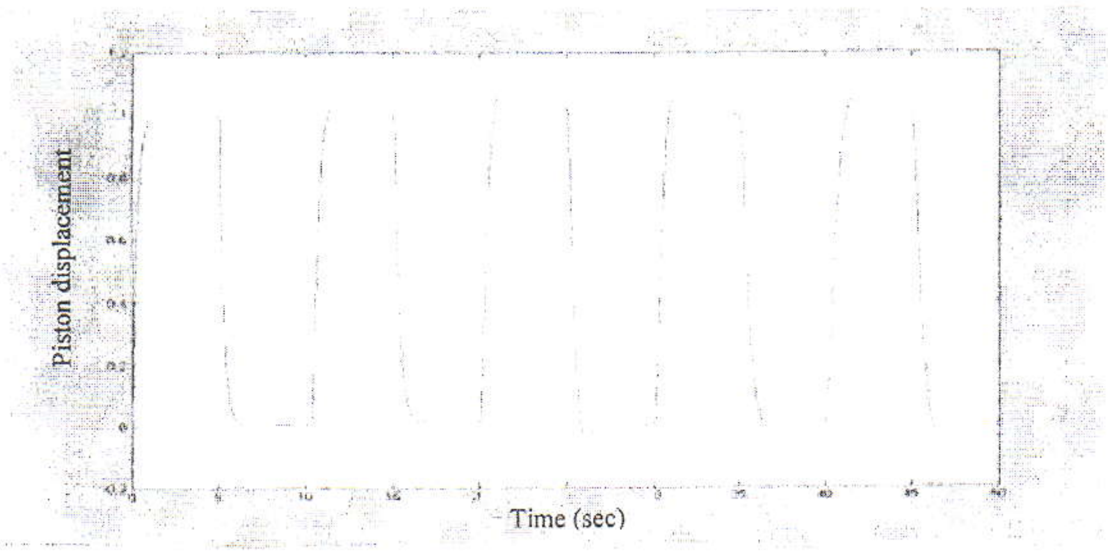

Fig. (11): The system output with MMC with hysteresis.

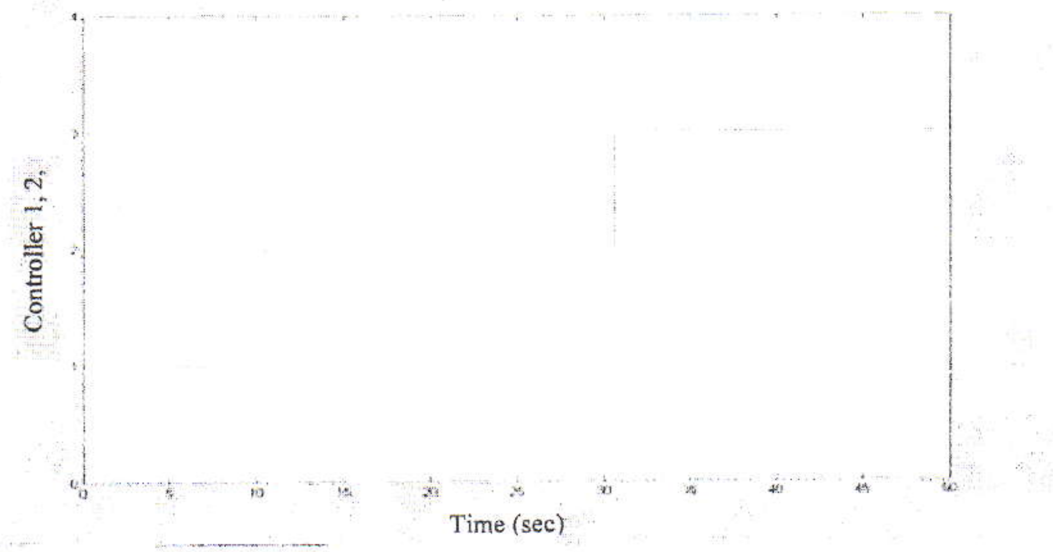

Fig. (12): Controller switching with hysteresis.