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Control Design of a High Speed Robot Arm

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abstracts

Controller design of a pneumatic servo-valve is discussed. This valve uses an electro-magnetic coil system to drive the spool, and relies on its control mechanism to adjust the spool position based on feedback characterizing the location of the robot arm. The configuration of the valve provides the benefit of high flow gains and stability. However, it puts high demands on the controller which needs to respond to the load changes inherent in a pick-and-place operation, while satisfying the criteria of moving the robot arm at high speed without overshoots.

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1 INTRODUCTION

Robot arms are widely used in pick-and-place applications such as assembly, die casting, forging, and injecting molding. In these applications fast and accurate positioning of robot arms may become difficult when the arm is subject to load changes as the result of different product weights. Accurate positioning of robot arms can be addressed by improving their structural stiffness, controller, motion trajectory, or actuator. Among the actuators used for robot arm motion, hydraulic and pneumatic actuators are of most utility. Hydraulic actuators provide speed and accuracy, but they are expensive. As such, they are only suitable for heavy duty applications. Pneumatic actuators, on the other hand, are attractive because of their simplicity and ease of operation. However, they are generally slow in response to command inputs. This results in lack of positioning accuracy, especially when the actuator is used to move the arm to various positions rapidly, as required in pick-and-place applications.

Traditionally, electro-pneumatic servo-valves consist of a coil armature that moves the valve spool back and forth against the compliant force of a spring. The valve is designed to linearly open and close in response to the voltage applied to the coil. The linearity between the applied voltage and coil motion depends on the electro-magnetic force generated by the coil, the resistance from the spring, and friction between the valve spool and its housing. Therefore, to maintain this linearity the spring force and mechanical tolerances of the moving parts must be tightly controlled and maintained over the life of the valve. These factors dramatically increase production costs and lower repeatability and reliability.

In this paper, a newly developed pneumatic servo-valve (HV-1000) is introduced that contains faster dynamics than available on traditional valves. This new valve incorporates a much smaller spring used for centering purposes only. Therefore, the valve has a simpler structure and is easier to manipulate. Because of these features, the valve is expected to have (1) a higher bandwidth, (2) no undesirable resonance, (3) a larger spool displacement, and (4) lower manufacturing costs. Of course, the replacement of the heavy spring with a lighter one, eliminates the linearity of motion for this valve. This imposes higher demands on the valve controller, which needs to move the spool accurately and rapidly in response to feedback information about the position of the arm.

The objective of this paper is to explore various control designs for this newly developed servo-valve. For testing purposes, the valve is used to move a piston to different positions. The pneumatic valve is quite nonlinear due to variations of air density with pressure and friction between the spool and housing. This, combined with inherent load changes on the arm during pick-and-place operations [9,3] and requirements for its smooth positioning impose serious limitations on traditional controllers such as PID. In this paper the application of Fuzzy Logic Control (FLC) and Self-Organizing Fuzzy Control (SOC) is considered and their performance is compared with PID.

2 VALVE STRUCTURE AND EXPERIMENTAL SETUP

The schematic of the developed pneumatic servo-valve is shown in Fig. 1. It incorporates a motor driven spool, where the motor armature consists of an electro-magnetic coil which drives the spool in response to a command voltage from the controller. The valve also uses a light weight spring on one side for centering purposes. The elimination of the traditionally used spring puts higher demands on control to position the spool in response to feedback about the position of the arm.

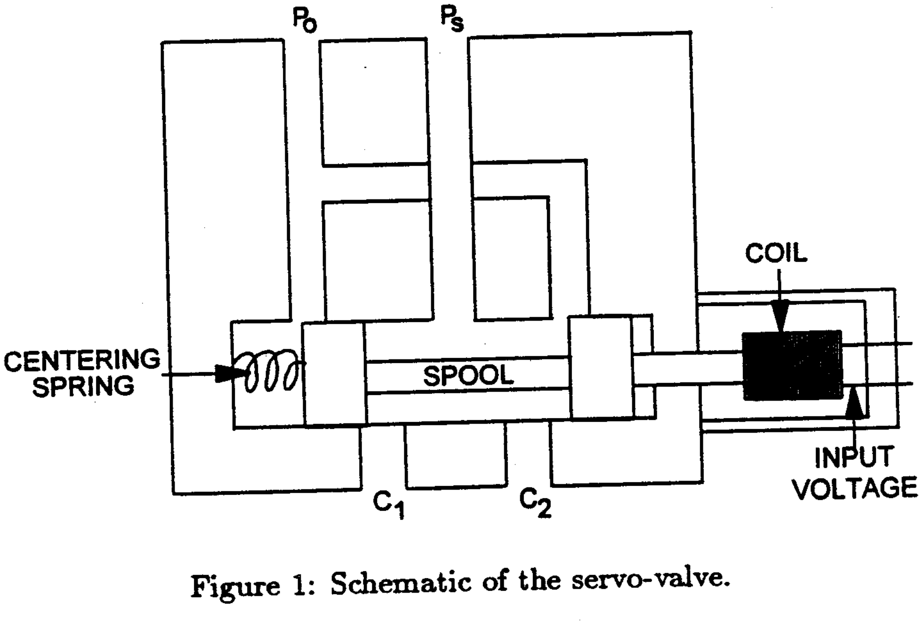

Figure 1: Schematic of the servo-valve.

The valve is implemented on a cylinder-piston system to test its dynamic response as well as the effectiveness of its controller. The schematic of the experimental setup is shown in Fig. 2. In this setup, the valve is used to position a piston with different loads at various locations. Control is performed by a 486 personal computer equipped with a 12 bit data acquisition board. For all the controllers, the sampling rate was set at 200 Hz, with the anti-aliasing filter set at 80 Hz. The command signal from the computer is amplified by a 24W current amplifier to provide translational force to the spool, which in turn controls the flow of air into the cylinder chamber (see Fig. 2). Feedback is provided by an encoder which measures the position of the piston through a cable which is connected to the end of the piston and wraps around the spool of a torquer.

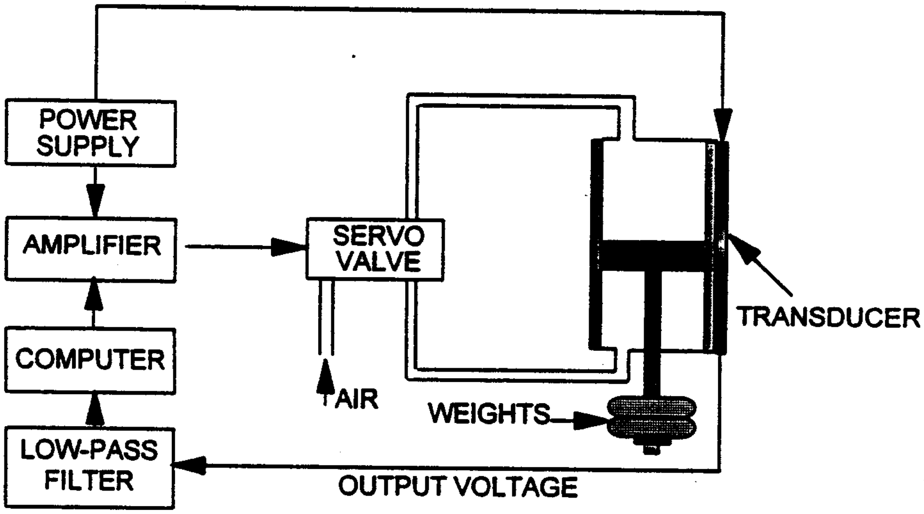

Figure 2: Schematic of the experimental setup.

In order to study the degree of linearity of the system to the air supply pressure, step responses of the pneumatic servo system to air supply pressures at 40 psi, 60 psi, and 80 psi were obtained (see Fig. 3). The results indicate that step responses contain an initial time delay which is almost the same for all of the three different air pressures. This time delay is suspected to be due to the initial accumulation of air pressure in the cylinder. The results also indicate that the piston responses are not proportional to the air pressures, reflecting the nonlinearity of the system.

The response of the system was also studied to various input voltages to the valve (see Fig. 4). Nonlinearity is also observed in these responses. The nearly similar response of the piston to 0.25 volts and 0.50 volts is suspected to be caused by the relatively high pressure difference on the two sides of the spool near its null position, which is overcome by the 1 volt input providing relatively a much faster response than the smaller inputs of 0.25 and 0.50 volts. The small ripples in the feedback signal are caused by fluctuations in the feedback mechanism. As observed from the response of the piston to various inputs, the system is a highly nonlinear system. This nonlinearity poses stringent limitations on the controller used for the valve.

3 CONTROLLER DESIGN

Accurate and rapid control is an essential part of every servo mechanism. To investigate the feasibility of various controllers for the valve, three different controllers were designed and implemented on the cylinder-piston system (see Fig. 2). These controllers were Proportional-Integral-Derivative (PID), Fuzzy Logic Control (FLC), and Self-Organizing Fuzzy Control (SOC). PID control is widely used in industry for its simplicity and low cost. However, it is known to have

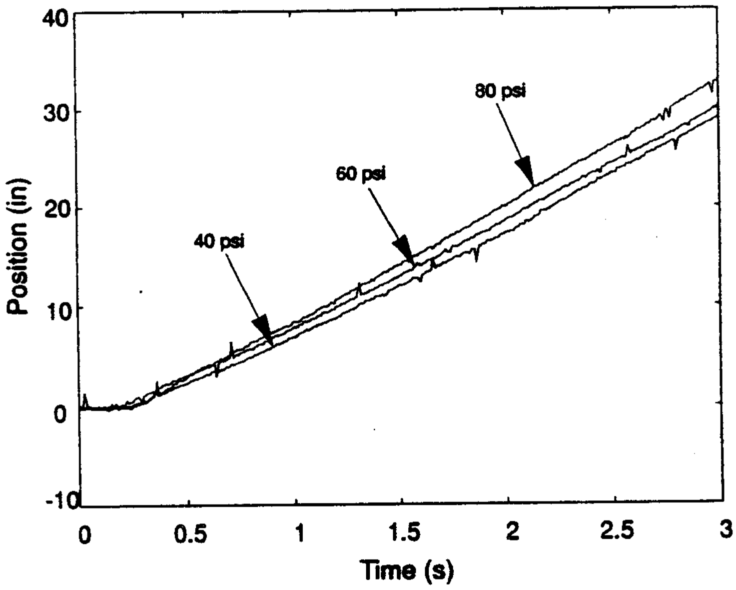

Figure 3: Step response of the system to various air supply pressures.

difficulty with systems that contain high levels of nonlinearity. The advantage of the FLC is that it is inherently a nonlinear controller, so it has a better potential to cope with load changes, variations in the operating conditions, and noise [6,7]. A fuzzy logic controller consists basically of a set of heuristic decision rules which are devised by an expert for various operating conditions. These rules are usually compiled *a-priori* and contained in a "look-up table" to avoid on-line computation [6,7,14]. Successful applications of fuzzy logic control are numerous, for example, they have been applied to servo motor position control [8], warm water plant control [10], automatic car parking [13], and hydraulic cylinder position control [4]. However, the rules in FLC do not have adaptation capability. A modified version of FLC that incorporates tuning is Self-Organizing Fuzzy Control (SOC), which adjusts the control rules on-line according to the system performance [12,5]. In order to test the robustness of the controllers to different operating settings of the valve, the piston was moved to two positions in consecutive steps.

3.1 PID Controller

As a first attempt to control the valve, PID control was used. Since the system behaved nonlinearly at different operating conditions [1], gain scheduling was adopted. Two PIDs were designed, one for each step of the two-step motion. The response of the piston moved by PID action to its two positions is shown in Fig. 5. The results indicate that the system has a relatively large settling time (about 2.3 seconds) and considerable initial delay. As expected, this is due to the inadequacy of PID to cope with nonlinearity.

3.2 Fuzzy Logic Control

Fuzzy set theory provides a suitable framework for control of nonlinear systems that need to operate under a wide range of operation conditions. The basic structure of fuzzy logic control is shown in Fig. 6. Unlike a regular analytic controller, a FLC is composed of rules which stipulate

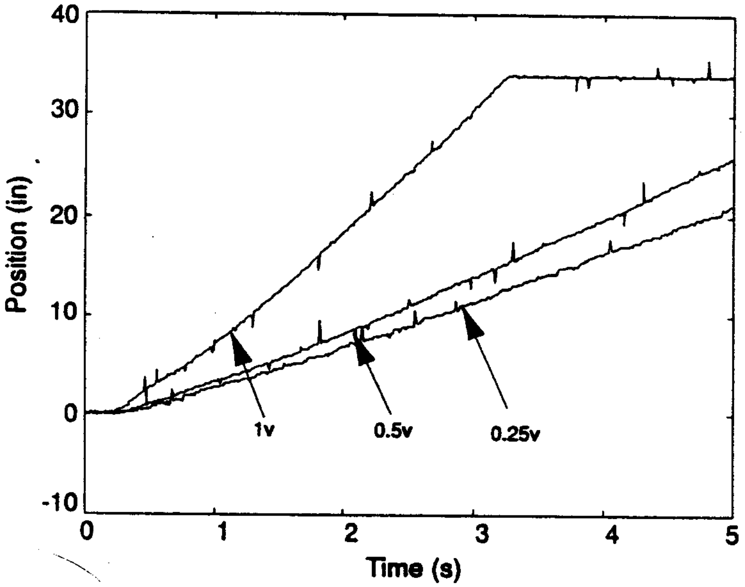

Figure 4: Step response of the system to various input voltages.

that a certain action be initiated should a special set of conditions occur. The generic form of such a rule is:

$$\text{if } A_i \text{ is } \tilde{A}_i \text{ and } B_i \text{ is } \tilde{B}_i \text{ then } C_i \text{ is } \tilde{C}_i$$

where A and B denote the input variables, C represents the FLC output, and $\tilde{A}$, $\tilde{B}$, and $\tilde{C}$ denote the linguistic qualifiers of A, B and C, respectively [14].

Usually in FLC, the error (e) and the rate of change of the error (ee), defined as

$$e(k) = y(k) - r(k) \tag{1}$$

$$ee(k) = e(k) - e(k-1) \tag{2}$$

are used as inputs to the controller, where k denotes the sample point, r is the reference input (target position of the piston), and y is the system output (actual position of the piston).

In order to develop the fuzzy rules, various possible values (*crisp values*) of inputs e and ee within the range of motion of the piston were quantized into the interval [-6,6] by the linear transformation:

$$X = \frac{12}{b-a}\left[x - \frac{a+b}{2}\right] \tag{3}$$

where X represents the quantized value of the variable, $x \in [a, b]$ denotes its crisp value, and a and b represent the limits of the quantization range. This procedure is referred to as *fuzzification*, and a and b are called *fuzzification parameters*. Once the variables are quantized, they need to be represented by linguistic qualifiers. In this problem, seven linguistic qualifiers were used: PL, PM, PS, ZO, NS, NM, NL, where N means negative, P means positive, L means large, M means medium, S means small, and ZO means zero. The triangular shape membership functions defining the association of these qualifiers with the quantized values of inputs are illustrated in Figs. 7 and 8, respectively. The membership values obtained from the membership functions for E and EE are shown in Table 1.

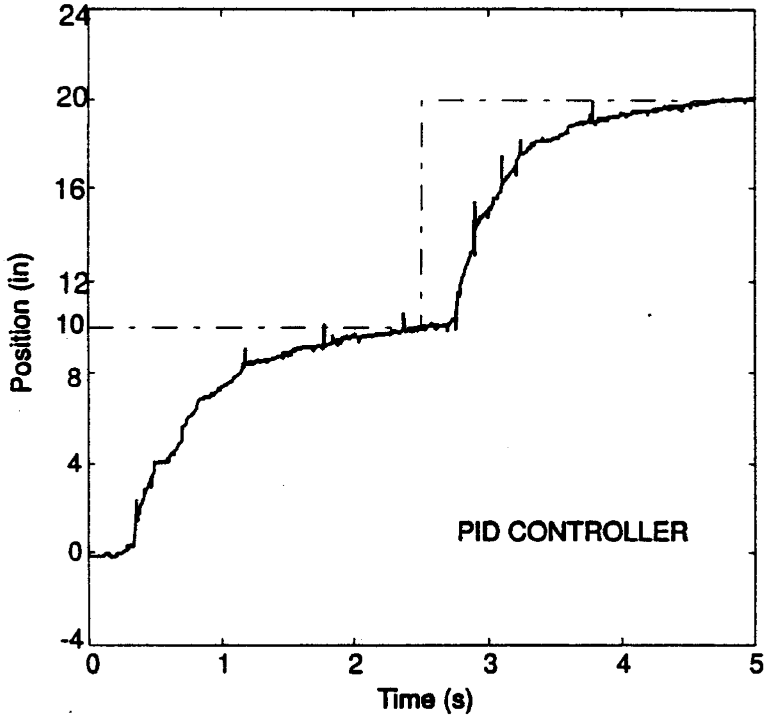

Figure 5: Two-step system response resulted from PID action.

In fuzzy logic control, the control action is decided based on experience in terms of the linguistic status of the inputs. These rules have the form, for example,

If E is PL and EE is ZO then U is NM;

The rules used for the piston control are summarized in the linguistic control decision Table 2. As can be seen from Fig. 7, each quantized value relates to two linguistic qualifiers, except when the quantized value is at the center of the membership function. As such, for each pair of control inputs, a maximum of four linguistic rules can be envoked. In order to integrate these rules such that they result into one control action, the center of the gravity method [11] was used. This method can be used to derive the control action on-line. However, in order to save computation time during on-line application, the control actions were derived *a-priori* and included in a lookup table (see Table 3).

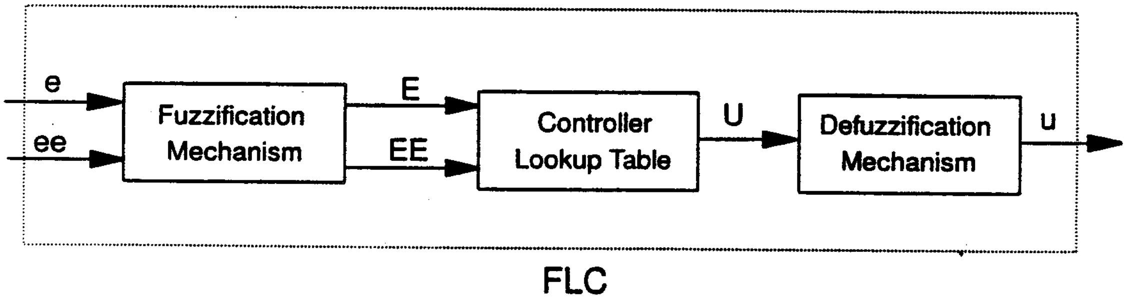

Figure 6: Structure of fuzzy logic control.

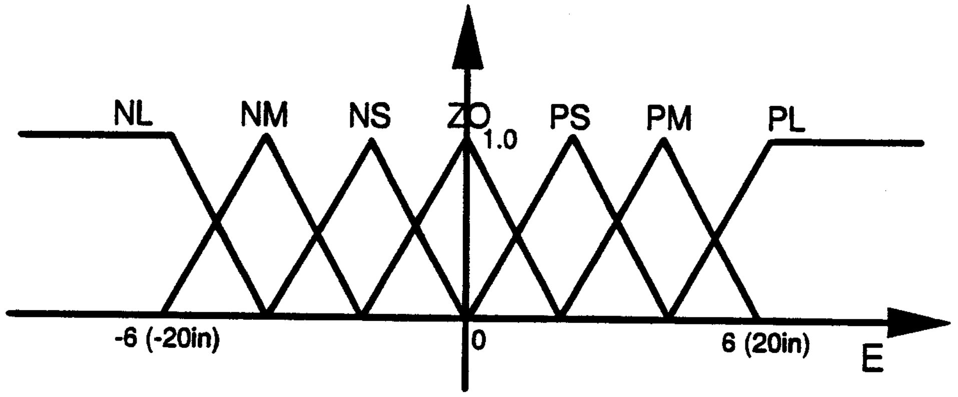

Figure 7: Membership functions used for error, *e*.

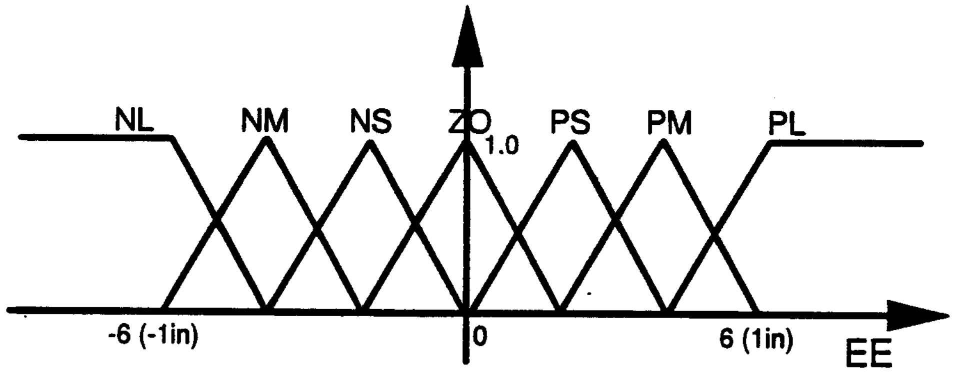

Figure 8: Membership functions used for the rate of error change, *ee*.

Once the quantized value of the control action is determined, it needs to be *de-quantized* so that its crisp value can be obtained. This was performed by the following formula:

$$u = U \times ua/6 \tag{4}$$

where u is the crisp value of the control action, U is its quantized value, and ua is a constant within the saturation limit of the control action determined based on the sampling rate of the system.

Control actions from Table 3 were applied to the piston. The response of the piston, as controlled by FLC is shown in Fig. 9 and compared with that obtained from PID. The results indicate that the response of the FLC-controlled piston is much improved in terms of both response time and steady state error. In order to test the robustness of this controller in presence of load changes, it was also applied to the piston with an added weight of 10 lb. The two-step responses of the piston with and without load resulted by FLC action are shown in Fig. 10. The results indicate that the piston response with load has a faster rise time and larger steady-state error. The faster rise time for the loaded piston was expected, since the load helps the piston move downward faster (see Fig. 2).

3.3 Self-Organizing Fuzzy Control

From the results of Fig. 10 it can be observed that the performance of the FLC is affected by load changes. In order to investigate the effect of adaptation on FLC's performance, a Self-Organizing Fuzzy Controller (SOC) [11,2] was considered. The concept behind adaptation in

	QUANTIZED LEVEL												
E or *EE*	-6	-5	-4	-3	-2	-1	0	1	2	3	4	5	6
NL	1	0.5	0	0	0	0	0	0	0	0	0	0	0
NM	0	0.5	1	0.5	0	0	0	0	0	0	0	0	0
NS	0	0	0	0.5	1	0.5	0	0	0	0	0	0	0
ZO	0	0	0	0	0	0.5	1	0.5	0	0	0	0	0
PS	0	0	0	0	0	0	0	0.5	1	0.5	0	0	0
PM	0	0	0	0	0	0	0	0	0	0.5	1	0.5	0
PL	0	0	0	0	0	0	0	0	0	0	0	0.5	1
	MEMBERSHIP VALUE												

Table 1: Membership values obtained from the membership functions of e and ee.

FLC is explained as follows. The control action in FLC is related to the error and the rate of change of error via the membership functions. That is, the control action will be larger when the membership functions are more compact, and vice versa (e.g., see Fig. 7). As such, the control action can be increased for small errors by reducing the quantization range $(b - a)$ as shown in Fig. 11. This is achieved by using the adaptation formula

$$b - a = k \times e \tag{5}$$

where $k > 1$ is a constant. The above adaptation rule, however, has the adverse effect of reducing control action for large errors such that a slow response is resulted when the error is large. Moreover, it may cause oscillations by producing large control actions near the target position when the error is small. In order to safeguard against these drawbacks, the control action was adjusted in the defuzzification process, as

$$ua = k_1/e \tag{6}$$

where $k_1 > 1$ when error $e > 0.1$ inch, otherwise $k_1 < 1$. Furthermore, in order to reduce the delay time and increase the initial speed of the system, the SOC was bypassed by using maximum control for large errors using a two-mode control architecture (see Fig. 12). With this control configuration, when the error was large ($|e| \geq |e_m|$), the output of the controller is set at its maximum level, so the SOC is only used when $|e| \leq |e_m|$.

The two-mode controller was applied to the system with $k = 1.2$, $k_1 = 1.25$ or 0.75 for SOC. The resulted two-step response of the system from this controller is shown in Fig. 13, along with the response obtained from the FLC. The results indicate that the two-mode controller reduced the initial delay by nearly 100% (from 0.4 s to 0.2 s) compared to the FLC by using the maximum control effort at the initial stage of the piston response. The results also indicate that the SOC provided a smoother response as the piston approached its target position, despite the considerable amount of noise present in the feedback signal. It also had a faster rise time during the second step, which was caused in part by the inclusion of small ripples in the control effort by the SOC. These ripples keep the valve spool constantly moving, thus, eliminate the need to overcome static friction during the initial stage of the second step response.

In order to investigate the robustness of the two-mode controller, it was also applied to the piston with load (see Fig. 14). The results indicate that the two-mode controller also improved the response of the loaded piston. The slow response of the piston as it approaches its target position is due to the delay in air accumulation, the same phenomenon observed during the initial stage of the piston response.

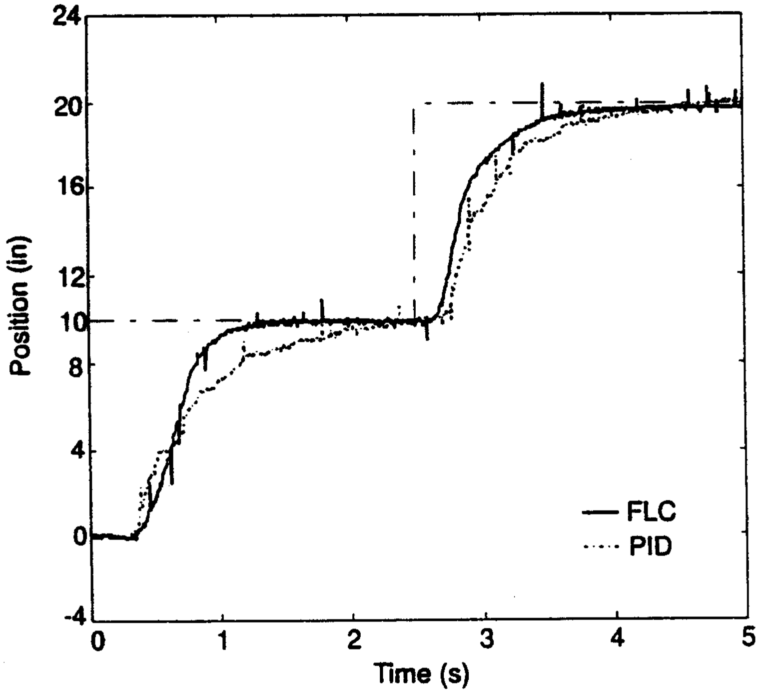

Figure 9: Two-step system responses resulted from PID and FLC actions.

4 DISCUSSION

The results obtained from the developed servo-valve control suggest that fuzzy logic control provides a more suitable structure for its control. The positioning accuracy obtained from the SOC is in the same order of magnitude as required in industrial applications, which is traditionally provided by hydraulic valves. The absence of any overshoots resulted from SOC is also important, as it is often required in pick-and-place operations. The adaptability of the SOC is an important feature of this controller, making it quite versatile in application to different loading situations.

ACKNOWLEDGMENT

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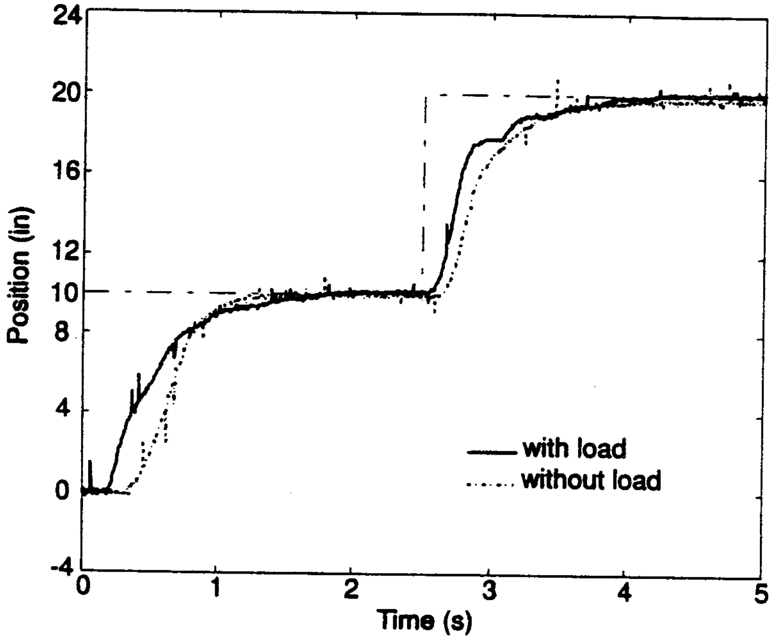

Figure 10: Two-step system responses with and without load by FLC action.

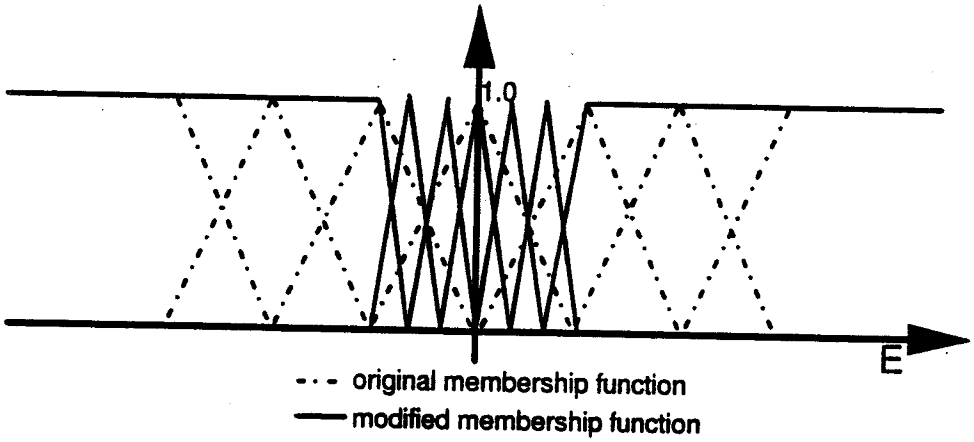

Figure 11: Adaptation of membership functions in SOC.

ERROR	ERROR CHANGE RATE						
	NL	NM	NS	ZO	PS	PM	PL
NL	PM	PS	PL	PL	PL	NM	NL
NM	PM	PS	PS	PM	PM	NM	NL
NS	PL	PM	PS	PS	PS	NM	NL
ZO	PL	PM	PS	ZO	NS	NM	NL
PS	PL	PM	NS	NS	NS	NM	NL
PM	PL	PM	NM	NS	NS	NS	NM
PL	PL	PM	NL	NM	NL	NS	NM
	CONTROL ACTION						

Table 2: Fuzzy logic control rules.

	ERROR CHANGE RATE												
ERROR	-6	-5	-4	-3	-2	-1	0	1	2	3	4	5	6
-6	6	5	6	5	6	6	6	3	3	1	0	0	0
-5	5	5	5	5	5	5	5	3	3	1	0	0	0
-4	6	5	6	5	6	6	6	3	3	1	0	0	0
-3	6	5	5	5	5	3	3	2	1	0	-1	-1	-1
-2	3	3	3	4	3	1	1	1	0	0	-1	-1	-1
-1	3	3	3	4	1	1	0	0	0	0	-1	-1	-1
0	3	3	4	4	1	0	0	-1	-1	-1	-3	-3	-3
1	1	1	1	1	0	0	-1	-1	-1	-1	-2	-2	-3
2	1	1	1	1	0	-1	-1	-1	-2	-2	-2	-3	-3
3	0	0	0	0	-1	-1	-2	-2	-3	-3	-3	-5	-5
4	0	0	0	-1	-2	-2	-2	-3	-4	-4	-5	-5	-6
5	0	0	0	-1	-2	-3	-5	-5	-5	-5	-5	-5	-5
6	0	0	0	-1	-3	-3	-6	-6	-6	-5	-6	-5	-6
	CONTROL ACTION												

Table 3: Crisp values of the control action.

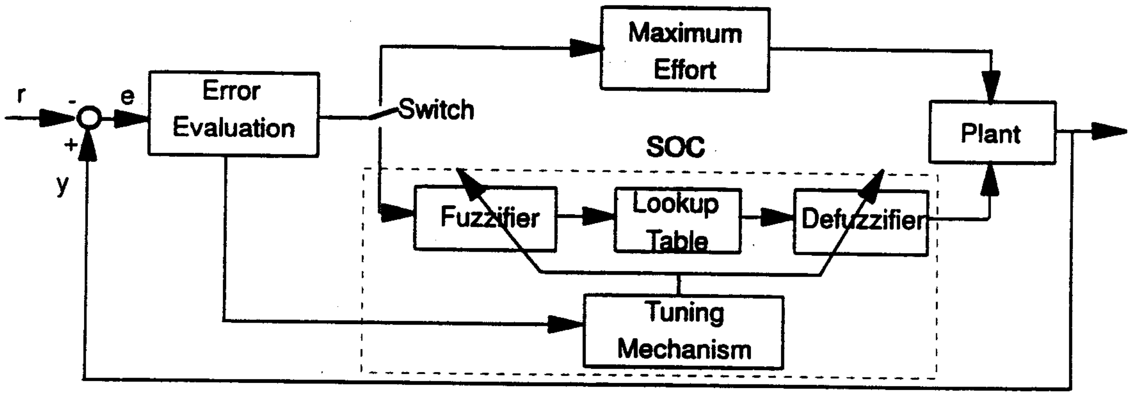

Figure 12: Two-mode control strategy used with SOC.

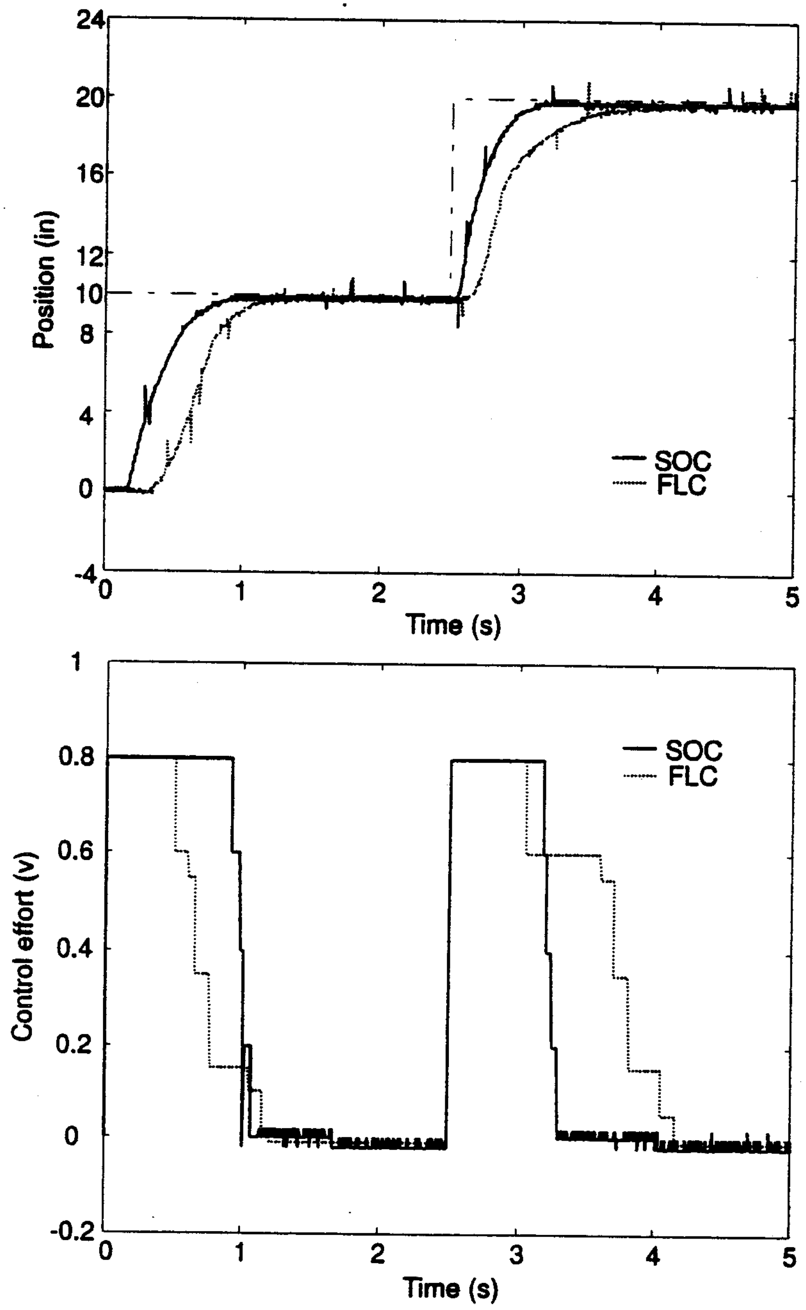

Figure 13: Two-step system responses without Load by two-mode control and FLC.

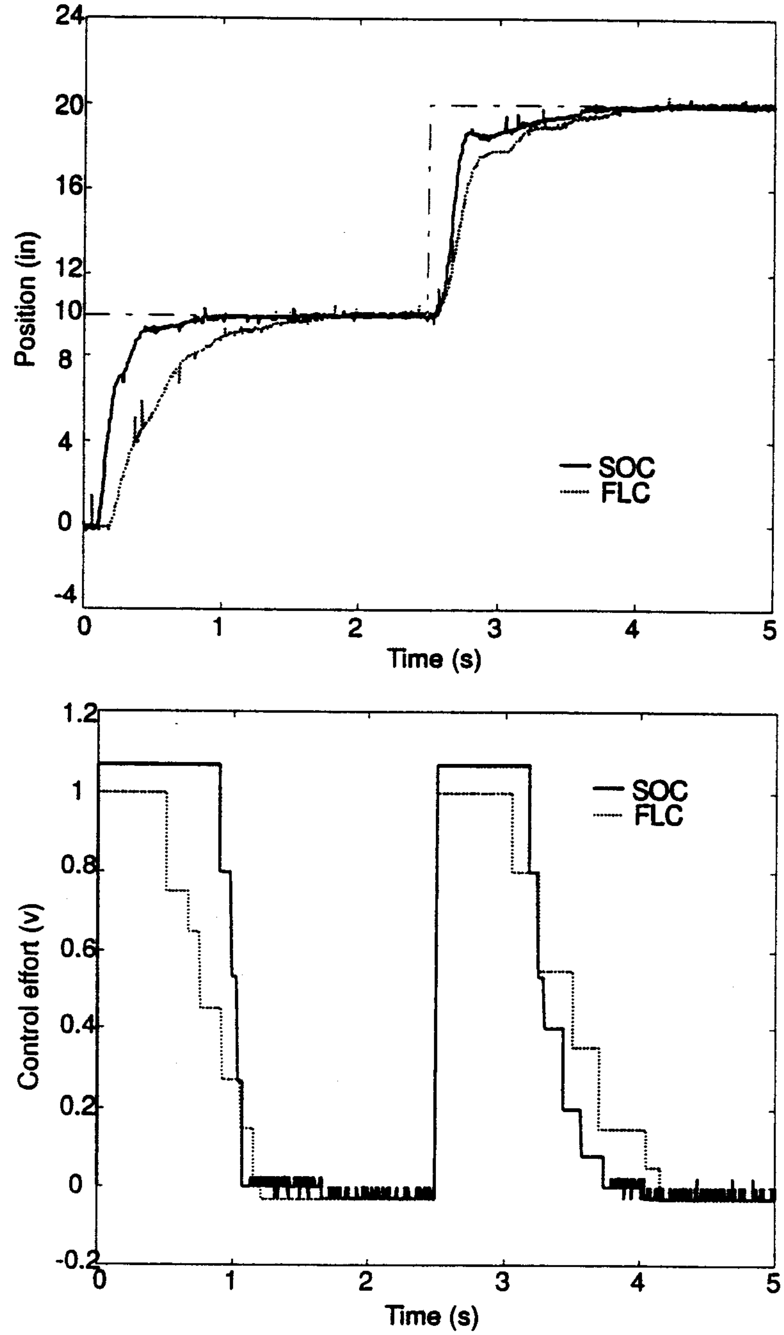

Figure 14: Two-step system responses with Load by two-mode control and FLC.

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