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# APPLICATION OF FUZZY GAIN-SCHEDULING IN POSITION CONTROL OF A SERVO HYDRAULIC SYSTEM WITH A FLEXIBLE LOAD

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**Abstract**

The control of hydraulic servo-systems has been the focus of intense research over the past decades. Hydraulic position servos with an asymmetrical cylinder are commonly used in industry. These kinds of systems are nonlinear in nature and generally difficult to control. Changing system parameters using the same gains will cause overshoot or even loss of system stability. The highly non-linear behaviour of these devices makes them ideal subjects for applying different types of sophisticated controllers. The paper is concerned with a second order model reference to positioning control of a flexible load servo-hydraulic system using fuzzy gain-scheduling. In the present study, to compensate the lack of damping in a hydraulic system, an acceleration feedback was used. To compare the results, a p-controller with feed-forward acceleration and different gains in extension and retraction is used. The design procedure for the controller and experimental results are discussed. The results suggest that using the fuzzy gain-scheduling controller decrease the error of position reference tracking.

**Keywords:** servo hydraulic, fuzzy controller, reference model, position control

## 1 Introduction

Electro-hydraulic servomechanisms are known for their fast dynamic response, high power/inertia ratio and control accuracy.

Many types of sophisticated controllers (adaptive, optimal, etc.) have been applied to hydraulic servomechanisms with varying degrees of success (Edge et al., 1987). Some of these controllers have limitations. The model reference adaptive controller, for example, generally requires a considerable amount of CPU time for effective compensation (Tamaki et al., 1986). In addition, the computational complexity of implementing the algorithms has restricted their applications.

Fuzzy logic has found control applications in all facets of engineering. In the fluid power area, fuzzy logic has been used for control, identification and modelling of the system (Zhao et al., 1995; Deticek, 2000). The popularity of fuzzy logic can be attributed to its ability to deal with nonlinear systems. Also, easy tuning of such systems makes them a good candidate to control hydraulic systems.

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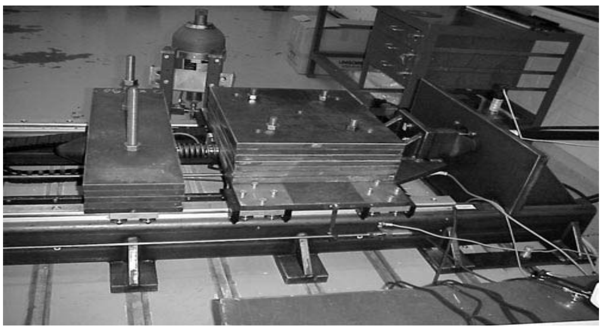

**Fig. 1:** *Setup of the servo hydraulic system*

Since the first fuzzy controller was invented by Mamdani, many applications of fuzzy controllers have been used this particular type of fuzzy inference system. Usually, the fuzzy controller is composed of linguistic control rules which are conditional linguistic statements of the relationship between inputs and outputs, so that the most attractive property of a fuzzy controller is its ability to emulate the behaviour of a human operator. Another important characteristic of a fuzzy controller is its applicability to systems with model uncertainty or even to an unknown model.

Classical approaches, like P or PD regulators for positioning of hydraulic drives, do not give satisfactory performance. For this reason, adaptive control techniques (reference model) are used.

A comparatively convenient method is to have a reference response model, which can be the tracking object of the control system. By using the model reference adaptive control theory (Franklin et al., 1986; Unbehauen et al., 1989), an adaptive reference model is used and implemented in the microcomputer to control a servo-hydraulic system.

Gain scheduling based on the measurements of the operation conditions of the process is often a good way to compensate for variations in the process parameters or the known nonlinearities of the process.

Owing to the low damping ratio in hydraulic systems, the performance of the controller is not satisfactory. To overcome this problem, generally load pressure or acceleration feed-forward is used.

In the study a fuzzy gain-scheduling to track the reference model is proposed. To verify the performance of the proposed controller, simulation and experimental results are compared with a commonly used p-controller. Both controllers are used in feed-forward acceleration to increase the system damping.

The paper is organised as follows. In Section 2 the dynamic equations of the system under study are presented. The nonlinear controllers are developed in Section 3. In Section 4 and 5 simulation and experimental results are discussed respectively and finally conclusions are drawn in Section 6.

## 2 Servo-Hydraulic System

The servo-hydraulic system with flexible load shown in Fig. 1 is comprised of a servo-valve, a hydraulic cylinder and two masses that are connected by a parallel combination of a spring and damper. The schematic diagram of the system is illustrated in Fig. 2.

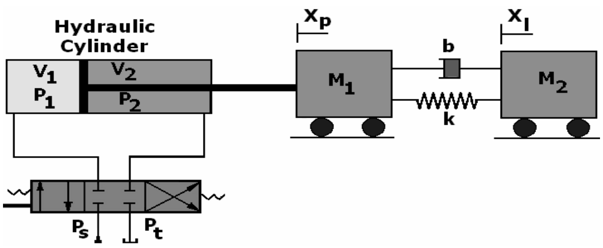


**Fig. 2:** *Schematic diagram of the system*

The nonlinearity of the constitutive equations as well as the sensitivity of the system's parameters to the sign of the voltage fed to the valve make the control of system complicated (Viersma, 1980).

### 2.1 System Model

In order to represent servo-valve approximation dynamics through a wider frequency range, a first or second order transfer function is used. For low frequencies (up to about 50 Hz), a first order approximation may be sufficient (Niksefat et al., 2001). The relation between servo-valve spool position $X_s$ and the input voltage $u$ can be written as,

$$\frac{X_s}{u} = \frac{\tau_1}{s+\tau_2} \tag{1}$$

Where, $\tau_1$ is the valve gain and $\tau_2^{-1}$ is the time constant.

Applying Newton's second law to each mass, taking friction into account yields,

$$\begin{aligned} m_1\ddot{X}_p &= -b(\dot{X}_p - \dot{X}_L) - k(X_p - X_L) \\ &\quad + p_1A_1 - p_2A_2 - F_f \end{aligned} \tag{2}$$

$$m_2\ddot{X}_L = -b(\dot{X}_L - \dot{X}_p) - k(X_L - X_p) \tag{3}$$

Friction in the hydraulic cylinder is taken into account as an external disturbance. Contact between two surfaces occurs at surface asperities (microscopic roughness) (Owen et al., 2003). Due to the tight sealing, hydraulic cylinders feature a strong dry friction effect. The behaviour of this friction force is rather complex (Lischinsky et al., 1999; Olsson et al., 1998). Friction is usually modelled as a discontinuous static mapping between the velocity and the friction force that depends on the velocity's sign. It is often restricted to the Coulomb and viscous friction components. However, there are several important properties observed in systems with friction which cannot be explained by static models only. This is basically due to the fact that the friction does not have an instantaneous response to a change in velocity, i.e. it possesses internal dynamics. Examples of these complex properties are: (1) stick-slip motion, characterised by large friction at rest and at low velocities, and small friction during rapid motion; (2) pre-sliding displacement, which shows that the friction behaves like a spring when the applied force is less than the static friction break-away force; (3) friction lag, which means that there is a hysteresis that characterises the relationship between the friction and velocity. All these static and dynamic characteristics of the friction are captured by the analytic model of friction dynamics proposed in (Wit et al., 1995), which is called the LuGre model and defined by,

$$\frac{dz}{dt} = \dot{X}_p - \frac{|\dot{X}_p|}{g(\dot{X}_p)} z \tag{4}$$

$$g(\dot{X}_p) = \frac{1}{\sigma_0}\left( F_c + (F_s - F_c) e^{-\left(\frac{\dot{X}_p}{v_s}\right)^2} \right) \tag{5}$$

$$F_f = \sigma_0 z + \sigma_1 \frac{dz}{dt} + k_v \dot{X}_p \tag{6}$$

where $\dot{X}_p$ is the piston velocity, and $F_f$ is the friction force described by a linear combination of $z$, d$z$/d$t$ and viscous friction. Equation (6) represents the dynamics of the friction where the internal state $z$, is not measurable. The function $g(\dot{X}_p)$ describes part of the "steady state" characteristics of the model for constant velocity motions: $v_s$ is the Stribeck velocity, $F_s$ is the static friction, $F_c$ is the Coloumb friction, $k_v$ is the viscous friction. Thus, the complete friction model is characterised by four static parameters and two dynamic parameters, a stiffness coefficient and a damping coefficient.

The friction parameters are difficult to estimate since they appear nonlinearly in the model and the average deflection of the bristles cannot be measured (Wit et al., 1995).

Continuity equations for the output ports of the servo valve results (Jelali et al., 2004),

$$\begin{cases} \dot{p}_1 = \dfrac{\beta_e}{V_1}(Q_1 - A_1\dot{X}_p + Q_{Li} - Q_{Le1}) \\ \dot{p}_2 = \dfrac{\beta_e}{V_2}(-Q_2 + A_2\dot{X}_p - Q_{Li} - Q_{Le2}) \end{cases} \tag{7}$$

Here, $Q_{Li}$ is the internal leakage flow. $Q_{Le1}$ and $Q_{Le2}$ are the external leakage flows for ports one and two, respectively. The internal leakage flow and external leakages (i.e., leakage from one port of the chamber to the other) can be calculated by (provided the flow is laminar),

$$\begin{cases} Q_{Li} = L_i(p_2 - p_1) \\ Q_{Le1} = L_{e1}(p_1 - p_t) \\ Q_{Le2} = L_{e2}(p_2 - p_t) \end{cases} \tag{8}$$

where, $L_i$ is the internal leakage flow coefficient, $L_{e1}$ and $L_{e2}$ are the external leakage coefficients.

There are lots of empirical formulas for the effective bulk modulus $\beta_e$, including the effect of mechanical compliance, based on direct measurements. The commonly used equation for the bulk modulus $\beta_e$ for a hydraulic cylinder in German literature is (Jalali et al., 2004),

$$\beta_e = a_1 E_{max} \log\left( a_2 \frac{p}{p_{max}} + a_3 \right) \tag{9}$$

Parameters $E_{max}$ = 18000 bar and $p_{max}$ = 280 bar. The other parameters, $a_1$, $a_2$ and $a_3$ must be identified.

The volumes between the valve and each side of the piston are calculated as,

$$\begin{cases} V_1 = A_1 X_p + v_{01} \\ V_2 = A_2(L - X_p) + v_{02} \end{cases} \tag{10}$$

Introducing two new parameters, $C_1$ and $C_2$, defined as,

$$C_1 = \frac{V_1}{\beta_e}, \; C_2 = \frac{V_2}{\beta_e} \tag{11}$$

Equation (7) can be written in the form of,

$$\begin{cases} \dot{p}_1 = \dfrac{1}{C_1}(Q_1 - A_1\dot{X}_p + Q_{Li} - Q_{Le1}) \\ \dot{p}_2 = \dfrac{1}{C_2}(-Q_2 + A_2\dot{X}_p - Q_{Li} - Q_{Le2}) \end{cases} \tag{12}$$

The nonlinear equations of flow rate of the valve can be written in the following form (Viersma, 1980),

$$Q_1 = \begin{cases} c_s X_s \sqrt{p_s - p_1} & u \geq 0 \\ c_s X_s \sqrt{p_1 - p_t} & u < 0 \end{cases} \tag{13}$$

$$Q_2 = \begin{cases} c_s X_s \sqrt{p_2 - p_t} & u \geq 0 \\ c_s X_s \sqrt{p_s - p_2} & u < 0 \end{cases} \tag{14}$$

The nonlinear equations of the system are used for designing the controllers in simulation mode. The natural frequency and damping ratio of the system is important from the point of view of controller design and they are calculated from the linearized state space model of the system.

The following equations are the standard linearized form of Eq. 13 and 14,

If $u \geq 0$

$$\begin{cases} Q_1 = K_{v1} X_s - K_{p1} p_1 \\ Q_2 = K_{v2} X_s + K_{p2} p_2 \end{cases} \tag{15}$$

Where the parameters of the linearized form are defined as,

$$K_{v1} = c_s \sqrt{p_s - p_{10}}, \; K_{v2} = c_s \sqrt{p_{20}} \tag{16}$$

$$K_{p1} = \frac{Q_{10}}{2(p_s - p_{10})}, \; K_{p2} = \frac{Q_{20}}{2 p_{20}} \tag{17}$$

The linearized equations for the rate of pressures by using Eq. 12 and neglecting the leakages can be written in the following forms,

$$\begin{cases} \dot{p}_1 = -\dfrac{A_1}{C_1}\dot{X}_p - \dfrac{K_{p1}}{C_1} p_1 + \dfrac{K_{v1}}{C_1} X_s \\ \dot{p}_2 = \dfrac{A_2}{C_2}\dot{X}_p - \dfrac{K_{p2}}{C_2} p_2 + \dfrac{K_{v2}}{C_2} X_s \end{cases} \tag{18}$$

The state space model of the linear system is as follows:

$$\begin{cases} \dot{\boldsymbol{X}} = \mathbf{A}\boldsymbol{X} + \mathbf{B}u \\ y = \mathbf{C}\boldsymbol{X} + \mathbf{D}u \end{cases} \tag{19}$$

The states of system are defined as,

$$\begin{cases} X_1 = X_s, \; X_2 = X_L, \; X_3 = \dot{X}_L \\ X_4 = X_p, \; X_5 = \dot{X}_p, \; X_6 = p_1 \\ X_7 = p_2 \end{cases} \tag{20}$$

For positive input voltage ($u \geq 0$) by neglecting the friction in Eq. 2 we have,

$$\dot{X}_7 = \frac{A_2}{C_2} X_5 - \frac{K_{p2}}{C_2} X_7 + \frac{K_{v2}}{C_2} X_1 \tag{21}$$

$$\dot{X}_6 = -\frac{A_1}{C_1} X_5 - \frac{K_{p1}}{C_1} X_6 + \frac{K_{v1}}{C_1} X_1 \tag{22}$$

$$\begin{aligned} \dot{X}_5 = & \frac{k}{m_1} X_2 + \frac{b}{m_1} X_3 - \frac{k}{m_1} X_4 \\ & - \frac{b}{m_1} X_5 + \frac{A_1}{m_1} X_6 - \frac{A_2}{m_1} X_7 \end{aligned} \tag{23}$$

$$\dot{X}_4 = X_5 \tag{24}$$

$$\dot{X}_3 = -\frac{k}{m_2}X_2 - \frac{b}{m_2}X_3 + \frac{k}{m_2}X_4 + \frac{b}{m_2}X_5 \tag{25}$$

$$\dot{X}_2 = X_3 \tag{26}$$

$$\dot{X}_1 = -\tau_2 X_1 + \tau_1 u \tag{27}$$

The **A** matrix will be,

$$\mathbf{A} = \begin{bmatrix} -\tau_2 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & -\frac{k}{m_2} & -\frac{b}{m_2} & \frac{k}{m_2} & \frac{b}{m_2} & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & \frac{k}{m_1} & \frac{k}{m_1} & -\frac{k}{m_1} & -\frac{b}{m_1} & \frac{A_1}{m_1} & -\frac{A_1}{m_1} \\ \frac{K_{v1}}{C_1} & 0 & 0 & 0 & -\frac{A_1}{C_1} & -\frac{K_{p1}}{C_1} & 0 \\ \frac{K_{v2}}{C_1} & 0 & 0 & 0 & \frac{A_2}{C_2} & 0 & \frac{K_{p2}}{C_2} \end{bmatrix} \tag{28}$$

In addition, Matrixes **B**, **C**, and **D** can be written in the following forms respectively,

$$\mathbf{B}^{T} = \begin{bmatrix} \tau_1 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \tag{29}$$

$$\mathbf{C} = [0\ 1\ 0\ 0\ 0\ 0\ 0] \tag{30}$$

$$\mathbf{D} = 0 \tag{31}$$

Using the setup parameters, the dominant poles of the system are at positions $-0.17 \pm 25.78i$. The minimum natural frequency of the system and the related damping ratio can be determined by,

$$\begin{aligned} \omega_n &= \sqrt{\mathrm{Re}^2 + \mathrm{Im}^2} \\ &= \sqrt{0.17^2 + 25.78^2} \\ &= 25.78 \end{aligned} \tag{32}$$

$$\varsigma = \left|\frac{\mathrm{Re}}{\mathrm{Im}}\right| = \frac{0.2}{25.78} = 0.0066 \tag{33}$$

The setup parameters are shown in Table 1. Notice that for very low damping the response of the system is oscillatory, while for large damping ($\varsigma$ near 1) the response consists of very little oscillation.

**Table 1:** Setup Parameters

| | |
|---|---|
| $m_1 = 210$ kg | $L = 1$ m |
| $m_2 = 80$ kg | $\tau_1 = 0.471$ |
| $A_1 = 8.04\times10^{-4}$ $m^2$ | $\tau_2 = 490$ 1/s |
| $A_2 = 4.24\times10^{-4}$ $m^2$ | $b = 418.33$ Ns/m |
| $v_{01} = 2.13\times10^{-4}$ $m^3$ | $P_s = 14$ MPa |
| $v_{02} = 1.07\times10^{-4}$ $m^3$ | $P_t = 0.9$ MPa |
| $c_v = 2.36\times10^{-5}$ $\frac{m^3}{sVPa^{1/2}}$ | $a_1 = 0.1972$ |
| $L_i = 1724\times10^{-13}$ $\frac{m^4 s}{kg}$ | $a_2 = 124.368$ |
| $L_{e1} = 2.4\times10^{-12}$ $\frac{m^4 s}{kg}$ | $a_3 = 26.814$ |
| $L_{e2} = 1.02\times10^{-13}$ $\frac{m^4 s}{kg}$ | $k = 42950$ N/m |
| $\sigma_0 = 97525.459$ m/s | $\sigma_1 = 385.167$ Ns/m |
| $K_v = 376.613$ Ns/m | $F_c = 247.804$ N |
| $F_s = 7485.084$ N | $V_s = 0.026318$ m/s |

Following this, the structures of the controllers for positioning tracking of the piston mass and load mass are proposed.

## 3 Controller Design

In the study, the aim of the controller is for position tracking of the reference model. Classical approaches, like P or PD regulators for positioning of hydraulic drives, do not give satisfactory performance. For this reason, adaptive control techniques, an adaptive reference model, and gain scheduling are used to improve the performance of the controllers (Franklin et al., 1986, Passino et al., 1998 and Jalali et al., 2004). The variations in the parameters depending on the change in the sign of voltage fed to the valve are compensated by a fuzzy gain scheduling block. The acceleration feed-forward improves the lack of damping in the hydraulic systems. Following this the reference model is designed.

### 3.1 Reference Model

The desired linear, second order reference model is selected to run parallel with the nonlinear system. From Eq. 32 the natural frequency, $\omega_n$ of this model is set equal to 26 rad/sec. The damping ratio is considered to be $\zeta = 0.866$.

$$G_{ref}(s) = \frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} \tag{34}$$

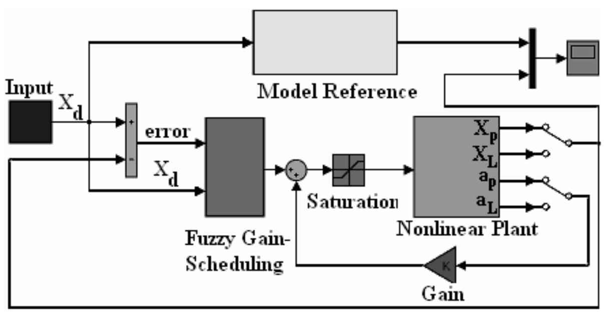


**Fig. 3:** *Schematic diagram of the proposed controller*

There are different kinds of methods to find a reference model such as ITAE or Bessel transfer functions. The natural frequency of the system was chosen in a manner so that the response of the system is as fast as possible. The chosen damping ratio is provided by using the second order Bessel transfer function (Franklin et al., 1986).

The proposed controller depicted in Fig. 3 is composed of feed forward acceleration and Fuzzy Gain Scheduling. The feed forward gain is chosen such that the vibration is as low as possible and at the same time the controller is as fast as possible. Note that by increasing the gain of acceleration the controller becomes slower and vibration is decreased. The gain is found by a trial and error method.

### 3.2 Fuzzy Controller Design

Figure 4 shows the fuzzy logic control system. It usually consists of four parts: Fuzzification, Rule Base, Inference Engine and Defuzzification (Passino et al., 1998; Jelali et al., 2004; Zimmermann, 2001). The first block inside the controller is Fuzzification, which converts each piece of input data to degrees of membership. The Fuzzification block thus matches the input data with the conditions of the rules to determine how well the condition of each rule matches that particular input instance. There is a degree of membership for each linguistic term that applies to that input variable.

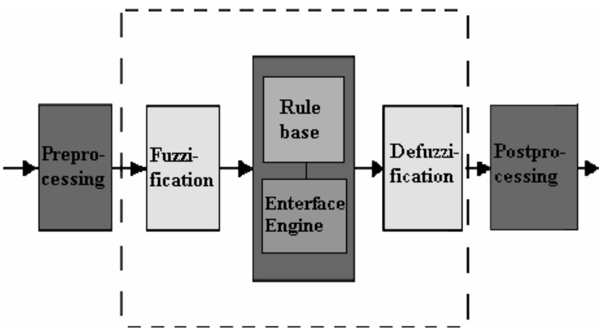


**Fig. 4:** *Fuzzy controller system*

The controller has two inputs and one output. The inputs are error and desired position. The output of the fuzzy controller is added to the feed forward acceleration which is multiplied with a gain to make the control effort. Following this, the membership functions are designed.

#### 3.2.1 The Input Membership Function

Generally, the PD-fuzzy controller is used in many industrial applications. Here, because of the sensitivity of the system's parameters to the sign of the voltage fed to the valve, if the error and change of error are used as the inputs of the controller, the performance of the controller in the extension and the retraction movements are not satisfactory. Two fuzzy inputs are used in the study: error and desired position. The reference position, $X_r$, is used for tracking and the position of the load (piston), is considered to make a system error. As depicted in Fig. 5, the inputs are error and desired input. When the input to the controller is the error, the control strategy is static mapping between the input and control signal. A dynamic controller has additional inputs, for example derivatives, integrals, and so on.

These inputs are created in the pre-processor, thus making the controller multi-dimensional, which requires many rules and makes it more difficult to design. Following this, the inputs are defined as error and desired input.

The idea of using different gains in extension or retraction by fuzzy controller is that the dynamics of the system depends on the position of piston, Eq. 7 and 10, and it varies during the piston movements. The control signal also contains an acceleration feed-forward term which affects the system's movement.

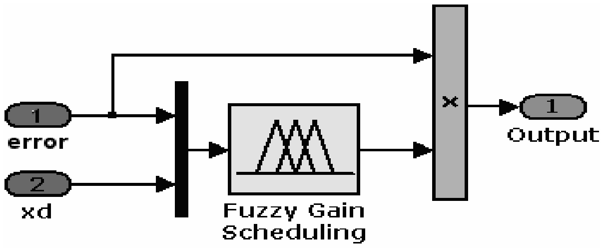


**Fig. 5:** *Inputs and output of the fuzzy controller*

The error quantity is defined as the difference between the desired position and the position of the load (piston). Figure 6 indicates the membership function of error.

Five membership functions are used to fuzzify of the error. Following this the required equations to calculate the membership function for vector $x$ in the case of a triangular or trapezoid membership function are provided.

$$f(x,a',b',c') = \max\left(\min\left(\frac{x-a'}{x-b'},\frac{c'-x}{c'-b'}\right),0\right) \quad (35)$$

$$f(x,a',b',c',d') = \max\left(\min\left(\frac{x-a'}{x-b'},1,\frac{d'-x}{d'-c'}\right),0\right) \quad (36)$$

In Eq. 35 the parameters $a'$ and $c'$ locate the feet of the triangle and the parameter $b'$ locates the peak and in Eq. 36 the parameters $a'$ and $d'$ locate the feet of the trapezoid and the parameters $b'$ and $c'$ locate the shoulders.

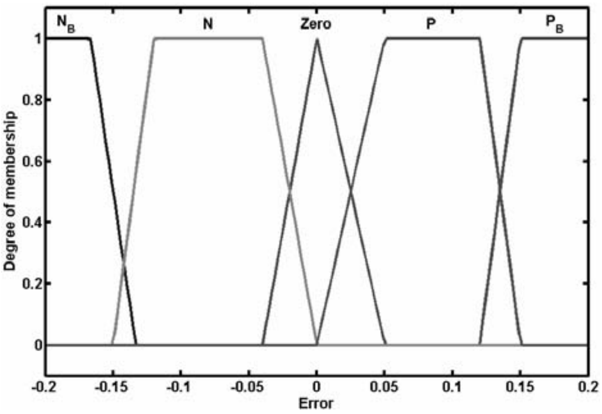


**Fig. 6:** *Membership functions of error*

The membership function of the second input (desired input) of the fuzzy controller is shown in Fig. 7. This input helps the controller to understand the direction of movement, so it can make the proper control effort. In the study, the special forms of trapezoid membership functions around the maximum of extension and the minimum of retraction (0.2 and 0.4 meter) are used. The singleton membership function has the same performance.

Here, the numbers of the second membership functions are equal to the number of desired extreme heights of the pulse input.

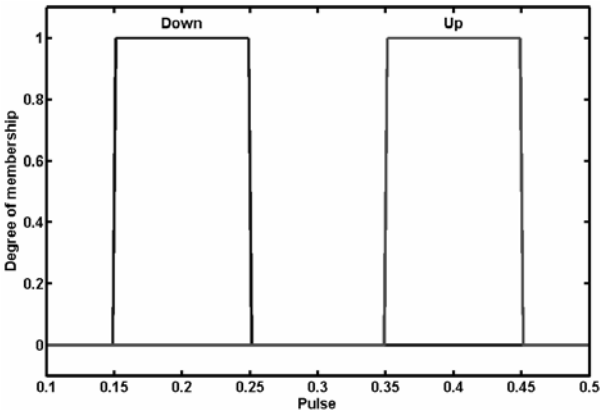


**Fig. 7:** *Membership functions of desired input*

### 3.2.2 The Output Membership Function

Figure 8 shows the output membership function of the fuzzy controller. Here there are four output membership functions. At the first the membership functions of the extension and retraction are chosen independently and then by trail and error in simulation mode the proposed functions are derived.

### 3.2.3 Fuzzy Rules

Rules provide a formal way of representing directives and strategies and are often appropriate when domain knowledge results from empirical associations or experience. Rule-based systems are built upon a set of rules and use a collection of facts to make inferences.

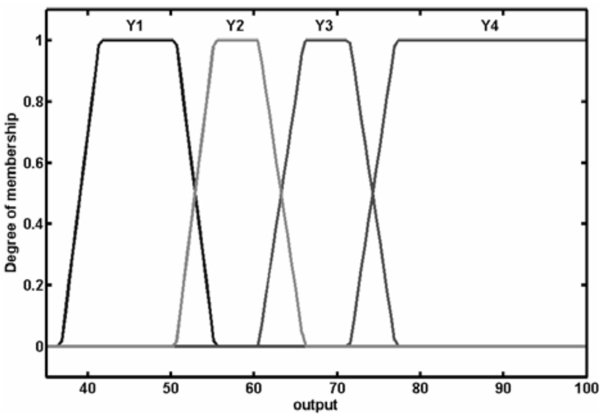


**Fig. 8:** *Membership functions of output*

According to the input membership functions, a total of $5 \times 2 = 10$ rules are needed to control the system. However, because some of the rules do not apply to the practical test, the number of rules is decreased to 6. The rules of the proposed fuzzy controller are as follows:

- 1-If (Error is N_B) and (Pulse id 0.2) then (output is Y4)
- 2-If (Error is N) and (Pulse id 0.2) then (output is Y3)
- 3-If (Error is Zero) and (Pulse id 0.2) then (output is Y2)
- 4-If (Error is P_B) and (Pulse id 0.4) then (output is Y2)
- 5-If (Error is P) and (Pulse id 0.4) then (output is Y1)
- 6-If (Error is Zero) and (Pulse id 0.4) then (output is Y2)

Note that because of the second input of the fuzzy controller is not a continuous parameter, the fuzzy surface is not clear to demonstrate.

The control effort is a summation of the fuzzy output and acceleration feed-forward, so the output of the fuzzy controller must be chosen in a way that oscillation is removed or minimized.

### 3.2.4 Defuzzification and Interface

Defuzzification is the process of producing a quantifiable result in fuzzy logic. There are several kinds of defuzzifications. In the study the Centre Of Area (COA), defuzzification method is used to make the outputs for the fuzzy controller as follow,

$$COA = \frac{\int \mu(Y) Y dY}{\int \mu(Y) dY} \tag{37}$$

where $Y$ is the output and $\mu(Y)$ is its membership function.

Fuzzy inference is the process of formulating the mapping from a given input to an output using fuzzy logic. The mapping then provides a basis from which decisions can be made. The process of fuzzy inference involves all of the subjects such as membership functions, fuzzy logic operators, and if-then rules. There are two types of fuzzy inference systems which are varying somewhat in the way that the outputs are determined. Here the Mamdani interface system is used.

The Following section deals with the design of a p-controller to verify the performance of the fuzzy controller.

### 3.2.5 Design of the P-Controller

Generally, to design a common p-controller the linearized equations of the system are used. The gain of the p-controller ($K_P$) is determined by using the Ziegler-Nichols method (Franklin et al., 1986) then the gain is modified and tuned if needed to work with the nonlinear model of system in simulation program. In this study two different controller gains for the extension and retraction movements are determined, so that the gain of the p-controller varies depending on the movement direction. Also in this case an acceleration feed-forward to compensate the lack of damping in the hydraulic system is used.

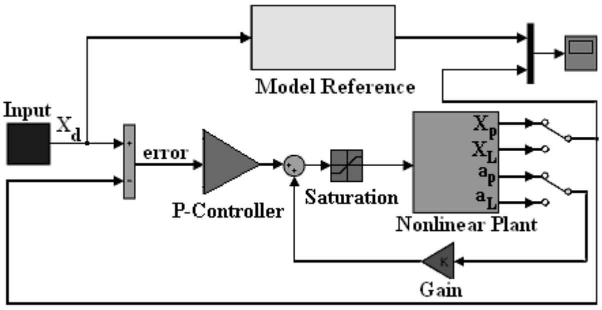


**Fig. 9:** *Schematic diagram of the P-Controller*

The schematic diagram of the p-controller is shown in Fig. 9. The acceleration feedback is also used to improve the dynamics of the system. Depending on the position control of the piston or flexible load, its related acceleration is used.

In the following section the results of the simulations for the proposed fuzzy controller and the improved version of the p-controller are described.

## 4 Simulation Results

In this section, the results of using the proposed controller and modified p-controller in tracking of the reference model for piston load ($m_1$) and flexible load ($m_2$) are compared.

The desired input ($X_d$) is a pulse input with an amplitude of 0.2 m and period of 4 sec.. The initial positions for the piston and flexible load are 0.2 m. To compare the results of simulation, first piston tracking is considered.

### 4.1 Piston Tracking

Figure 10 is the piston tracking of the reference model using fuzzy gain scheduling. Figure 11 is its control signal.

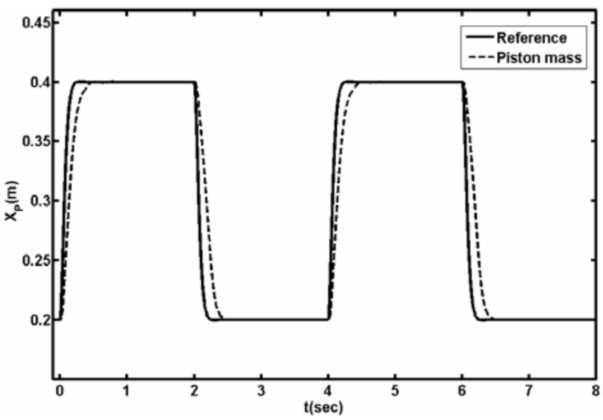


**Fig. 10:** *Simulation result of piston mass tracking using fuzzy gain scheduling*

The maximum and minimum voltages fed to the valve are limited to +-10 Volts (limit of servo valve). As shown in Fig. 11, the maximum and minimum voltages are banded by -10 to 10 volts. In the simulation model, a saturation element was used to avoid the controller output going beyond the limits.

Figure 10 shows that the fuzzy gain scheduling has very good performances in extension and retraction movement. The proposed controller tracks the reference model very well. Figure 11 shows the control signal of the controller is smooth without any oscillation.

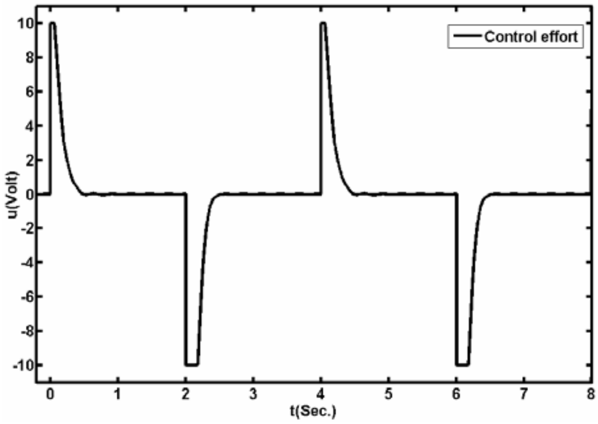


**Fig. 11:** *Control signal using the proposed controller*

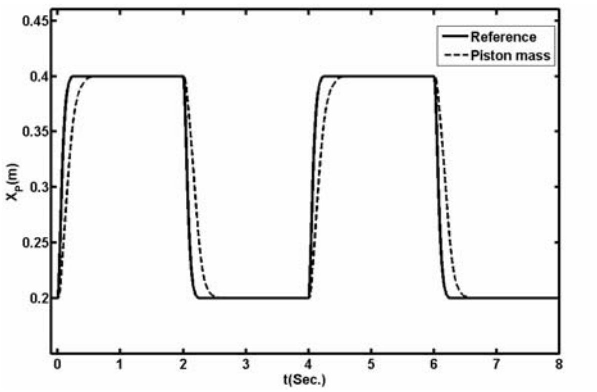


**Fig. 12:** *Simulation result of piston mass tracking using P-Controller*

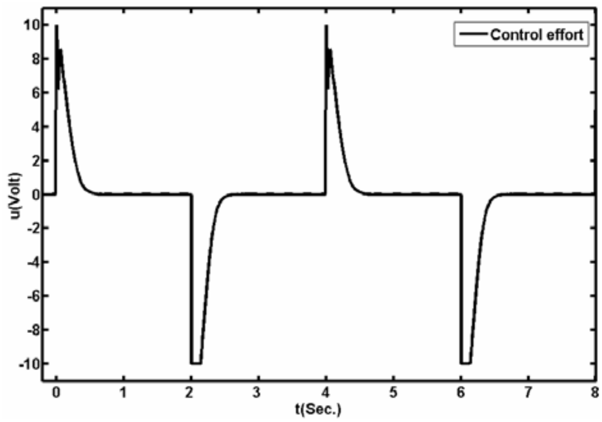


**Fig. 13:** *Control signal using the P-Controller*

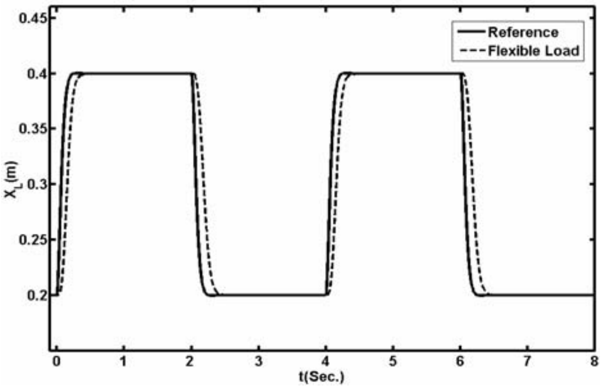


**Fig. 14:** *Simulation result of flexible load tracking using the proposed controller*

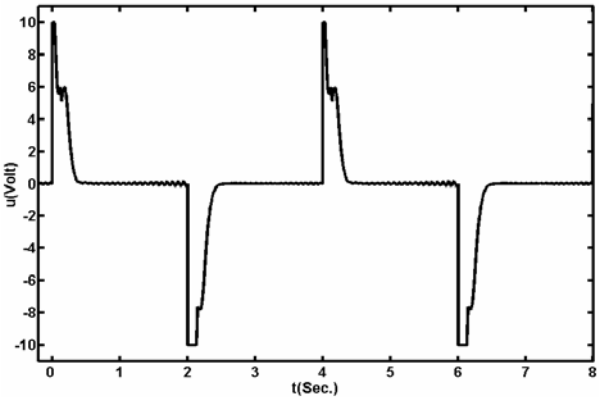


**Fig. 15:** *Control signal using the proposed controller*

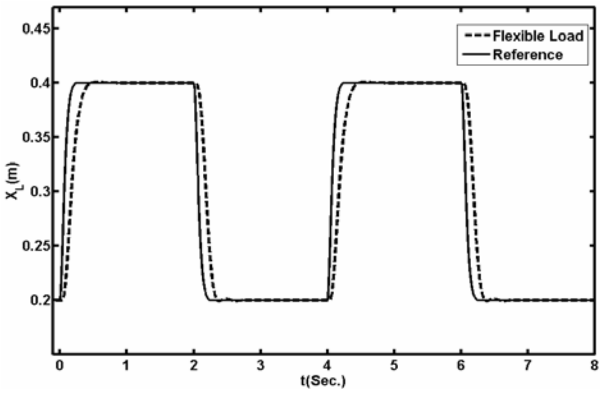


**Fig. 16:** *Simulation result of flexible load tracking using a P-Controller*

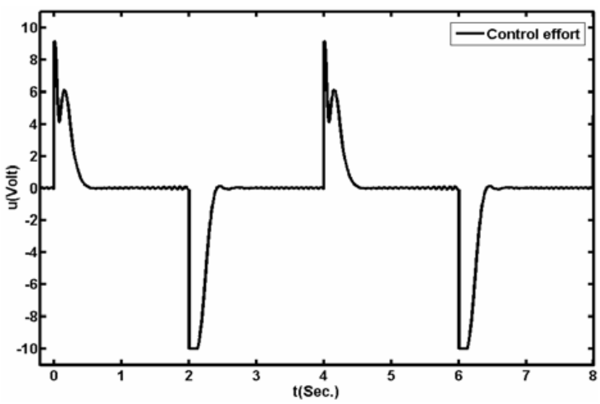


**Fig. 17:** *Control signal using the P-Controller*

Figures 12, 13 show that using the modified p-controller, the piston tracks the reference model in the extension and retraction movements well, but the proposed controller is faster than the modified p-controller.

### 4.2 Flexible Load Tracking

In this section, the simulation results are compared using the two controllers to track the reference model with the flexible load. Figure 14 is the tracking of the reference model by the flexible load using the proposed controller.

The result is satisfactory and illustrates the capability of the fuzzy controller in tracking of the reference model. Figure 15 shows the control signal of the proposed controller.

Figures 16 and 17 indicate the behaviour of the P-Controller in tracking the reference model.

As illustrated in Fig. 16, the p-controller has good tracking in the extension and retraction movements, but there is a very small vibration. The Summation of the Absolute Amount of Error (SAAE) during one period (4 sec.) for each controller is calculated. The SAAE amounts are 69.7 and 74.17 for the proposed controller and modified p-controller respectively, so the proposed controller has 6.4 percent decrease in the amount of SAAE.

## 5 Experimental Results

In the experimental test, an Ultra Hydraulics servo valve (Bosch 4661) with nominal flow rate of 40 liters per minute has been used. The dimensions of the hydraulic cylinder are 32/22/1000 millimetres.

The structural damping is neglected because it is not important in this model. Some states of the system, $X_p$ ($X_L$) and $a_P$ ($a_L$) are directly measured by the position and accelerometer sensors respectively.

The physical system application was driven in such a way that the proposed controller was implemented in Simulink and then the derived algorithm was transferred to C/C++ code , which is generated by RTW (Users Guide of Real Time Workshop, 2005) for dSPACE©'s digital signal processor (DSP) to use in real-time. The control effort voltage was fed into the servo valve using a DS1103 I/O card. The computational time step for the each controller was 1 ms.

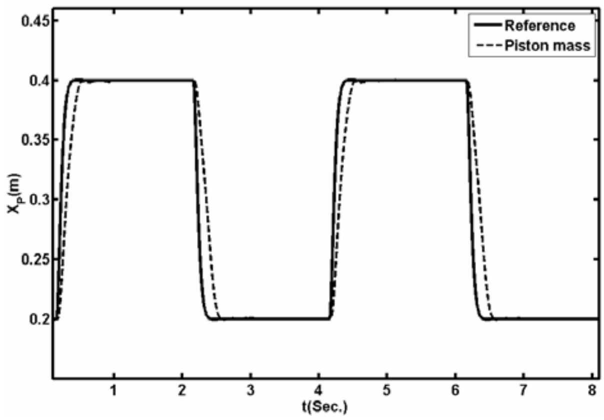


**Fig. 18:** *Experimental result of piston mass tracking using the proposed controller*

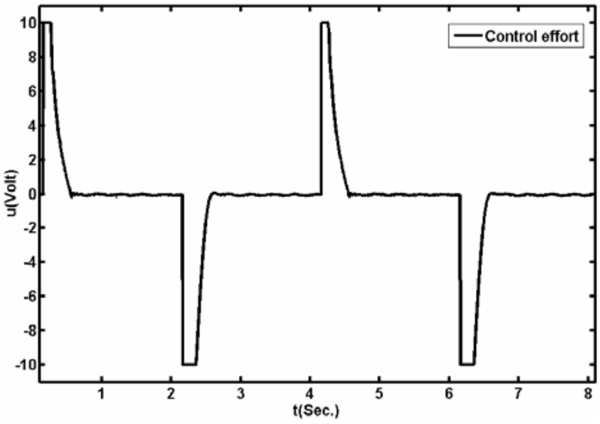


**Fig. 19:** *Control signal using the proposed controller*

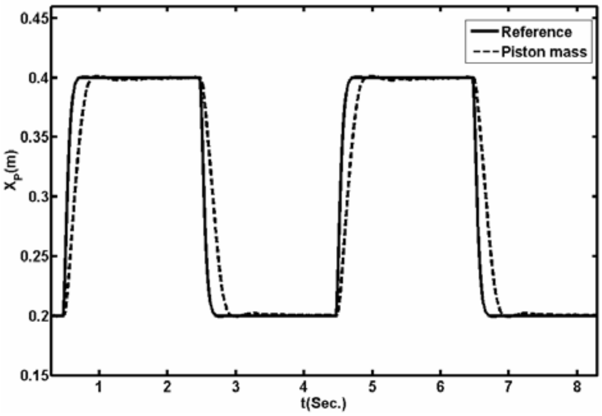


**Fig. 20:** *Experimental result of piston mass tracking using the P-Controller*

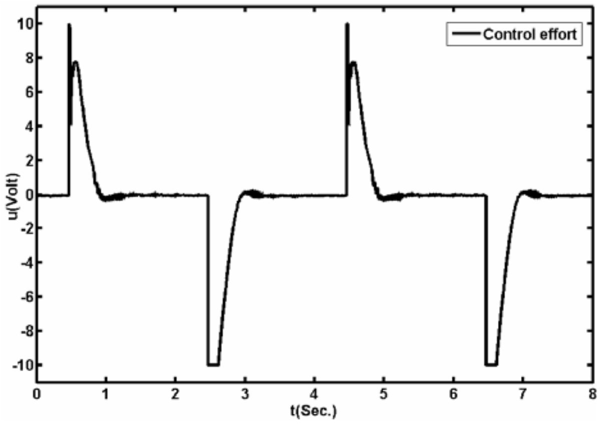


**Fig. 21:** *Control signal using the P-Controller*

Figures 18 - 25 are the results of the experimental test in tracking of the reference position. It is clear that the performance of the proposed controller is better than the modified p-controller.

Note that there is a small deviation between the simulation and the real test results. The reasons for this are summarized as follows; the first reason is the noise that is present in all of the electro-mechanical systems. The second reason is the supply pressure is constant with an amount of 14 MPa in the simulation, but during the real test that amount is varying and its minimum amount is 12.5 MPa. This pressure drop is normal in many industrial applications. In a practical test if the pressure drop is high, an accumulator is used.

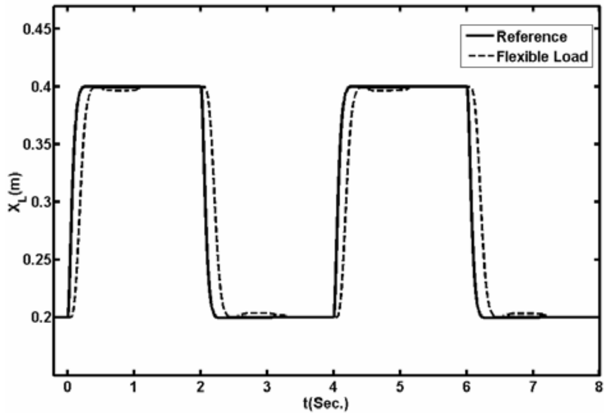


**Fig. 22:** *Experimental result of flexible load tracking using the proposed controller*

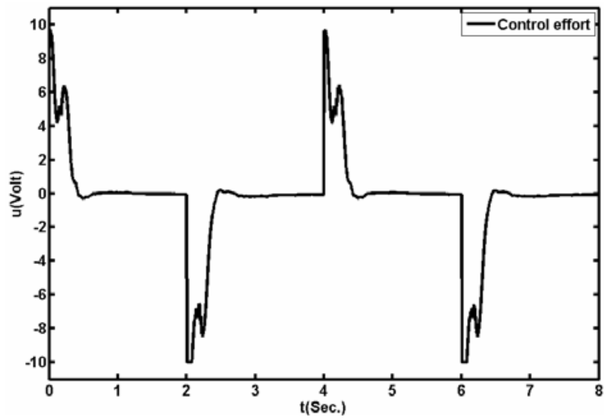


**Fig. 23:** *Control signal using the proposed controller*

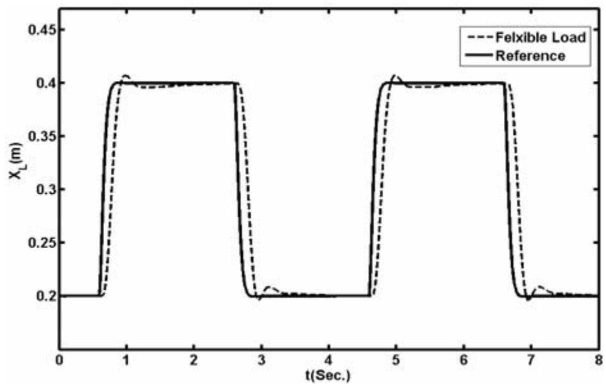


**Fig. 24:** *Experimental result of flexible tracking using the P-Controller*

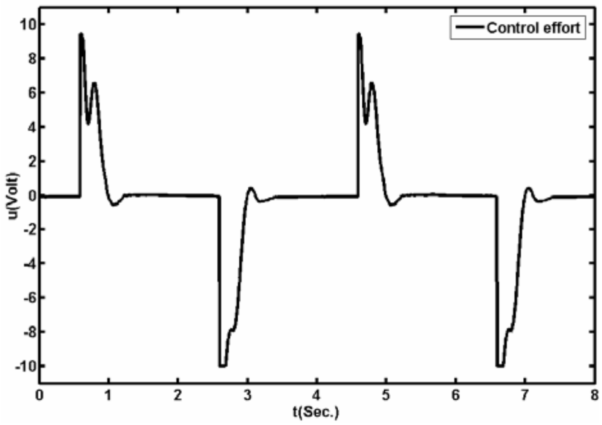


**Fig. 25:** *Control signal using the P-Controller*

## 6 Conclusions

In the present paper, the application of the fuzzy gain-scheduling controller in a servo hydraulic system was studied. The results suggest that the proposed controller with an acceleration compensator has better performance than the modified p-controller.

The proposed controller was designed in a way that the speed of tracking for a pulse input is as fast as possible, so the application of controller is limited to the pulse inputs.

The architecture of the fuzzy controller is essential to overcoming the change in the system´s dynamics during the extension and retraction tracking.

In light of the above, using feed-forward acceleration to compensate the dynamics of the hydraulic systems is an important factor to improve the performance of controllers. The results suggest that the proposed controller is faster and more accurate than the P-controller in the tracking the reference model. The mount of SAAE for the proposed controller is 6.4 percent less than for the modified p-controller.

The comparison between the simulation results and the experimental results suggest that fuzzy gain scheduling is one of the best controllers for servo hydraulic systems in position tracking of pulse inputs.

## Nomenclature

| | | |
|---|---|---|
| $A_1$ | piston area in chamber one | [m²] |
| $A_2$ | piston area in chamber two | [m²] |
| $a'$ | Membership parameter | [-] |
| b | viscous friction coefficient | [Ns/m] |
| $b'$ | Membership parameter | [-] |
| $c'$ | Membership parameter | [-] |
| $c_v$ | flow coefficient | $[\text{m}^3/\text{sVPa}^{1/2}]$ |
| $d'$ | Membership parameter | [-] |
| $F_f$ | cylinder friction | [N] |
| $F_s$ | static friction | [N] |
| k | spring constant | [N/m] |
| $k_p$ | proportional gain | [-] |
| $K_v$ | viscous friction | [Ns/m] |
| $L$ | stroke | [m] |
| $L_i$ | internal leakage coefficient | [m$^4$s/kg] |
| $L_{e1}$ | external leakage coefficient | [m$^4$s/kg] |
| $L_{e2}$ | external leakage coefficient | [m$^4$s/kg] |
| $m_1$ | mass of rigid body | [Kg] |
| $m_2$ | mass of flexible load | [Kg] |
| $p_1$ | pressure in chamber one | [Pa] |
| $p_2$ | pressure in chamber two | [Pa] |
| $p_s$ | pressure supply | [Pa] |
| $p_T$ | tank pressure | [Pa] |
| $Q_1$ | flow rate in chamber one | $[\text{m}^3/\text{s}]$ |
| $Q_2$ | flow rate in chamber two | $[\text{m}^3/\text{s}]$ |
| $u$ | control effort signal | [V] |
| $u_G$ | proportional controller signal | [V] |
| $u_N$ | feed-forward signal | [V] |
| $V_1$ | compressed volume | $[\text{m}^3]$ |
| $V_2$ | compressed volume | $[\text{m}^3]$ |
| $V_s$ | stribeck velocity | [m/s] |
| $X_L$ | load position | [m] |
| $X_p$ | piston position | [m] |
| $X_d$ | desired position | [m] |
| $X_r$ | reference position | [m] |
| $\dot{X}_p$ | piston velocity | [m/s] |
| $X_s$ | spool position | [m] |
| $y$ | state space output(s) | [-] |
| $Y$ | fuzzy output | [V] |
| $z$ | internal state | [-] |
| $\Delta p_n$ | nominal pressure difference | [Pa] |
| $\beta_e$ | effective bulk modulus | [Pa] |
| $\tau_1$ | valve gain | [-] |
| $\tau_2^{-1}$ | valve time constant | [s] |
| $\mu(Y)$ | output membership function | [-] |
| $\sigma_o$ | stiffness coefficient | [m/s] |
| $\sigma_1$ | damping coefficient | [Ns/m] |
| $v_0$ | dead volume | $[\text{m}^3]$ |
| $\omega_n$ | natural frequency | [rad/s] |

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