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Simulation and hybrid fuzzy-PID control for positioning of a hydraulic system

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Abstract Accuracy and precision of position control of hydraulic systems are key parameters for engineering applications in order to set more economical and quality systems. In this context, this paper presents modeling and position control of a hydraulic actuation system consisting of an asymmetric hydraulic cylinder driven by a four way, three position proportional valve. In this system model, the bulk modulus is considered as a variable. In addition, the Hybrid Fuzzy-PID Controller with Coupled Rules (HFPIDCR) is proposed for position control of the hydraulic system and its performance is tested by simulation studies. The novel aspect of this controller is to combine fuzzy logic and PID controllers in terms of a switching condition. Simulation results of the HFPIDCR based controller are compared with the results of classical PID, Fuzzy Logic Controller (FLC), and Hybrid Fuzzy-PID controller (HFPID). As a result, it is demonstrated that Hybrid Fuzzy PID Controller with Coupled Rules is more effective than other controllers.

Keywords Hydraulic systems · Position control · Hybrid fuzzy-PID · Coupled rules

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1 Introduction

Due to their high force to weight ratio, small size, flexible, and esthetic properties, ease of setting speed, force, and torque, high precision control, etc., hydraulic systems are convenient for many industrial areas such as manufacturing, automotive technology, aerospace industry, mobile vehicles, robotics, and mechatronics [1–3]. The accurate design and high performance of the hydraulic systems have importance as an engineering requirement, in order to create more economical and high quality systems. In this situation, computational modeling studies assist to obtain high-level production quality. Furthermore, modeling and simulation studies have the potential to guide technological development and to reduce costs.

The nonlinearities of hydraulic systems have considerable effects on the model accuracy and arise from compressibility of the hydraulic fluid, the complex flow properties of the servovalve, and frictional forces [2]. In recent times, significant progress has been made on the modeling of hydraulic systems in the literature. For example, Pfeiffer [4] developed a new modeling scheme for hydraulic systems and illustrated the performance on a large industrial example. Eyres et al. [5] studied on several possible methods of modeling and dynamic response of a passive hydraulic damper with a bypass tube. A more complex model incorporating the dynamics of the internal spring and fluid compressibility is derived. Sağırlıet al. [6] obtained a theoretical model of a spatially actuated telescopic rotary crane by the Bond Graph method. Compressibility of hydraulic fluid in cylinders is included in the model. Scheidl and Manhartsgruber [7] investigated the nonlinear dynamic behavior of servo-hydraulic drives for position control, which is a partially singularly perturbed system. Bonchis et al. [8] studied on the effect of friction nonlinearities on position control of hydraulic servo systems. Kalyoncu and Haydim [9] obtained a mathematical model considering the internal leakage in servovalve and actuator.

In hydraulic systems, cylinders are crucial component converting the fluid power into linear motion and force. Recently, hydraulic cylinders are used widespread with the proportional and servo-valves, which is electronically controlled. One of the important required functions is accurate and precise position control. In order to achieve this, it is necessary to apply convenient control strategies. In industrial applications, conventional control methods such as PD, PID are commonly used. However, conventional control methods have linear characteristics. Therefore, they are insufficient to overcome nonlinearities which exist in the nature of hydraulic systems [2, 3].

Several control methods have been proposed to cope with these nonlinearities in hydraulic systems [8, 10–21]. For instance, Liu and Daley [10] designed an optimally tuned nonlinear PID controller for hydraulic systems. An estimated process model was used for tuning optimal PID parameters in the case of variable process parameters. Bonchis et al. [11] compared ten position control scheme for tracking control of a hydraulic servo system with single ended cylinder driven by a proportional directional valve. Best results were reached by FRID (experimental friction model), MRAC (model reference adaptive controller), and VSC (variable structure with sliding mode). Li and Khajepour [12] designed a robust controller to regulate both flexural vibrations and rigid body motion of a hydraulically driven flexible arm. Kim et al. [13] investigated a robust velocity control problem for a hydraulic elevator including cylinder friction, pump friction, and pump leakage. Nakkarat and Kuntanapreeda [14] designed a nonlinear controller based on backstepping approach for controlling force of a single rod hydraulic actuator. A PI observer was used to estimate the states of the system. Sohl and Bobrow [2] proposed a nonlinear controller for tracking trajectory of a hydraulic servo system force. Lyapunov function used to guarantee the stability of the system. Chen et al. [15] conducted a sliding mode controller for tracking control of a hydraulic servo system. Deticek [16] proposed a control strategy based on fuzzy logic and conventional control approach. Truong and Ahn [17] designed a novel parallel controller based on self-tuning quantitative feedback theory technique and applied to electro-hydrostatic load simulator for position and force control. Knohl and Unbehauen [18] studied on adaptive position control using artificial neural networks for a hydraulic system consisting of 4/3 way proportional valve, a differential cylinder and a variable load force. Kalyoncu and Haydim [8] applied fuzzy logic control to a hydraulic servo system for position control. They investigate the effect of internal leakage on the control performance. Chen et al. [19] proposed a fuzzy controller to achieve a synchronous positioning objective for a dual-cylinder electro hydraulic lifting system with unbalanced loading, system uncertainties and disturbances. Controller system consists of one fuzzy coordination controller and one fuzzy tracking controller. Lee and Kopp [20] developed an adaptive fuzzy controller for a hydraulic forging machine in which the process is nonlinear, nonstationary, and time invariant. Lee and Cho [21] studied a new fuzzy controller using phase plane for an electro-hydraulic fin servo-system of a missile. A phase plane used to describe the system trajectories.

When focused on the literature studies mentioned above, it could be deduced that intelligent and robust nonlinear controllers are necessity for hydraulic actuators in order to obtain accurate and precise control performance. Modeling issue is also important for position control of hydraulic systems. Therefore, bulk modulus is considered variable and a new controller named as Hybrid Fuzzy PID Controller with Coupled Rules (HFPIDCR) is applied for position control of a hydraulically actuated system.

In this paper, simulation studies of a hydraulic actuator driven by a proportional valve are carried out to illustrate the effectiveness of Hybrid Fuzzy PID Controller with Coupled Rules (HFPIDCR) on position control. The most important characteristic of this controller is that it has two parts: Fuzzy control and PID control. The fuzzy controller and the PID controller are switched when the piston is near desired position. The rest of the paper is organized as follows. Mathematical model of hydraulically actuated system, which is one of the objects of this study, is presented in Sect. 2. Controllers, which are applied to the system, represented in Sect. 3. In Sect. 4, numerical re-

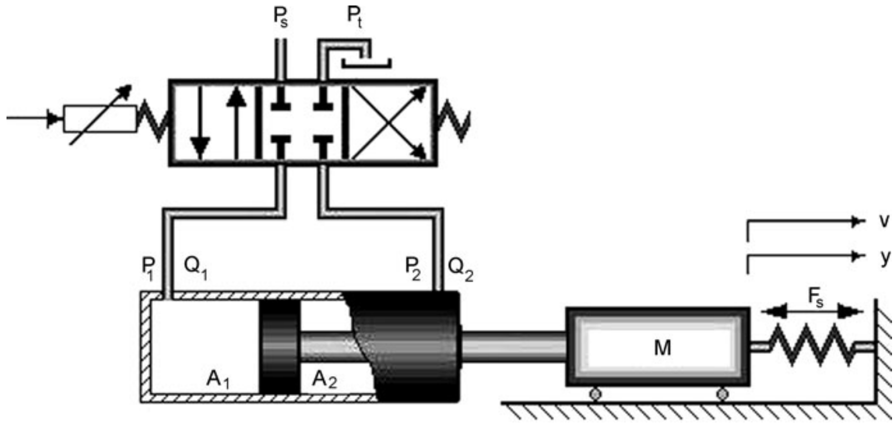


Fig. 1 Asymmetric hydraulic cylinder driven by a four way three position proportional valve

sults are considered. Finally, in Sect. 5, conclusions are presented in the light of simulation results.

2 Mathematical model of the hydraulic system

The physical model of the hydraulic actuation system consisting of an asymmetric hydraulic cylinder driven by a four way, three position proportional valve is shown in Fig. 1. The asymmetric hydraulic cylinder is operated through pressurized fluid, which is controlled by the valve. The piston separates the asymmetric cylinder in two divisions. When the hydraulic fluid is pumped into the divisions of the asymmetric cylinder the hydraulic pressure acts on the piston moving it back and forth. This creates the hydraulic cylinder force which moves the attached objects. The hydraulic pressure of the oil moves the piston rod; the piston rod moves the mass and spring at the end of the piston rod.

In the Fig. 1, P_s is the supply pressure (N/m^2) P_t is reservoir pressure (N/m^2), P_1 and P_2 are pressures in side 1 and side 2 of the cylinder, respectively (N/m^2), Q_1 ve Q_2 are fluid flows (m^3/s), A_1 is the piston area of side 1 (m^2), A_2 is the piston area of side 2 (m^2), M is the actuated mass (kg), F_s is the spring force (N).

While the mathematical model is obtained, hydraulic pipe and valve dynamics are neglected and it is considered that there is no leakage between the piston and the cylinder. Besides, it is assumed that the supply pressure is constant and reservoir pressure is zero. Valve spool displacement over maximum valve spool displacement is defined as $\varepsilon = u/x_{\max}$ and underlapped valve gap over maximum valve spool displacement is defined as $\psi = x/x_{\max}$. In this study, ψ is 1% of maximum valve displacement.

Effective bulk modulus β_v is variable. Fluid is an important element of hydraulic systems and enables power transmission, and hence it can influence the dynamic behaviors of the overall and the control system. The bulk modulus of nonaerated hydraulic oil depends on temperature and pressure. For mineral oils with additives, its value ranges from 1200 to 2000 MPa. Moreover, system pressure and entrapped air affect the bulk modulus value. If a hydraulic hose is used rather than a steel pipe, the bulk modulus of this section may be considerably reduced. Owing to these reasons, the parameters influencing bulk modulus value must be included in the system model for more accurate system dynamics [22].

The equation which gives the variable bulk modulus of fluid-air mixture in a flexible container is as follows [23, 25]:

$$\frac{1}{\beta_v} = \frac{1}{\beta_f} + \frac{1}{\beta_h} + \frac{V_a}{V_t}\frac{1}{\beta_a} \tag{1}$$

where the subcripts a, f, and h refer to air, fluid, and hose, respectively. It is assumed that the initial total volume $V_t = V_f + V_a$, and that $\beta_f \gg \beta_a$. Thus bulk modulus will be less than any β_f, β_h, or $(V_t/Va)\beta_a$. The bulk modulus of the fluid β_f is obtained from the manufacturer's data. The adiabatic bulk modulus used for air is $(C_p/C_v)P = 1.4P$. With these assumptions, (1) can be rewritten as in

$$\frac{1}{\beta_v} = \frac{1}{\beta_f} + \frac{1}{\beta_h} + \frac{s}{1.4P} \tag{2}$$

where, s is entrapped air percent in the total volume ($s = V_a/V_t$).

Flows occurred from backward and forward cylinder motions and compressibility of the fluid are considered in the system equations. Due to the fact that the system equations are nonlinear, an applied control algorithm is examined on this structure.

Using Newton's second law, the piston force balance results in a motion differential equation as in (3). Applying the continuity equation to both sides of the cylinder, the following (4) and (5) can be derived [23–25].

$$\ddot{y} = (P_1 A_1 - P_2 A_2 - f_v v - F_s)/M \tag{3}$$

$$\dot{P}_1 = \frac{\beta_1}{A_1.Y}(Q_1 - A_1 v) \tag{4}$$

$$\dot{P}_2 = \frac{\beta_2}{A_2(Y - S_L)}(A_2 v - Q_2) \tag{5}$$

Flow equations derived from underlapped valve characteristics are given in (6–11) in Bernoulli form in terms of valve cross-section [24]. These flow equations are derived for three different valve positions and flow directions. $k_{1,2,3,4}$ are the valve coefficients obtained from valve leakage characteristics [24].

Flow equations for $\varepsilon \geq \psi$ are (P_s is applied on side 1 of the cylinder, P_t is at side 2):

$$Q_1 = k_1(\varepsilon + \psi)\mathrm{sign}(P_s - P_1) \times \sqrt{(P_s - P_1)\mathrm{sign}(P_s - P_1)} \tag{6}$$

$$Q_2 = k_2(\varepsilon + \psi)\mathrm{sign}(P_2 - P_t) \times \sqrt{(P_2 - P_t)\mathrm{sign}(P_2 - P_t)} \tag{7}$$

Flow equations for $-\psi < \varepsilon < \psi$ are (both sides of the cylinder are closed, there exists only leakage flow):

$$\begin{aligned} Q_1 = {} & k_1(\varepsilon + \psi)\mathrm{sign}(P_s - P_1) \\ & \times \sqrt{(P_s - P_1)\mathrm{sign}(P_s - P_1)} \\ & + k_4(\varepsilon - \psi)\mathrm{sign}(P_1 - P_t) \\ & \times \sqrt{(P_1 - P_t)\mathrm{sign}(P_1 - P_t)} \end{aligned} \tag{8}$$

$$\begin{aligned} Q_2 = {} & k_2(\varepsilon + \psi)\mathrm{sign}(P_2 - P_t) \\ & \times \sqrt{(P_2 - P_t)\mathrm{sign}(P_2 - P_t)} \\ & + k_3(\varepsilon - \psi)\mathrm{sign}(P_s - P_2) \\ & \times \sqrt{(P_s - P_2)\mathrm{sign}(P_s - P_2)} \end{aligned} \tag{9}$$

Flow equations for $\varepsilon \leq -\psi$ are (P_t is at side 1, P_s is applied on side 2 of the cylinder):

$$Q_1 = k_4(\varepsilon - \psi)\mathrm{sign}(P_1 - P_t) \times \sqrt{(P_1 - P_t)\mathrm{sign}(P_1 - P_t)} \tag{10}$$

$$Q_2 = k_3(\varepsilon - \psi)\mathrm{sign}(P_s - P_2) \times \sqrt{(P_s - P_2)\mathrm{sign}(P_s - P_2)} \tag{11}$$

3 Different control schemes

In the industrial applications, conventional control methods such as PD and PID are used. However, conventional control methods have linear characteristics. Fuzzy control provides a formal methodology for representing, manipulating, and implementing a human's heuristic knowledge about how to control a system [26]. Several types of the Fuzzy Logic controllers have been studied in the literature. We use three types of controller based on FLC for controlling of the hydraulic cylinder. The first one is the well-known fuzzy logic controller (FLC) which uses the error and the change rate of the error for determining the control action. The second one is the hybrid fuzzy-PID controller (HFPID) which uses fuzzy logic controller or PID controller according to distance to target position. Finally, HFPIDCR uses fuzzy logic controller and PID with coupled rules (HFPIDCR) which combines both PI and PD actions.

3.1 PID control

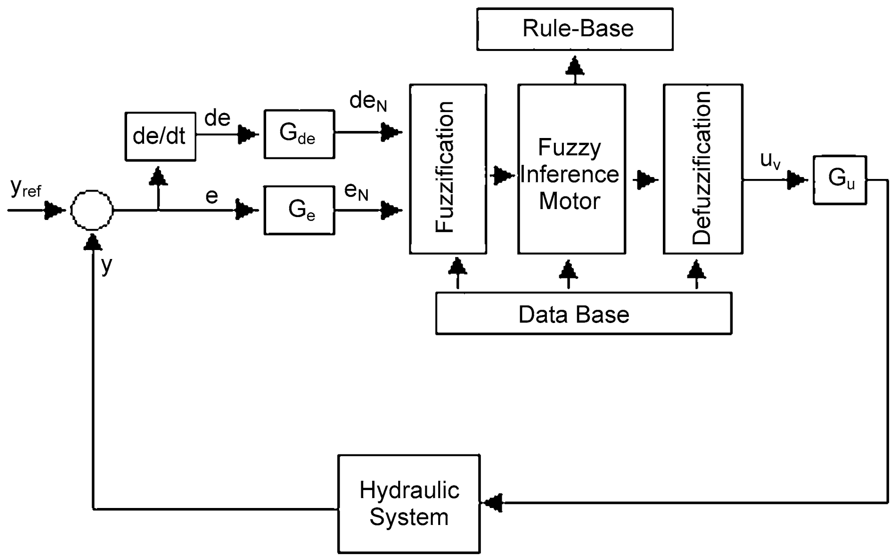
PID control is preferred in industrial applications because it is effective on system behavior for decreasing overshoot, raising damping, and decreasing rise and settling time and it is a simple characteristic controller. In the hydraulic system shown in Fig. 1, hydraulic cylinder piston position that is required to settle on reference value is measured continuously and compared with reference value. This carries on until the reference value and measured value are the same. PID control structure is given in (12) and (13).

$$e(t) = y_{\mathrm{ref}} - y \tag{12}$$

$$U(t) = K_P e(t) + K_I \int_0^t e(\tau)\,d\tau + K_D \dot{e}(t) \tag{13}$$

where e is the error between reference position (y_{ref}) and position (y). K_P is proportional gain, K_I is integral gain and K_D is derivative gain.

Fig. 2 Fuzzy logic control structure



3.2 Conventional fuzzy control

FLC has two inputs and one output variables. These variables are error (e), change rate of the error (de) and valve input signal (u_v), respectively. Rules consisted of these variables are written in a rule base. Fuzzy controller scheme is shown in Fig. 2.

All of the variables are implied as linguistic values and defined with the 7 linguistic values. These values are: NB-negative big, NM-negative medium, NS-negative small, Z-zero, PS-positive small, PM-positive medium, PB-positive big. Each linguistic value is represented as triangular membership function. Thus, e, de, and u_v variables are converted to fuzzy logic linguistic variables. Variables are defined in interval of $[-1, 1]$. Real interval of variables is obtained by using scaling factors which are G_e, G_{de} ve G_u. Rule base was determined by using experience and engineering mentality. For instance, one of the possible rule is: IF $e =$ PS and $de =$ NB THAN $u_v =$ NM. This rule can be explained as following: If error is small, piston position is around the reference position. Tremendously, negative big value of derivative of error shows that the piston is approaching the reference position rapidly. Consequently, controller output should be negative small to prevent overshoot and to make a brake effect. These rules are written in a rule base look-up table. The rule base look-up table used in this study is presented in Table 1.

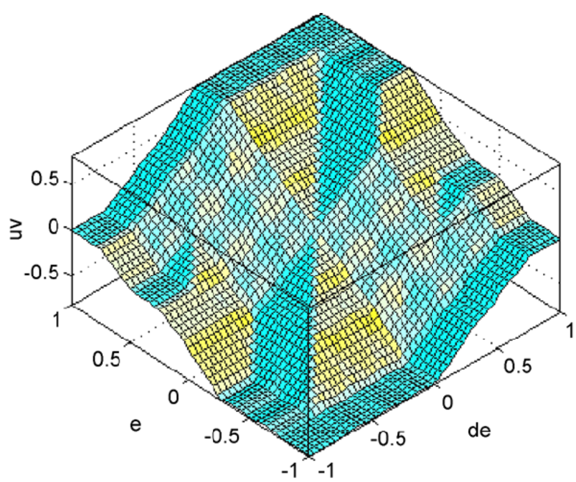


Fig. 3 u_v variation with e and de

Nonlinear characteristic of rule base consisted of these rules can be seen apparently in Fig. 3. As a rule inference method, Mamdani method is selected [27]. Moreover, centroid method was used for defuzzification [28].

3.3 Hybrid fuzzy-PID control

As mentioned above, because of their effectiveness on system behavior and simple characteristic, PID control is preferred in industrial applications. However, they are not robust to parameter changes. Fuzzy

Table 1 Rule base for fuzzy logic control

de/*e*	NB	NM	NS	Z	PS	PM	PB
NB	NB	NB	NB	NM	NM	NS	Z
NM	NB	NB	NM	NS	NS	Z	PS
NS	NB	NM	NS	NS	Z	PS	PM
Z	NM	NS	NS	Z	PS	PS	PM
PS	NM	NS	Z	PS	PS	PM	PB
PM	NS	Z	PS	PS	PM	PB	PB
PB	Z	PS	PM	PM	PB	PB	PB

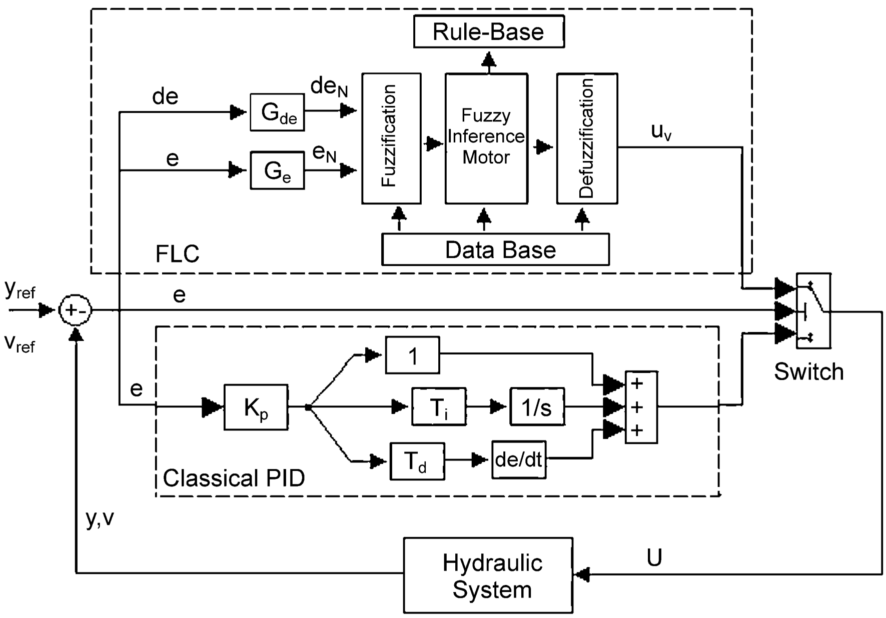


Fig. 4 Hybrid Fuzzy—PID control structure

logic controller is more adaptable against the parameter changes. It can be used in complex processes. However, there are some difficulties in the design of fuzzy logic controllers. The hybrid fuzzy PID controller (HFPID) takes advantage of the nonlinear characteristics of the fuzzy and the accuracy near a set point is guaranteed by the classical PID [29]. The fuzzy controller is used when the piston is far from desired position. Otherwise, PID controller is used when the piston is close to the desired position. The overall structure of used controller is shown in Fig. 4.

Fuzzy Logic controller (FLC) used in this paper is based on two input FLC structure. The input variables are defined as; $e = y_{\text{ref}} - y$ and $de = v_{\text{ref}} - v$. Real interval of variables is obtained by using scaling factors which are G_e, and G_{de}. The fuzzy control rule is in the form of: IF $e = E_i$ and $de = dE_j$ THAN $U_V = U_V(i, j)$. These rules are written in a rule base look-up table, which is shown in Table 1. The rule base structure is Mamdani type.

3.4 Hybrid fuzzy-PID control with coupled rules

The hybrid fuzzy-PID controller with coupled rules (HFPIDCR) consists of FLC, PID controller and a tuning scheme, which is called coupled rules. The overall structure of used controller is shown in Fig. 5. Controller signal that is determined by HFPID controller as explained in Sect. 3.3, is coupled with PI and PD actions by a tuning scheme.

Tuning scheme used in this paper is based on coupling of PI and PD actions [26, 30]. Therefore, the output of hybrid controller is formed by combining both

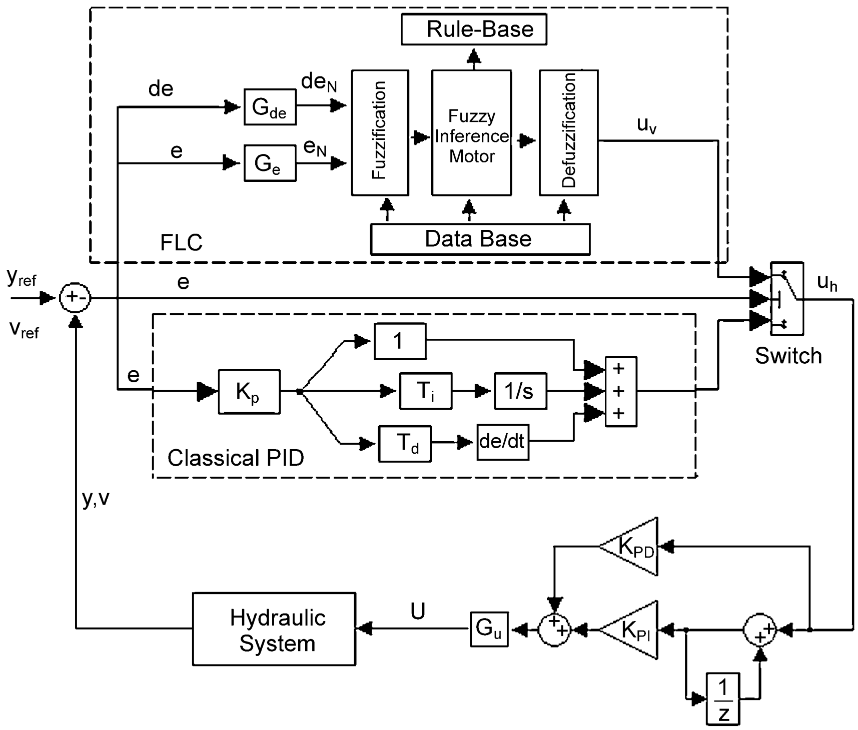


Fig. 5 Hybrid Fuzzy—PID control with coupled rules structure

PI and PD actions. The final controller signal can be given as

$$U(t) = G_U \left[K_{\mathrm{PI}} \sum_{i=0}^{t} U_h(i) + K_{\mathrm{PD}} U_h(t) \right] \tag{14}$$

4 Simulation results and discussion

Computer model is developed using system equations of the hydraulic system given in Sect. 2 in order to verify the effectiveness of the used controllers. MATLAB-Simulink environment is used for this modeling aim [31]. MATLAB ode45 solution function based on Runge–Kutta and Dormand Prince Couple fulfills the integration method in this Simulink model. MATLAB-Fuzzy Logic Toolbox is used for Fuzzy Logic based controllers. The values in the Table 2 were used for all of the simulation studies of the hydraulic system.

To show the performance of the Hybrid Fuzzy PID with Coupled Rules (HFPIDCR), comparisons are made with the other control schemes. Simulation is carried out with a single square function of 0.15 m. Two types of motion of the asymmetric cylinder are examined: forward and backward motion. The motion of the cylinder changes from forward to backward at 1.5 s. In the hydraulic system shown in Fig. 1, the position of hydraulic cylinder piston, which is required to settle on reference value, is measured continuously and compared with reference value. This carries on until the reference value and measured value are the same. Hydraulic piston position responses are given in Fig. 6 for interval of [0−3] s.

From this figure, it is easy to see that, both HFPIDCR and HFPID achieved significantly good performance on the position control of hydraulic cylinder. However, HFPIDCR controller has also smaller rise time, settling time and better IAE and ITAE values. Comparisons of velocity, pressure, fluid flow and control signal of cylinder are depicted in Figs. 7, 8, 9, and 10, respectively. Because of the cylinder we used here is asymmetric, those characteristics of the system are different in forward and backward motion of the cylinder. For example, velocity of the cylinder is set around 1 m/s but at backward motion, cylinder moves a bit slowly comparing to opposite motion. However, the spring at the end of the cylinder, which has an increasing force at forward motion and decreasing force

Table 2 Simulation input parameters

Parameter	Value	Unit
Mass (M)	50	(kg)
Pressure supply (P_s)	75×10^5	(N/m^2)
Bulk modulus of hose (β_h)	1.49×10^9	(N/m^2)
Bulk modulus of fluid (β_f)	47×10^7	(N/m^2)
Degree of entrapped air (s)	1	(%)
Coefficient of fluid viscosity (f_v)	2000	(Ns/m)
Nominal mass flow rate of valve (Q)	1.33×10^{-3}	(m^3/s)
Cylinder sizes	40/20	(mm)
Cylinder stroke (S_L)	1000	(mm)
Piston area of side 1 (A_1)	13×10^{-4}	(m^2)
Piston area of side 2 (A_2)	9.43×10^{-4}	(m^2)
Valve coefficient (k_1)	0.55×10^{-6}	$(\mathrm{m^3/s}\sqrt{\mathrm{N/m^2}})$
Valve coefficient (k_2)	0.55×10^{-6}	$(\mathrm{m^3/s}\sqrt{\mathrm{N/m^2}})$
Valve coefficient (k_3)	0.57×10^{-6}	$(\mathrm{m^3/s}\sqrt{\mathrm{N/m^2}})$
Valve coefficient (k_4)	0.45×10^{-6}	$(\mathrm{m^3/s}\sqrt{\mathrm{N/m^2}})$
Stiffness coefficient of spring	10	kN/m

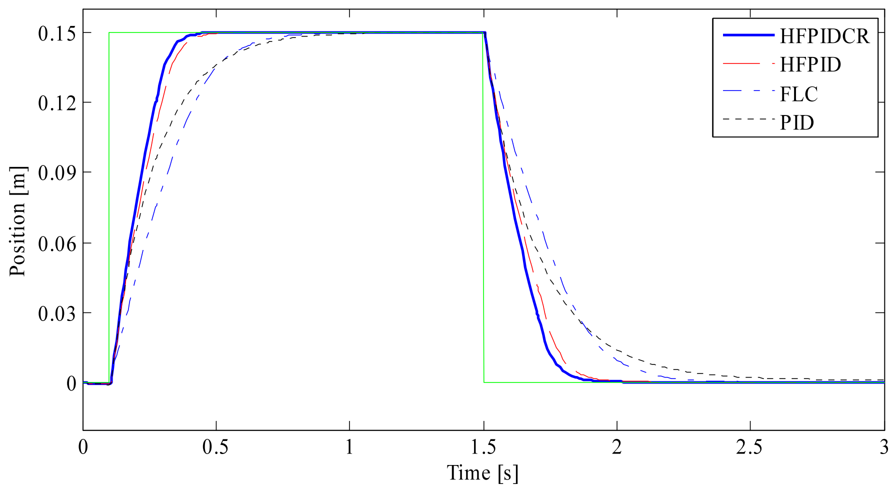


Fig. 6 Comparison of position responses

at backward motion, closes up these responses for a single square input. Besides, effective bulk modulus of the hydraulic system changes as a function of pressure. This interacted variation can also be tracked easily from Figs. 8 and 10. It is seen clearly from these mentioned figures that the applied control algorithms do not have effect adversely on overall dynamics of hydraulic system and the all important system characteristics remain in system limits.

Consider four criteria, rise time, settling time, integral absolute error (IAE) and integral of time multiplied absolute error (ITAE) that have different aspects on performance of controllers. In Table 3, comparisons of used controllers in terms of rise time, settling time, IAE and ITAE can be seen. As mentioned above, performances of both HFPIDCR and HFPID are better than both FLC and PID controller. In comparison with HFPID, HFPIDCR controller reduces rise time 13% and 15% at forward and backward motion respectively. HFPIDCR controller has also 9% and 15% better settling time than HFPID. From the table, it can be see that FFPIDCR control scheme

Fig. 7 Velocity response and control signal

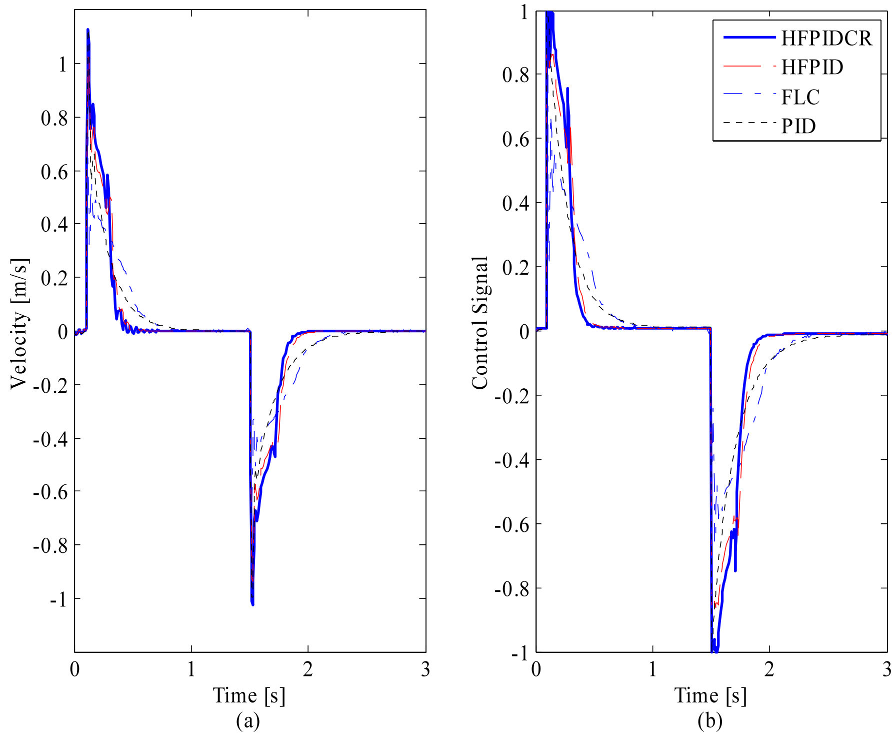


Fig. 8 Pressure response of hydraulic cylinder

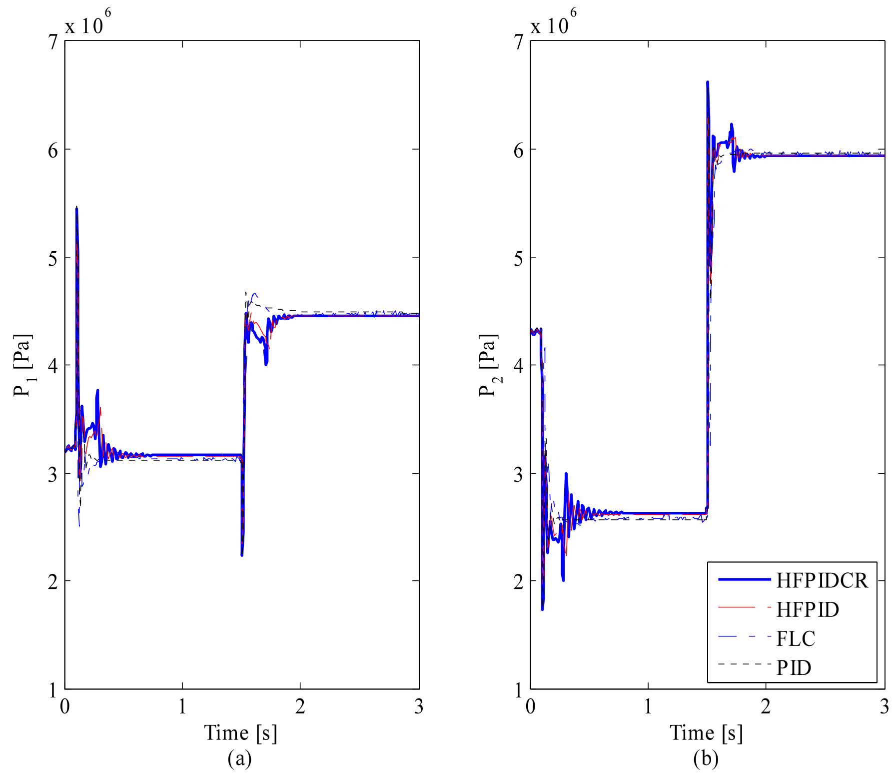


Fig. 9 Flow response of hydraulic cylinder

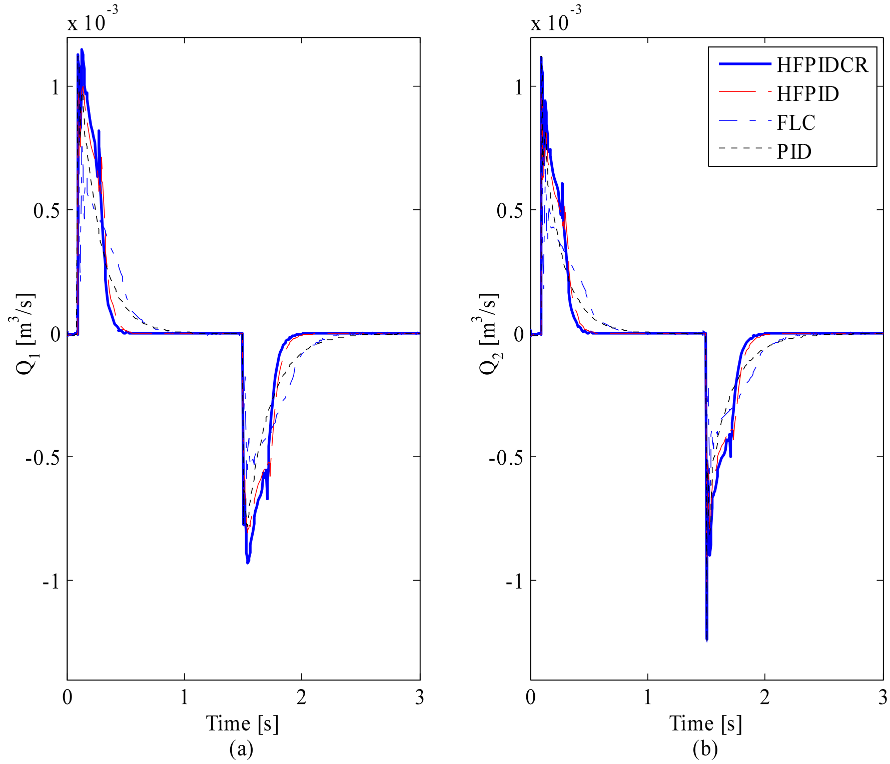


Fig. 10 Variation of Bulk modulus

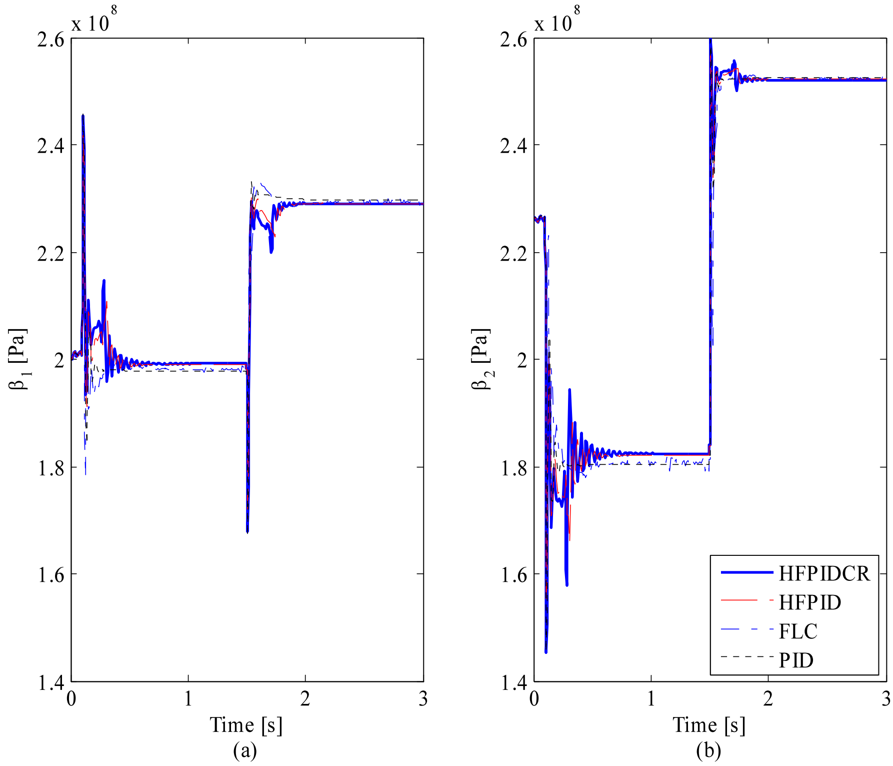


Table 3 Comparisons of controllers

Performance	Forward				Backward			
	FLC	HFPIDCR	HFPID	PID	FLC	HFPIDCR	HFPID	PID
Rise Time (s)	0.4687	0.2216	0.2506	0.4953	0.4990	0.2616	0.2719	0.631
Settling Time (s)	0.5876	0.3375	0.3682	0.6112	0.5160	0.2786	0.3190	0.6480
IAE	0.0307	0.0164	0.0184	0.0259	0.0332	0.0192	0.0218	0.0321
ITAE	0.0081	0.0029	0.0035	0.0069	0.0558	0.0309	0.0354	0.0556

provides smaller IAE and ITAE values than HFPID does.

FLC provides smaller rise and settling time than PID. FLC have 6% and 26% smaller rise time than PID at forward and backward cases, respectively. FLC also reduces the settling time 4% and 26% in each direction according to PID controller. Although PID has a good performance on IAE and ITAE values at forward motion, these performance indexes are nearly equivalent at backward motion.

Comparing forward and backward situations, with the exception of PID all of the controllers provide nearly the same performance. For example, while HFPIDCR, HFPID, and FLC have smaller settling time at backward motion than at forward motion, PID controller has larger one. Because, HFPIDCR, HFPID, and FLC which uses nonlinear rule base, run more efficiently on variation of parameters while PID controller uses the tuned gains on condition of $y_{\mathrm{ref}} = 0.15$. In addition, PID controller has a steady state error of 0.8% at backward case. This arises from the effect of the spring force, which is attached at the end of the piston.

5 Conclusions

The mathematical model of a hydraulically actuated system consisted of an asymmetric hydraulic cylinder driven by a four way, three position proportional valve is obtained. Interesting point of this model is that the bulk modulus parameter is considered as variable. HFPIDCR is proposed for positioning of the hydraulically actuated system. Subsequently, HFPID, FLC, and PID controller are applied to this model and position control simulations are achieved. According to simulation results, the overall performance of HFPIDCR is better. Thus, its rise and settling time are shorter than the other controllers. HFPIDCR controller has also smaller IAE and ITAE values. The most important reason of this FLC makes short rise time and PID control can control the system with high accuracy around set point. Consequently, it is observed that Hybrid Fuzzy-PID with Coupled Rules (HFPIDCR) controller can be applied successfully on position control of a hydraulic system.

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