Performance Analysis of Different Controller for a 2 DOF Electro-Hydraulic Motion Simulator

Bibhuti Bhusan Ghosh
Aerosystems Laboratory
Senior Scientist, CMERI
Durgapur, India
bibhutighosh@gmail.com

Bikash Kumar Sarkar
Assistant Professor, Mechanical Engg. Dept.
NIT Meghalaya
Shillong, India
bikash_ju2001@yahoo.co.in

Rana Saha
Assistant Professor, Mechanical Engg. Dept.
Jadavpur University
Kolkata, India
rsaha@mech.jdvu.ac.in

***Abstract* — In the present work a two degree of freedom (2-DOF) electrohydraulic motion simulator platform has been developed. The heave motion of the platform has been controlled by a real time controller. Three different types of controller, namely proportional integral derivative (PID) controller, Hybrid Fuzzy-PID Controller and self-tuning Fuzzy-PID controller have been designed. The real time control performance of the controllers has been studied for 0.05m, 0.1m and 0.15m step demand. In order to achieve the required performance of the controller, suitable values for the control parameters are tuned by different method for different controller. The self-tuned Fuzzy-PID controller shows best control response compare to PID and hybrid Fuzzy PID controller.**

Keywords— *Electrohydraulic system, control design; real time control*

I. INTRODUCTION

Electrohydraulic systems are characterized by high power to weight ratio, self lubrication property, heat transfer property of the hydraulic oil, very good controllability, high stiffness, small position error, stable etc. The applications of the electrohydraulic systems are increasing rapidly with its reliability and satisfactory performances. The electrohydraulic systems are used in industries, automobiles, aircraft, highly precision machining technology, elevator, excavator, manipulator due to precise control of position, velocity and force as per requirement. The main challenges of the electrohydraulic systems are highly nonlinear [1] due to oil compressibility, nonlinear flow pressure relationship, piston friction hysteresis behaviour, valve deadband. Other disadvantages associated with the hydraulic system are high cost of the hydraulic components, loss of power due to leakage. The main challenges in designing a perfect controller for the electrohydraulic system are, difficult to identify the exact model of the electrohydraulic system, inherent nonlinearity, parameter variations with change in the working environment. To overcome these difficulties different types of control design approach like, nonlinear model based or using artificial intelligence (without modeling of the system) are attempted by the researchers in the last decade.

The simple proportional integral derivative (PID) controller mostly used in industrial application, because of easy to design i.e. simple structure and robustness [2]. The controller has transient and steady-state error reducing capacity at the same time. In case of industrial application the control gains K_P, K_I and K_D of the PID controller kept constant. Consequently the PID controller cannot perform satisfactorily, [3, 4] when the system have hard nonlinearity, parameter uncertainty, variable set point, and requirement of fast response. To improve the PID controller performance the classical optimal gain tuning method like Ziegler-Nichols and Cohen-Coon can be applied. The disadvantage of these methods is that, to get the optimal value of parameter it takes long time and because of oscillation the nonlinear real system may lead to instability [5, 6].

There are numerous control strategies used in control system design as applied in classical control, modern control and intelligent control systems. Every control system technique has its advantages and disadvantages. Therefore, the trend nowadays is to implement hybrid systems consisting of more than one types of control technique. The ideal controller would be robust against parameter variations and lead to better performances.

Feedback control system design using PID controller has been commonly used due to its simplicity. However, the PID method is not suitable for controlling a system with large amount of lag, parameter variations and uncertainty in models. Thus, PID control method cannot accurately control position in a hydraulic system due to valve dead band, piston friction, oil compressibility. The control performance of the controller can be improved with some modification [7], like feedforward plus PID controller, friction compensation controller [8]. To improve PID control performance, many researchers have integrated Fuzzy Logic Control technique to tune the PID parameter [9]. Song and Liu [10] used self-tuning Fuzzy-PID controller to control switched reluctance motor. In this paper the design and performance analysis of the PID controller, hybrid Fuzzy PID controller and self-tuning Fuzzy-PID controller has been discussed.

II. SETUP DESCRIPTION

Two Degrees of Freedom (2 DOF) electro-hydraulic motion simulator test set up has been developed for real time control experimentation. The circuit diagram and photographic view of the experimental setup have been shown in Fig. 1 and Fig. 2 respectively. The setup consists of two hydraulic cylinders S1 and S2, two proportional solenoid operated direction control valves SV1 and SV2, one pressure relief valve RV, one gear pump P, motor M, one NI-cRIO 9014 real time system having 9215 input module and a 9263 output module, two linear variable differential transducers LVDT1 and LVDT2 a Host PC. The double acting single rod linear hydraulic actuators CD250B40/20-00A1X/01/CGDM1-1M from Rexroth attached to base and the piston rod connected to a common platform, as shown in Fig. 1 and Fig 2. Top end of one piston rod is connected with the platform by one universal joint whereas top end of another piston rod is connected to the same platform by a pivot joint to allow pitch motion in combination with heave motion for the platform. If there is no pitch motion, we can get pure heave motion of the platform. The cylinders are fixed with the vertical pillar as shown in the Fig. 1 and Fig. 2. The platform will move vertically, keeping the face horizontal when both the cylinders are move together at same velocity. Two proportional solenoid direction control valves PV1 and PV2, 4WRE-6E1-32-21/G24-K4/V from Rexroth is connected to the each cylinder separately. The magnitude and direction of flow through these valves depends on command voltage signal supplied to the solenoid of the valve. The command signal, which has been processed in the Host computer as a digital signal through Labview software, is based on control algorithm and error signal fed to the controller. The digital signal has been communicated between Host PC and NI-cRIO through a Ethernet cable. The command signal has been send to the analogue amplifier card (AAC) of the PV through the output module of the NI-cRIO. The command voltage signal is amplified in the ACC and drives the solenoid of the PV. The command voltage signal to the solenoid causes the movement of the spool of the PV, which led to port opening of the PV, hence the pressurized oil flows to the cylinder and low pressure oil flows to the tank from the piston chamber.

The direction and the magnitude of the oil flow caused the direction and magnitude of the cylinder velocity thereby the cylinder position. The acquired LVDT analogue signal is fed to the host computer as digital signal through the input module NI-9263 of the NI-cRIO. The conversion of the analogue signal to digital signal and digital signal to analogue signal takes place in NI-cRIO processor. A solenoid driven pilot operated pressure relief valve from Yuken has been installed to set the operating pressure of the system. The system pressure has been set at 3 MPa during the real time experimentation. Two LVDTs from Gefran LT-M-0200-S-XL0202 having range 0 to 0.2m. are used to sense the positions of two piston rods, hence the orientation of the platform.

TABLE I. NOMENCLATURE

SYMBOL	DESCRIPTION	UNIT
K_P	Proportional gain	V/m,
K_I	Integral gain	V/ms,
K_D	Derivative gain	Vs/m
K_{PSi}	Proportional gain scale factor	-
K_{ISi}	Integral gain scale factor	-
K_{DSi}	Derivative gain scale factor	-
y_{di}	Demand piston displacement	m
y_{LVDTi}	Feedback piston displacement	m
y_{ei}	Displacement error	m
v_{ei}	Velocity error	m
LVDT	Linear Variable Differential Transducer	
SV	Spool Valve (Proportional)	
RV	Relief Valve	
PID	Proportional Integral Derivative	
CRIO	Compact reconfigurable input output	
DOF	Degree of freedom	
COA	Center of area.	

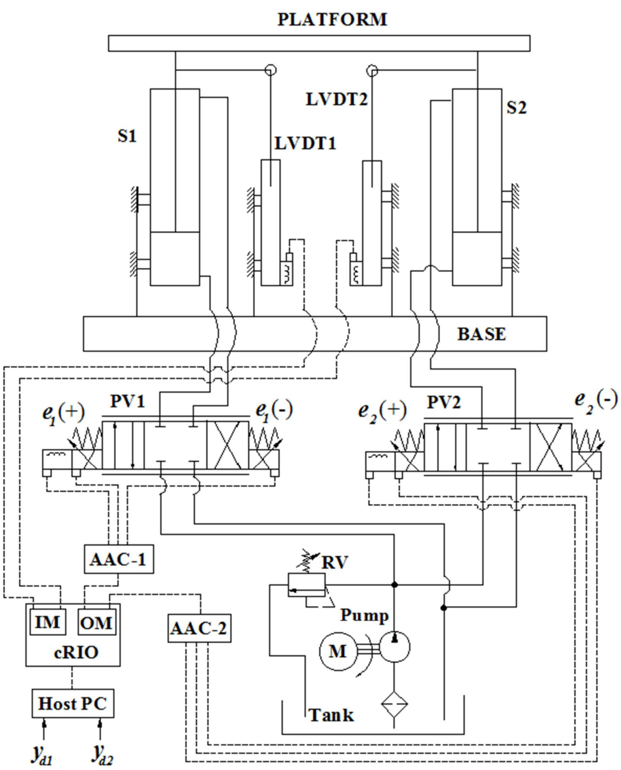

Fig. 1. Block diagram of the experimental setup.

A Dowty make Gear pump of 30 lpm has been used for oil supply to the system. It is possible to get different orientations of the platform by adjusting the magnitude and sign of the voltage signal to the solenoid of the each proportional valve. If negative command signal given to the solenoid of PV, the spool moves to the left side as shown in the Fig. 1, the pressurized oil will go to the lower chamber of the cylinder and low pressure oil from upper chamber of the cylinder flows to the tank, hence the piston will move upward, i.e. the platform will move upward. Similarly if negative voltage signal given to the solenoid of PV the platform will move downward.

Fig. 2. Photographic view of the experimental setup.

III. Controller Design

The real time controller has been designed in Labview graphical user interface (GUI) with NI-cRIO data acquisition system and the field programmable gate array (FPGA) architecture has been used for embedding the control design. Three different types of the controllers, like PID, Fuzzy-PID hybrid and Fuzzy based self tuned Fuzzy-PID controller has been designed for real time control of the hydraulic platform. The structures of the three types of controller have been represented in Fig. 3, Fig. 4 and Fig. 5 respectively.

A. PID Controller Design

The real time PID control block diagram for 2DOF electro hydraulic motion simulator shown in the Fig 3. The LVDT1 which is connected with Cylinder S1 give the position feedback y_{LVDT1} and LVDT2 which is connected with Cylinder S2 give the position feedback y_{LVDT2} . demand y_{di} and response. The PID controllers generates the control signal e_i with respect to the position received from the LVDT y_{LVDTi} , where i=1 for controller 1 and i=2 for controller 2. The controller 1 and 2 generates the control on the basis of the Eqn. 1.

$$e_i = K_{Pi}(y_{di} - y_{LVDTi}) + K_{Ii}\int (y_{di} - y_{LVDTi})dt + K_{Di}\frac{d}{dt}(y_{di} - y_{LVDTi}) \qquad (1)$$

Where K_{Pi} , K_{Ii} and K_{Di} are the proportional, integral and derivative gain respectively.

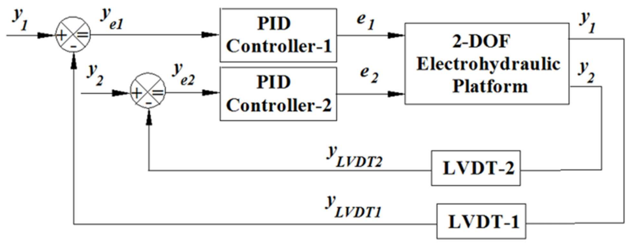

Fig. 3. PID control block diagram for the 2DOF twin hyfraulic cylinder platform..

B. Fuzzy-PID Hybrid Controller Design

The block diagram of the Fuzzy-PID hybrid controller architecture represent in the Fig 4. The Fuzzy-PID hybrid controller has two components, i.e. PID controller and Fuzzy controller component. The two controllers coupled with a switch, shown in the Fig.4.

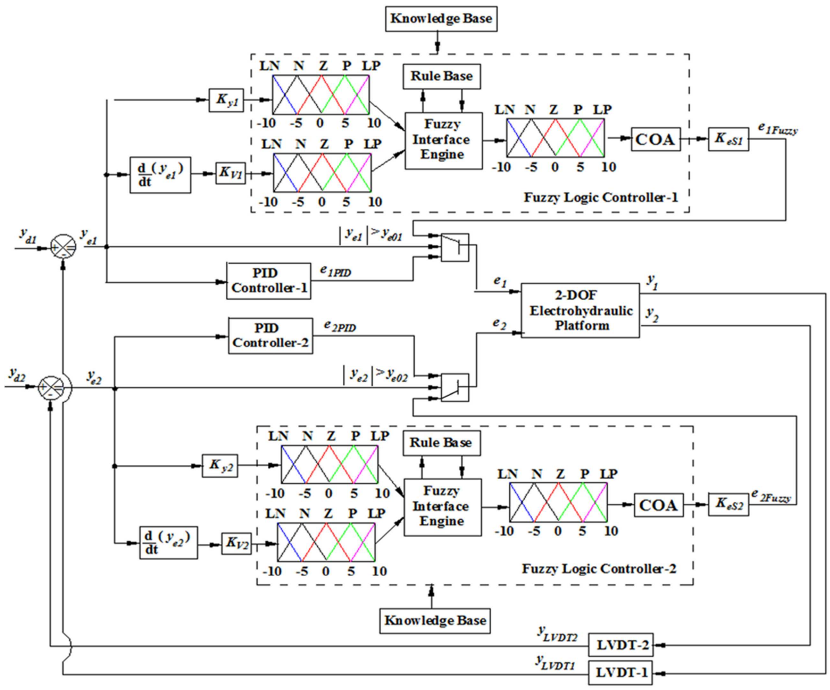

Fig. 4. Hybrid PID-Fuzzy block diagram.

The multiple input single output (MISO) type fuzzy Mamdani control structure has been used for Fuzzy controller component of the hybrid Fuzzy-PID controller. Fuzzy controller and PID controller are generates control signal separately, but which component of the control signal will go the plant decide by the Eq. (2).

$$\text{If } |y_{ei}| > y_{e0i},\ e_i = e_{iFuzzy} \text{ else } e_i = e_{iPID} \quad (2)$$

Where, y_{ei} is the displacement error has been fed to the controller and y_{e0i} manually chosen threshold value of the displacement error. The value of i=1 for controller 1 and i=2 for controller 2.

TABLE II. FUZZY RULE OF THE HYBRID PID-FUZZY CONTROLLER

y_e \ v_e	LN	N	Z	P	LP
LN	LN	LN	N	N	Z
N	LN	N	N	Z	P
Z	N	N	Z	P	P
P	N	Z	P	P	LP
LP	Z	P	P	LP	LP

Two input variable of the fuzzy controller are displacement error y_{ei} and the velocity error v_{ei}. The scaled crisp input variable is fuzzyfied with respect to triangular membership function like LN, N, Z, P, LP as shown in the Fig 4. The rule of the fuzzy part of the hybrid Fuzzy-PID controller has been shown in the table I. Based on that rule and the input variable, the output variable has been estimated. The output variable has been defuzzyfied by the centre of area (COA) method. Hence the crisp output variable command signal has been scaled and sent to the PV

C. Self Tuning Fuzzy-PID Controller Design.

The self tuning Fuzzy-PID controller has been developed for position control of the twin cylinder hydraulic platform. In this control scheme, Fuzzy Mamdani rule base has been used for tuning of the PID gain as shown in Fig 5.

The control gain has been estimated based on the position error y_{ei} and the velocity error v_{ei} .The position of the hydraulic cylinder has been measured by the LVDT. Based on the position demand and the LDVT feedback, the position error has been estimated. The velocity error has been estimated by getting the derivatives of the position error through a inbuilt derivative block exist in the Labview GUI. The scaling of the position error and velocity error has been done by multiplying the position error and velocity error by the gain K_{yi} and K_{vi} .

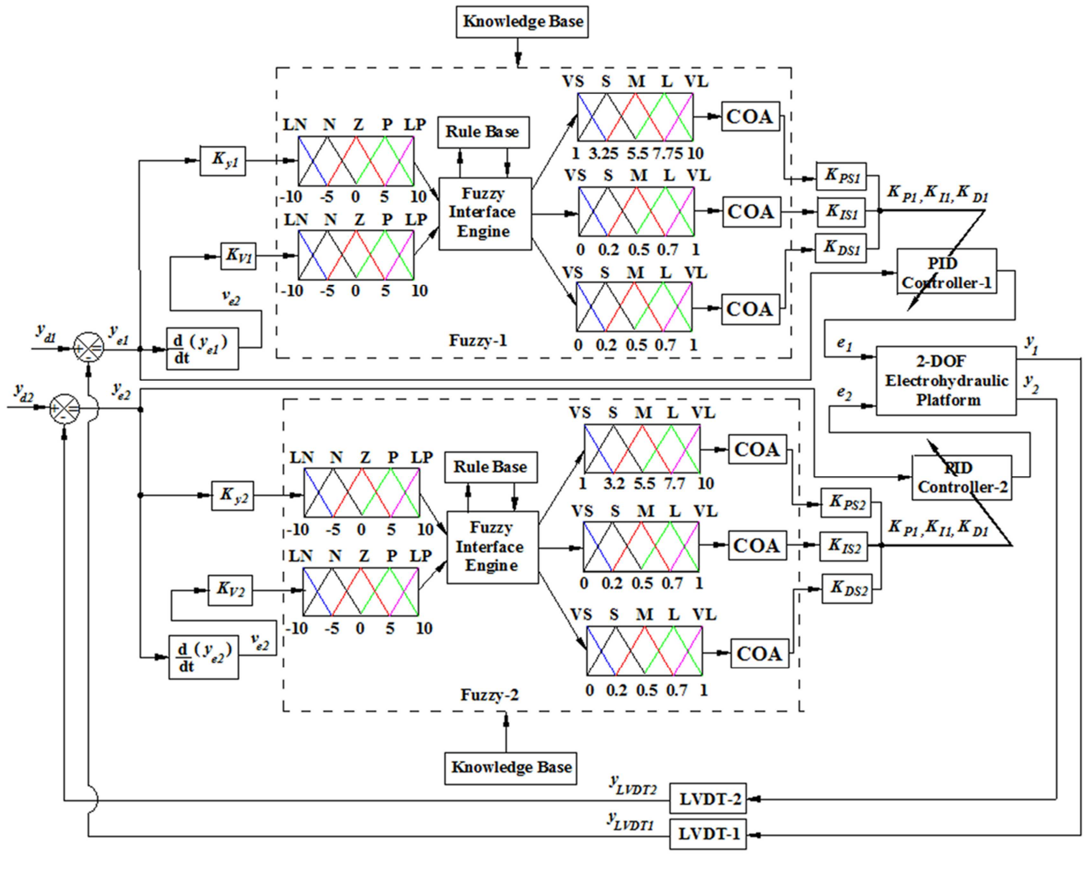

Fig. 5. Self-tuning Fuzzy-PID control block diagram.

For self-tuning Fuzzy-PID controller the membership function for the input variable, i.e. displacement error, y_{ei} and velocity error, v_{ei} has been chosen as, LN, N, Z, P, LP and the membership function for the output variable, i.e. proportional integral and derivative gain K_{Pi}, K_{Ii}, and K_{Di} has been chosen as, VS, S, M, L, VL.

The scaled position and velocity error are fuzzyfied. So the crisp value of the position error and velocity error has been converted in to the fuzzy variable μ_y and μ_v. Based on the fuzzy variable position error and the velocity error μ_y and μ_V and the rule base conditional statement like "IF y_{ei} is P AND v_{ei} is N THEN K_{Pi} is M ", the interface engine gives the output of proportional integral and derivative gain K_{Pi}, K_{Ii}, and K_{Di} as fuzzy variable. The estimated fuzzy output variables are de-fuzzyfied by centre of area method (COA). The de-fuzzyfied output variable has been scaled by the scale factors K_{PSi}, K_{ISi} and K_{DSi}. Hence the PID control gain K_P, K_I and K_D, is estimated and send to the PID controller.

The PID controller computes the control command signal, e_i with respect to the feedback position error and the estimated control gain. The estimated control signal send to the PV through the NI-cRIO, which lead to the movement of the hydraulic cylinder, hence reduce the position error. The rules of the control gain K_P, K_I, and K_D are shown in the Table II, III and IV respectively.

TABLE III. FUZZY RULE OF THE K_P GAIN FOR SELF TUNING FUZZY-PID CONTROLLER

y_e \ v_e	LN	N	Z	P	LP
LN	VS	VS	S	S	M
N	VS	S	S	M	L
Z	S	S	M	L	L
P	S	M	L	L	VL
LP	M	L	L	VL	VL

TABLE IV. FUZZY RULE OF THE K_I GAIN FOR SELF TUNING FUZZY-PID CONTROLLER

y_e \ v_e	LN	N	Z	P	LP
LN	VS	VS	S	S	M
N	VS	S	S	M	L
Z	S	S	M	L	L
P	S	M	L	L	VL
LP	M	L	L	VL	VL

TABLE V. FUZZY RULE OF THE K_D GAIN FOR SELF TUNING FUZZY-PID CONTROLLER

y_e \ v_e	LN	N	Z	P	LP
LN	LN	LN	N	N	Z
N	LN	N	N	Z	P
Z	N	N	Z	P	P
P	N	Z	P	P	LP
LP	Z	P	P	LP	LP

IV. RESULTS AND DISCUSSION

The comparisons of PID, hybrid Fuzzy-PID and self-tuning Fuzzy-PID control response of the twin cylinder hydraulic platform for 0.05 m step demand have been illustrated in Figs 6 and Fig. 7. Fig. 6 represents the controller performance for the cylinder S1 and Fig. 7 represents the performance for the cylinder S2 The dark continuous line represent as the demand, the dashed dot line represent as the PID control response, the gray continuous line represent as the performance of the hybrid Fuzzy-PID control response and the dashed line represent as the performance of the self-tuning Fuzzy-PID control response. The PID controller gains are selected as KP=50, KI=20 and KD=2 for the all the cases. The control gains are chosen manually by trial and error method to obtain the desire control performance.

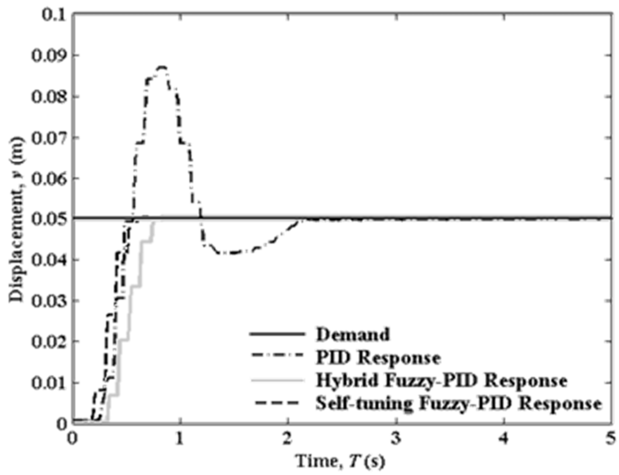

Fig. 6. Comparison of PID, Hybrid Fuzzy-PID & self-tuning Fuzzy-PID control response of cylinder 1 for 0.05 m step demand.

The PID controller has been found to depict poor response compare to the hybrid Fuzzy-PID controller and self-tuning Fuzzy-PID controller. It shows a great amount of overshoot and the time taken to come at steady state also large. The hybrid Fuzzy-PID controller achieves the steady state position without overshoot but it is slower compare to the self-tuning Fuzzy-PID controller.

From the Fig. 6 and Fig. 7 it is evident that the cylinder S1 and S2 are not moving together at equal velocity, the overshoot of the cylinder S2 is slightly more. The behavior of the two cylinders are not same because the difference of flow transient. Cylinder S1 achieves the demand faster than the cylinder S2.

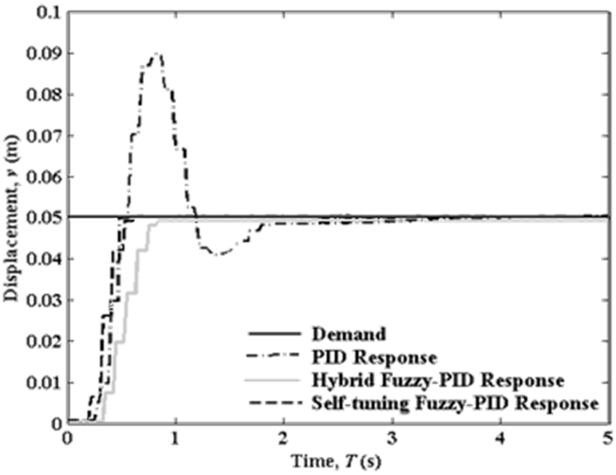

Fig. 7. Comparison of PID, Hybrid Fuzzy-PID & self-tuning Fuzzy-PID control response of cylinder S2 for 0.05 m step demand.

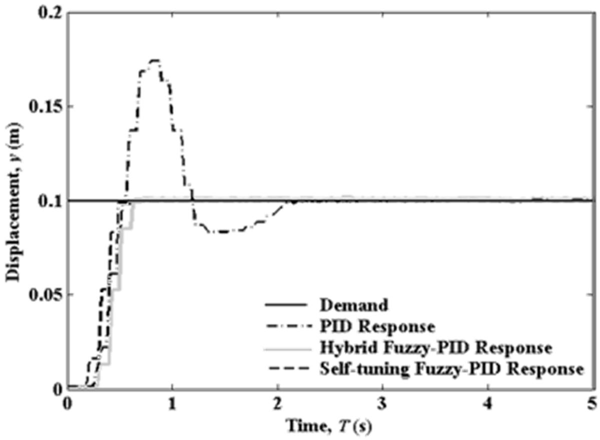

Fig. 8. Comparison of PID, Hybrid Fuzzy-PID & self-tuning Fuzzy-PID control response of cylinder S1 for 0.1 m step demand.

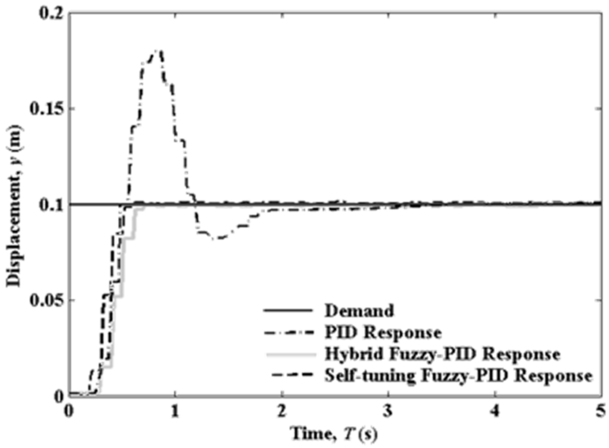

Fig. 9. Comparison of PID, Hybrid Fuzzy-PID & self-tuning Fuzzy-PID control response of cylinder S2 for 0.1 m step demand.

Fig. 8, and Fig. 9 shows the control performances of the PID, hybrid Fuzzy-PID and self-tuning Fuzzy-PID control response of the twin cylinder hydraulic platform for 0.1 m step demand. The performance comparisons of the different controllers are similar as 0.05 m step demand.

The control responses of PID, hybrid Fuzzy-PID and self-tuning Fuzzy-PID controller of the twin cylinder hydraulic platform for 0.15 m step demand for heave motion have been illustrated in Figs 10 and Fig. 11 for cylinder S1 and cylinder S2 respectively. For the PID controller, the cylinder S1, with overshoot moves to the saturated position with some steady state error however, the cylinder S2 has been found to achieve the demand finally. The performance of the hybrid Fuzzy-PID and self-tuning Fuzzy-PID are similar to the other cases.

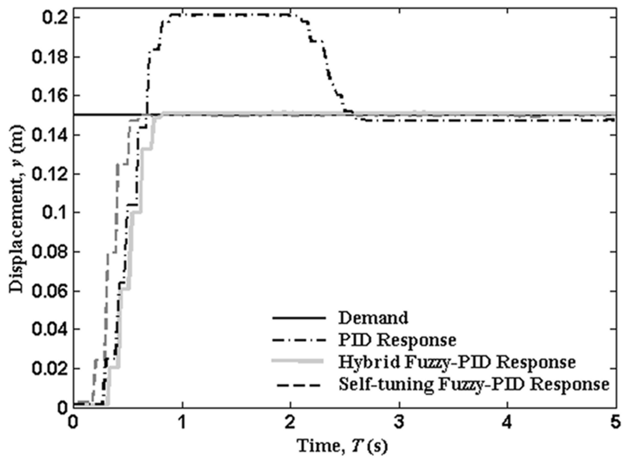

Fig. 10. Comparison of PID, Hybrid Fuzzy-PID & self-tuning Fuzzy-PID control response of cylinder S1 for 0.15 m step demand.

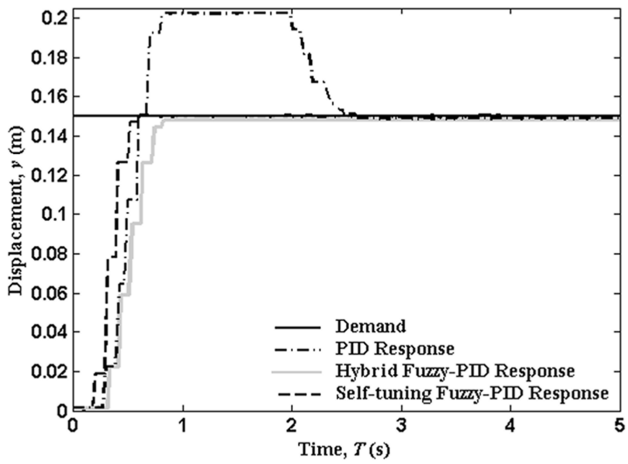

Fig. 11. Comparison of PID, Hybrid Fuzzy-PID & self-tuning Fuzzy-PID control response of cylinder S2 for 0.15 m step demand.

V. CONCLUSION

In the present work a twin cylinder hydraulic platform setup has been developed and different types of controller have been designed for that system. The PID, hybrid Fuzzy-PID and self-tuned Fuzzy-PID controller are used to study the control performance of the twin cylinder hydraulic cylinder for heave motion. The control performance of the three types controllers are compared for 0.05m, 0.1m and 0.15m step demand respectively. The PID controller achieves the demand with a high overshoot and small oscillation. To achieve the demand it takes about four second. On the other hand the Fuzzy-PID hybrid controller and the self tuning Fuzzy-PID controller achieve the demand without any overshoot, and it takes about 0.6 second which is very fast. The performance of the self tuned Fuzzy-PID controller is best compare to these three controllers and it is satisfactory.

ACKNOWLEDGMENT

The authors acknowledge the financial support from Council of Scientific and Industrial Research (CSIR), India for installing the test set-up.

REFERENCES

[1] H. E. Merritt, *Hydraulic control systems.* New York: Wiley, 1967..

[2] N. Ishak, M. Tajjudin, H. Ismail, M. H. F. Rahiman, Y. Md Sam, R. Adnan "PID Studies on Position Tracking Control of an Electro-Hydraulic Actuator," International Journal of Control Science and Engineering, 2012, pp. 120-126.

[3] Parnichkun, M. C. Ngaecharoenkul., "Hybrid of fuzzy and PID in kinematics of a pneumatic system," Proceeding of the 26th Annual Conference of the IEEE Industrial Electronics Society, Japan, 2000, pp: 1485-1490.

[4] P. Pratumsuwan, S. Thongchai, S. Tansriwong., "A Hybrid of Fuzzy and Proportional-Integral-Derivative Controller for Electro-Hydraulic Position Servo System" Energy Research Journal, vol. 1,issue 2, 2010, pp. 62–67.

[5] V. Bobal, J. Bohm, J. Fessl, and J. Machacek, Digital Self-tuning Controllers: Algorithms, Implementation and Applications, Springer, 2005.

[6] Ismail Algelli Sassi Ehtiwesh, and Željko Đurović, "Comparative Analysis of Different Control Strategies for Electro-hydraulic Servo Systems", World Academy of Science, Engineering and Technology 32 2009.

[7] B. K. Sarkar, J. Das, R. Saha, S. Mookherjee, D. Sanyal, "Approaching Servoclass Tracking Performance by a Proportional Valve-Controlled System" IEEE/ASME transaction on Mechatronics., vol. 18, no. 4, pp.1425-1430, 2013.

[8] G. A. Sohl and J. E. Bobrow, "Experiments and simulations on the nonlinear control of a hydraulic servosystem," *IEEE Trans. Control Syst.Technol.*, vol. 7, no. 2, pp. 238–247, Mar. 1999

[9] Ki. Sinthipsomboon, W. Pongaen , P. Pratumsuwan "A Hybrid of Fuzzy and Fuzzy self-tuning PID Controller for Servo Electro-hydraulic System" ,6th IEEE Conference on Industrial Electronicsand Applications (ICIEA) , pp. 220-225, 2011.

[10] S. Song, W. Liu.,"Fuzzy parameters self-tuning PID control of switched reluctance motor based on Simulink/NCD," CIMCA-IAWTIC'06, IEEE, 2006.