Jurnal
Teknologi

Full Paper

PID PARAMETERS OPTIMIZATION USING PSO TECHNIQUE FOR NONLINEAR ELECTRO HYDRAULIC ACTUATOR

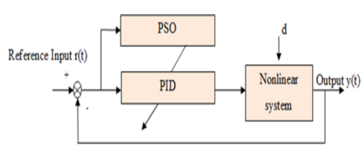
Siti Marhainis Othman, Mohd Fua'ad Rahmat*, Sahazati Md. Rozali, Sazilah Salleh

Department of Control and Mechatronic Engineering, Faculty of Electrical Engineering, Universiti Teknologi Malaysia, 81310 UTM Johor Bahru, Johor, Malaysia

Article history
Received
1 June 2015
Received in revised form
13 August 2015
Accepted
29 September 2015

*Corresponding author
fuaad@fke.utm.my

Graphical abstract



Abstract

Electro-hydraulic actuator (EHA) system inherently suffers from uncertainties, nonlinearities and time- varying in its model parameters which cause the modeling and controller designs are more complicated. Proportional Integral Derivative (PID) control scheme has been proposed and the main problem with its application is to tune the parameters to its optimum values. This study will look into an optimization of PID parameters using particle swarm optimization (PSO). Simulation study has been done in Matlab and Simulink.

Keywords: Electrohydraulic Actuator, PID, PSO, Optimization, Nonlinear System

Abstrak

Over recent years, there has been an explosive growth of interest in the development of novel gel-phase materials based on small molecules. It has been recognised that an effective gelator should possess functional groups that interact with each other via temporal associative forces. This process leads to the formation of supramolecular polymer-like structures, which then aggregate further, hence gelating the solvent. Supramolecular interactions between building blocks that enable gel formation include hydrogen bonds, interactions, solvatophobic effects and van der Waals forces

Kata kunci: Sistem servo penggerak hidrolik;PID;PSO;pengoptiman;Sistem Tidak Linear

1.0 INTRODUCTION

Electro-Hydraulic Actuator (EHA) system is one of the important drive systems in industrial sectors and most engineering practices due to its high power to weight ratio and stiffness response being good, smooth and fast. Recently, with the research and development of mathematics, control theory, computer technology, electronic technology and basic theory of hydraulics, hydraulic control technology has been developed and used widely in many applications such as manufacturing systems, material testing machines, active suspension systems, mining machineries, fatigue testing, flight simulation, paper machines, ships and electromagnetic marine engineering, injection moulding machines, robotics, and steel and aluminium mill equipment [1]. Due to its applications, the highest performance of the electro-hydraulic actuators in terms of position, force or pressure is needed. However, the system is highly nonlinear due to many factors, such as leakage, friction, and especially, the fluid flow expression through the servo valve[2].

A suitable controller needs to be designed in order to acquire good performance of the electro-hydraulic actuator. The controller design requires the best mathematical model of the system under control[3]. The mathematical model is established through a modeling process where the system is identified based on the conservation laws and property laws. This process is crucial since a controller is design solely based on this mathematical model. Thus, an accurate equation must be derived in order for the controller to response accordingly. This work presents an investigation of performance comparison between conventional (PID) and optimized PSO-PID for an electro-hydraulic actuator in terms of trajectory tracking. The dynamic model and design requirement have been taken from National Institute for Aerospace Research, Romania[4]. Comparative assessment of both control schemes to the system performance is presented and discussed.

2.0 DYNAMIC MODEL

The servo valve and hydraulic cylinder are the two important components of an EHA system. Basically, EHA is a double-acting hydraulic cylinder with the single - rod or single ended piston. A single load is normally attached at the end of the piston without the inclusion of spring and damper [5]. The schematic diagram of a typical EHA system is shown in Figure 1. In this figure, ps is the hydraulic supply pressure and p_r is the return pressure. x_v is the spool valve displacement, Q1 and Q2 are the fluid flow from and to the cylinder, and p_1 and p_2 are the fluid pressure in the upper and lower cylinder chambers of the actuator.

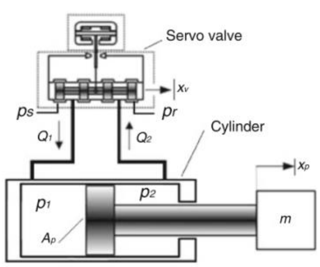


Figure 1 The schematic diagram of typical EHA systems

The actuator dynamic equation of electro-hydraulic actuator servo system is expressed as[4]

$$m\ddot{x}_p = SP_L - f\dot{x}_p - kx_p - F_L \tag{1}$$

where, m is load at the rod of the system, x_p is the displacement of the piston, P_L is the difference in pressure between two chambers, k is the coefficient of aerodynamic elastic force, f is the coefficient of viscous friction, S is the piston area and F_L is the external disturbance injected into the system's actuator. With the assumption that a high-response servo valve is used in the system, the control applied to the spool valve is proportional to the spool position. Its equation is given as:

$$x_v = k_v u \tag{2}$$

where x_v is the opening of the valve, k_v is the coefficient of the servo valve and u is the input voltage.

Assume that the system is a symmetrical cylinder, therefore, both piston area and volume for each port are similar. Thus, the dynamics of cylinder oil flow can be expressed as follows:

$$Q_L = \dot{P}_L + \frac{2\beta s}{v}\dot{x}_p \tag{3}$$

where Q_L is the difference between supplied flow rate to the chambers, v is the volume of the chamber and β is the effective bulk modulus of the fluid. Thus the difference of the flow rate to the chambers is given as:

$$Q_L = \frac{2\beta}{v}\left[c_d w\sqrt{\frac{P_a - P_L}{\rho}}x_v - k_l P_L\right] \tag{4}$$

where c_d is the coefficient of the volumetric flow of the valve port, P_a is the supply pressure, ρ is the oil density and k_l is the coefficient of internal leakage between the cylinder chambers. Let $x_1 = x_p$, $x_2 = \dot{x}_p$ and $x_3 = P_L$.

$$\dot{x}_1 = \dot{x}_p = x_2 \tag{5}$$

$$\dot{x}_2 = \ddot{x}_p \tag{6}$$

Referring (1),

$$\ddot{x}_p = \frac{s}{m}P_L - \frac{f}{m}\dot{x}_p - \frac{k}{m}x_p - \frac{F_L}{m} \tag{7}$$

Thus,

$$x_2 = \frac{s}{m}P_L - \frac{f}{m}\dot{x}_p - \frac{k}{m}x_p - \frac{F_L}{m} \tag{8}$$

$$\dot{x}_3 = \dot{P}_L \tag{9}$$

From (3),

$$P_L = Q_L - \frac{2\beta s}{v}\dot{x}_p \tag{10}$$

Substituting (4) into (10), thus (10) becomes

$$P_L = \frac{2\beta}{v}c_d w\sqrt{\frac{P_a - P_L}{\rho}}x_v - \frac{2\beta}{v}k_l P_l - \frac{2\beta}{v}\dot{x}_p \tag{11}$$

As a result, the differential equations governing the dynamics of electro-hydraulic actuator servo system with external disturbance injected to its actuator is given as

$$\dot{x}_1 = x_2 \tag{12}$$

$$\dot{x}_2 = -\frac{k}{m}x_1 - \frac{f}{m}x_2 + \frac{s}{m}x_3 - \frac{F_L}{m} \tag{13}$$

$$\dot{x}_3 = -\frac{s}{k_c}x_2 - \frac{k_l}{k_c}x_3 + \frac{c}{k_c}\sqrt{\frac{P_a - x_3}{2}}k_v \tag{14}$$

where x_1 is the displacement of the load, x_2 is the load velocity and x_3 is the differential pressure $\rho_1 - \rho_2$ between the cylinder chambers caused by the load.

F_L is the external disturbance given to the system and it can be constant or a time varying signal. Table 1 shows the parameters of electro hydraulic actuator servo system which are represented by (12), (13) and (14).

.

Table 1 Parameter of EHA Servo System

Bil	Parameters	Symbol	Value	Unit
1	Load at the EHA rod	m	0.33	Ns^2/cm
2	Piston Area	S	10	cm^2
3	Coefficient of viscious friction	f	27.5	Ns/cm
4	Coefficient of aerodynamic elastic force	k	1000	N/cm
5	Valve port width	w	0.05	cm
6	Supply pressure	P_a	2100	N/cm^2
7	Coefficient of volumetric flow of the valve port	c_d	0.63	-
8	Coefficient of internal leakage	k_l	2.38x10-3	cm^5/Ns
9	Coefficient of servo valve	k_v	0.017	cm/V
10	Coefficient involving bulk modulus and EHA volume	k_c	2.5x10-4	cm^5/N
11	Oil density	ρ	8.87x10-7	Ns^2/cm^4

By substituting all the parameters into (12), (13) and (14), the system equations can be represented in state–space form as below:

$$\dot{x}(t)_{3\times1} = A_{3\times3}x(t)_{3\times3} + B_{3\times1}u(t) + C_{3\times1}$$

where

$$\dot{x}(t) = \begin{bmatrix}\dot{x}_1(t) & \dot{x}_2(t) & \dot{x}_3(t)\end{bmatrix}^T$$

$$A = \begin{bmatrix} 0 & 1 & 0 \\ -3030.3 & -83.3 & 30.3 \\ 0 & -40000 & -9.52 \end{bmatrix}$$

$$x(t) = \begin{bmatrix}x_1(t) & x_2(t) & \end{bmatrix}^T$$

$$B = \begin{bmatrix} 0 & 0 & 3196\sqrt{\dfrac{2100 - x_3}{2}} \end{bmatrix}^T$$

$$C = \begin{bmatrix}0 & -3.03F_L & 0\end{bmatrix}^T$$

3.0 CONTROLLER DESIGN AND SIMULATION

In this section, two control schemes PID and PSO-PID are proposed and described. The following design requirements have been made to examine the performance of both control strategies.

1) *The system overshoot (%OS) of displacement of the piston, x_1 is to be at most 10%.*
2) *The rise time (T_r) of displacement of the piston, x_1 is to be less than 50 s.*
3) *The settling time (T_s) of displacement of the piston, x_1 to be less than 200 s.*

F_L is an external disturbance that will be injected to the system's actuator as perturbations to the EHA system. Different signal of F_L is used to examine the response of both controllers. Constant value of signal F_L=10000 N is added as a perturbation to system actuator in the first case, while the second case is considered as benchmark with F_L=0 N. The detail of the disturbance signals is listed as below:

1) Case 1 : F_L=10000 N
2) Case 2 : F_L=0

3.1 PID Controller

PID controller is most commonly used in process control applications because of their relative ease of operation and satisfied performances. Users can modify the dynamic properties of this controller by adjusting the three parameters: proportional, integral, and derivative[6]. By tuning the value of K_p, K_i and K_d of the PID controller, the performance of the system such as rise time, overshoot, settling time and steady state error can be improved [7].The critical gain, K_{cr} and critical period of oscillation, T_{cr} need to be determine before the tuning process. The value of K_p, K_i and K_d is adjusted from this two parameters based on Ziegler-Nichols tuning rules[8]. In this model, the critical gain attained is K_c = 0.4 with critical period, T_c = 7.15. From calculation based on Ziegler-Nichols tuning method and manual adjustment, the parameters of PID controller, Kp, Ki and Kd are 0.01, 0.01 and -0.1 respectively. Figure 2 shows the simulation model of the EHA with PID controller

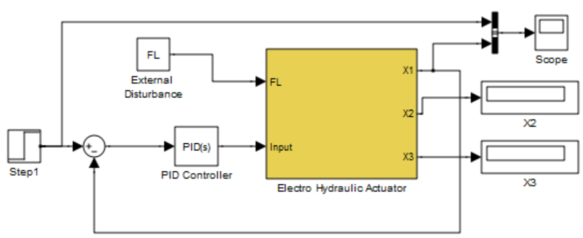


Figure 2 The simulation model of EHA with PID Controller

3.2 Particle Swarm Optimization

Particle Swarm Optimization (PSO) is motivated by swarming behaviors observed in flocks of birds, schools of fish, or swarms of bees. PSO is a population-based optimization tool, which could be implemented and applied to solve various function optimization problems, or the problems that can be transformed to function optimization problems. This method was developed through simulation of a simplified social system, and has been found to be robust in solving continuous nonlinear optimization problems[9]. The parameters used in the PSO are as follows:

Number of particles: 30,

Dimension of the problem: 3,

Number of maximum iterations: 50,

Figure 3 illustrates the block diagram of the PID controller with PSO algorithm.

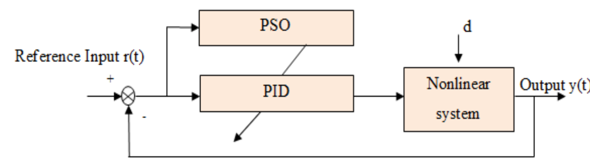


Figure 3 Block diagram of the PID Controller with PSO algorithm

From PSO optimization, the parameters of PID controller, Kp, Ki and Kd are 0.0152, 0.00557 and-0.114265 respectively. The performance comparison between PID and PSO-PID will be discussed in Section 4.

4.0 RESULTS AND ANALYSIS

In this section, the simulation results of the proposed controller are presented. A relative assessment of both control strategies to the system performance is also deliberated in details in this section.

Nonlinear electro-hydraulic actuator system with PID and PSO-PID controller block diagram produced three responses, displacement of piston x_1, piston velocity x_2 and the differential pressure between the cylinder chamber x_3. However, only the responses of the piston position, x_1 will be examined and presented in detail. As show in the simulation model earlier, the desired value of the piston position was set to follow the step signal. With the presence of different types of external disturbance, the response of the system with two different controllers will be compared. Figure 4 shows the comparison of the EHA system piston position response when F_L=0N injected to the system's actuator, between PID and PSO-PID controller graphically.

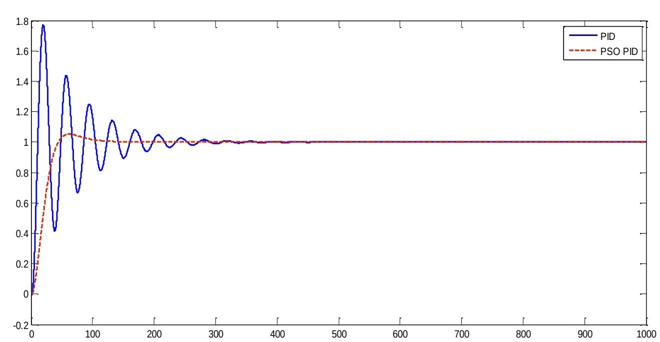


Figure 4 The response of the system with PID and PSO-PID when F_L=0N

In this figure, the response of the system with a PID controller is presented by blue color line or a straight line and the response of the system with PSO-PID controller is presented by red color line or a dotted line. The same color label also implemented in Figure 5.

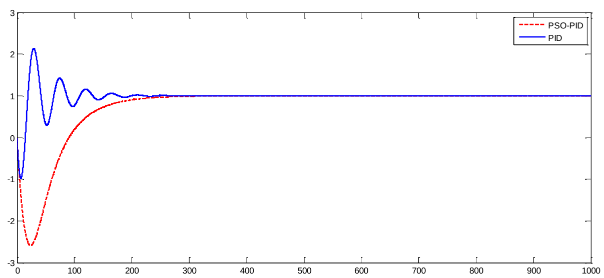


Figure 5 The response of the system with PID and PSO-PID when F_L=10000N

All figures show that both of controllers are capable to control the piston position, however with the presence of different types and value of external disturbance, the summary of the performance characteristics was tabulated in Table 2 and the comprehensive assessment will be discussed.

Table 2 shows the summary of the performance characteristics of the piston position between PSO-PID and PID controller quantitatively. Based on the data tabulated, PSO-PID has the fastest settling time with 130 compared to the PID controller with 263 seconds for F_L equal to 0 N. However, for F_L equal to 10000 N the performance of PSO-PID controller is lightly better and almost same with 263 seconds compared to the PID controller with 270 seconds. Both controllers with the presence of difference FL manage to drive the system response to achieve the desired reference given. The response of the system with PSO-PID has acceptable minimum overshoot and undershoot while the system with PID produced large overshoot with 76.6 and 113 percent. Performance characteristics for both controllers are represented in bar chart form as shown in Figure 6.

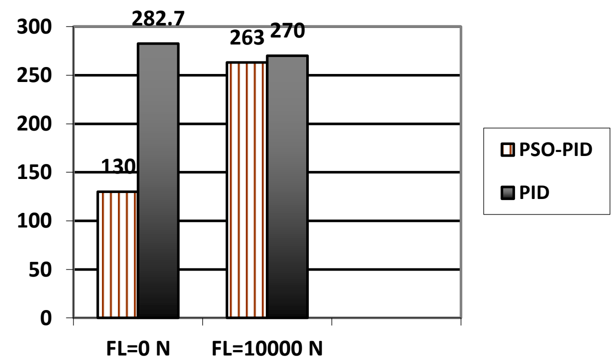


Figure 6 The response of the system with PID and PSO-PID controller when F_L=0 N and F_L=10000 N

5.0 CONCLUSION

In this paper, two controllers such as PSO-PID and PID are successfully designed. Based on the results and the analysis, a conclusion has been made that both the controllers, optimized PSO-PID and conventional controller (PID) are capable of controlling the nonlinear electro-hydraulic actuator displacement of the piston and fulfilled all the design requirements. The responses of each controller were plotted in one window and are summarized in Table 2. Simulation results and bar charts in Figure 4, 5 and 6 show that optimized PSO-PID controller has better performance compared to the PID controller in term of settling time and percentage of overshoot.

Acknowledgement

The author would like to thank the Malaysia Ministry of Higher Education (MOHE) for financial assistance while conducting this project.

Table 2 Summary of performance characteristics

Time Response Specification	FL=0N		FL=10000N	
	PSO-PID	PID	PSO-PID	PID
Settling Time	130sec	282.7 sec	263sec	270sec
Steady state error	0	0	0	0
Percentage of overshoot	5.1	76.6	Under-damped	113

References

[1] R. Ghazali, Y. M. Sam and M. F. Rahmat. 2012. Simulation and Experimental Studies on Perfect Tracking Optimal Control of an Eledrohydraulic Actuator System. *J. Control Sci. Eng*. 1-8.

[2] C. Kaddissi and J. Kenn. 2007. Identification and Real-Time Control of an Electrohydraulic Servo System Based on Nonlinear Backstepping. *IEEE Trans. Mechatronics*. 12(1): 12-22.

[3] R. Adnan, M. Tajjudin and N. Ishak. 2011. Self-tuning Fuzzy PID Controller for Electro- Hydraulic Cylinder. *IEEE 7th Int. Colloq. Signal Process. its Appl. Self-tuning*. 395-398.

[4] I. Ursu, F. Ursu and F. Popescu. 2006. Backstepping Design For Controlling Electrohydraulic Servos. *J. Franklin Inst.* 343(1): 94-110.

[5] Z. Has, M. F. Rahmat, A. R. Husain, and K. Ishaque. 2013. Robust Position Tracking Control of an Electro-Hydraulic Actuator in the Presence of Friction and Internal Leakage. *Arab. J. Sci. Eng.* 39(4): 2965-2978.

[6] M. A. A. A. N. K. Nasir, R. M. T. Raja Ismail. 2010. Performance Comparison Between Sliding Mode Control (SMC) and PD-PID Controllers for a Nonlinear Inverted Pendulum System. *World Acad. Sci. Eng. Technol*. 358-363

[7] X. Ma, X. Song, and H. Li. 2010. Tuning of the PID Controller Based on Model Predictive Control. *Chinese Control and Decision Conference*.1091-1096

[8] S. Md. Rozali. 2010. PID Controller Design for an Industrial Hydraulic Actuator with Servo System. *IEEE Student Conf. Res. Dev. (SCOReD 2010)*. 13-14.

[9] D. M. Wonohadidjojo and M. Y. Hassan. 2013. Position Control of Electro-Hydraulic Actuator System Using Fuzzy Logic Controller Optimized by Particle Swarm Optimization. *Int. J. Autom. Comput*. 10(3): 181-193.