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POSITION CONTROL FOR AN ELECTROHYDRAULIC VERTICAL LAUNCHING SYSTEM BASED ON PSO-PID STRATEGY

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Abstract

Concerning the positioning control problem of electrohydraulic vertical rocket launchers, which are mostly due to servo-valve nonlinearities. The model possesses time varying parameters with nonlinear functions that produces model uncertainties. Therefore, the control of such a dynamic system is still a problem. To achieve perfect behavior of the rocket launcher system, a tracking control policy for positioning of the rocket launcher is suggested. The particle swarm proportional–integral–derivative (PSO-PID) controller is developed, and the PSO algorithm is adjusted to search for the optimal control gains for the rocket launcher positioning model. The results illustrate that the PSO-PID strategy can guarantee the response speed and accuracy. Also, the nonlinear characteristics of a system can be effectively suppressed.

***Keywords*: PID; electrohydraulic; vertical elevation system; rocket launcher; PSO.**

Introduction

Electrohydraulic control systems possess many important merits of small size with compact structure, good weight to power ratio, high acceleration, stability, and ranging from heavy-duty applications to small price machine tools. So, it has important applications in the defense industry like rocket launchers and many industries such as hydraulic manipulators. However, the servo-valve dynamic behavior of electrohydraulic circuits is highly nonlinear. It depends on the flow-pressure relation, the control volume variations, and the friction, [1].

To grantee good tracking of the vertical elevation system, researchers have done a lot of scientific research papers to overcome the system uncertainty and disturbance rejection of electrohydraulic rocket launchers, [2, 3]. The linear control strategies are popular in these research work, [4, 5]. In such research, the design of these controllers is based on a linearized model at the operating point and may be ignoring some uncertain dynamic information.

Also, many studies of different control strategies are used for controlling the electrohydraulic servo system such as state observer based backstepping, optimal-tuning PID controller, h∞ position control synthesis, adaptive fuzzy sliding mode method and disturbance observer for position tracking system. The above-mentioned control policies cannot guarantee the required tracking characteristics because of the shortening of control gains with the system nonlinearities and external disturbances, [6-10]. So, it is significantly clear the needing of different techniques as nonlinear control policy that is adequate for such electrohydraulic vertical launching system, [11-12].

In this article, the PSO-PID controller is designed. In electrohydraulic system, a price description of the system model utilized all complicates. The dynamics equations of the vertical launching model are used and controller parameters are identified by numerical simulation.

The outline of the manuscript is as: in section 2, the problem of positioning the vertical launching system is formulated. In section 3 the mathematical dynamics equations are shown. In section 4, the proposed electrohydraulic control system is designed. The last section is devoted to the results and conclusions.

Problem Formalization of Vertical Launching Positioning System

The electrohydraulic positioning of the vertical launching system transfer arm as illustrated in Fig.1. It is a high-dynamic system, and the payload on the actuator varies with actuator position and rocket mass. Aiming to study how to guarantee good dynamic behavior.

For precise positioning of an electrohydraulic servo, the controlled volume flow rate reacts on the proportional relief valve to regulate the pressure relief valve ratio and realize active system compensation. Concerning the balance, a controller is designed; utilizing the control optimal algorithm, position the electrohydraulic system is calculated and the controller signal will regulate the system position by the proportional servo-valve.

The simulation results illustrate that compared with classical PID control and POS-PID control, the controller has excellent tracking performance.

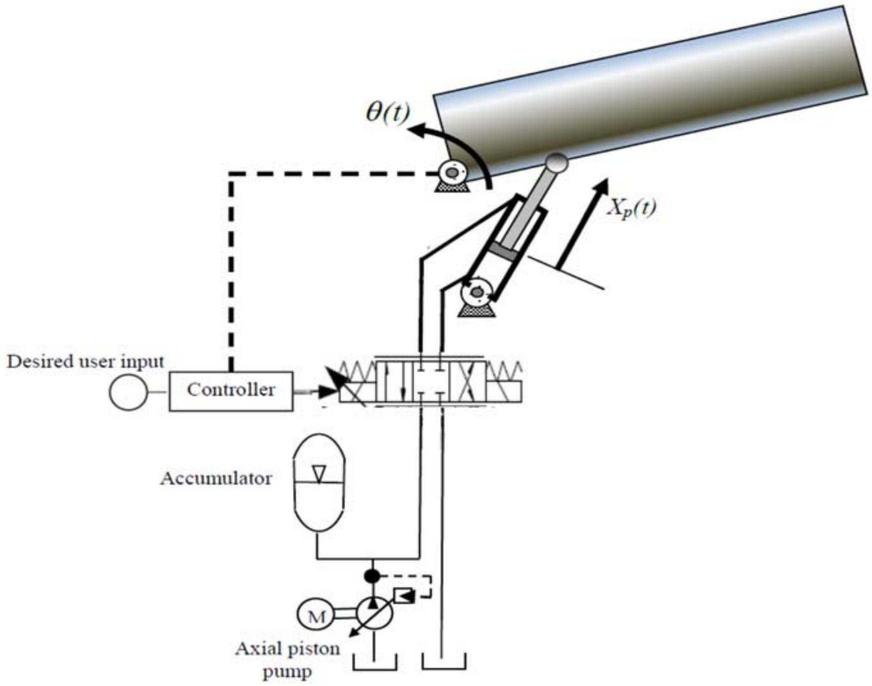

Fig. 1. Schematic of electrohydraulic vertical launching system

System Mathematical Model

The vertical launching system has a mechanical structure, electrohydraulic part, and control unit. The hydraulic actuator rises the launching-cover, and the linear variable differential transducer (LVDT) measures the angle of launcher position. As the angle maximumly open, the launching -cover is in position, locks the launching-cover, and loads the missile launch. The system-cover is unlocked after the missile is launched, and the cover is closed. It is often to model the hydraulic servo valve as a 2nd order transfer function, [13].

$$\frac{d^2x_v}{dt^2} + 2\zeta_v\omega_v\frac{dx_v}{dt} + \omega_v^2 x_v = \omega_v^2 k_v i_v \qquad (1)$$

where x_v is the servo valve spool displacement, i_v is the input current, k_v is the servo valve gain, ω_v is the servo valve natural frequency, and ζ_v is the servo valve damping ratio.

The dead-zone is often unknown, and hard nonlinearity in actuators of industrial control systems. It is between the hydraulic cylinder dynamics and the servo valve dynamics. It may reduce the control systems stability and performance. The simple form to represent the dead zone as:

$$i_v = \begin{cases} i - I_1 if i > I_1 \\ 0 if |i| \le I_1 \\ i + I_1 if i < -I_1 \end{cases} \qquad (2)$$

where i_v is the control signal and I_1 the dead zone width, load pressure $P_L = P_1 - P_2$ and load flow $Q_L = Q_1 - Q_2$.
For symmetric orifice, load pressure P_L and load flow Q_L relationship can be represented as [14]:

For negative x_v $$q_f = C_d W x_v sgn(P_f)\sqrt{\frac{2}{\rho}|P_f|}, q_n = C_d W x_v sgn(P_s - P_n)\sqrt{\frac{2}{\rho}|P_s - P_n|} \qquad (3)$$

For positive x_v $q_f = C_d W x_v\, sgn(P_s - P_f)\sqrt{\frac{2}{\rho}\left|P_s - P_f\right|}$, $q_n = C_d W x_v\, sgn(P_n)\sqrt{\frac{2}{\rho}\left|P_n\right|}$ (4)

where x_v is the spool valve position, C_d is the discharge coefficient, w is area gradient, ρ is fluid density and P_s is the supply pressure. The fluid flow in each chamber of hydraulic actuator, the next equation can be given, [15]:

$$q_{le} = K_l \frac{A_e}{A_f}\left[\frac{1+\left(\frac{A_n}{A_f}\right)^2}{1+\left(\frac{A_n}{A_f}\right)^3}\right] p_{le} + A_e \dot{X}_p + \frac{2A_e}{A_f}\frac{\dot{P}_{le}}{4B}\left[\frac{V_f+\left(\frac{A_n}{A_f}\right)^2 V_n}{1+\left(\frac{A_n}{A_f}\right)^3}\right] \quad (5)$$

where $P_{le} = \frac{p_f A_f - p_n A_n}{A_e}$, $q_{le} = \frac{q_f + q_n}{2} \rightleftarrows$, $A_e = \frac{A_f + A_n}{2}$, P_{le}, q_{le}, A_e, k_l, B, X_p, V_n, V_f, A_n, and A_f are the effective load pressure, the effective load flow rate, the effective piston area, the oil bulk modulus, the leakage coefficient of the piston, the piston displacement, the annular cylinder side volume, the full cylinder side volume, the cylinder annular area and is the cylinder full area respectively.

The governing equation of the launching system is formed as:

$$P_{le} A_e = M_e \ddot{X}_P + B_e \dot{X}_P + F_d \quad (6)$$

where M_e, B_e and F_d are the mass of both the variable inertia load and the piston, the viscous damping coefficient, the friction forces, [13,15].

The mathematical model of friction behavior in hydraulic actuators affected by:

- Presence of rubbers or composite materials O-rings and V-type seals in dynamic connections.
- The temperature influence in the higher thermal expansion coefficient sealing materials than metal elements.
- Contaminations of solid deposition on the piston.

The friction description is modeled by the stick-slip law as:

$$F_f = \begin{cases} F_{st}\, for\, 0 \le V_P < \Delta V_P \\ F_{sl}\, for\, V_P \ge \Delta V_P \end{cases} \quad (7)$$

where F_{sl}, F_{st} and V_p are the slip friction, the stick friction, and the piston speed. The time delay equation can describe the hoses and pipes dynamics as transport lag equation as:

$$H(s) = e^{-sT_d} \quad (8)$$

Transport delays are nearly 1st order lag.

$$e^{-sT_d} \cong \frac{1}{\left(T_d/2\right)s+1} \quad (9)$$

where T_d is the delay time. A LVDT transudes the piston displacement with gain of 0.1 m/V. The system physical parameters are illustrated in Appendixes A.

The control problem of an electrohydraulic system has many difficulties to deal with nonlinear models by classical linear PID control method. In the end, the strategy based on the PID-PSO strategy is suggested to optimally find the PID gains.

Controller Design

PID-controller has many popular applications in industry, and it has been stated in many scientific references that the PID-control has been utilized in electrohydraulic systems [7, 16-17]. According to the system dynamics nonlinearities, control requirements can be satisfied. The effective nonlinearities are in servo valves, equivalent load, and friction in the moving parts. The nonlinearities of hydraulic servo systems are in the forward loop as shown in Fig. 2. They are the main source of error in systems position and limit its performance.

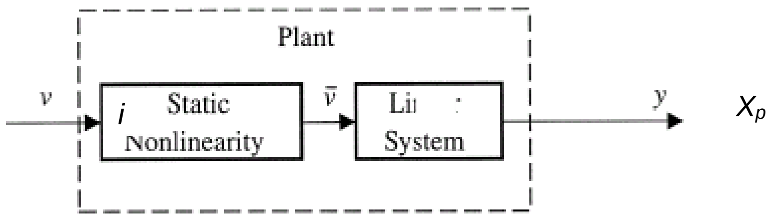

Fig. 2. Hydraulic operate system

As described, representation of the launching system would not be determined easily so it is difficult to be utilized in controller tuning. The popular way is making an approximation to the price nonlinear equation to be linearized and finally tune the PID control gains based on such as (Z-N) method, [18]. The Laplace PID form is illustrated as:

$$U(s) = \left[k_p + \frac{k_i}{s} + k_d s\right] E(s) \tag{10}$$

where k_p, k_i, k_d $U(s)$, and E(s) are the proportional, integral, derivative PID parameters, the control signal and position error, respectively. Based on the research of different scholars in the domain of electrohydraulic systems, [7, 16 -18] the following conclusions can be made:

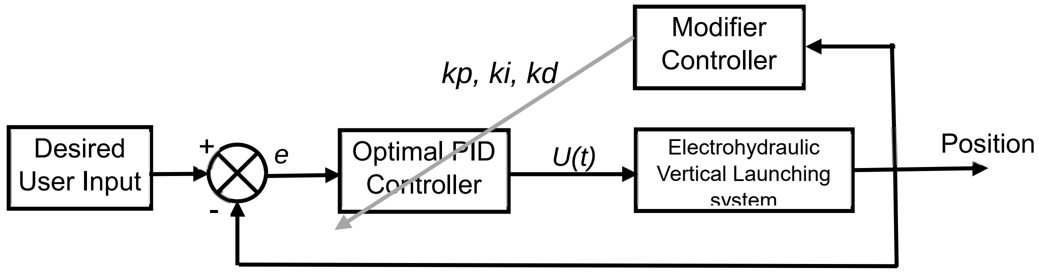

Fig. 3. Schematic of lunching system control system

Another strategy is designing PID control parameters based on optimization policy as in eq. (11). To grantee, closed loop response, the next cost equation should be deemed throughout the tuning of the PID control gains:

$$J(k_p, k_i, k_d) = ISE \tag{11}$$

where *ISE* is the integral square position error. Finally, it is implementation of the optimized PID control in the nonlinear model.

If the reference points are changeable, the dynamic characteristics will be different and price position regulation is needed, it is difficult to design a controller via classical methods. A PSO-PID controller to guide the nonlinear launcher is proposed as presented in Fig. 3. PSO strategy is one of artificial optimization techniques. It is a soft concept and can be coded easily, [19].

4.1 *Particle Swarm Optimization*

Evolution of soft computation algorithms is based on the expansion and spreading of pluralism inspections for the maximum fruitful productive position. A "swarm" is a clearly torn set of dynamic distributives that produce sheaf simultaneously whilst all solitary proceed in a disparate manner. It employs different particularistic that set up a swarm stirring on every side in the searching domain locking in the preferable configuration, [20]. Every particle is treated to modulate its moving using the flying experiments of the other particles. Each one presents its directions in the domain which belong to the lowest cost (minimum square error) that has been achieved currently. This estimate is p_{best}. More fit estimate that is routed by the PSO is the best estimate acquired so far by any mote in the neighbors. This estimate is g_{best}, [21-23].

The *PSO* concept embrace of the speed change of every particle in the direction of its *pbest* and the *gbest* location at any move. Every particle attempts to regulate its current place and pace as per the remoteness between its existent situation and *pbest*, and the remoteness between its existting situation and *gbest* as illustrated the next: all headway *n*, by the individual best position, *pbest*, and global best position, *gbest*, a recent speed for the i^{th} particle is renewed.

The speed is ranged to $[-v_{max}, +v_{max}]$. Variable speed during this range authorizes the i^{th} particle to find its thematic best placement, *pbest*, and global best placement, *gbest*. According to the recent speed, all particle modifies its *placement* as:

$$p_i(n) = p_i(n-1) + v_i(n) \tag{12}$$

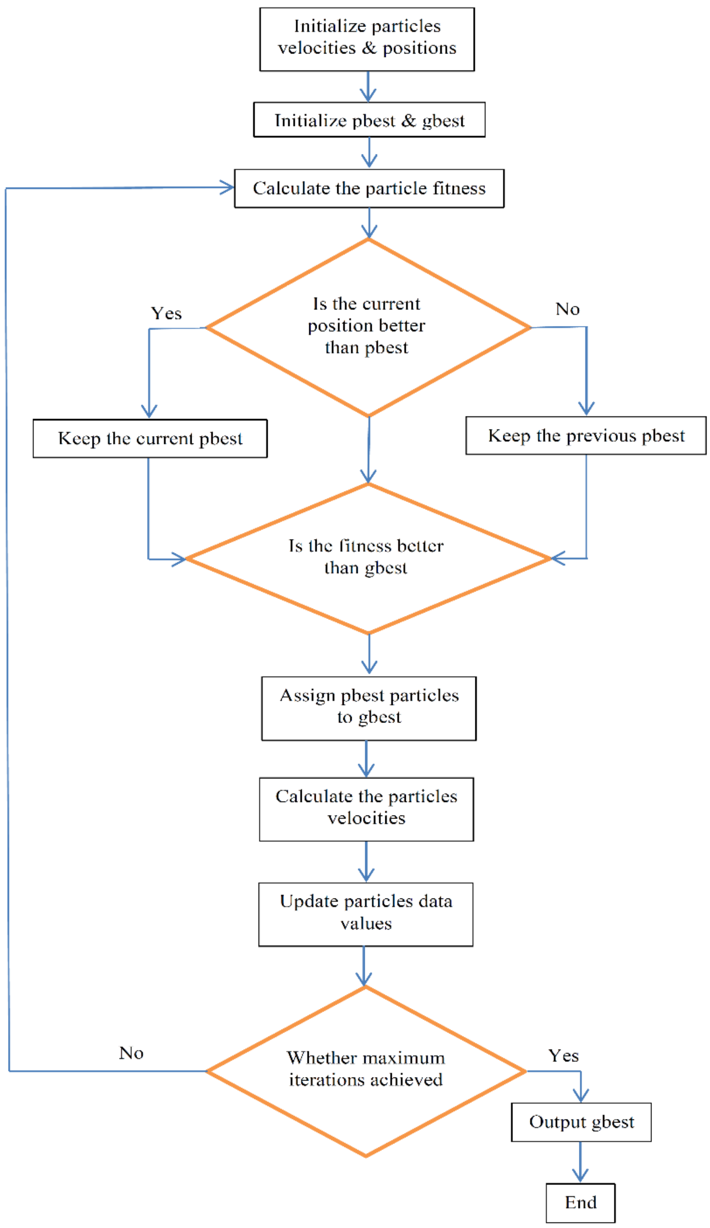

Fig. 4. Flow diagram illustrating the PSO algorithm.

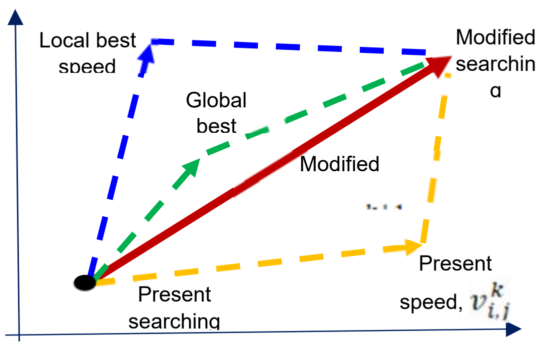

Fig. 5. Changes of searching point for PSO strategy

Figure 6 illustrates the particle position procession through the searching steps for k_p, k_i and k_d. Figure 7 shows the particle speed parade through the repetition execution. This process is kept while most cycle limit are obtainable as presented in Fig. 8.

PSO tries to find the totality of the preamble and outcome parameters in 50 dimensional spaces. The instruct of a particle is presented as:

$$p_i = [\sigma_{11} c_{11} \ \ldots \ldots \sigma_{15} c_{15} \ \sigma_{21} c_{21} \ \ldots \ldots \sigma_{nn} c_{nn} \ ] \qquad (13)$$

where the σ_{ij} and c_{ij} are the midst and aberration of the MFs parameters. The implied evaluation of particles is chaotically generated in the 1st creation.

The effective cross in carrying out *PSO* is to look for the best fitness function which computes the particle cost. When the tuning process is working with *PSO*, *MAE* is employed as a cost function, which is presented in Fig.6. The *MAE* for i^{th} particle is evaluated as:

$$MAE = E(j) = \frac{1}{N}\sum_{i=1}^{n}|e(i)| \qquad (14)$$

where $e\ (i)$, i^{th}, N and j are the error, sample, sample number and iteration number, respectively.
The population size is set to be *10* particles for limiting the evaluation time, the MAE is evaluated for every particle and *pbest* and *gbest* are calculated at definitive time (t_f). A particle speed is determined for all particle and a particle place as:

$$v_{i,j}^{k+1} = v_{i,j}^{k} + c_1 * r_1 * \left(pbest_{i,j}^{k} - x_{i,j}^{k}\right) + c_2 * r_2 \ * \left(gbest_{i,j}^{k} - x_{i,j}^{k}\right) x_{i,j}\,\hat{}\,(k+1)$$
$$= x_{i,j}\,\hat{}\,(k) + v_{i,j}\,\hat{}\,(k+1) \qquad (15)$$

The fitness distribution is computed by (15) as shown in Fig.9. From the fitness territory concerning generation number plots, the near-optimal values of feedback parameters can acquire within 10 generations, which equals to about 200 experiments. The optimal parameters are clearly assigned in every parameter domain. These figures also illustrate that it would not be easy to find optimal parameters by manual acquiring, by the disparity attitudes of the control gains.

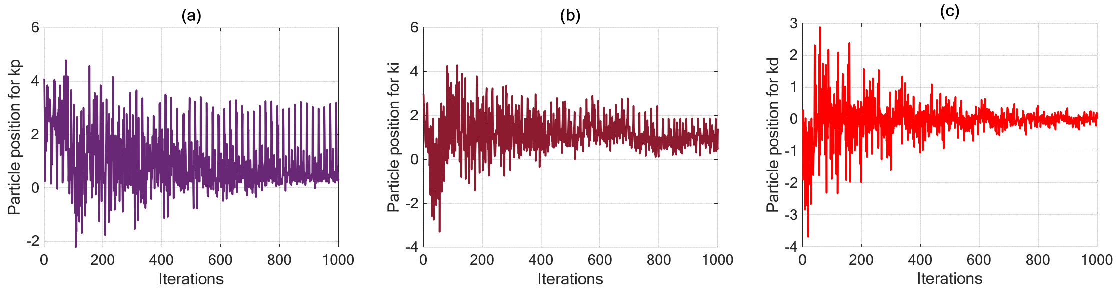

Fig. 6. Particle position trip over the iteration steps: (a) for k_p; (b) for k_i ; *(c)* k_d

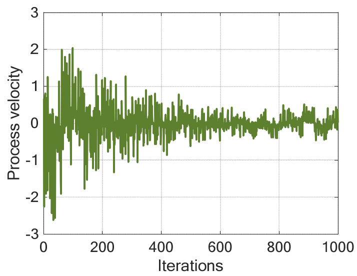

Fig. 7. Velocity trip over the iteration steps

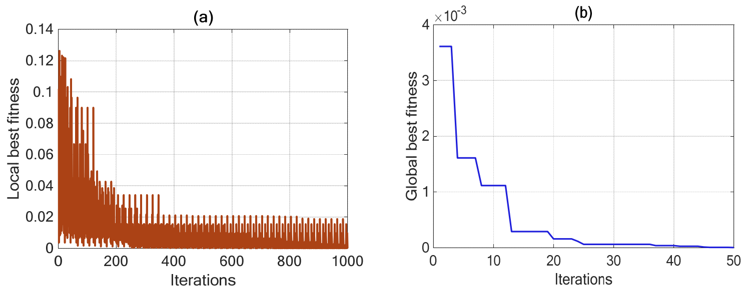

Fig. 8. Best fitness *MAE*: (a) local; (b) global

Simulation Verification

The nonlinear dynamics of the electrohydraulic model is not easy to outline precisely, so it is important to utilize the procedure of model identification to achieve the nearly accurate model. The control goal of the electrohydraulic launcher is to minimize the error $e(t)$ between the actual state $x(t)$ and the reference state $x_i(t)$ to be close to zero.

Simulink/MATLAB is utilized to realize the implementation of the PSO-PID control. The control system loop was computed utilizing numerical integration method with 0.001 s sampling time.

Figure 9 illustrates the step input response of the cylinder by the optimized control gains. The PID parameters were tuned by three different way one of them is done based on the linearized system model using the standard Z-N technique as a comparison base. The second technique is an optimally-tuning based on ISE function to improve the previous PID gains to meet the specifications of no overshoots and keep minimum steady state error. However, the rise time increased which means the system is slower than before. To minimize the error and overshoots in the response and improve its rise time, the PID controller gains are optimized using the soft computation strategy of PSO to reduce the MAE function as in eq. (15). The evaluated PID parameters are: $k_p = 2.250$, $k_i = 0.002$ and $k_d = 0.199$. it is noticed the value of k_i gain is less clear than the other gains because of the integration nature of the electrohydraulic system. Electrohydraulic system is, however, a type 1 system, and this means that there is already an integrator in the forward loop. Table 1 presents the parameters values for each PID controller. The settling time of the proposed cost function MAE is clearly less than that obtained by the other methods. One of the advantages of any control policy is its robustness with system modeling errors. Figure 8(b) shows the control effort obtained by the proposed control method. For dynamics alerting in the electrohydraulic systems, the cylinder position is changed, the responses also changed as illustrated in Fig. 10. The system keeps its stability during these variations. Table 2 illustrates the performance analysis for different PID controllers.

PID	**Z-N**	**ISE**	**PSO**
Kp	0.887	0.420	2.250
Ki	0.464	0.001	0.002
Kd	0.089	0.003	0.199

Table 1. Parameters value of PID controllers

PID	**Z-N**	**ISE**	**PSO**
Overshoot percentage	15.1	0.0	0.0
settling time (s)	8.08	3.21	1.407
Steady state error (m)	0.0001	0.0002	0.0001
Rise time (s)	0.447	1.7	0.754

Table 2. Performance analysis of PID controllers

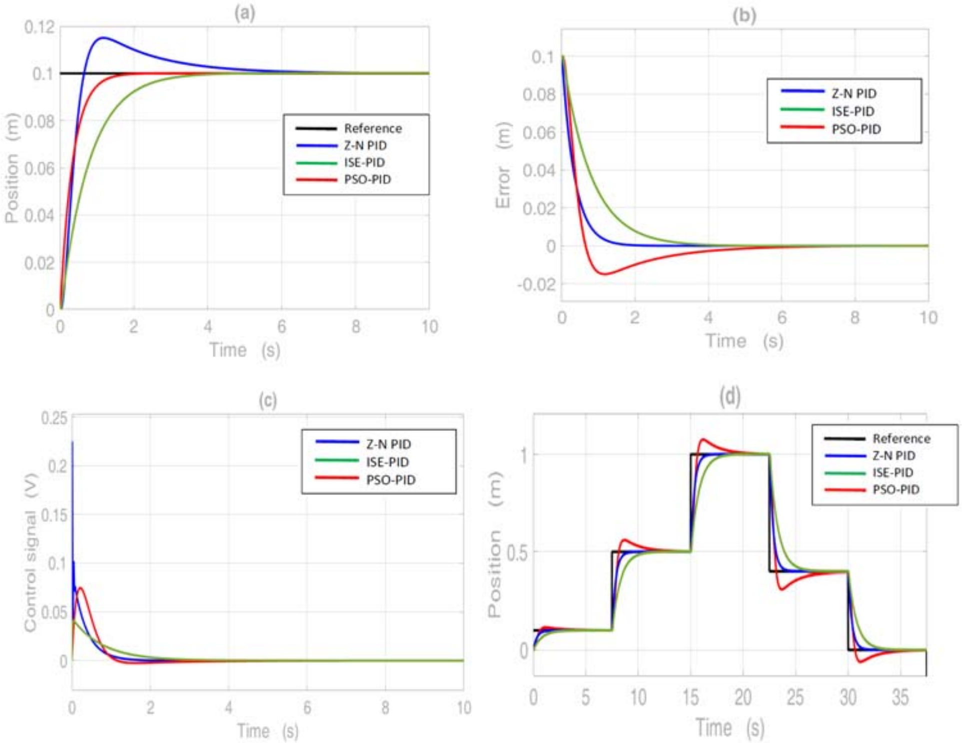

Fig. 9. Step tracking response with different control technique: (a)step refence of 0.1 m; (b) error signals; (c) control effort; (d) multi-step levels response.

Another good robustness test is the application of step disturbance to the system. The results assure the ability of the proposed controller for disturbance rejection. As presented in Fig. 10, step disturbance is applied where its amplitude is 10% of the reference values in the process. The steady-state error of three PID techniques is close to zero, the settling time of PSO-PID is 1.4s, tuned-PID and Ziegler-Nichols-PID is 3.7s and 4.6s, respectively. In Fig. 11, different reference signals have been also used with this system and nearly similar results are achieved each time. Another robustness test for proposed control techniques, the trajectory response curve of Fig. 12 shows sine wave response, the maximum error caused by PSO-PID is 0.008m, 003s and 0.017 by ISE-PID and Z-N-PID, respectively. The PSO-PID strategy gives more convenient and best performance in tracking control under stable conditions.

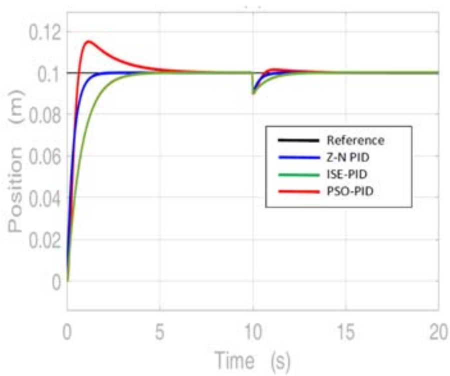

Fig. 10. Positioning error with external loading disturbances.

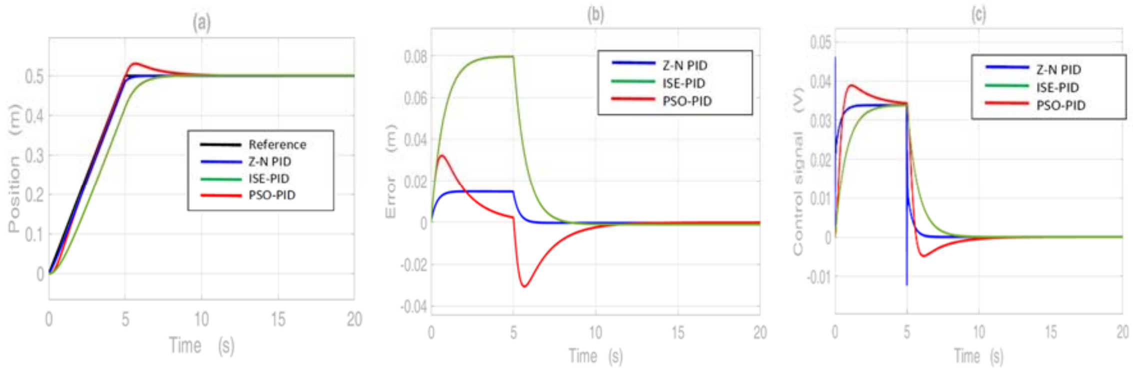

Fig. 11. Ramp tracking response with different control technique: (a) ramp refence of 0.1 m/s; (b) error signals; (c) control effort.

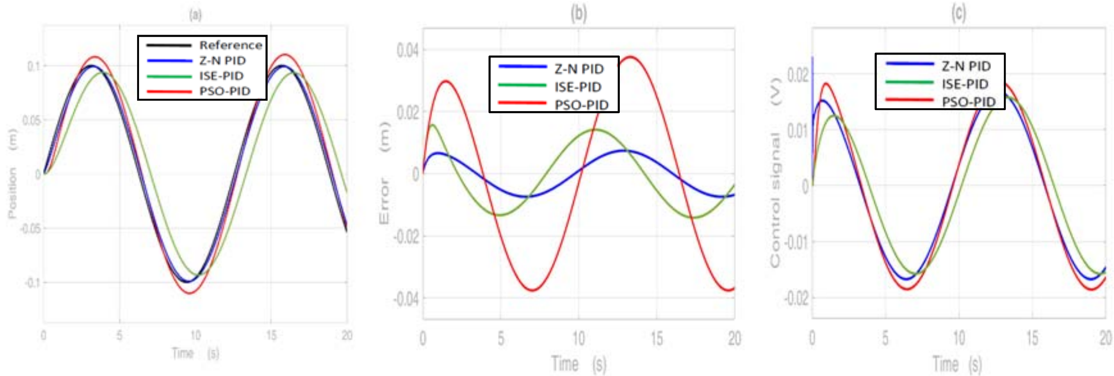

Fig. 12. Sin tracking response with different control technique: (a) sinusoidal tracking [0.1m amplitude, 0.5 rad/s frequency]; (b) sinusoidal tracking error signals; (c) control effort curves.

Conclusion

The current study investigated the tracking problem of controlling electrohydraulic vertical launching systems by PSO as a search technique with limited system information as the defined cost function. Classical PID control design cannot release the required specifications of the desired response with reference input changes and external disturbance. The optimized PSO-PID control is designed to manipulate with the trajectory tracking problem regardless of its nonlinearities. It has robust response due to system reference trajectory changes and external disturbances.

It is illustrated that the PSO-PID improved the response characters of the electrohydraulic vertical launching system to obtain less settling time with removing overshoot and closely zero steady state error. For future works, one can implement the proposed control strategy experimentally to confer its excellence to both modeling imprecision and external disturbances.

Conflicts of Interest: The authors declare that they have no conflicts of interest.

Appendix A. System Parameters

Parameters	Symbols	Values	Units
Servo valve torque gain	k_v	$3.75\text{x}10^{-4}$	m/mA
Servo valve natural frequency	ω_v	1068	rad/sec
Servo valve damping ratio of	ζ_v	0.5	-
Rated flow rate	Q_v	$0.333\text{x}10^{-3}$	m^3/sec
Total leakage coefficient	C_t	$1.0\text{x}10^{-10}$	m^5 /(sec N)
Supply pressure.	P_s	140	bar
Oil bulk module.	B	$7.0\text{x}10^{8}$	Pa
Oil mass density.	ρ	900.0	kg /m
Mass load	M_e	100	kg
Diam. of rod	d_{rod}	0.12	m
Diam. of piston	d_{piston}	0.14	m
Max. stroke of actuator	X_p	1.2	m
Length of pipeline and hoses	L	5.0	m
Actuator Coulomb friction force	F_{fc}	200	N

Table A.1. System Parameters

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Authors Profile

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