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# PID Controller Optimization by GA and Its Performances on the Electro-hydraulic Servo Control System

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**Abstract**

A proportional integral derivative (PID) controller is designed and attached to electro-hydraulic servo actuator system (EHSAS) to control the angular position of the rotary actuator which control the movable surface of space vehicles. The PID gain parameters are optimized by the genetic algorithm (GA). The controller is verified on the new state-space model of servo-valves attached to the physical rotary actuator by SIMULINK program. The controller and the state-space model are verified experimentally. Simulation and experimental results verify the effectiveness of the PID controller adaptive by GA to control the angular position of the rotary actuator as compared with the classical PID controller and the compensator controller.

*Keywords:* PID controller; electro-hydraulic servo control system; genetic controller; GA

## 1 Introduction

Proportional integral derivative (PID) controllers are still the most popular ones in the processing industries. They are simple in structure, reliable in operation and robust in performances. One key factor for their success is that they act in the processes under control in a manner closely similar to human's natural responses to outside stimuli, that is the combined effects of spontaneity (proportional action), post training (integral action) and projection into future (derivative action).

The objectives of the PID controller are to control and improve the angular position response (the output) of the hydraulic rotary actuator. The PID controller implements the state-space model of the electro-hydraulic servo actuator system (EHSAS) to control the movable surface of space vehicle. The simulation, performed by SIMULINK program to check the effects of the PID controller on the EHSAS, uses the intelligent control, genetic algorithm (GA), to optimize the controller gain parameters $K_p$, $K_i$, and $K_d$ on line. The experiment was carried out to verify the new state-space model which was used to design the controllers.

## 2 System Description

The system consisted of a rotary actuator and a two-stage electro-hydraulic servo-valve with mechanical feedbacks as shown in Fig.1.

**System Operation** If an electrical control signal is applied to the coils (first stage), the resultant torque is proportional to the applied current. This torque causes the flapper plate to move and the throttle area of two regulating jets (nozzles) to change. For example, flapper's moving to the right increases the area of the left jet and decreases that of the right jet, and thus the pressure $P_1$ decreases while $P_2$ increases. The pressure difference ($\Delta P = P_2 - P_1$) increases in proportion to the flapper displacement. This pressure difference is the valve

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output. The spool valve (second stage) is driven hydraulically by the pressure difference produced by the flapper valve. Thus the spool is displaced to the left connecting the port $A$ of the rotary actuator to the high pressure and port $B$ to the low pressure which rotate the wheel counterclockwise via the rightward displacement of the piston. The angular position is proportional to the input current. The maximum angular position for the rotary actuator is ±10°.

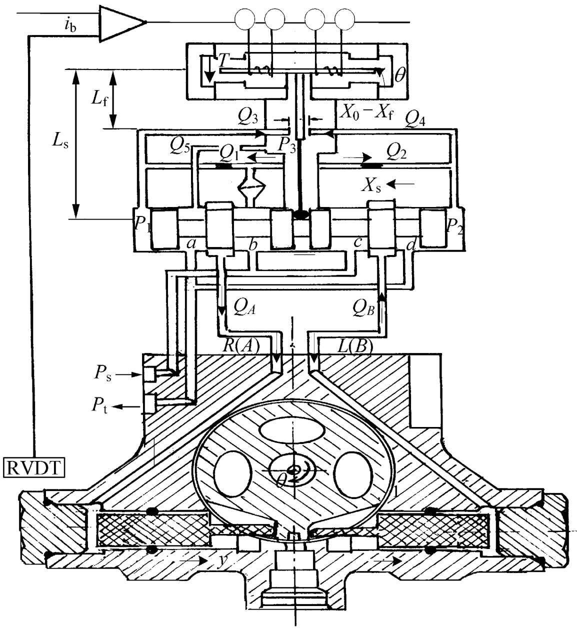


Fig.1 Cross-section of the rotary actuator and the two-stage servo-valve with mechanical feed-backs.

## 3 Mathematical Analysis and Model of Electro-hydraulic Servo-system

The design of stabilizing controllers used in nonlinear electro-hydraulic servo-systems has been extensively studied in recent years. Fig.1 shows a laboratory experimental electro-hydraulic servo-system driven by a hydraulic motor (rotary actuator) controlled by a servo-valve. In the system, the aerodynamic load acting on the fins' surfaces represents the outside load. The objective of the controller is to keep the angular position of the motor following a desired trajectory as precisely as possible. The entire system can be described by following equations.

### 3.1 Mathematical description of the motor torque

The torque of electromagnetic motors depends mainly on the coil currents and armature displacements. Neglecting the effects of magnetic hysteresis and the small term $K_\vartheta \vartheta$, the following expression that describes the motor torque mathematically is given by Ref.[1].

$$T = K_i i_e \tag{1}$$

The equation of motion of the armature includes the moment of inertia, damping coefficient, stiffness of flexural tubes, torque due to flapper displacements, limiter torque due to pressure forces and feedback torque.

$$T = J\frac{d^2\vartheta}{d^2 t} + f_\vartheta \frac{d\vartheta}{dt} + K_T\vartheta + T_L + T_P + T_f \tag{2}$$

Suppose $|X_f| < X_{fL}$ then $T_L = 0$. The motor torque can be represented by

$$K_i i_e = J\frac{d^2\vartheta}{d^2 t} + f_\vartheta \frac{d\vartheta}{dt} + K_T\vartheta + A_n P_L L_f + (\vartheta L_s + X)K_s L_s \tag{3}$$

### 3.2 Flow rate equations of flapper valve

In the steady state, the ports $A$ and $B$ are closed by the stationary spool, and then the steady state flow rates in the valve are $Q_1$, $Q_2$, $Q_3$, $Q_4$, and $Q_5$[2]. Assuming the supplied pressure is constant[2-3], the valve load pressure can be derived from the continuity equation:

$$\dot{P}_L = \frac{2\beta}{V_0} C_{12}\sqrt{\frac{P_s}{2}}\left[(1+\frac{X_f}{X_0})\sqrt{1-\frac{P_L}{P_s}} - \sqrt{1+\frac{P_L}{P_s}}\right] - \frac{2\beta A_s}{V_0}\frac{dX}{dt} \tag{4}$$

### 3.3 Mathematical description of the spool

The motion of the spool can be described by applying Newton's second law.

$$m_s \frac{d^2 X}{dt} = A_s P - f_s \frac{dX}{dt} - (\vartheta L_s + X)K_s \tag{5}$$

Neglecting the transmission lines connecting the valve to the rotary actuator, the flow rates are $Q_a$, $Q_b$, $Q_c$, and $Q_d$[2]. Assuming the supplied pressure is