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Motion control of an aircraft electro-hydraulic servo actuator

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Abstract. The use of the Fly-by-Wire (FBW) systems for flight control of aeronautical vehicles has increased steadily in the recent years. Such systems use electro-hydraulic servo-actuators to position the manoeuvring surfaces of airplanes or movable thrust vector controls of space vehicles. The marriage between electronics and hydraulic power systems has led to many powerful and precise control systems, saving much energy and money. This paper deals with the motion control of aircraft integrated electro-hydraulic servo-actuator (ISA) via development of a detailed nonlinear mathematical model and a computer simulation program using MATLAB/SIMULINK package. The ISA mainly consists of two separate active hydraulic power systems, used to supply the ISA with the required power. The studied ISA incorporates two electro-hydraulic servo-valves, a twin-symmetrical-hydraulic actuating cylinder, and a smart design of built in directional control valves with a feedback system. The output linear motion of the actuating cylinder of the ISA is presented and controlled by obtaining the transient response of the ISA. A classical PID controller is designed and tuned by Zeigler-Nichols method according to the Integral Square Error (ISE) criteria to minimize the difference between the obtained system output feedback and the desired set input by adjusting the system control parameters. The system stability and precision requirements are insured by applying the tuned PID controller.

1. Introduction

Fly-By-Wire (FBW) control system technology has greatly enhanced the flexibility of the parameters in airplanes design. This control systems provide a new interface for controlling airplane using digital technology to replace mechanical components. This system has enabled the development of a variety of innovative airplanes designs from aerodynamically unstable airplane to autonomous air vehicles. The digital computer-based system was able to provide improved manoeuvrability and weight reduction in both military and commercial applications at the cost of reduced reliability [1]. This concept is used in the electro-hydraulic servo systems, which have the functions of electric and hydraulic systems.
The electro-hydraulic servo systems play an important role in industrial applications, especially in-flight simulators and actuating systems of the aircraft. The main reason of using hydraulic systems in many applications is that, they can provide a high torque and high force. In addition, the electric systems

produce a high controllability and precision actuating motion. So, these systems are used to control the position of the maneuvering surfaces of the airplanes.

The electro-hydraulic servo system refers to the control system which combined two control modes of electrical and hydraulic. Transmitting the signal by use of electronic and electric parts, driving the load with hydraulic transmission in the electrohydraulic servo control system. So it can use an electrical system for its efficiency and suitability, and use of hydraulic system for its rapid response speed, big load stiffness and accurate positioning characteristics to make the whole system more adaptable [2]. Generally, the main objective of controlling of the aircraft means to direct its orientation (attitudes) during its flight path according to the command motions with respect to the inertial or reference frame. To achieve this goal, controller is used. The most common controller is the proportional-integral-derivative (PID) controller. PID controller is the most common controller class of feedback systems. It is widely used in industrial control fields as stated by [3] and [4]. It has been used to minimize the difference between the measured system output feedback and the desired input by adjusting the system control parameters [5].

Many techniques have been developed for optimally tuning the controller's parameters. These range from trial and error [6], root locus [7] and artificial intelligence techniques [8]. The performance indices, which are usually used in the optimization are Integral Absolute Error (IAE), Integral Square Error (ISE) and Integral Time Absolute Error (ITAE) [9].

PID controller is one of the earliest developed control strategies. The design algorithm and control structure involved in PID controller are simple, and are suitable for engineering application background. Also, PID control scheme does not require accurate mathematical model of controlled object, and the control effect of PID control is commonly satisfactory. So PID controller in industry is one of the most widely used control strategies, and is more successful. According to statistics, PID controller applied to more than 90% in the industrial control of the controller [10].

PID controller with proportional, integral, and differential actions of three parameters need to design and tune, the sampling period of system parameters need to choose. Therefore, it is very difficult to design the ideal parameters. Traditional methods are usually used algorithms, such as Ziegler-Nichols tuning methods, Chien-Hrones-Reswick tuning technique, the optimal parameters for empirical formula and the genetic algorithm technique [11].

In this work, the output linear motion of the actuating cylinder of the ISA is presented and controlled by obtaining the transient response of the ISA. A classical PID controller is designed and tuned by Zeigler-Nichols method according to the Integral Square Error (ISE) criteria to minimize the difference between the obtained system output feedback and the desired set input by adjusting the system control parameters.

2. Mathematical modelling

The studied ISA incorporates two electro-hydraulic servo-valves (EHSV) and a smart design of four built-in direct operated directional control valves (DCV) controlled by four electrical solenoids and a switching DCV works as ON-OFF switch of the EHSV and a twin-symmetrical-hydraulic actuating cylinder with a feedback system as shown in figure 1.
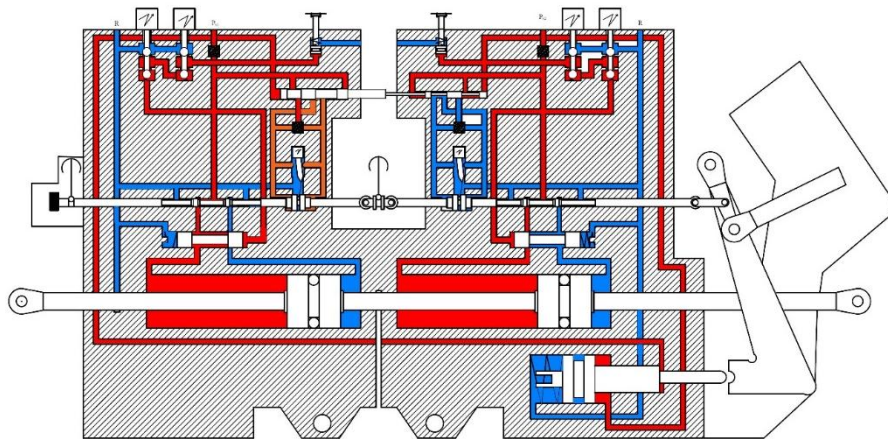
Figure 1. Integrated servo actuator scheme

The ISA is divided into two modules; first, the servo-valve (SV) module. Second, the actuating cylinder module. The mathematical relations that describe the system are deduced as follow:

2.1. Servo valve module
The detailed non-linear mathematical model that describes this module is given clearly in my paper [12], which includes the mathematical model, the utilized computer simulation program and a discussion about the different failure modes of the system components and their effect on the transient response of the SV.

2.2. Actuating cylinder module
This module consists of two typical interconnection valves and a twin-symmetrical-hydraulic actuating cylinder with a feedback system as shown in figure 2. The detailed mathematical modelling of this module is deduced as follows:
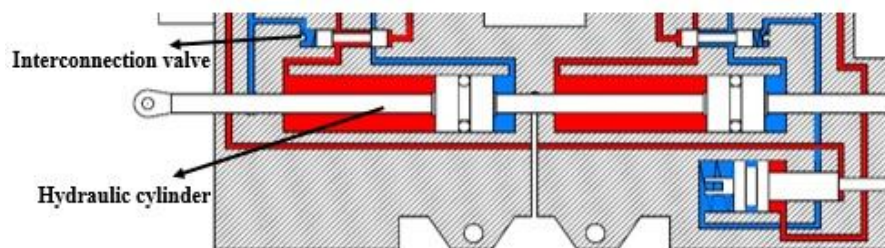

Figure 2. Actuating cylinder module

2.2.1. Interconnection valve model. This valve is shown in figure 3. The detailed mathematical model is deduced as follow:
  • *Equation of motion*
The motion of this valve is described by this equation of motion. This valve works as a by-pass valve as shown in figure 3.

$$P_{s1}A_{Pz} + m_z\ddot{z} + f_z\dot{z} + K_{sp}(z - z_0) + F_{sR} = 0 \qquad (1)$$
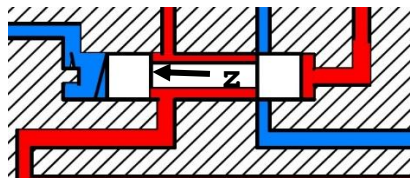
Figure 3. Interconnection valve scheme

where

P_{s1}	Supply pressure of the main system of ISA, 300 bar
A_{Pz}	Piston area of the by-pass valve, m^2
m_z	Mass of the by-pass valve, 0.02 kg
f_z	Interconnection valve damping coefficient, 50 Ns/m
K_{sp}	Spring stiffness, 15000 N/m
Z	Interconnection valve displacement, m
z_0	Spring pre-compression distance, 0.003 m
F_{sR}	Seat reaction forces of the interconnection valve, N

- *Seat reaction force*

$$F_{SR} = \begin{cases} 0 & z > 0 \\ \\ f_{sm} * \dot{z} + K_{sm} * (z - z_o) & z \leq 0 \end{cases} \tag{2}$$

where

K_{sm}	Seat material stiffness, $1 * 10^7$ N/m
f_{sm}	Seat material structural damping coefficient, 5000 Ns/m

2.2.2. Actuating cylinder of motion. The actuating cylinder is a cylinder of twin symmetrical type as shown in figure 4. Its motion is performed against the pressure, inertia, viscous, and the seat reaction forces which described by the following relations:
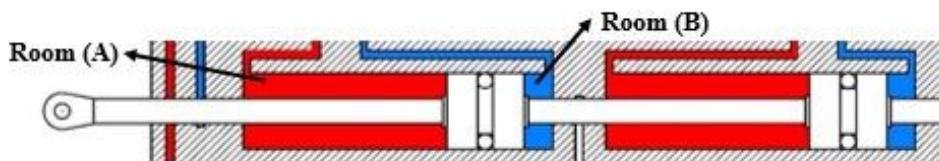

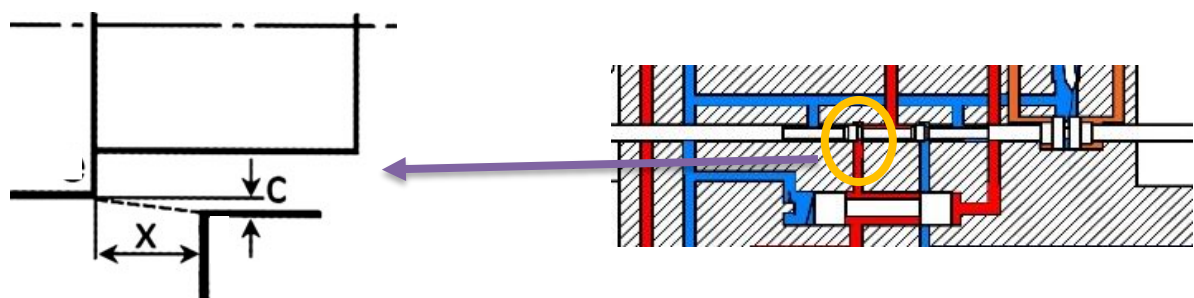**Figure 4.** Actuating cylinder scheme

- Restriction areas

Figure 5. Restriction areas

4

The restriction areas through the spool valve are described as follows (figure 5):

$$\left.\begin{array}{l} A_a = A_c = \omega\, c \\[2mm] A_b = A_d = \omega\sqrt{x^2 + c^2} \end{array}\right\} \qquad for\ x \geq 0 \tag{3}$$

$$\left.\begin{array}{l} A_a = A_c = \omega\sqrt{x^2 + c^2} \\[2mm] A_b = A_d = \omega\, c \end{array}\right\} \qquad for\ x < 0 \tag{4}$$

$$A_{BP} = \omega * |d_i - z| \tag{5}$$

where

$A_{a,b,c,d}$	Throttling areas of the port a, b, c and d, m2
ω	Spool port width, 0.002 m
c	Spool radial clearance
x	Spool displacement, m, [12]
A_{Bp}	By-pass area, m2
d_i	Transmission line diameter, 0.025 m

• *Flow rates through the spool valve*
The rates through the restriction areas of the spool valve are described as follow:

$$Q_a = C_d A_a \sqrt{\frac{2(P_A - P_t)}{\rho}} \tag{6}$$

$$Q_b = C_d A_b \sqrt{\frac{2(P_{s1} - P_A)}{\rho}} \tag{7}$$

$$Q_c = C_d\, A_c \sqrt{\frac{2(P_{s1} - P_B)}{\rho}} \tag{8}$$

$$Q_d = C_d\, A_d \sqrt{\frac{2(P_B - P_t)}{\rho}} \tag{9}$$

$$Q_a = C_d A_a \sqrt{\frac{2(P_A - P_t)}{\rho}} \tag{10}$$

$$Q_{BP} = C_d\, A_{BP} \sqrt{\frac{2(P_A - P_B)}{\rho}} \tag{11}$$

where

Q	Flow rate, kg/m^3
C_d	Discharge coefficient, 0.611
ρ	Oil density, 900 Kg/m^3
P_t	Return tank pressure, 2 bar

• *Continuity equations applied to the cylinder chambers*
The actuating cylinder has two chambers, (A and B). Neglecting the internal and external leakage.
The continuity equations of each chamber are described as follows:

$$Q_b - Q_a - Q_{BP} - A_p\dot{y} = \frac{V_c + A_p\, y}{B} \frac{dP_A}{dt} \tag{12}$$

$$Q_d - Q_c + Q_{BP} + A_p\dot{y} = \frac{V_c - A_p\, y}{B} \frac{dP_B}{dt} \tag{13}$$

where

A_P	Cylinder piston area, $12.5*10^{-4}$ m^2
V_c	Initial volume of the cylinder chamber, $100*10^{-6}$ m^3
B	Bulk modulus of oil, 1.9 Gpa

- *Seat reaction force*

The piston displacement in the closure direction is limited mechanically. When reaching its seat, a seat reaction force takes place due to the action of the structural damping of the seat material. There are four components of seat reaction force effects on the cylinder piston:

$$F_{SRC} = F_{cR1} + F_{cR2} - F_{cL1} - F_{cL2} \tag{14}$$

$$F_{cR1} = F_{cR2} = \begin{cases} f_{sm}\dot{y} + K_{sm}(y - y_{ciR}) & y \geq y_{ciR} \\ \\ 0 & y < y_{ciR} \end{cases} \tag{15a}$$

$$F_{cL1} = F_{cL2} = \begin{cases} -f_{sm}\dot{y} - K_{sm}(y + y_{ciL}) & (y + y_{ciL}) \leq 0 \\ \\ 0 & (y + y_{ciL}) > 0 \end{cases} \tag{14b}$$

where

y	Actuating hydraulic cylinder displacement, m
y_{ciR}	Right initial position of the cylinder piston, 0.04 m
y_{ciL}	Left initial position of the cylinder piston, 0.04 m
F_{SRC}	Seat reaction force of the cylinder, N

- *Equation of motion*

The motion of the piston under the action of pressure, viscous friction, inertia, and external forces is described by the following equation, assuming unloaded piston:

$$(P_{AL} - P_{BL} + P_{AR} - P_{BR})A_P = m_P\ddot{y} + f_P\dot{y} + K_p y + F_{SRC} \tag{15}$$

where

m_p	Piston mass, 10 kg
f_P	Friction coefficient on piston, 1000 Ns/m
K_p	Piston loading coefficient, 0 N/m

2.2.3. Feedback system. The piston displacement is picked up by a displacement transducer and feedback to the electronic controller, which generates the corresponding error signal. The feedback loop can be described by the following equations:

$$i_e = i_c - i_f \tag{16}$$

$$i_f = K_{FB}\, y \tag{17}$$

where

i_e	Torque motor input current, A
i_c	Control current, A

i_f	Feedback current, A
K_{FB}	Feedback gain, 0.25 A/m

3. Use of PID controller

The PID controller consists of Proportional, Integral and Derivative action. PID controller algorithm is used in feedback loops and can be implemented in many forms. Classical PID controller has good static performance, simple designing technique, reliability and robustness [13]. Design and tuning of PID controllers have been large research areas ever since Ziegler and Nichols presented their methods in 1942. A standard PID controller is also known as "three-term" controller. The utilized PID controller is characterized by three parameters K_p, K_i and K_d and the main task is to find the optimal values for these parameters according to the Integral Square Error (ISE) criteria, figure 6.
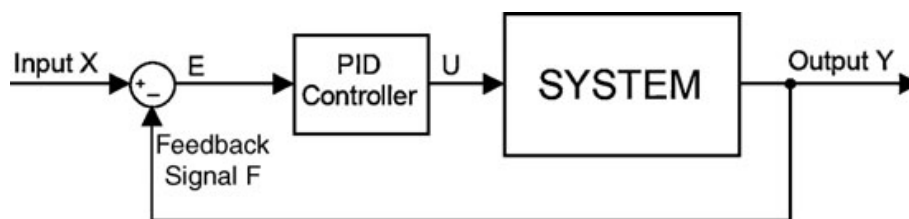

Figure 6. Connection of the PID controller in the closed loop system

3.1. PID controller design

The PID controller parameters are the proportional gain K_p, the integral gain K_i and the derivative gain K_d . The most well-known methods for estimating the PID parameters are those developed by Ziegler and Nichols method according to Table 1. They have had a major influence on the practice of the PID control for more than half a century. The general transfer function of the PID controller is:

$$G_C = K_P + \frac{K_i}{S} + K_d S \qquad (18)$$

Table 1. Ziegler-Nichols first method rules for estimating the PID controller parameters [14]

Controller	Symbol	Gain K	T_i [s]	T_d [s]
Proportional	P	T / L	-	-
Proportional Integral	PI	0.9 T / L	L / 0.3	-
Proportional Integral Derivative	PID	1.2 T / L	2 L	0.5 L

3.2. Tuning of PID controller

Usually, the design of PID controller, based upon the first estimated parameters does not give satisfactory results. Therefore, a tuning process is recommended for each of the three PID controller parameters K_p, K_i and K_d. This tuning process aims to finding the optimum values of these parameters which minimize the performance index according to ISE criteria. The PID with tuned parameters gives usually quite acceptable results.

3.3. Performance index

The main objective is to tune the gains K_p, K_i and K_d that can minimize the performance index according to ISE criteria. The equation of the performance index that used in this work is illustrated as follows [15]:

$$ISE = \int_0^\infty e(t)^2 \, dt \qquad (19)$$

$$e(t) = y_{ss} - y(t) \qquad (20)$$

where y_{ss} the required steady state value, and $y(t)$ is the actual response.

4. Simulation results

The simulation program is built considering numerical values of a typical electro-hydraulic servo actuator (EHSA) which is stated in [16]. Equations from (1) to (19) describe the dynamic performance and the transient response of the ISA system. These equations represent a detailed non-linear mathematical modelling of the system that used to develop a computer simulation program using MATLAB/SIMULINK package. This transient response is obtained by the simulation program to a 10 mA step input current with a step time of 2 second. The hydraulic cylinder response is over-damped with a steady state value of 4 cm, a settling time of 9.584 second, and a rise time of 3.173 second, overshoot of 0.265 % and with steady state error (performance index) of 0.005867 %. The transient response and the block diagram of the ISA are shown in figure 7.
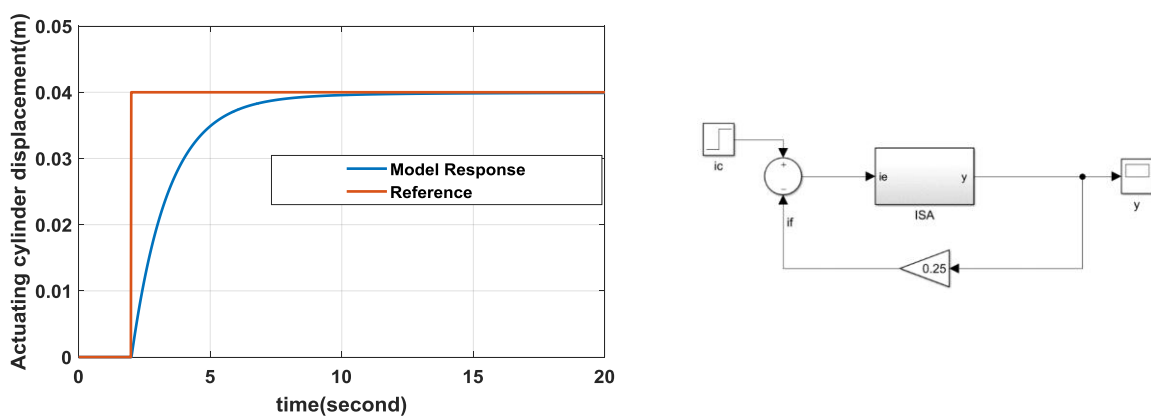

Figure 7. Step response and the block diagram of the ISA without controller

In order to improve the system performance, a classical PID controller should be used. The first estimated values of the PID controller is calculated using Zeigler-Nichols method. These parameters are $K_p = 3.6$, $K_i = 6$ and $K_d = 0.54$. Figure 8 shows the transient response and the block diagram of the ISA with the first estimated PID control parameters.
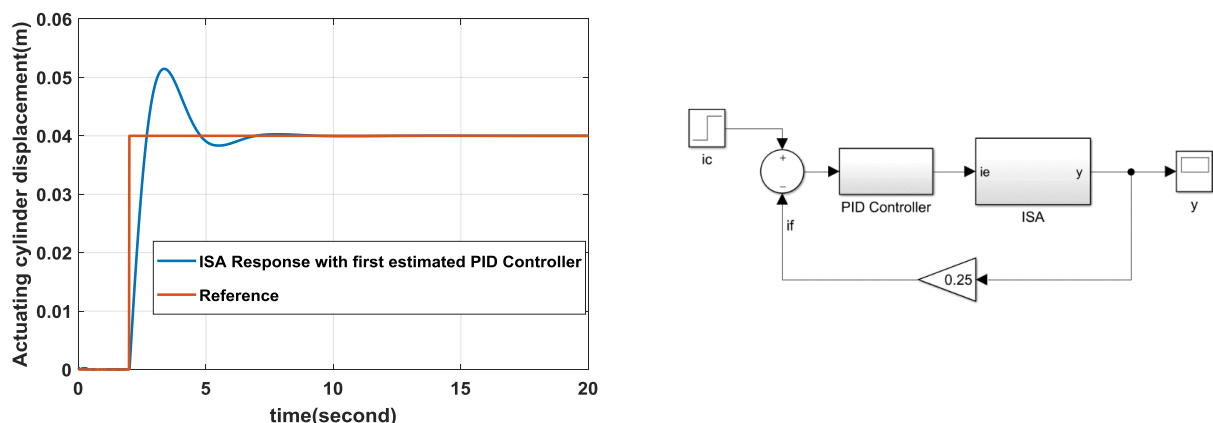

Figure 8. Step response and the block diagram of the ISA with the PID controller

The tuning technique is applied to the system according to the ISE criteria for obtaining the minimum steady state error between the output feedback and the input current signal. The tuned PID control parameters are $K_p = 5$, $K_i = 0.15$ and $K_d = 0.3$. After applying the tuned PID controller in the system,

the performance of the system is improved. Figure 9 shows the transient response and the block diagram of the ISA with the tuned PID control parameters.
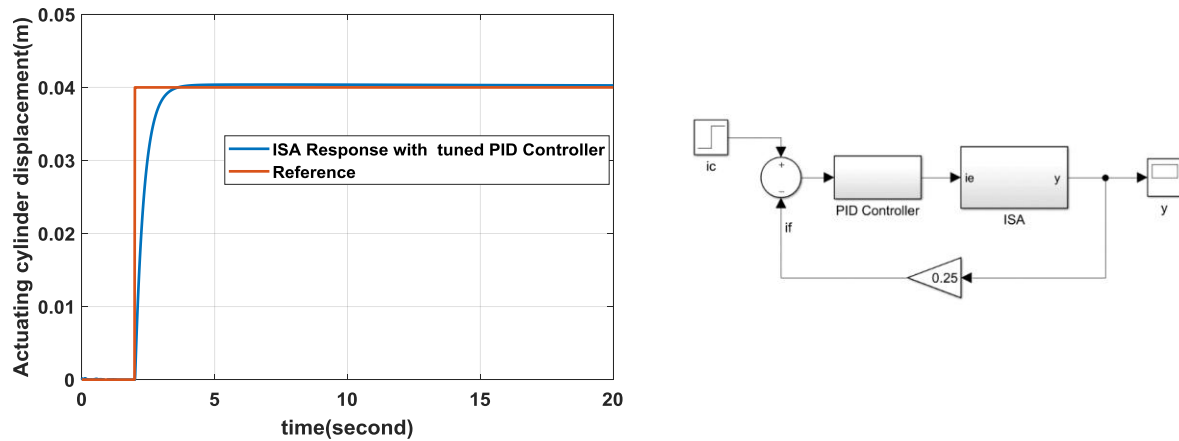

Figure 9. Step response and the block diagram of the ISA with the tuned PID controller

The transient response parameters of the system such that the settling time, the rise time, the overshoot percentage, and the performance index are obtained. These results are collected in Table 3 and Figure 10.

Table 2. Simulation results and the ISA transient response parameters

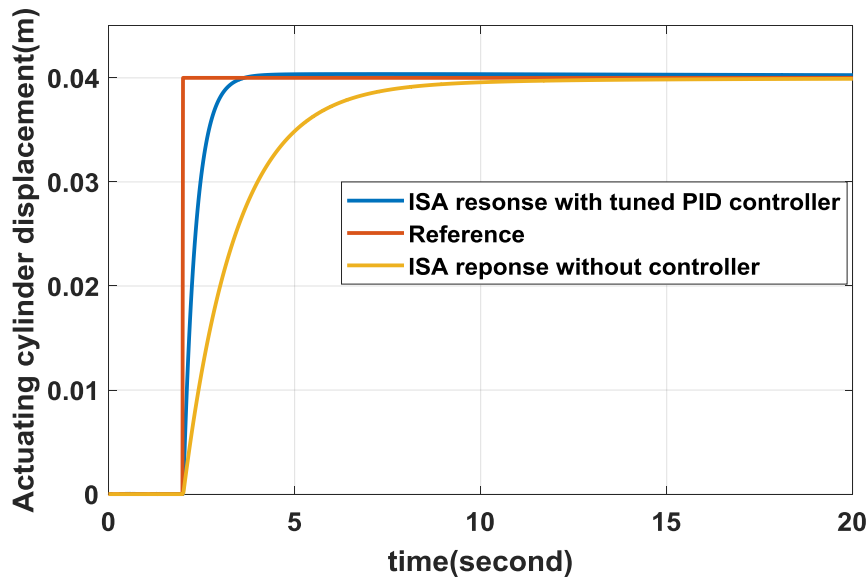| Control System	PID parameters			Settling Time (second)	Rise Time (second)	Overshoot %	Performance index (ISE)
	K_p	K_i	K_d				
ISA Model Response	0	0	0	9.584	3.173	0.265 %	0.005867
ISA response with first estimated PID control parameters	3.6	6	0.54	7.572	0.5312	24.37 %	0.005399
ISA response with tuned PID control parameters	5	0.15	0.3	6.3411	1.056	0.505 %	0.005314

Figure 10. Step response of the ISA with and without PID controller

5. Conclusion

The proportional integral derivative controller (PID) is the most common form of feedback. It was an essential element of early governors. Today, the majority of industrial systems use the PID or PI type controllers. The PID controllers have survived many changes in technology, from mechanics and pneumatics to microprocessors via electronic tubes, transistors and integrated circuits.

The major work of this paper contains three aspects. First, a non-linear mathematical model and a computer simulation program are developed for the hydraulic actuating cylinder module with its feedback system. Second, a classical PID controller is designed and tuned using Ziegler and Nichols method according to the ISE criteria. Third, the effect of adding PID controller to the system was presented. The transient response of the ISA system is obtained and discussed. The simulation results showed that the ISA presented has an acceptable transient response. The required output such that the main driven spool valve displacement and the actuating hydraulic cylinder displacement are insured.

Adding the PID controller to the system improves the transient response parameters as the following results:

- Reducing the rise time by 66.7 %
- Reducing the settling time by 33.8 %
- Reducing the steady state error by 9.4 %
- With 0.505 % overshoot of the system

This study shows that the system with PID controller has better performance and better transient parameters than the system without PID controller. The ISA control system with PID controller converge rapidly to the required steady state value, with very small steady state error.

In future work, different controllers such as Fuzzy logic controller (FLC) or Genetic Algorithm (GA) could be used to validate the system. It is necessary to develop a suitable hardware system to verify the simulation results in this paper via experimental setup.

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