Hardwer-in-the-Loop Testing of an Electrohydraulic Servo System

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Abstract: Hardware-in-the-Loop (HIL) test is an effective platform for developing and testing complex real-time systems and allows the engineer to test their control units with greater test coverage compared to physical testing alone. Expensive and unique electrohydraulic servo systems are hard to test but HIL systems provides the complexity of the plant under control using mathematical representation of all related dynamic systems. The servo system consist a double ended symmetrical double-acting hydraulic cylinder loaded with a mechanical system, servo valve, a hydraulic power supply with pump, accumulator and relief valve. The real-time servo system HIL simulator produces real-world electrical interactions for the test of the system. In this paper HIL test was used to examine the dynamic behavior of an electrohydraulic servo system while internal leakage changes at cylinder's piston seal.

Keywords: HIL test, hydraulic servo system, modelling, simulation

1 Introduction

Hardware-in-the-Loop (HIL) simulation is a technique that is used increasingly in the development and test of complex real-time systems so the purpose of HIL simulation is to provide an effective platform for developing and testing real-time systems. Hardware-In-the-Loop is a form of real-time simulation. Hardware-In-the-Loop differs from pure real-time simulation by the addition of a real component in the loop and simulation is achieving a highly realistic simulation of equipment in an operational virtual environment. HIL simulation provides an effective platform by adding the complexity of the plant under control to the test platform.

The examined and tested plant is an electrohydraulic servo system which is driven by a Texas Instruments DSP control unit.

2 Hardware-in-the-Loop Test

The current industry definition of a Hardware-In-the-Loop system is shown in Figure 1. It shows that the plant is simulated and the control unit is real. The purpose of a Hardware-In-the-Loop system is to provide all of the electrical stimuli needed and a typical HIL system includes sensors, actuators to receive data from the control system, actuators to send data, a controller to process data, a human-machine interface (HMI) and a development post-simulation analysis platform. The value of each electrically emulated sensor is controlled by the plant simulation and is read by the embedded system under test. Likewise, the embedded system under test implements its control algorithms by outputting actuator control signals. Changes in the control signals result in changes to variable values in the plant simulation.

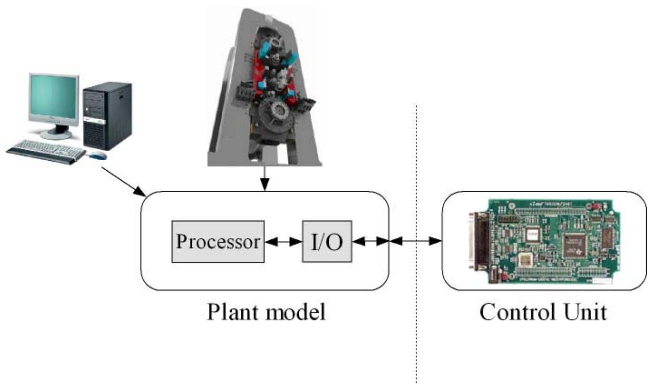

Figure 1
Hardware-in-the-Loop System

Advantages of HIL systems:

- enable testing the hardware without building a “plant prototype”
- supports reproducible test runs that can assist in uncovering and tracking down hard to find problems
- enables testing less risk of destroying system
- provides cost savings by shortened development time
- complete, consistent test coverage
- supports automated testing
- simulator performs test outside the normal range of operation

The most evident advantage of HIL simulation is that real-world conditions are achieved without the actual risks involved. HIL simulation is achieving a highly realistic simulation of equipment in an operational virtual environment. With HIL, you can test the control units with extreme conditions that might not be feasible in the real world. HIL enables you to isolate deficiencies in the control unit even if they occur only under certain circumstances. Robust, high-fidelity real-time HIL simulations not only enable shorter time to market by reducing the development period, but also reduce cost by eliminating the need for actual hardware during testing, as well as associated maintenance costs. With the power and flexibility of today's computers, engineers and scientists are increasingly using PC-based systems for HIL simulation applications. A key element of the development of such a system is the integration of signal generation/acquisition I/O functions with the software used to simulate the system. A normal desktop PC was used as hardware of the HIL simulator with a National Instruments PCI-6251analog-digital data acquisition card. The real-time operating system solution for the plant model was xPC Target real-time kernel.

2 Plant and Control Unit

Plant is an electrohydraulic servo system shown in Figure 1 and 2. Electrohydraulic servo actuators are widely used in industrial applications because it has high moving force, power/volume ratio and they proof against environmental impacts so it can be used as built-in element at the acting location directly. Electrohydraulic servos are capable of performance superior to that of any other type of servo. Large inertia and torque loads can be handled with high accuracy and very rapid response. A typical position controlled hydraulic system consists of a hydraulic power supply, flow control valve, linear actuator, displacement transducer, and electronic servo-controller. [1]

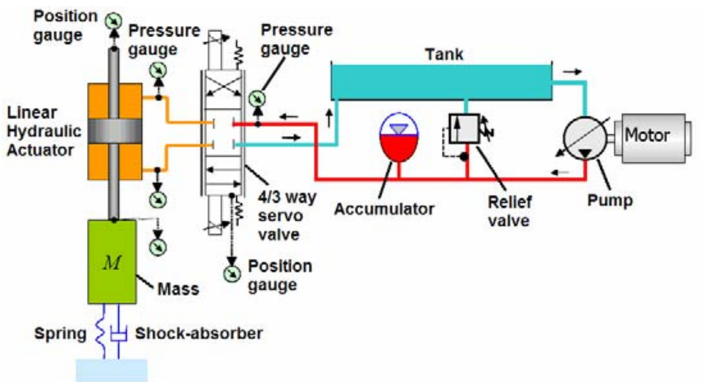

Figure 2
Electrohydraulic Servo System

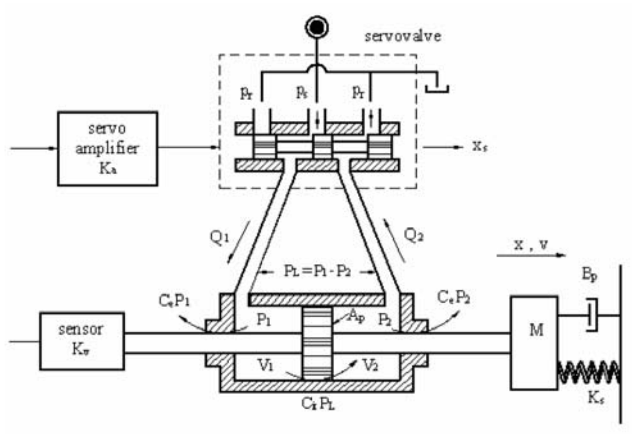

Figure 3
Electromagnetic Driven Servo Valve and Double-acting, Double-ended Linear Hydraulic Actuator

The control unit F2812 eZdsp™ card is a stand alone module with a single chip parallel port to JTAG scan controller. The eZdsp F2812 allows developers to get started using the F2812 DSP. The C28x core is the world highest performance DSP core for digital control applications. The 32-bit F2812 has on board flash, 64K words on board RAM and runs at 150MHz, making it capable of numerous sophisticated control algorithms in real-time. The module can be operated without additional development tools such as an emulator. The combination of a bundled version of Code Composer and an on-board JTAG interface means that the eZdsp can be used to develop and debug code without the requirement for an external emulator or debugger. [2]

Figure 4
eZdsp F2812 Development Package

3 Plant Mathematical Model

A servo-valve is a complex device which exhibits a high-order non-linear response, and knowledge of a large number of internal valve parameters is required to formulate an accurate mathematical model. When modeling complex servo-valves, it is sometimes possible to ignore any inherent non-linearities and employ a small perturbation analysis to derive a linear model which approximates the physical system. Such models are often based on classical first or second order differential equations, the coefficients of which are chosen to match the response of the valve based on frequency plots taken from the data sheet. The electrical characteristics of the servo-valve torque motor may be modeled as a series L-R circuit. The transfer function of a series L-R circuit is [4]:

$$L_C \cdot \frac{di}{dt} + R_C \cdot i = K_a \cdot u \tag{1}$$

where LC is the inductance of the servo valve coil, and RC the combined resistance of the servo valve coil and the current sense resistor, K_a the amplification of the servo amplifier, u and i are the control voltage and current. The lateral force on the valve spool is proportional to torque motor current, but oil flow rate at the control ports also depends upon the pressure drop across the load. The dynamic model of the valve spool may be approximated by a second order transfer function without serious loss of accuracy [4]:

$$\frac{d^2 x_s}{dt^2} + 2 \cdot \zeta_s \cdot \omega_s \cdot \frac{dx_s}{dt} + \omega_s^2 \cdot x_s = \omega_s^2 \cdot k_t \cdot i \tag{2}$$

where ω_s is the natural frequency and ζ_s damping ratio of the spool, k_t proportionality coefficient between the control current and valve spool displacement x_s.

The servo-valve delivers a control flow proportional to the spool displacement for a constant load. For varying loads, fluid flow is also proportional to the square root of the pressure drop across the valve. Control flow, input current, and valve pressure drop are related by the following equations [3,4]:

$$Q_a = Q_{1s} = c_s \cdot w \cdot x_s \cdot \sqrt{\frac{2}{\rho} \cdot (P_S - P_a)} \tag{3}$$

$$Q_b = Q_{2s} = c_s \cdot w \cdot x_s \cdot \sqrt{\frac{2}{\rho} \cdot (P_b - P_T)} \tag{4}$$

$$A_s = w \cdot x_s \tag{5}$$

where c_s is the volumetric flow coefficient and w the valve-port width – area gradient, A_s the size of flow cross-section of the valve, ρ volumetric density of the oil, P_S, P_T are system pressure and tank pressure, P_a and P_b are the load and return pressure, Q_a and Q_b the load and return flow of the valve.

The compressibility of the oil creates a "spring" effect and it can be modeled using the flow continuity equation from fluid mechanics [3,4]:

$$Q_a - Q_b = \frac{dV}{dt} + \frac{V}{\beta} \cdot \frac{dP}{dt} \tag{6}$$

where V is the internal fluid volume (in pipe and cylinder) and β the fluid bulk modulus. This equation can be used if the mechanical structure is perfectly rigid. The pressures in cylinder chambers are [1,3,4]:

$$\frac{dP_a}{dt} = \frac{dP_1}{dt} = \frac{\beta}{V} \cdot \left(Q_a - \frac{dV_a}{dt} \right) \tag{7}$$

$$\frac{dP_b}{dt} = \frac{dP_1}{dt} = \frac{\beta}{V} \cdot \left(Q_b - \frac{dV_b}{dt} \right) \tag{8}$$

$$\frac{dV_a}{dt} = -\frac{dV_b}{dt} = A_P \cdot \frac{dx}{dt} = A_P \cdot v \tag{9}$$

where x and v are the position and speed of the piston and A_p the active area of the piston annulus. If the sealing is not perfect there is an additional leakage oil flow Q_i between the chamber "a" and "b" [1]:

$$Q_i = \frac{P_1 - P_2}{R_i} = C_i \left(P_1 - P_2 \right) \tag{10}$$

$$Q_{e1} = \frac{P_1}{R_{e1}} = C_{e1} P_1, \; Q_{e2} = \frac{P_2}{R_{e2}} = C_{e2} P_2 \tag{11}$$

$$C_i = \frac{1}{R_i}, \; C_{e1} = \frac{1}{R_{e1}}, \; C_{e2} = \frac{1}{R_{e2}} \tag{12}$$

where R_i and R_e are the internal and external cylinder's leakage resistance, C_i and C_e the internal and external cylinder's leakage coefficient. Using equations (7)-(11) the load and return flow Q_1 and Q_2 of the hydraulic actuator and considering the internal fluid volume V is:

$$V = V_0 + A_P \cdot x \tag{13}$$

$$\frac{dP_1}{dt} = \frac{\beta}{V_0 + A_P \cdot x} \left[Q_a - A_P \cdot v - Q_i - Q_{e1} \right] \tag{14}$$

$$\frac{dP_2}{dt} = \frac{\beta}{V_0 - A_P \cdot x} \left[-Q_b + A_P \cdot v + Q_i - Q_{e2} \right] \tag{15}$$

where V_0 is the internal fluid volume when the piston is in middle position.

The net force acting on the piston (F_P) can be calculated by the differential pressure:

$$F_P = A_P \cdot (P_1 - P_2) \tag{16}$$

Force equation of the piston can be calculated when the load is a mass M, spring with stiffness coefficient K_S and a shock absorber with damping ratio B_P:

$$F_P = (M + M_P) \cdot (a + g) + B_P \cdot v + K_S \cdot x + \mathrm{sign}(v) \cdot F_f \tag{17}$$

where M_P is the mass of the actuator piston and rod of the actuator, g gravity, F_f frictional force, a acceleration, v speed and x position of the piston.

4 Test Environment

Plant model runs on a PC using xPC Target real-time kernel and a laptop with MATLAB to generate the code from mathematical model of the electrohydraulic servo system. The PCI-6251 A/D card voltage input is used to control the servo valve and voltage outputs to examine the time functions of the system variables. The control signal of the servo valve was generated by the F2812 DSP board.

Figure 5
Test Environment

The parameters of the electrohydraulic servo system are:

Symbol	Description	Value
K_a	Amplification of the servo amplifier	50
L_C	Inductance of the servo valve coil	0.6 H
R_C	Combined resistance of the servo valve coil and the current sense resistor	100 Ω
ζ_s	Damping ratio of the spool	0.9
ω_s	Natural frequency of the spool	200 rad/s
k_t	Proportionality coefficient between the control current and valve spool displacement	0.01 m/A
c_s	Volumetric flow coefficient	0.6
w	Valve-port width	10^{-3} m

Table 1

Parameters of the Flow Control Servo Valve

Symbol	Description	Value
V_0	Internal fluid volume when the piston is in middle position	$300 \cdot 10^{-6}$ m^3
A_p	Active area of the piston annulus	$8 \cdot 10^{-4}$ m^2
β	Hydraulic fluid bulk modulus	$1.4 \cdot 10^9$ N/m^2
R_i	Internal cylinder's leakage resistance	10^{12} Ns/m^5
R_{e1}, R_{e2}	External cylinder's leakage resistance	$4 \cdot 10^{12}$ Ns/m^5
M_P	Mass of actuator piston and rod	5 kg
x_{pmax}	Total stroke of the piston	0.3 m

Table 2

Parameters of the Actuator

Symbol	Description	Value
P_S	System pressure	$210 \cdot 10^5$ Pa
P_T	Tank pressure	0 Pa
ρ	Volumetric density of the oil	890 kg/m^3

Table 3

Parameters of the Power Supply

Symbol	Description	Value
M	Load mass	50 kg
B_P	Damping ratio	$2 \cdot 10^3$ Ns/m
K_S	Spring stiffness coefficient	$5 \cdot 10^4$ N/m
Ff	Frictional force	10 N
g	Gravity	9.81 m/s^2

Table 4
Parameters of the Load

5 Test Results

The test execution time was 2s and step response was examined. The observed variables are u servo valve control voltage, x_s servo valve spool position (these are independent of R_i), Q_a, Q_b load and return flow of the valve, P_1, P_2 load and return pressure, x position of the piston and mass load and F_p acting force of the hydraulic actuator. Time functions of the electrohydraulic system are shown in the next figures (Fig. 6-12).

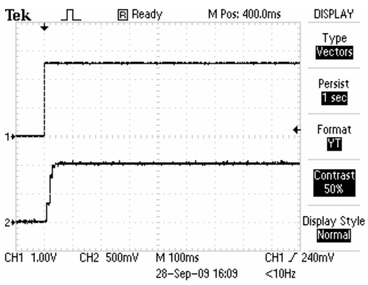

Figure 6
CH1: control voltage u(t) K_u=4, CH2: position of servo valve spool $x_s(t)$ K_{xs}=0,005m/V

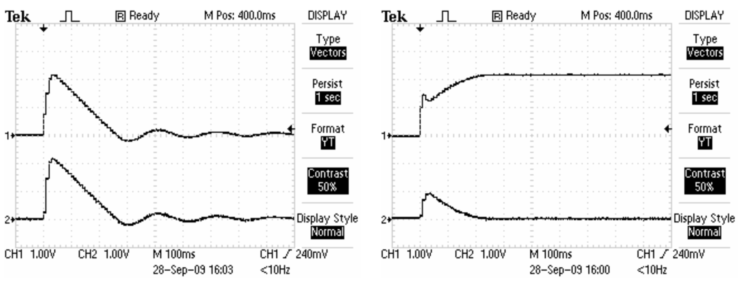

Figure 7 a,b
CH1a: $Q_a(t)$, CH2a: $Q_b(t)$ $K_Q=10^{-3}$ $m^3/s/V$, CH1b: $P_1(t)$, CH2b: $P_2(t)$, $K_p=10^7$ Pa/V at $R_{i3}= 10^{12}$

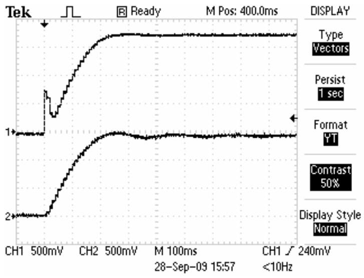

Figure 8

CH1: Fp(t) K_{Fp}= 10^4N/Vand CH2: x(t) K_x=0,25m/V at R_{i3}= 10^{12}

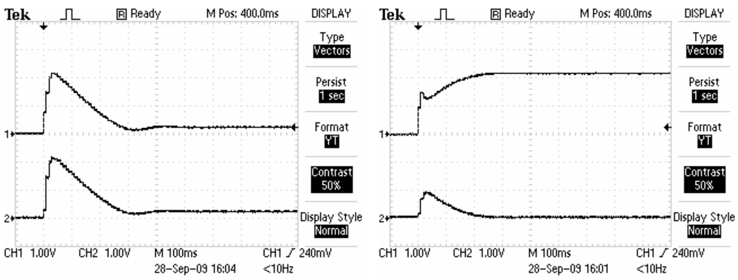

Figure 9 a,b

CH1a: Q_a(t), CH2a: Q_b(t) K_Q=10^{-3} m^3/s/V, CH1b: P_1(t), CH2b: P_2(t), K_p=10^7 Pa/V at R_{i3}= 10^{11}

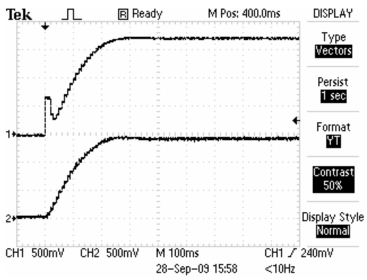

Figure 10

CH1: Fp(t) K_{Fp}= 10^4N/Vand CH2: x(t) K_x=0,25m/V at R_{i3}= 10^{11}

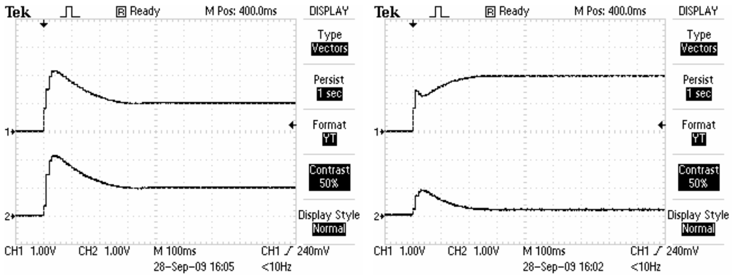

Figure 11 a,b

CH1a: $Q_a(t)$, CH2a: $Q_b(t)$ $K_Q=10^{-3}$ $m^3/s/V$, CH1b: $P_1(t)$, CH2b: $P_2(t)$, $K_p=10^7$ Pa/V at $R_{i3}= 10^{10}$

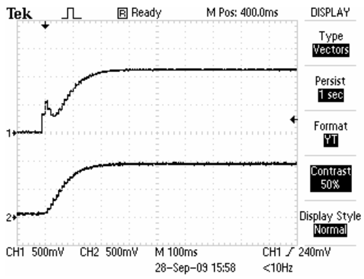

Figure 12

CH1: Fp(t) $K_{Fp}= 10^4$N/Vand CH2: x(t) K_x=0,25m/V at $R_{i3}= 10^{10}$

Conclusions

It is shown in the step response in Figure 6 that position of servo valve pool rapidly follows the valve control voltage and opens the flow channel to the hydraulic actuator. When the servo valve opens, the oil flow immediately increasing (see Figure 7a) and it causes growing pressure drop in the control valve therefore the load pressure growing period is cracked (see P1 in Figure 7b). When the quantity of oil flow goes on decreasing, the load pressure starts to grow again and reaches the system pressure. Because of the acting force is proportional to load pressure the time function of acting force is similar to load pressure (see Figure 8). After the control valve opening the position of the actuator piston suddenly starts to move therefore the return pressure starts to grow. When the speed of the piston is decelerating the pressure is decreasing (see Figure 7b, P2). It is shown in Figure 8 that the piston is moving rapidly and stops with decreasing oscillation at 0,35m.

It is shown in the step response that 10% cylinder's leakage resistance changing the system's behavior from periodic to aperiodic (See Fig 10). Additionally 20% cylinder's leakage resistance changing decreases the acting force with 30% so the movement of the actuator is less than 0,25m (Fig. 12). Furthermore it is shown in Fig 11a that high quantity, more than 10^{-3} m^3/s oil flows from the actuator cylinder chambers “a” to “b” and the maximum pressure in cylinder chamber “a” is less than $200 \cdot 10^5$ Pa (Fig. 11b).

The presented plant model is capable to examine the dynamic behavior of the described electrohydraulic servo system in real-time, and this model can give a basis for examination of these systems, for example how the system behavior changes when sealing is damaged.

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