HARDWARE-IN-THE-LOOP SIMULATION EXPERIMENTS
WITH A HYDRAULIC MANIPULATOR MODEL

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ABSTRACT

Hardware-in-the-loop simulation (HILS) is a technique that allows a simulation model to interact with real-world components, that is, one part of a system can be a simulation model, when other parts of the system are the actual hardware.

This paper presents the development of a hydraulic SCARA manipulator model with the purpose of using it in hardware-in-the-loop simulation experiments. Most of the mathematical models are semi-empirical in order to reduce the required simulation time and to perform real time simulations in small cost platforms. Two small cost acquisition boards from National instruments, installed in a PentiumIII PC, were used to connect the real time simulation to the real world. The real time simulations run in the xPC platform from Mathworks. Position control experiments, following typical robot trajectories, were made with PD controllers implemented with operational amplifiers. The results obtained in the HILS experiments were very satisfactory, considering that semi-empirical models were used.

Keywords: hydraulic, modelling, hardware-in-the-loop simulation

1 INTRODUCTION

The hydraulic systems have been for a long time often used in industrial manufacturing and in heavy machinery. The hydraulic hardware has a great evolution during last years, from hydro-mechanical devices to sophisticated electro-hydraulic systems controlled by microprocessors. The use of electronics and microprocessors contributes to improve the dynamic performance and to increase the traditional systems with new features that can be used as an alternative to electro-mechanical servo systems. In addition the use of new control schemes, usually applied to non-linear systems, such as adaptive control,
fuzzy logic control or neural networks control, just to name a few, becomes possible because of the technological evolution of hydraulic hardware.

Applying new control schemes on real hydraulic systems is, nevertheless, complicated due to the cost and/or size of hardware. However, it is important that for a given application the relative merits of different control schemes can be evaluated, being the computer simulation one of best evaluation tools [Edge97]. In fact, a simulated environment is the cheapest and fastest way to test control algorithms.

Hardware-in-the-loop simulation refers to a technology where some of the components of a pure simulation are replaced with actual hardware [Maclay97]. One of the main uses of these techniques is the test and parameterization of real controllers that, instead of controlling real plants, with the associated risks of any experiment, control a real time simulation of a plant model. This technology provides a means for testing control systems over the full range of operating conditions, including failure modes. Testing a control system prior to its use in a real plant can reduce the cost and the development cycle of the overall system. Hardware-in-the-loop simulation has been used, with success, in the aerospace industry and is now emerging as a technique for testing electronic control units. This procedure has been applied to solve some specific problems but is seldom used as a platform to test the real time behaviour of hardware components.

2 HYDRAULIC PLATFORM: SCARA MANIPULATOR

The hydraulic platform consists on a SCARA manipulator, figure 1. The manipulator has two hydraulically actuated rotational axes and one hydraulically actuated linear axis. The motion control is accomplished using servo-solenoid proportional valves. The manipulator is also equipped with several sensors to measure the axes positions and accelerations and also to measure the actuators chamber pressures. The hydraulic circuit is shown in figure 2.
3 SCARA MANIPULATOR MODEL

This section presents the dynamic model for the SCARA manipulator and the models for hydraulic circuit components.

SCARA dynamic model

In this manipulator the linear axis dynamics can be decoupled from the dynamics of the other two arms. Only its mass distribution (actuator + load) influence the inertia matrix, through the variation of the centre of mass of the arm 2. For that reason its dynamics can be treated separately. Thus it’s the case of a manipulator with two degrees of freedom with rotary joints (figure 3).

Through the application of the Lagrange method, and considering null the potential energy the motion torques applied to the joints can be expressed by [Gomes de Almeida93]:

\[
\tau_1 = \frac{m_1 L_1^2 \ddot{\theta}_1 + l_1 \dot{\theta}_1 \ddot{\theta}_1 - l_1 \dot{\theta}_1 \dot{\theta}_2}{l_1^2 + l_2^2 - 2 l_1 l_2 \cos \theta_2},
\]

\[
\tau_2 = \frac{m_2 L_1^2 \ddot{\theta}_2 + l_2 \dot{\theta}_1 \ddot{\theta}_2 + m_2 L_2 \ddot{\theta}_2 + l_2 \dot{\theta}_2 \ddot{\theta}_2 - 2 l_1 l_2 \dot{\theta}_1 \dot{\theta}_2 - m_2 L_2 \dot{\theta}_2}{l_1^2 + l_2^2 - 2 l_1 l_2 \cos \theta_2}.
\]
\[ \tau = J(\theta)\dot{\theta} + c(\theta, \dot{\theta}) \]  

(1)

where \( \theta \) is the joint positions vector, \( \theta = [\theta_1 \ \ \theta_2]^T \), \( J \) is the inertia matrix of the manipulator, \( c \) is the torque vector introduced by the Coriolis and centripetal forces, and \( \tau \) is the vector of the torques applied to the joints, \( \tau = [\tau_1 \ \ \tau_2]^T \).

The components of \( \tau \) are expressed by:

\[
\begin{align*}
\tau_1 &= J_{11}\ddot{\theta}_1 + J_{12}\ddot{\theta}_2 + h_1 \\
\tau_2 &= J_{21}\ddot{\theta}_1 + J_{22}\ddot{\theta}_2 + h_2
\end{align*}
\]

(2)

where

\[
\begin{align*}
J_{11} &= m_1l_{11} + I_1 + m_2(l_1^2 + l_2^2 + 2L_1l_2\theta_2 \cos(\theta_2)) + I_2 \\
J_{12} &= J_{21} = m_2L_1l_{11} \cos(\theta_2) + m_2l_2^2 + I_2 \\
J_{22} &= m_2l_2^2 + I_2 \\
h_1 &= -2\dot{\theta}_1\dot{\theta}_2m_2L_1l_2 \sin(\theta_2) - \dot{\theta}_2^2m_2L_1l_2 \sin(\theta_2) \\
h_2 &= m_2L_1l_2 \sin(\theta_2) \dot{\theta}_1^2
\end{align*}
\]

(3)

The angular accelerations of the joints are given by:

\[
\begin{align*}
\ddot{\theta}_1 &= \frac{J_{22}(\tau_1 - h_1) - J_{12}(\tau_2 - h_2)}{J_{11}J_{22} - J_{12}J_{21}} \\
\ddot{\theta}_2 &= \frac{-J_{21}(\tau_1 - h_1) + J_{11}(\tau_2 - h_2)}{J_{11}J_{22} - J_{12}J_{21}}
\end{align*}
\]

(4)

for \( J_{11}J_{22} - J_{12}J_{21} \neq 0 \)

The above manipulator model don’t incorporates the actuator dynamics (including the friction forces) and is valid for any type of actuation. In the present case (hydraulic actuation), the motion torques \( \tau_1 \) e \( \tau_2 \) are given by the hydraulic actuators models.

**Hydraulic circuit model**

This section presents the models for the actuators and valves (see figure 2). The source pressure, \( P_s \), is considered constant and the tank pressure, \( P_t \), is considered to be zero.

The valve is modelled by assuming that the volume of the valve chambers are negligible compared to the overall volume of the system, and that fluid flow due to the compressibility of the hydraulic fluid can be included in the model of the hydraulic cylinder. These assumptions mean that the flows through the valve can be calculated by algebraic equations [Merrit67].
The area of the valve orifice opening versus the relative position of the valve spool, \( x_s \in [-1,1] \) could be modelled by pseudo-sections [Quintas99]. It is assumed that the valve is matched and symmetric (the pseudo-sections are equal two-by-two). The orifice sections are modeled in such a way that the flow through the valve is always turbulent, the laminar flow being implicitly modeled in the pseudo-sections [Ferreira02].

The tank flow \( Q_t \), source flow \( Q_s \) and outlet flows \( Q_1 \) and \( Q_2 \) may be formulated as follows:

\[
\begin{align*}
Q_s &= q_{s1} + q_{s2} \\
Q_t &= q_{t1} + q_{2t} \\
Q_i &= q_{s1} - q_{t1} \\
Q_2 &= q_{2t} - q_{s2}
\end{align*}
\]

(5)

The flow rates through the valve given by:

\[
\begin{align*}
q_{s1} &= \text{sign}(\Delta P_{s1}) \cdot A_{p}(x_s) \cdot \sqrt{\Delta P_{s1}} \\
q_{s2} &= \text{sign}(\Delta P_{s2}) \cdot A_{p}(x_s) \cdot \sqrt{\Delta P_{s2}} \\
q_{t1} &= \text{sign}(\Delta P_{t1}) \cdot A_{t}(x_s) \cdot \sqrt{\Delta P_{t1}} \\
q_{2t} &= \text{sign}(\Delta P_{2t}) \cdot A_{p}(x_s) \cdot \sqrt{\Delta P_{2t}}
\end{align*}
\]

(6)

where \( \Delta P_{ij} = P_i - P_j \) is the pressure drop between the two ports and \( \text{sign}(\Delta P_{ij}) \) is the sign of \( (P_i - P_j) \). \( A_{xy} \) are the pseudo-sections.

Hyperbolic functions were used to model the pseudo-sections. The main reason for this is the possibility of characterizing two well-defined asymptotes for the valve. For a symmetrical and matched valve the hyperbolic functions can be described by:
\[ A_p(x) = k_1 \cdot x + k_2 + \sqrt{k_3 \cdot x^2} + k_4 \cdot x + k_5 \]
\[ A_p(x) = -k_1 \cdot x + k_2 + \sqrt{k_3 \cdot x^2} - k_4 \cdot x + k_5 \]

where \( k_i \in \mathbb{R} \). The valve parameters can be calculated through the use of the static valve characteristics (pressure gain, leakage flow rate, flow gain and nominal flow rate) [Ferreira 02].

**Hydraulic actuators**

The inlet and outlet chamber flows are related to the actuator velocity and the compressibility effects. These phenomena are expressed by known as the continuity equation:

\[ \sum Q_{in} - \sum Q_{out} = \frac{dV_0}{dt} \left( \frac{V_0}{\beta_e} \right) \frac{dP}{dt} \]  

(8)

Considering the diagram presented in figure 5, the cylinder flow rate can be stated by applying (8) to each cylinder chamber:

\[ Q_1 = g_{ke} (P_1 - P_2) + A_1 \cdot v_p + \frac{V_{L1} + A_1 x_p}{\beta_e} \frac{dP_1}{dt} \]  

(9)

\[ Q_2 = g_{ke} (P_1 - P_2) + A_2 \cdot v_p - \frac{V_{L2} + A_2 (L - x_p)}{\beta_e} \frac{dP_2}{dt} \]  

(10)

The volume in chamber 1 is \( V_{01} = V_{L1} + A_1 x_p \) and in chamber 2 this volume is \( V_{02} = V_{L2} + A_2 (L - x_p) \); \( V_{L1,2} \) represents the line volume plus the cylinder dead volumes and respective valve chamber volume; \( \beta_e \) is the effective bulk modulus; \( v_p \) is the piston velocity. It is assumed that the cylinder has no external leakages being the internal leakage flow rate given by \( q_{ke} = g_{ke} (P_1 - P_2) \), where \( g_{ke} \) is the leakage conductance.

The cylinder dynamics is defined as:
\[ M \cdot \dot{v}_p = P_1A_1 - P_2A_2 - k_vv_p + F_L \]  

(11)

where \( F_L \) is the total load force, \( k_v \) is the viscous friction coefficient and \( M \) is the total mass in motion (load, piston, rod).

**Modeling the vane actuator**

The output torque, assuming 100\% of efficiency, is given by:

\[ \tau_{out} = D(P_1 - P_2) \]  

(12)

where \( D \) is the volume per unit of angle

By applying (8) to the vane actuator chambers, figure 5, the inlet and outlet flow rates are:

\[ Q_1 = g_{lv}(P_1 - P_2) + D \cdot w + \frac{V_{l1}}{\beta_e} \frac{dP_1}{dt} \]  

(13)

\[ Q_2 = g_{lv}(P_1 - P_2) + D \cdot w - \frac{V_{l2}}{\beta_e} \frac{dP_2}{dt} \]  

(14)

\( Q_1 \) and \( Q_2 \) are the inlet and outlet flow rates; \( P_1 \) and \( P_2 \) are the actuator chambers; \( g_{lv} \) is the leakage conductance; \( \beta_e \) is the effective bulk modulus; \( \theta_{max} \) is the maximum displacement and \( \theta \) is the angle; \( V_{l1} \) and \( V_{l2} \) represent the same volumes as in the linear actuator; \( w \) is the angular velocity.

The vane dynamics is given by:

\[ J\ddot{\theta} = \tau - k_v \cdot \omega \]

where \( J \) is the total inertia attached to the vane and \( k_v \) is the viscous friction coefficient.

When modelling the dynamics of the actuators, special attention must be placed on the physical limits of motion because the model is to be used in real time simulations. The effect on system dynamics when the actuator reaches either of the physical limits is crucial if the model is to provide realistic results for all possible operational conditions. The physical limit model that was used is based on a finite state machine, with three distinct operational zones, right limit or StopRight, left limit or StopLeft, and the normal operating range, Normal [Ferreira03]. Figure 6 shows the example of the finite state machine for the linear actuator where \( L \) is the right limit of the piston’s movement, and \( f \) is the available force of the actuator.

![Finite state machine operational zones](image)
The Normal operating zone governs system dynamics and is used when the cylinder is between the upper and lower limits. The StopLeft and StopRight zones govern system dynamics when the cylinder reaches the left and right limits. The velocity is initialized when StopLeft or StopRight are activated and the acceleration is set to zero as long as these states are active. At Normal state the acceleration is calculated with Newton second law.

4 HARDWARE-IN-THE-LOOP SIMULATION EXPERIMENTS

The model of the SCARA manipulator, presented in the above sections, was implemented in Simulink and simulated in real time, in a computer where the xPC kernel is installed (figure 7). In this computer the xPC only uses the memory and the CPU. Through command lines from Matlab, installed on the host computer, the real time simulation can be operated and monitored. Two low cost data acquisition cards (LabPC+) from National Instruments were used to connect the real time simulation to the real controllers.

The overall manipulator was simulated with a 4th order Runge-Kutta integration algorithm with a fixed step of 0.25ms. The reference trajectories, used in the HILS experiments, for the manipulator positions are typical point-to-point trajectories. The positions controllers, shown in figure 8, and are PD (proportional+derivative) controllers implemented with operational amplifiers.
Figures 9 and 10 show the results for the manipulator first arm (angle $\theta_1$) and for the linear arm (position $x_p$). The desired trajectories exceed purposely the actuator limits to evaluate the limits implementation. $u_{1,3}$ are the valve inputs and (ref) means the reference.

5 CONCLUSIONS

A hydraulic manipulator model, with some components modeled semi-empirically, was setup in the Simulink environment. The model describes all the manipulator operation regions, including the motion limits. The motion limits are implemented with a state machine that consider all the impact fast dynamics as instantaneous changes.

The xPC real time kernel from Mathworks, was used for real time simulation of the overall SCARA model and to perform hardware-in-the-loop simulation experiments. Some HILS experiments have been performed by using real controllers, implemented with operational amplifiers, attached to the simulation through low cost data acquisition cards from National Instruments.
From the HILS experiments can be concluded that, if semi-empirical models are used for some hydraulic components and phenomena, hydraulic systems, that traditionally originate stiff equation systems can be simulated in real time and used to perform HILS experiments in low cost hardware platforms.

REFERENCES


APPENDIX

Parameter values for the manipulator models:

\[ D_1 = 3 \times 10^{-5} m^3 rad^{-1} \quad D_2 = 3 \times 10^{-5} m^3 rad^{-1} \quad \theta_{\text{max}} = 4.71 \text{rad} \quad g_{\text{like}} = g_{\text{rev}} = 1 \times 10^{-12} m^3 s^{-1} Pa^{-1} ; \]

\[ V_{10} = V_{20} = 1 \times 10^{-6} m^3 \quad \beta_v = 1 \times 10^3 Pa \quad k_v = 1000 N sm^{-1} \quad P_s = 140 \times 10^3 Pa \quad P_l = 0 Pa ; \]

\[ k_1 = -2.142 \quad k_2 = 5.818 \times 10^{-3} \quad k_3 = 4.604 \quad k_4 = -5.554 \times 10^{-2} \quad k_5 = 1.534 \times 10^{-2} \quad m_1 = 27.0 Kg ; \]

\[ I_1 = 0.675 Kg m^2 ; \quad l_1 = 0.31 m \quad m_2 = 33.2 Kg \quad I_2 = 1.027 Kg m^2 ; \quad l_2 = 0.32 m ; \]

\[ L_2 = 0.417 m ; \]