CLOSED LOOP PARAMETRIC IDENTIFICATION OF A ROCKET THRUST CONTROL SYSTEM

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Abstract. *This paper focuses the dynamic modeling and identification of a hydromechanical servomechanism with electric command and its application to the attitude flight control system of a solid-propellant satellite launch vehicle (VLS). The vector thrust control (VTC) is used to maintain the desired flight track and to have a stable trajectory against unknown external disturbances. In the VTC, an electro hydraulic servo actuation system is used to position the thrust vector direction and achieve the desired rocket attitude. The VTC system is obtained by a movable nozzle actuated by two hydraulic pistons. In the current system, the rocket engine nozzle is fixed to the engine body by an elastic coupling element allowing angular movement of the nozzle in two orthogonal directions. The angular motion of the nozzle is actuated by a two-stage electro-hydraulic servovalve with a direct mechanical feedback between the flapper-nozzle electro-valve and the piston output. The flapper-nozzle system is actuated by a variable reluctance linear motor, controlled by an electric signal from the rocket embedded flight command computer system. A dynamic model of the complete electro-hydro servomechanism is proposed here. The model parameters are obtained by closed loop parametric identification process in the time domain using the indirect approach. Data for the identification process was obtained in an experimental setup, which is also presented in the paper. Model validation is performed through comparison with the real system behavior.*

Keywords*: electro hydraulic control system, vector thrust control, hydraulic parameter estimation*

1. INTRODUCTION

Thrust vector control (TVC) of space flights solid rockets generally employs a movable nozzle attached to the rocket body by a flexible joint which allows the angular displacements of the nozzle. The nozzle angular displacements are driven by a servo actuator as shown in Figure 1.

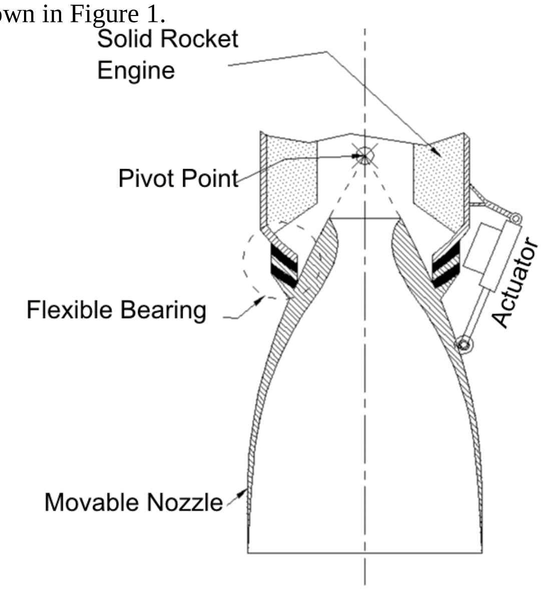

Figure 1. Scheme of thrust vector control by movable nozzle.

The vast majority of the servo actuators employed on space rockets is of the eletrohydraulic type which receives an electrical signal as control signal and uses hydraulic power to handle their loads. Almost all eletrohydraulic servo actuator is composed by a linear hydraulic piston or rotary actuator, which handles the load, and by an electrohydraulic servovalve which, by receiving an electric control signal, controls the high energy flows and pressures necessary the move the hydraulic actuator. For hydraulic pistons, as in this paper case, there is the option of 3 way and 4 way valve scheme, as presented in Figure2.

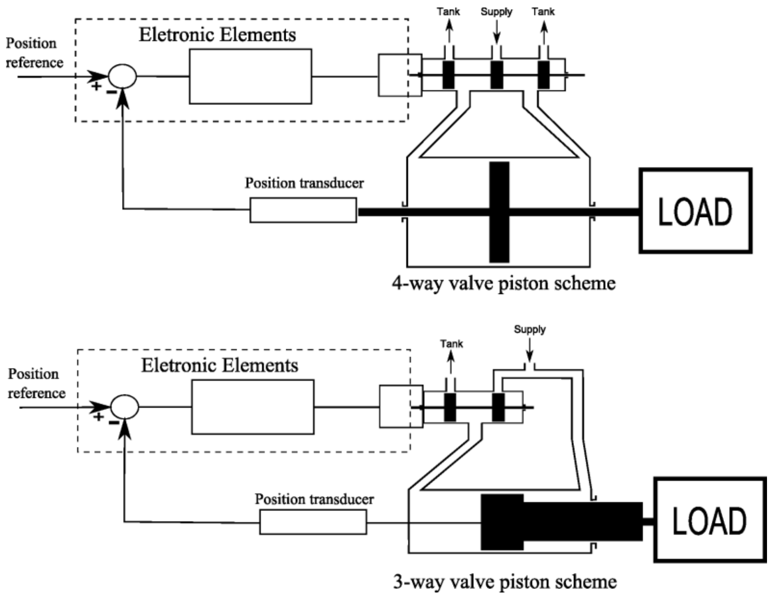

Figure 2. General three-way and four-way valve piston schemes.

Usually the proportional valve actuator control employs a closed loop system with feedback of the piston position signal and electronic devices for position error detection and control signal implementation. but some servo actuators employ a mechanical feedback mechanism. This is the case of the servo actuator studied in this work. More details are presented in section 3.

Understanding the servo actuator dynamic behavior is very important since this component guarantees the correct attitude control actuation of the rocket launcher. This means that to have a reliable model of this component, the servo actuator model has to be developed by applying the proper physic principles or it can be experimentally identified assuming an a priori model structure and applying system identification methods. When the open loop model is unstable or it is unsecure to operate a device in open loop, this system dynamic model identification must be performed in closed loop (Ljung, L. 1987). The indirect identification method (Forssell, U. and Ljung, L., 1997) is one of the several strategies available to perform closed loop parameter identification.

Input, output data for system identification is obtained by applying a persistently exciting signal to the servo actuator, and collecting its response. It was employed a PRBS type signal (Aguirre, L. A., 2000) to perform an informative data for system identification.

2. OBJECTIVES

This work has the objective to present a closed loop identification procedure applied to identification of a servo actuator. The identification process relies on experimental data generated by an experiment with a real servo actuator. Section 3 presents a description of the servo actuator and its proposed model structure. In this work, the servo actuator do not has a movable nozzle and flexible joint load attached to its rod, e. g., the actuator without external load.

The closed loop parametric identification procedure is presented together with the results obtained with this procedure.

3. SERVO ACTUATOR DESCRIPTION

The servo actuator consists of a eletrohydraulic servo actuator controlled by an input current, i_c, with the output hydraulic piston rod displacement, y_{at}, proportional to i_c. Figure 3 shows the servo actuator scheme together with its main components.

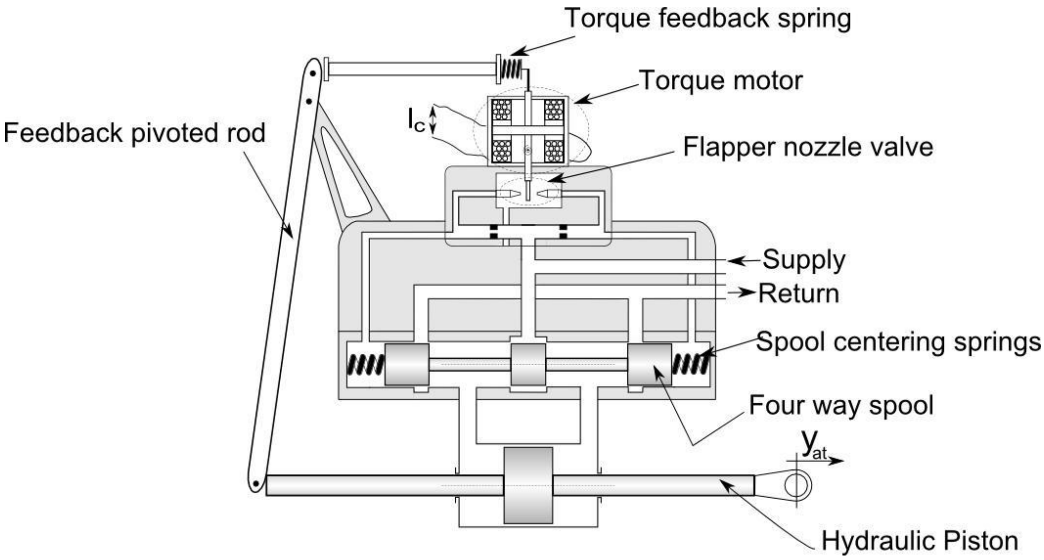

Figure 3. Servo actuator scheme.

The servo actuator consists of a hydraulic piston driven by a two stage eletrohydraulic servovalve. The first servovalve stage is composed of a flapper nozzle valve which drives the second stage, comprised of a four way spool type hydraulic valve. The flapper is driven by a variable reluctance torque motor which receives the control current i_c and creates a drive torque T_M over the flapper and causes its angular displacement. The balance rod senses the piston displacement y_{at} and together with the torque feedback springs performs the position feedback. Torque balance happens because the feedback system creates a feedback torque, T_{fb}, proportional to piston position, y_{at}, which opposes the torque created by the torque motor acting over the flapper.

In this work one consider a linear model structure for the identification process. So issues as dead time, dry friction and other mechanical nonlinearities are not considered.

The proposed model structure for this servo actuator is shown in Figure 4.

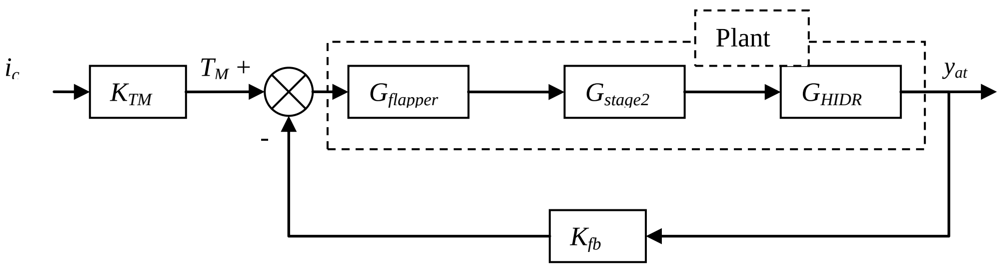

Figure 4. Proposed model structure for the servo actuator.

The blocks identified as G_{stage2} and G_{HIDR} describe the dynamics of the second stage where the flapper nozzle valve controls the position of the spool valve and the dynamics of the hydraulic piston driven by the spool valve, respectively. They presented together in Eq(1), see Merrit (1966) and Loureiro and Goes (2007).

$$G_{stage2}.G_{HIDR}=\frac{X_{spool}(s)}{X_{flapper}(s)}\frac{Y_{at}(s)}{X_{spool}(s)}=\left[\frac{\dfrac{K_{qflapper}}{A_{spool}}}{\left(s+\dfrac{K_{cflapper}.K_{spool}}{A_{spool}^2}\right)\left(\dfrac{s^2}{\omega_{spool}^2}+\dfrac{2\delta_{spool}}{\omega_{spool}}s+1\right)}\right].\left[\frac{\dfrac{K_{qspool}}{A_p}}{s\left(\dfrac{s^2}{\omega_{hp}^2}+\dfrac{2\delta_{hp}}{\omega_{hp}}s+1\right)}\right] \quad (1)$$

In Eq.(1) A_p is the piston area, ω_{hp} means the natural frequency of the hydraulic piston and is function of the oil bulk modulus and piston mass, see Merrit (1966). The hydraulic damping coefficient of the piston is denoted by δ_{hp} and is caused by the internal leakages of the spool and piston. K_{qspool} is the flow gain of the spool, Eq.(2), with Q_L the flow through the spool and x_{spool} its displacement. Also A_{spool} is the spool area, ω_{spool} means the natural hydraulic frequency of the spool (Merrit,1966). The hydraulic damping coefficient of the spool is denoted by δ_{spool} and is caused by the internal leakages of the flapper nozzle valve. $K_{qflapper}$ is the flow gain of the flapper nozzle, Eq.(3), with Q_{Lc} meaning the control flow which drives the spool.

$$K_{qspool}=\frac{\partial Q_L}{\partial x_{spool}} \quad (2)$$

$$K_{qflapper}=\frac{\partial Q_{Lc}}{\partial x_{flapperl}} \quad (3)$$

Equation (4) shows the linear displacement of flapper tip between the nozzles, $x_{flapper}$. $G_{flapper}$ describes the flapper dynamic as response to a torque input. The feedback mechanism is a pure gain denoted by K_{fb}.

$$x_{flapper}(s)=\left(K_{TM}.i_c-K_{fb}.y_{at}\right).G_{flapper}=\frac{\left(K_{TM}.i_c-K_{fb}.y_{at}\right)}{\left(\dfrac{s^2}{\omega_{flapper}^2}+\dfrac{2\delta_{flapper}}{\omega_{flapper}}s+1\right)} \quad (4)$$

Figure 4 shows that the plant model structure presented in Eq.(5)

$$G_{plant}=\frac{1}{\left(\dfrac{s^2}{\omega_{flapper}^2}+\dfrac{2\delta_{flapper}}{\omega_{flapper}}s+1\right)}\left[\frac{\dfrac{K_{qflapper}}{A_{spool}}}{\left(s+\dfrac{K_{cflapper}.K_{spool}}{A_{spool}^2}\right)\left(\dfrac{s^2}{\omega_{spool}^2}+\dfrac{2\delta_{spool}}{\omega_{spool}}s+1\right)}\right].\left[\frac{\dfrac{K_{qspool}}{A_p}}{s\left(\dfrac{s^2}{\omega_{hp}^2}+\dfrac{2\delta_{hp}}{\omega_{hp}}s+1\right)}\right] \quad (5)$$

The identification of a transfer function with such a high order would require the measurement of y_{at} at frequencies above 500 Hz and with very low amplitude. Merrit (1966) suggests that the dynamics of the hydraulic actuator dominates the servo actuator behaviour because the servovalves have the fastest responses. So, it is suggested to use a a reduced order plant model approximated by the model of Eq.(6).

$$G_{simple} = \left[\frac{\frac{K_{qflapper}}{A_{spool}} \cdot \frac{K_{qspool}}{A_p}}{s\left(\frac{s^2}{\omega_{hp}^2} + \frac{2\delta_{hp}}{\omega_{hp}} s + 1 \right)} \right] \tag{6}$$

4.CLOSED LOOP PARAMETER IDENTIFICATION

The objective of identification process is to find the parameters of the simplified open loop model G_{simple}. The Figure 3 show that the servo actuator mechanism do not allow open loop tests, e. g, without the feedback mechanism. So it was necessary to apply a closed loop systems identification technique. Taking *r* as the reference signal, F_y as the controller transfer function, G_0 as the plant transfer function, *v* as disturbance input, *u* as plant input signal and *y* as system response, Figure 5 presents a general closed loop system for the closed loop model identification.

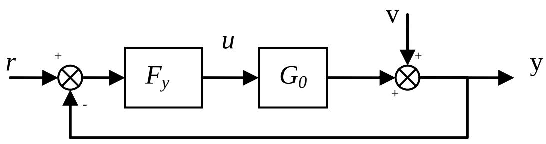

Figure 5. General closed loop system.

The closed loop parametric identification procedures objective is to estimate G_0 from available closed loop test data. According Forssell and Ljung (1997), closed loop identification procedures can be listed as direct approach, when G_o estimation takes *u* and *y* measured data; indirect approach, when G_o estimation takes *r* e *y* measured data and it is supposed F_y known; and the Joint Input – Output Approach when it is considered the input *u* and the output *y* jointly as the output from a system driven by the reference signal *r* and noise and it is used some available method to determine the open-loop parameters from an estimate of this system.

Considering the characteristics of servo actuator mechanism, it is possible to apply only the indirect approach since it is possible to measure only *r* and *y*.

Forssell and Ljung (1997) states that the indirect approach procedure takes two steps: identification of the estimated closed loop model $\hat{G}_{cl}$ from the measured *r* and *y* data, and determination of the estimated open loop plant model $\hat{G}_0$ using $\hat{G}_{cl}$ and the previously known F_y. As $\hat{G}_{cl}$ is given by using Eq.(7).

$$\hat{G}_{cl} = \frac{\hat{G}_0}{1 + F_y \hat{G}_0} \tag{7}$$

The estimated $\hat{G}_0$ may be found by Eq.(8)

$$\hat{G}_0 = \frac{\hat{G}_{cl}}{1 - F_y \hat{G}_{cl}} \tag{8}$$

An issue of this process is that the estimated $\hat{G}_0$ may have a high order since we have the degree of G_{cl} plus de degree of F_y in the denominator.

If it is possible to parameterize G_{cl} using plant parameters θ of $G_0(q,\theta)$, then $\hat{G}_{cl}$ can be estimated using only one step process, according to Eq.(9)

$$G_{cl}(q,\theta) = \frac{G(q,\theta)}{1 + F_y G(q,\theta)} \tag{9}$$

The servo actuator model is shown in Figure 6 in a modified way. In Figure 6 K_{driver}, Eq.(10), is the servo amplifier driver gain which relates i_c, in amperes, to a reference displacement y_r, in meters, as servo actuator input allowing easier input output comparisons.

$$K_{driver} = \frac{i_c}{y_r} = \frac{0{,}050[A]}{0{,}023[m]} = 2{,}1739[\mathrm{A/m}] \tag{10}$$

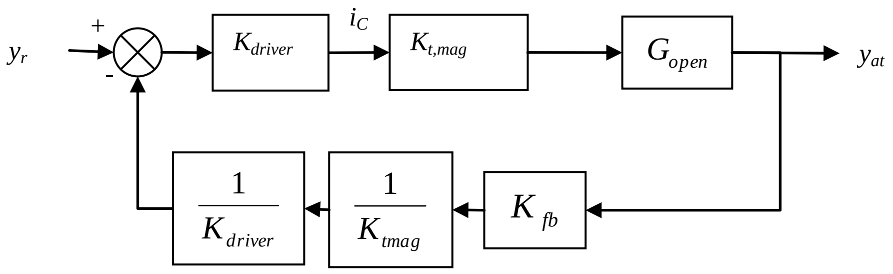

Figure 6. Servo actuator model diagram for identification.

As it is desired to estimate the poles of $G_{simples}$, as described in Eq. (6), the poles was taken as parameters of the closed loop identification. The continuous closes loop system is given by Eq.(11).

$$G_{at} = \frac{y_{at}}{y_r} = \frac{G_{simple}}{1 + K_{fb}.G_{simple}}.K_{tmag}.K_{driver} \tag{11}$$

For the identification numerical procedure, G_{simple} needs to be converted to discrete time transfer function. This was done by taking 1000 samples/s as sampling frequency and a bilinear Tustin continuous to discrete time model conversion method. The discrete time model G_d obtained is given by Eq. (12) which has 3 poles and 3 zeros.

$$G_d = \frac{b_3 z^3 + b_2 z^2 + b_1 z + b_0}{a_3 z^3 + a_2 z^2 + a_1 z + a_0} \tag{12}$$

As K_{fb}, K_{TM}, and K_{driver} are all pure gains, G_{at} in Eq(11) has the same 3 poles and 3 zeros structure.

5. THE EXPERIMENTAL APPARATUS

In order to get de input-output, *r* and *y* measured data it was provided an experiment with a real servo actuator. The testing set are shown in Figure 7. The servo actuator experiment.

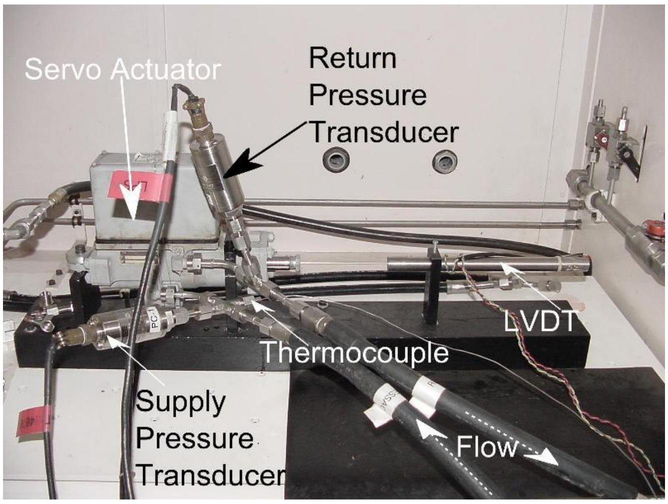

Figure 7. The servo actuator experiment.

A NI DAC system, composed of software LabView and aquisition board was employed to generate the input reference signal, *r*, acquire and record the outputs into a file: y_r , y_{at} (servo actuator rod displacement), P_s (servo actuator supply pressure), P_t (servo actuator return pressure) and T_s (oil temperature at supply line). Figure 8 presents the signals flow, e.g, the signals generated and acquired by the DAC system.

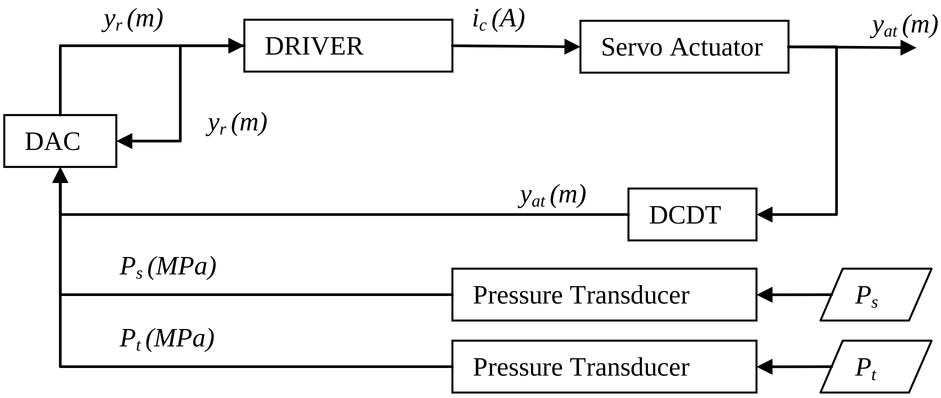

Figure 8. Signals flow of the experiment.

The y_{at} data was measured with a DCDT and the pressure signals with extensometric pressure transducers. The pressures P_S, P_t were acquired in order to estimate their contribution as external disturbance sources to the servo actuator model.

The Figure 9 shows the measured data at engineering units (m, MPa, etc), but physically the acquisition board generate and acquire signals in volts. As the servo actuator is controlled by the command current i_c *(A)*, the input reference voltage (related to y_r) is transformed in i_c by the driver.

In order to have consistent data for identification, it was employed a PRBS as input signal. Following the indication of Aguirre (2000) about the PRBS clock and, after a quick step input test with the servo actuator, it was choose a PRBS generation clock of 60 Samples/s. The data were acquired at a high sample rate and then decimated to improve numerical calculation of parameters.

Figure 9 presents the measured data from the experiment.

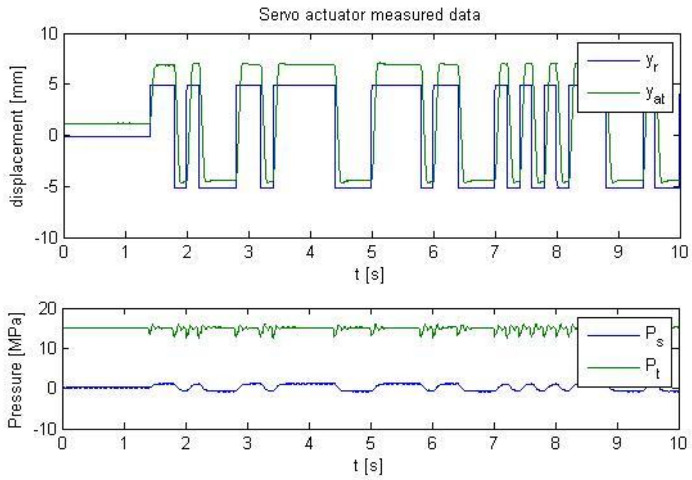

Figure 9. Signals acquired.

6. CLOSED LOOPS IDENTIFICATION RESULTS AND MODEL PERFORMANCE

The PRBS type signal was choose as an estimation and validation signal. The complete acquired signals were divided in two parts, one for estimation and other for validation. As PRBS has a random pattern, it was judged that another kind of signal, like sinusoidal or random noise would be applicable, but it was not necessary.

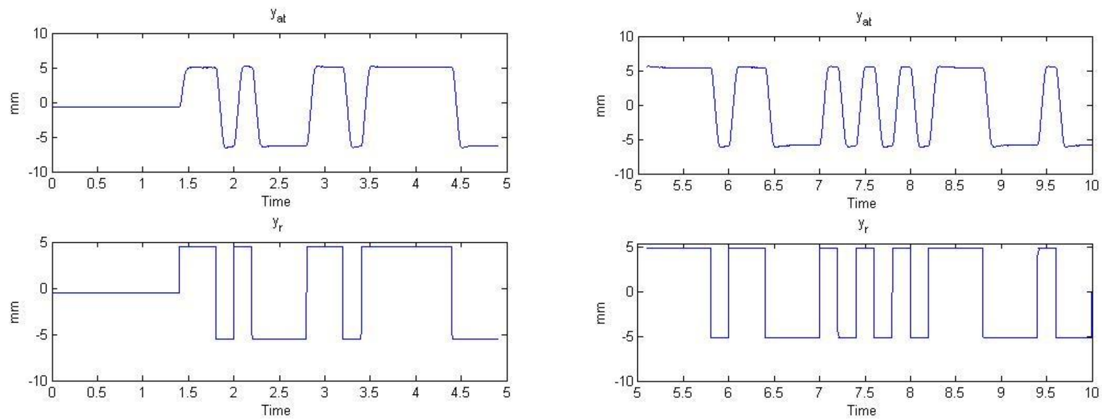

Figure 10. Estimation and Validation data, respectively.

The best close loop estimative $\hat{G}_{at}$ was an ARX model structure with $n_a = 3$, $n_b = 3$ and $n_k = 0$. Figure 11 presents the comparison of the simulated model response with the output measured data.

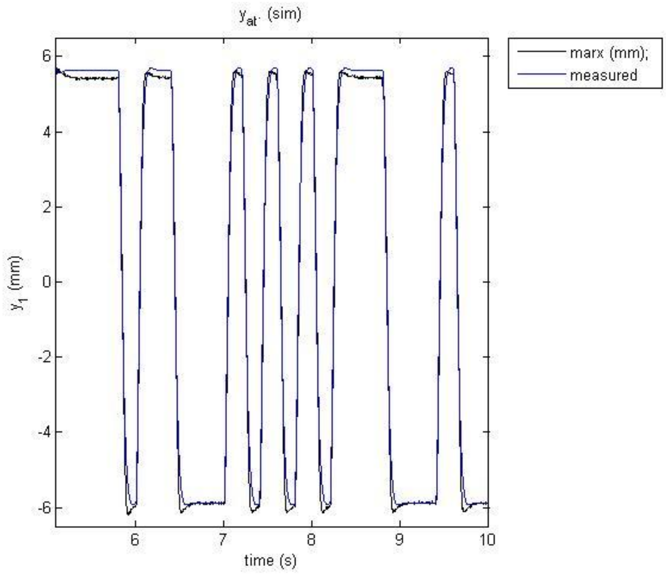

Figure 11. Validation of ARX (3,3,0) structure.

Applying Eq.(8) to $\hat{G}_{at}$ and transforming the result in continuous time transfer function, at first it was estimated the following plant model, $\hat{G}_{simples}$

$$\hat{G}_{simples} = \frac{9900135}{s(s+7104)(s+6019)} \tag{13}$$

As it can be seen a transfer function was obtained with the 3 poles for $\hat{G}_{simples}$ and the expected s = 0 pole was found. But the other two poles were different from what expected, since $G_{simples}$ was expected with two complex poles. The closed the loop obtained with Eq(11) is,

$$\hat{G}_{at} = \frac{9900135}{(s+7104)(s^2+60+1224)} \tag{14}$$

Figure 12 shows the simulation of $\hat{G}_{at}$ using the same y_r as input signal and a comparison of the closed loop model with the measured response.

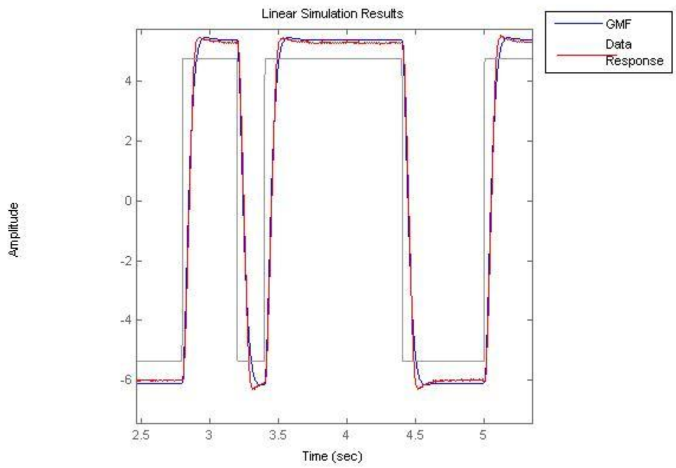

Figure 12. $\hat{G}_{simples}$.response to y_r.

The difference in expected and obtained model structure may be explained by the interpretation that as the servo actuator has no external load, the little mass of the actuator piston, about 0,500 kg, makes the complex poles of hydraulic actuator dynamics not expressive enough for the identification algorithm. The specific servo actuator tested is designed to handle loads of about 1500 kg.

Also, non linear effects, like dry friction on the hydraulic piston, may have changed the real response from the idealized linear one expected in ed (6).

7. CONCLUSIONS

The estimated open loop system for the servo actuator has given a good performance model. The simulated closed loop model response was very close to the real actuator response.

The pole structure obtained was different from the expected. The reason may be associated with the little external load. Future experiments adding a heavier mass load to the actuator and acquiring the modified actuator response may clarify this point.

Also, an analysis of the non linearity influence, such as dry friction, will clarify the differences from obtained and the previously adopted model.

8. REFERENCES

Loureiro,L.P and Goes, L.C.S, 2007,“Modelo Dinâmico de Sistema de Servo Atuação para Sistemas de Controle de Vetor de Empuxo para Foguetes”, CD CIBIM 8 2007, Cusco - Perú, pp 19-41.

Forssell, U. and Ljung, L., 1997, “Properties and Usage of Closed-loop Identificaton Methods”, Linköping University, Linköping, Sweden.

Merrit, H.E. 1966, “Hydraulic Control Systems”, New York: John Wiley & Sons, 358 p.

Aguirre, L. A., 2000; “Introdução a Identificação de Sistemas: técnicas lineares e não-lineares aplicadas a sistemas reais, 1ª Ed.”; Belo Horizonte: Ed. UFMG.

Ljung, L. 1987 , “System Identification, Theory for the User, 1ª Ed.”, Prentice-Hall, Inc.

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