FORCE CONTROL FOR HYDRAULIC FATIGUE TEST MACHINE BASED ON FEEDBACK LINEARIZATION

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Abstract. *Hydraulic actuators are used in many applications because its ability in driving large forces with low inertia and little vibration for a long period of time. However, the main problem in controlling this kind of systems concerns its dynamics, which presents several nonlinearities and parameters variations. Thus, to control hydraulic systems, appropriated nonlinear models and complex control techniques to achieve a stable force regulation with a specified performance are necessary. The purpose of this work is the application of a feedback linearization scheme in the design of a force controller for a hydraulic actuator used in a fatigue test machine. The main control objective considered regards the achievement of sinusoidal force reference tracking. With this aim the internal model principle is applied by using a dynamic compensator (containing imaginary poles with the same frequency of the force reference) in an outer regulation loop. A state feedback control law, considering both the states of the feedback linearized hydraulic system and the ones of the dynamic compensator, is therefore designed in order to stabilize the whole closed-loop system. Experimental model identification and simulation control results are presented and discussed.*

Keywords: *feedback linearization, hydraulics, force control.*

1. INTRODUCTION

Mechanical systems are widely used in industrial automation for different applications with the objective to increment productivity. After the second war hydraulic actuators were used in a wide variety of industrial application because of its ability in driving large forces at high speed, low inertia, little vibration and presents the advantage of working for long periods of time. Unfortunately, the dynamic behavior of these systems are highly nonlinear and its characteristics may change during the performed task, which difficult the control design. The nonlinearities of the system arise from the compressibility of the hydraulic fluid, nonlinear flow/pressure characteristics of the servovalve and friction in the hydraulic cylinder.

The traditional and widely used approach to the control of a hydraulic system is based on the local linearization of the nonlinear model about a nominal operating point (see, for instance (Merritt, 1967) and (Furst, 2001)) which provides only local stability. Also, uncertainties of the hydraulic model increase according to the operational point move away from the nominal point (Cunha, 1997). In order to achieve both stability and reference tracking, several force control approaches have been developed such as the cascade control proposed in (Cunha, 1997) and (Perondi, 2002), using feedback linearization and Liapunov functions. Also Canudas *et al.*(1997) proposed the use of friction compensation while (Chiriboga *et al.*1995) propose input-output feedback linearization control. In Niemela and Virvalo (1994) it is proposed fuzzy logic control.

In this work, the main control objective regards the achievement of sinusoidal force reference tracking. With this aim the internal model principle is applied by using a dynamic compensator (containing imaginary poles with the same frequency of the force reference) in an outer regulation loop. A state feedback control law, considering both the states of the feedback linearized hydraulic system and the ones of the dynamic compensator, is therefore designed in order to stabilize the whole closed-loop system. Experimental model identification and simulation control results are presented and discussed.

2. SYSTEM DESCRIPTION

A hydraulic system is a set of physical elements conveniently associated that, using a fluid as way of energy transference, allows the transmission and control of forces and movements (Linsingen,2001).

The schematic diagram of a typical hydraulic system is sketched in Fig. 1. It consists of a proportional servovalve and a hydraulic cylinder (actuator) coupled to a load. The load may be modeled by a damping coefficient B_e, a spring constant K_e and a mass M_e. The actuator is the element that applies force to the load; it is modeled by a mass M_c and a damping coefficient B_c . The actuator is responsible for the execution of work in association with the linear or oscillating movement. The servovalve with the actuator transforms the hydraulic energy into mechanic energy. The P_s pressure is considered constant (supplied by a hydraulic pump), P_r is the pressure of the pump reservatory and it is also supposed constant. The servovalve controls the flow q_{C1} and $q_{C2}$of the hydraulic fluid which are proportional to the displacement x of the spool and, at the same time, proportional to the electric signal u. F is a force applied by the hydraulic actuator. F_e is the applied force on the load.

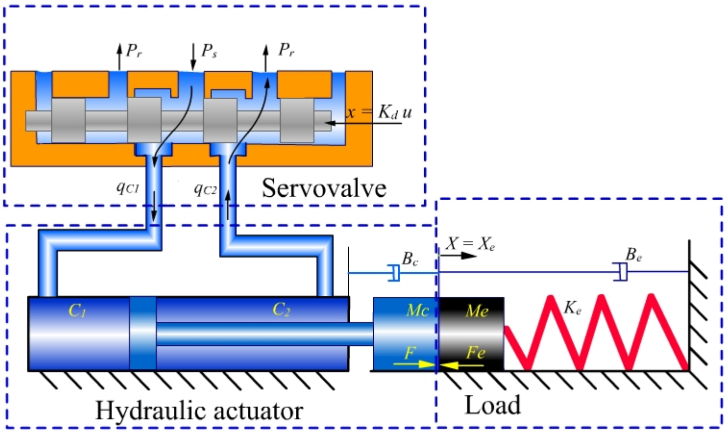

Figure 1. Schematic view of a hydraulic system that interacts with a mechanical load

2.1 Fatigue test machine description

In many laboratories of metallurgy hydraulic fatigue test machines, with electronic control, are used for process developments, educational works and researches. The objective of a fatigue test machine is to submit a test body to dynamical load efforts. The actuator of these machines applies accurate dynamical forces on the test bodies in each experiment. It should be notice that the system parameters depend on the physical characteristics of the test body (e.g. form, size, material,etc.). In Fig. 2 it can be seen a photo of a fatigue test machine and its main parts.

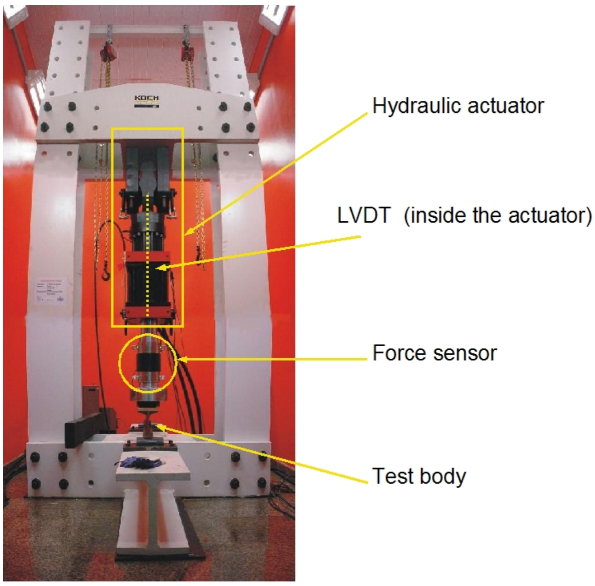

Figure 2. Fatigue test machine and its main components

The displacement sensor can not be seen in Fig. 2 because it is inside of the hydraulic actuator.

The fatigue test machine has a steel structure where it is allocated the hydraulic actuator that is coupled to a test body, which is fixed in the steel structure. This structure must support the dynamic efforts applied to the test body by the hydraulic actuator. The test body is fixed between the structure and the extremity of the actuator.

To locate the piston rod end of the hydraulic actuator close to the test body it is used a LVDT displacement sensor that is located inside the hydraulic cylinder.

Aiming at measuring the force F applied by the hydraulic actuator it was used pressure sensors placed in the chambers C_1 and C_2 of the cylinder.

To implement a force control on a hydraulic system it is necessary to know the displacement of the hydraulic actuator and the force applied on the test body, or the difference of pressure between the chambers of the cylinder. This information is obtained from signals provided by the respective sensors placed in the fatigue test machine. The controller is basically a processor that uses a control algorithm to calculate the value of the electric signal u in function of the signal values proceeding from the fatigue test machine sensors.

3. SYSTEM ANALYSIS AND MODELING

The system dynamics can be studied by describing mathematically the given hydraulic system of Fig. 1. To model this hydraulic system it is used a double acting cylinder coupled to a load and a bidirectional flapper-nozzle servovalve of two stages and 4 ways. In this type of servovalves, the hydraulic flow is proportional to the spool position and the displacement of the spool is proportional to the input command u.

For simplicity of the model it is considered that the servovalve and the cylinder presents symmetrical construction without constructive imperfections, that is, the servovalve has a zero lapped center and there is not leakages in the valve nor in the cylinder. Thus, $q_{C1} = q_{C2} = q_C$, where q_C is the load flow.

The spool displacement x is caused by a solenoid which receives an electrical input signal u from the controller. The relation between x and u could be modeled by a first-order transfer function described in Cunha (2000), but since the maximum close-loop bandwidth of the hydraulic cylinder is less than 1Hz, it is assumed that the spool position x is proportional to the input control u approached by a constant k_d. Note that in such case, the dynamic of the servovalve is fast enough to be neglected, as assumed by Perondi (2002).

Moving the spool to the left $(x > 0)$ would cause a communication between the line of pressure P_s and the chamber C_1 of the cylinder, resulting in a fluid flow q_{C1}. For the same situation $(x > 0)$, it will also have a communication between the line P_r and the chamber C_2 creating a fluid flow q_{C2} of the same value that q_{C1}. As $P_s > P_r$, the pressure P_1 of the chamber C_1 will be greater than the pressure P_2 of the chamber C_2, thus, the actuator will be displaced X in decorrence of a force F equal to the difference of pressure ΔP multiplied by the area A of the piston.

$$\Delta P = P_2 - P_1 \tag{1}$$

Some of the reasons that cause the resultant model of a hydraulic system be represented by nonlinear equation are due to the laminar and turbulent fluids, geometry of the pipes and friction. Parameters of hydraulic systems depend on the relationship between speed and pressure of the fluid as well as the viscosity of the oil that varies considerably with the temperature, (Lischinsky *et al.*, 1999).

The mathematical model used to analyze the hydraulic system is obtained from Bernoulli's law, continuity law and the second law of Newton, as described in Negri (2001). Thus, the mathematical formulation that represents the system is shown in the equations (2) (3) e (4) bellow.

$$\Delta \dot{P} = \frac{\beta v}{\left(\frac{v}{2}\right)^2 - (AX)^2} \left[K_d u \sqrt{(P_s - sign(u)\Delta P)} - A\dot{X}\right] \tag{2}$$

$$F = \Delta P A = (M + M_e)\ddot{X} + (B_c + B_e)\dot{X} + K_e X \tag{3}$$

$$F_e = y = M_e \ddot{X} + B_e \dot{X} + K_e X \tag{4}$$

where v is the volume of the hydraulic cylinder $[m^3]$, β is fluid bulk modulus $[Pa]$, B_c is the constant damping of the actuator $[Nm^{-1}s]$, B_e is the constant damping of the test body $[Nm^{-1}s]$, K_e is the elastic constant of the test body $[Nm^{-1}]$ and F_e is the applied force $[N]$ on the test body that is equivalent to the system output y.

Generally, force controllers use pressure sensors to control hydraulic actuator. For special applications that require enough accuracy in the force control it will be necessary to consider inertia forces and friction forces. It should be noticed that the force F is not the actual force applied on the test body due to friction force effects that appear because of the contact of the piston ring with the interior of the cylinder, (Perondi, 2002).

3.1 System representation in state variables

To develop a state-space representation, it is selected the following state-vector $z = [X\dot{X}\Delta P]$ and the output $y = Fe$. Then, the system equation (2), (3) and (4) can be re-written as follows:

$$\dot{z_1} = z_2 \tag{5}$$

$$\dot{z_2} = -\frac{K_e}{(M+M_e)}z_1 - \frac{(B_e+B_c)}{M+M_e}z_2 + \frac{A}{(M+M_e)}z_3 \tag{6}$$

$$\dot{z_3} = \frac{\beta v}{\left(\frac{v}{2}\right)^2 - (Az_1)^2}\left[K_d u\sqrt{(P_s - sign(u)z_3)} - Az_2\right] \tag{7}$$

$$y = \frac{K_e M}{M+M_e}z_1 + \frac{B_e M + B_c M_e}{M+M_e}z_2 + \frac{AM_e}{M+M_e} \tag{8}$$

4. FEEDBACK LINEARIZATION

The traditional approach to model a hydraulic system is based on local linearization about a nominal operating point of equation (7) (Cunha, 1997) as it is shown in the follow equation:

$$\dot{z_3} = \frac{4\beta}{v}\left(K_Q u - K_C z_3 - Az_2\right) \tag{9}$$

where $K_Q = K_d\sqrt{P_s}$ is called the flow-gain $[m^3(Vs)^{-1}]$ and $K_C = Kd\frac{\sqrt{P_s}}{2P_s}|u|$ is called the flow-pressure coefficient $[m^5(Ns)^{-1}]$. This parameters are estimates for a nominal operating point which is generally considered as the origin. The nonlinearities and parameters variations due to deviations from the nominal operating condition are then considered as plant uncertainties. Applying Laplace transform at equations (5), (6) and (9) allows the calculation of the transfer function of the linearized system as follows.

$$H(s) = \frac{F_e(s)}{U(s)} = \frac{4\beta K_Q}{v}\frac{\left[\frac{AMe}{(M+Me)}s^2 + \frac{2AMKe}{(M+Me)^2}\right]}{s^3 + \left[\frac{4\beta Kc}{v} + \frac{(Bc+Be)}{(M+Me)}\right]s^2 + \left[\frac{4\beta Kc}{v}\frac{(Bc+Be)}{(M+Me)} + \frac{4\beta KcA^2}{v} + \frac{Ke}{(M+Me)}\right]s + \frac{4\beta KcKe}{v(M+Me)}} \tag{10}$$

where $F_e(s)$ is $\pounds\{F_e(t)\}$, $U(s)$ is $\pounds\{u(t)\}$ and $\pounds$ is the Laplace operator. The linearized system around a point can be represented in Fig. 3.

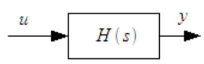

Figure 3. Linearized system around an equilibrium point.

where the output y is the force applied on the test body (F_e) and u is the input of the servovalve.

Feedback linearization is an approach to nonlinear control design. The central idea of this approach is to algebraically transform a nonlinear system into a linear one, (Slotine, 1991). So that, a linear control technique can be applied to the resulting linear model. Feedback linearization differs completely from other traditional local linearization techniques, because it linearize the system for all operating points.

Feedback linearization techniques can be viewed as a way of transforming original system models into equivalents models of a simpler form, canceling the nonlinearities of the original model to linearize it, (Then *et al.*, 1995). Figure 4 presents a block diagram of the feedback linearization technique applied to a nonlinear system. There are two important loops in this scheme, one to perform the plant linearization and another one to control it.

In Fig. 4, $G(s)$ is the transfer function of the linearized system using feedback linearization, and $F(s)$ is the close-loop transfer function of $G(s)$. For unit feedback $F(s) = \frac{G(s)}{1+G(s)}$.

For the cancellation of the plant model nonlinearities it is always recommendable to use its companion form representation. A system is represented in the companion form if:

$$z^{(n)} = f_{(z)} + b_{(z)}u \tag{11}$$

where $z^{(n)}$ is n^{th} derivative of the state z.

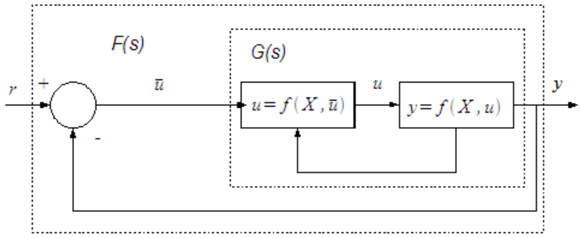

Figure 4. Schematic block diagram for Feedback Linearization.

For the hydraulic system previously modeled there is only one equation (2) thats presents nonlinearities and, at the same time, relates the derivate of one of the states ($\dot{z_3}$) with the input u. This equation can be represented in the following form:

$$\dot{z_3} = f_{(z)} + b_{(z)}u \tag{12}$$

To cancel the nonlinearity it is proposed the following control law:

$$u = \frac{1}{b_{(z)}}\left[h_{(z)} - f_{(z)}\right] \tag{13}$$

where $h_{(z)}$ is a polynomial that depends on the states of the plant and that defines the dynamic of the system. The input u will always be valid for $b_{(z)}$ different of zero. To apply feedback linearization in the nonlinear hydraulic model, $f_{(z)}$ and $b_{(z)}$ will be defined as follows:

$$f_{(z)} = -\frac{\beta v A z_2}{\left(\frac{v}{2}\right)^2 - (Az_1)^2} \tag{14}$$

$$b_{(z)} = \frac{\beta v K_d \sqrt{P_s - sign(u) z_3}}{\left(\frac{v}{2}\right)^2 - (Az_1)^2} \tag{15}$$

Finally, the input u of the system will be computed from the following equation:

$$u_{(z)} = \frac{\left(\frac{v}{2}\right)^2 - (Az_1)^2}{\beta K_d v \sqrt{P_S - sign(u) z_3}}\left[h_{(z)} + \frac{\beta v K_d A z_2}{\left(\frac{v}{2}\right)^2 - (Az_1)^2}\right] \tag{16}$$

It is important to remark that it will not be possible to calculate u when $z_1 = \frac{v}{2A}$ nor when $\Delta P = P_s$, due to indetermination of the equation (15). Thus, it will not be possible to control hydraulic actuator for the maximum force nor for maximum displacement.

The resultant equation of feedback linearization is:

$$\dot{z_3} = h_{(z)} \tag{17}$$

where the resultant dynamic of the linearized system will be defined by the polynomial $h_{(z)}$:

$$h_{(z)} = -k_1 z_1 - k_2 z_2 - k_3 z_3 + \bar{u} \tag{18}$$

where k_i $i = 1, ..3$ are chosen so that the pole location of $h_{(z)}$ is in the left half-plane in order to get an exponentially steady-state dynamic and $\bar{u}$ is the new input of the system.

Thus, the obtained linear system is given by:

$$\begin{bmatrix} \dot{z_1} \\ \dot{z_2} \\ \dot{z_3} \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ -\frac{K_e}{(M+M_e)} & -\frac{(B_c+B_e)}{(M+M_e)} & -\frac{A}{(M+M_e)} \\ -k_1 & -k_2 & -k_3 \end{bmatrix} \begin{bmatrix} z_1 \\ z_2 \\ z_3 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 1 \end{bmatrix} \bar{u} \tag{19}$$

$$y = \begin{bmatrix} \frac{K_e M}{(M+M_e)} & \frac{B_e M - B_c M_e}{(M+M_e)} & \frac{A M_e}{(M+M_e)} \end{bmatrix} \begin{bmatrix} z_1 \\ z_2 \\ z_3 \end{bmatrix} \tag{20}$$

5. REFERENCE TRACKING AND DISTURBANCE REJECTION

With the aim to improve reference tracking and disturbance rejection it is proposed the block diagram of Fig. 5, where it can be observed the iteration between a dynamic compensator and the linearized system $G(s)$. The dynamic compensator projected is based on the internal model principle (Chen, 1999). Now, the linear model studied is the following one:

$$\begin{aligned} \dot{z} &= Az + Bu + B_q q \\ y &= Cz + Eu + E_q q \end{aligned} \tag{21}$$

where q is the disturbance due to noise in the measured signals.

Let as introduce the following dynamic compensator for tracking and/or disturbance rejection:

$$\begin{aligned} \dot{z_d} &= A_d z_d + B_d e \\ y_d &= z_d \end{aligned} \tag{22}$$

where the matrix A_d and B_d depends on the type of reference signal and disturbance signal, and can be calculated as shown in Chen (1999), considering (22) the following augmented system that models the scheme of Fig. 5, is obtained:

$$\begin{bmatrix} \dot{z} \\ \dot{z_d} \end{bmatrix} = \begin{bmatrix} A & 0 \\ -B_d C & A_d \end{bmatrix} \begin{bmatrix} z \\ z_d \end{bmatrix} + \begin{bmatrix} B \\ B_d E \end{bmatrix} u + \begin{bmatrix} 0 \\ B_d \end{bmatrix} r + \begin{bmatrix} B_q \\ -B_d E_q \end{bmatrix} q \tag{23}$$

$$u = \begin{bmatrix} K & K_d \end{bmatrix} \begin{bmatrix} z \\ z_d \end{bmatrix} \tag{24}$$

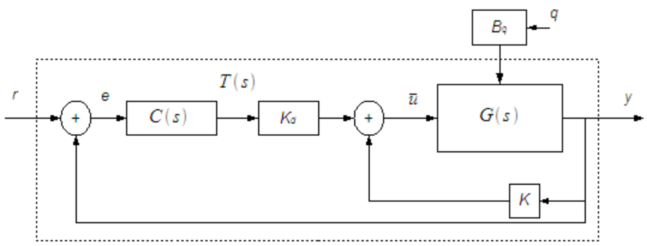

Figure 5. Block diagram of the increased system for reference tracking and disturbance rejection

As mentioned before, to calculate matrix A_d and B_d it is necessary the knowledge of the type of reference signal and disturbance signal. In this work, reference signals are considered sinusoidal and, in the same form, are considered disturbance signals. Thus, it is important to know the frequency of the reference signal (ω_r) and the frequency of the disturbance signal (ω_o). Gains K and K_d are then computed in order to ensure the close-loop stability as well as reference improvement of the resultant closed-loop system represented by the transfer function $T(s) = r(s)/y(s)$.

Another point of view of the effects introduced by the dynamic compensator is that for the obtained $T(s)$ one gets $|T(j\omega_r)| = 1$. Note that if we consider just a simple unit feedback, without (22), as in Fig. 4, one probably obtain $|F(j\omega_r)| \neq 1$ and $\angle F(j\omega_r) \neq 0$. Then the perfect tracking is not achieved.

6. FORCE CONTROLER DESIGN

6.1 Parameter estimation for the hydraulic system

The following parameters were measured:

Table 1. Parameters of the hydraulic system measured

M_e	$825[Kg]$	M	$768.32[Kg]$
β	$6 \cdot 10^8[Pa]$	P_s	$2.06x107[Pa]$
v	$0.0321[m^3]$	B_e	$0[Nsm^{-1}]$
A	$0.1279[m^2]$		

The value used for β was recommended in Merritt (1967). Many experiments were devised to identify other parameters of the model, such K_e and B_c. K_e depends on the test body. It was obtained from a static traction experiment made on the test body measuring force F and position X. Plotting F versus X results an approximated straight line whose slope is the elastic constant K_e. For the hydraulic system study with an specific test body, we obtained $K_e = 163447400[Nm^{-1}]$

Friction has significant effects on controllert's performance, (Perondi, 2002), (Cunha, 1997). One standard experimental method is to model friction as a function of velocity, by measuring the friction force required to move the piston without load at constant velocity and then develop a model such $g(\dot{X})$. A simple open-loop constant control signal was used to move the piston at constant velocity and register the data. Due to the low velocity of the hydraulic system, B_c was calculated making a median of all the values registered for positive velocity and later it was calculated in the same way for negative velocity, obtaining $B_c = 14906[Nm^{-1}s]$ for $X < 0$ and $B_c = -6.6799 \times 10^3[Nm^{-1}s]$ for $X > 0$.

Although equations used to model hydraulic systems are well known, there are no way to measure the parameters K_C , K_Q and K_d, because of dynamic variations along the time and because they depend on the structure and physic properties of each load. In this section it is proposed the use of a statistical technique base on linear regression to estimate the values of the non measurable parameters by data acquisition of several experiments of the physical system, (Montgomery, 1991). In this physical system, the available data to be measured are x, ΔP and u. So, this variables were registered along the time for different experiments. These experiments consists in introducing sine waves with different amplitudes and frequencies in the input u of the fatigue test machine. It was used a steel piece as test body along the experiments. To make the parameter estimation in an off-line data processing it is necessary the knowledge of all the states of the model along the time, thus, F was calculated as $A\Delta P$ and $\dot{X}e\ddot{X}$ were calculated based on x derivative. Then, linear regression technique was used to estimate K_C and K_Q of the linear equation (9) and also to estimate K_d of the nonlinear equation (7). Parameter estimation results were:

Table 2. Parameters of the hydraulic system estimated

K_C	K_Q	Kd
$1.57 \cdot 10^{-12}[m^4N^{-1}s]$	$2.59 \cdot 10^{-5}[m^3s^{-1}v]$	$-1 \cdot 10^{-10}[m^3v^{-1}Pa^{-0.5}]$

Once all the parameters were calculated and estimated, the system transfer function $H(s)$ can be obtained from equation (10) as follows:

$$H(s) = \frac{6.967 \times 10^5 s^2 + 1.361 \times 10^{11}}{50.99s^3 + 1067s^2 + 9.33 \times 10^6 s + 6.467 \times 10^4} \tag{25}$$

This transfer function is used to obtain the poles of the open loop system, considering a linearization around the equilibrium point (origin). These poles are given by:

λ_1 = -0.01
λ_2 = -10.46 + 427.63i
λ_3 = -10.46 + 427.63i

This transfer function has a pair of zeros in the imaginary axis, a real pole near the origin and a pair of complex poles located at the left half side of the complex plane and very far of the real axis, meaning low damping.

Once the pole location of the original open loop system is known, it is suggested to apply Laplace Transform at equations (19) and (20) that allows the calculation of the transfer function $G(s)$ of the linearized system obtained by the application of the feedback linearization, as follows:

$$G_{(s)} = C(sI - A)^{-1}B \tag{26}$$

$$G_{(s)} = \frac{\frac{AM_e}{(M+M_e)}s^2 + \left[\frac{AB_e}{(M+M_e)}\right]s + \frac{K_eA}{(M+M_e)}}{s^3 + \left[\frac{B_c+B_e}{(M+M_e)}\right]s^2 + \left[\frac{k_3(B+B_e)+Ak_2+K_e}{(M+M_e)}\right]s + \frac{Ak_1+k_3K_e}{(M+M_e)}} \tag{27}$$

With the objective to increase the dynamic and the damping ratio of the close-loop system $F(s)$, it is proposed to move the poles far away from the imaginary axis and, at the same time, to approach it to the real axis maintaining the natural frequency ω_n.

6.2 Feedback linearization design

It was devised a script in MATLAB to calculate the coefficients k_i depending of the pole placement desired for the close-loop system. Using the SIMULINK/MATLAB to implement the block diagram of Fig. 6 it was simulated the response of the hydraulic system with feedback linearization.

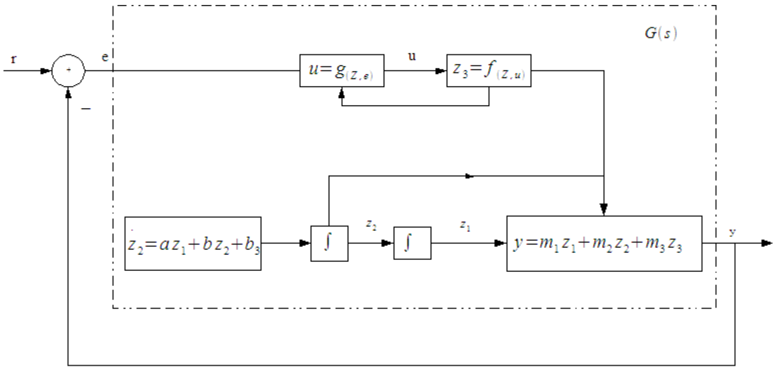

Figure 6. Block diagram implemented to simulate feedback linearization applied to a nonlinear system

The performance of the system was analyzed for many sine wave force references, performing several simulation until finding a pole placement that makes the system faster enough with a damping ratio bigger than its original. For each simulation it was taken into account the input saturation because the system input is limited in ± 10 Volts for the real system.

The obtained pole position and coefficients k_i for the close-loop linearized system are shown in Table 3.

Table 3. Pole position and coefficients k_i calculated

λ_1	-10	k_1	$6.7240 \cdot 10^{12}$
λ_2	$-400 - 50i$	k_2	$1.0625 \cdot 10^{9}$
λ_3	$-400 + 50i$	k_3	830.8397

which corresponds to the following transfer function:

$$F_{(s)} = \frac{0,06622s^2 + 1,312 \cdot 10^4}{s^3 + 810s^2 + 1,705 \cdot 10^5 s + 6,25 \cdot 10^8} = \frac{G(s)}{1 + G(s)} \quad (28)$$

6.3 System dynamic analysis

In Fig. 7 (B) it is observed the simulation response of the close loop linearized system $F(s)$ for a sine wave force reference with a maximum force $F_{max} = 2000[KN]$ and a minimal force $F_{min} = 200[KN]$ and with a frequency ($\omega = 0.088[rads^{-1}]$) without input saturation. The input signal u of this simulation is observed in Fig. 7 (A).

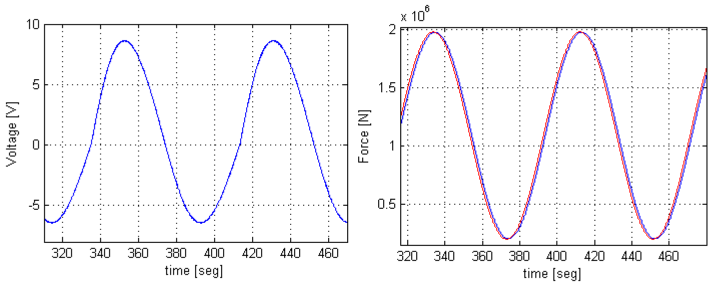

Figure 7. Simulation of the linearized system response for a sine wave reference

Note a little reference tracking erro.

6.4 Tracking and disturbance rejection

With the aim to test the dynamic compensator to improve the reference force tracking of a sinusoidal reference of frequency ω_r and disturbance rejection of frequency ω_0 added to the pressure signal, it was simulated the linearized increased system of equations (22) and (23) $T(s)$ using the block diagram of Fig. 5. The matrix A_d , B_d and E_q of the increased system were calculated in function of ω_r and ω_0. This procedure was based on the internal model principle.

The hydraulic actuator tested on this work with feedback linearization can drive a nominal force of $2500[KN]$ limited by a frequency $\omega_r = 0.088[rads^{-1}]$. In such conditions, the simulations of the linearized system does not presents error tracking to sinusoidal force reference. Due to the natural behavior of hydraulic systems, to obtain sinusoidal responses of higher frequency, the amplitude of the sinusoidal force reference may be decreased to the servovalve input not saturate, this means that the applied force F_e must be decreased when the reference frequency ω_r is increased. Thus, it was simulated the linearized system without dynamic compensator for a sinusoidal reference with minimal force of $150[KN]$ and maximum force of $1500[KN]$ to increase the reference frequency at $\omega_r = 0.35[rads^{-1}]$, taking care that system input does not saturate. Hence, Fig. 8 (A) shows the input of the system simulated (servovalve input) and Fig. 8 (B) shows the reference force tracking of the simulated response which presents erro tracking. Then, the linearized system $G(s)$ was simulated using a dynamic compensator $C(s)$ designed to track references with $\omega_r = 0.35[rads^{-1}]$ and reject a sinusoidal disturbance of frequency $\omega_0 = [370rads^{-1}]$ (noise that usually are induce by electric line). The improvement of the resultant system is observed in Fig. 8 (D) which presents null tracking erro for steady state. Figure 8 (C) shows the system input obtained with the dynamic compensator. It can be seen that the input amplitude has increased, comparing with Fig. 8 (A), because of the dynamic compensator improves reference tracking.

The gains K_c used to the feedback of the dynamic compensator were calculated to make the dynamic compensator faster enough to reject disturbance signals up to $\omega_0 = 370[rads^{-1}]$ and to stabilized the close-loop system $T(s)$. The pole position of the dynamic compensator is shown as follows:

$\lambda_{c1} = -20$
$\lambda_{c2} = -25$
$\lambda_{c3} = -30$
$\lambda_{c4} = -35$

Other simulation was made for the same conditions but adding sinusoidal noise, injected in the signal pressure sensor how is shown in Fig. 9 (C). The frequency of the disturbance was $\omega_0 = 370[rads^{-1}]$ with an amplitude of 1 per cent of the reference amplitude obtaining reference tracking null observed in Fig 9 (B). Simulation of the system input, shown in Fig. 9 (A) present longer amplitude and harmonics of frequencies ω_0, thus, dynamic of the servovalve may be considered. This simulation were made for another frequencies $\omega_0 < 370[rads^{-1}]$ obtaining reference null tracking erro. Thus, using the dynamic compensator, the reference tracking and the disturbance rejection for disturbances of frequencies less than ω_0 are improved. In Fig. 9 (D) can be observed a zoom of the Fig. 9 (B).

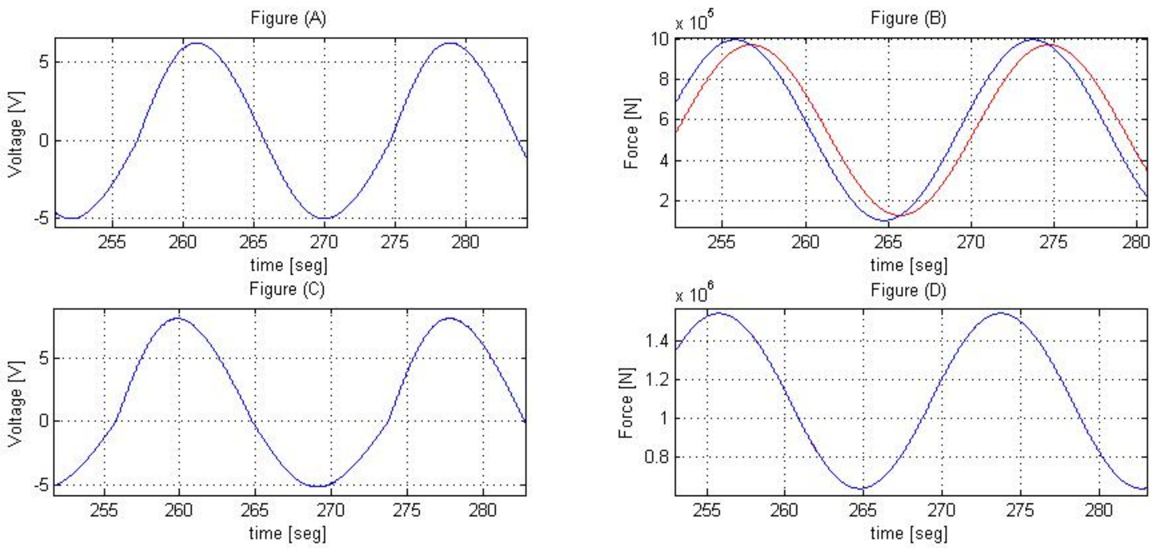

Figure 8. simulation of reference tracking without dynamic compensator

Parameter uncertainties were also introduced in simulations, varying up to 50 per cent of the coefficients used in the nonlinear model of the system. The system output responses were achieved reference tracking null for uncertainties in any parameter, but for some parameters like A and K_d system input was considerable increased its amplitude to compensate this uncertainties. Thus, parameters A and K_d may be carefully measure and estimated in order to not compromise the performance of the controller.

After several simulations made with the dynamic compensator for different disturbance amplitudes it was observed that the reference force tacking error of the close loop system does not change but the dynamic compensator increase the system input amplitude to compensate the disturbances as shown in Fig. 9(A), compared with Fig. 8(C). Then, for larger

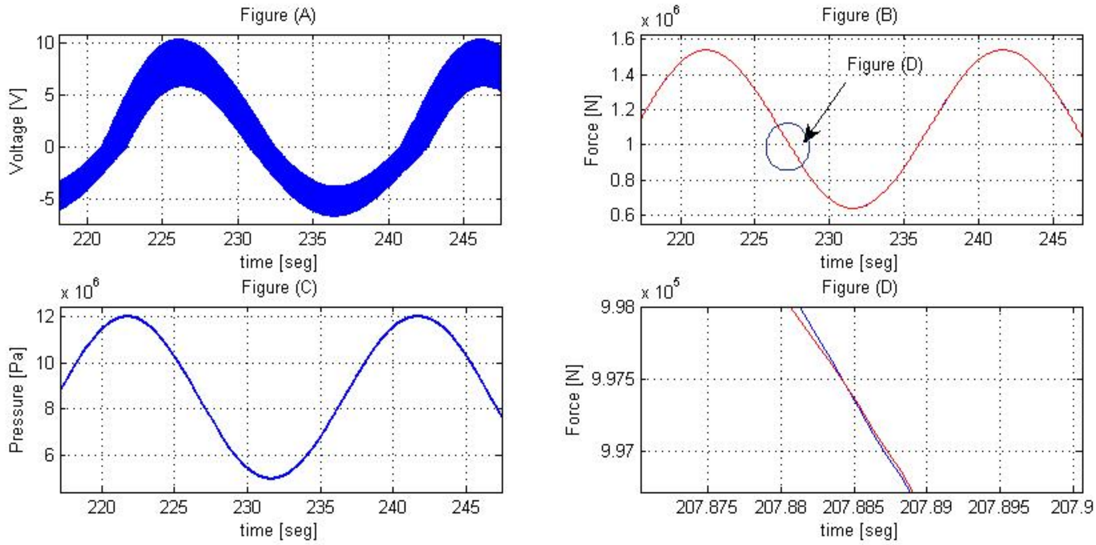

Figure 9. simulation of the input system with dynamic compensator and disturbance in the pressure sensor signal

disturbances the system input may be saturated, which can difficult the control.

7. CONCLUSION AND PROPOSALS FOR FURTHER WORK

The propose of this work was the application of a feedback linearization scheme in the design of a force controller for a hydraulic actuator used in a fatigue test machine. The main control objective considered regards the achievement of sinusoidal force reference tracking. With this aim the internal model principle is applied by using a dynamic compensator in an outer regulation loop.

Considering several simulation it was observed that the control law improves disturbance rejection of the hydraulic system and perform reference force tracking for sinusoidal references. The dynamic compensator increases the amplitude of the hydraulic system input for disturbance rejection, thus, in order to avoid input saturation in a fatigue test machine it is recommended the selection of the frequency ω_r in function of the reference amplitude. Parameter uncertainties were also introduced in simulations observing good force tracking for closed-loop system but, system input amplitude must be increased to compensate this uncertainties. It is important a good parameter estimation to avoid the performance degradation of the controller.

For disturbance rejection, it was shown in simulations that the input system presented harmonics of frequencies around ω_0, thus, servovalve dynamic may be considered.

As future work it is proposed to implement this control law in the electronic controller of the hydraulic system to obtain experimental results. Another proposal is the simulation of disturbance rejection considering the servovalve dynamics.

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