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Modeling of an Electrohydraulic Servo System with Friction Load Based on Experiments

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--*ABSTRACT*--
The paper presents research on modeling of an electrohydraulic servo system with friction load by using experimental data. According to the step command input signal on the system and the velocity output of the cylinder piston, the order and the delay time of the servo system model were determined. And then the discrete-time linear models with and without friction load were identified by means of the Least Squares (LS) method. Finally, the validation of the constructed model was performed and results shown the proposed models to be excellent.

KEYWORDS: *Electrohydraulic servo system, friction load, system identification, modeling*

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I. INTRODUCTION

Hydraulic systems are one of important power sources in modern industry, principally because they have a high power/mass ratio, fast response and high stiffness [1]. Therefore, investigating the servo control of position, velocity or force outputs should be of great interest to both the academic and industrial fields [2, 3]. However, the friction existence makes the servo control become more difficult and complex [4, 5]. And also, it is one of the common nonlinearities in the servo systems [6, 7].

This paper emphasizes on constructing the model of the electrohydraulic servo system with friction load. At first, experiments of an electrohydraulic servo system with the step command input signal and the velocity response output were implemented to obtain input and output data of the system. And then, according to the input signal and the response output signal of the system, the system models without and with friction load were identified by means of the Least Squares (LS) method. In order to validate the estimated model, other sets of measured data were applied to it. Results show that the predictive error is small, which testifies the availability of the estimated model.

II. EXPERIMENT DESCRIPTION

The layout photo and scheme of the experiment are illustrated in Fig. 1. Experiments were implemented with the step voltage input on the servovalve (2) and the velocity output of the cylinder piston (3). The computer produces the control signal to control cylinder motion by means of the DAQ board, amplifier and servovalve. The cylinder drives slip plane (6) which contacts with another friction material (7 rubber in the experiment). When the cylinder piston (3) moves, friction between the slip plane (6) and the friction material (7) can be measured by strain gauge (5) and sliding velocity also can be attained by the linear velocity transducer (4 LVT) at the same time. The load cell (8) is used to measure the normal force which is produced by the adjustable helical structure (9). The friction force, the piston velocity and the normal force are measured via LabVIEW platform.

III. MODELING WITHOUT FRICTION LOAD

It is possible to derive the meaningful transfer function for the electrohydraulic servo system and quantities of research works have been done. Unfortunately, the electrohydraulic servo system actually is a complex system and has many nonlinear characteristics which are significant in its operation [6]. As a result, it is very complicated that the system model would require to be derived precisely and completely. However, it is found in the practical application and measurement that some simplified models possess enough precision to describe the system dynamic characteristics. The servovalve is the most important and complex component of the electrohydraulic servo system. Generally, the whole features of the system depend upon that of the servovalve. The flow control servovalve was adopted in the experiment. Within the frequency range to about 50 Hz the servovalve could be expressed by the following first-order transfer function as shown in Eq. (1).

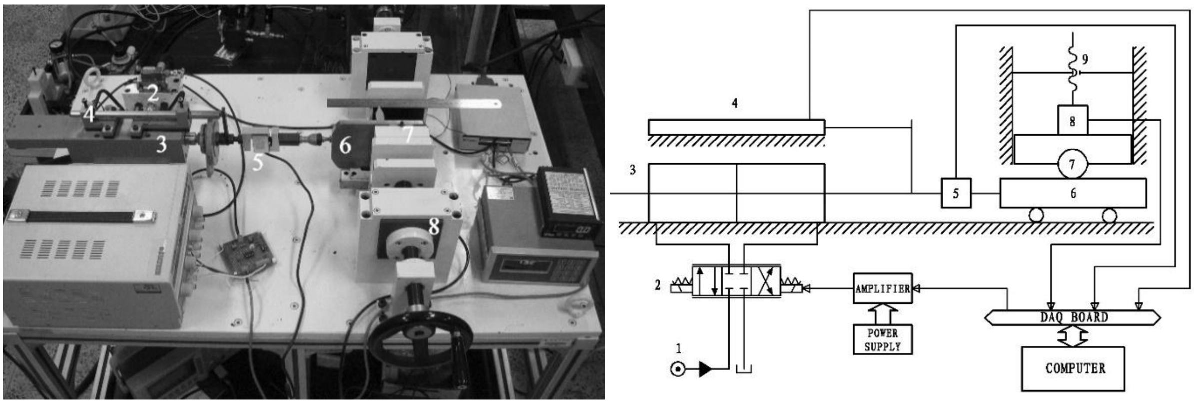


Fig. 1 Experiment scheme
1 hydraulic source, 2 servovalve, 3 double action double rod cylinder, 4 linear velocity transducer (LVT), 5 strain gauge, 6 slip plane, 7 friction material, 8 load cell, 9 helical structure

$$\frac{Q_L(s)}{I(s)} = K\left(\frac{1}{1+\tau s}\right) \tag{1}$$

where

Q_L: servovalve control flow

I: torque motor current

K: servovalve static flow gain at zero load pressure drop

τ: apparent servovalve time constant

If it is necessary to represent servovalve dynamics through a wider frequency range, a second-order response could be used as below.

$$\frac{Q_L(s)}{I(s)} = \frac{K}{1+\left(\frac{2\xi}{\omega_n}\right)s+\left(\frac{s}{\omega_n}\right)^2} \tag{2}$$

where

ω_n: apparent natural frequency

ξ: apparent damping ratio

In this paper the two-stage flow control servovalve is considered as a third order system with respect to flow rate output. And under the condition of no load, namely no load pressure, the cylinder is considered as a proportional transfer function with flow rate input and piston velocity output. Thereby, based on this assumption the block diagram of the whole electrohydraulic servo system can be described as Fig. 2 [8].

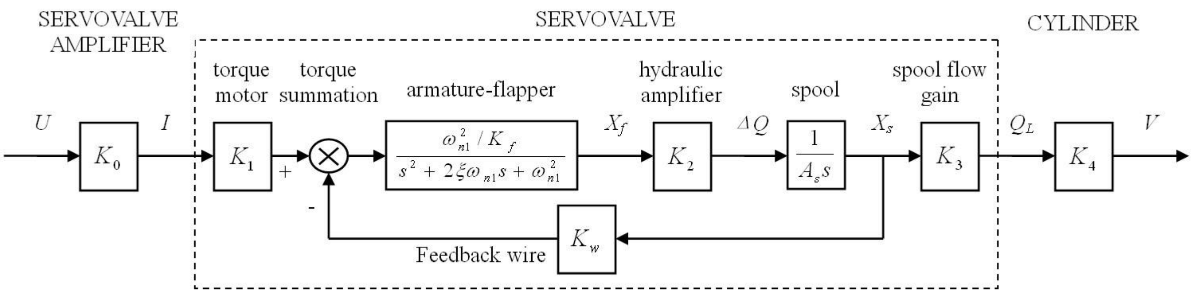


Fig. 2 A block diagram of the electrohydraulic servo system without load

Therefore, the entire transfer function of the system is

$$\frac{V(s)}{U(s)} = \frac{K_0K_1K_2K_3K_4\omega_{n1}^2}{K_fA_ss^3+2\xi\omega_{n1}K_fA_ss^2+K_fA_s\omega_{n1}^2s+K_2K_w\omega_{n1}^2} \tag{3}$$

where

K_0: servovalve amplifier gain

K_1: torque motor gain
ω_{n1}: natural frequency of first stage
K_f: net stiffness on armature/flapper
ξ: damping ratio of first stage
K_2: hydraulic amplifier gain
A_s: spool end area
K_w: feedback wire stiffness
K_3: flow gain of spool/bushing
K_4: cylinder gain without load

The system identification method [3 9] is adopted in the experiment to build system model. In the system identification there are many types of excitation signal, such as impulse, step, sinusoid or pseudo random signal. The unit step (1volt voltage input) as excitation signal for the servovalve and the relative response, the piston velocity as shown in Fig.3, are utilized for modeling the electrohydraulic servo system.

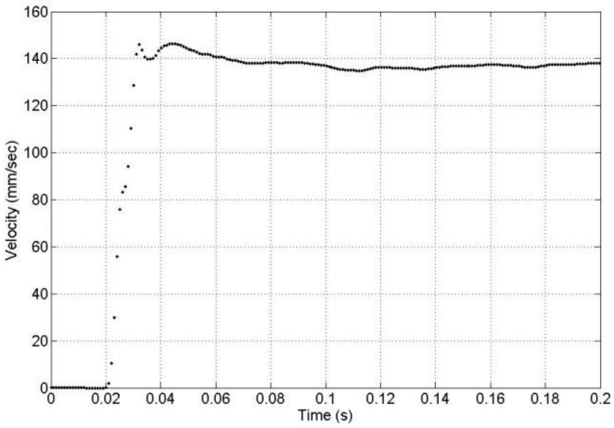


Fig. 3 The piston velocity response of the electrohydraulic servo system without friction load

It is considered that the system is represented by a linear difference equation of the following form.

$$y(k) + a_1 y(k-1) + \cdots + a_n y(k-n) = b_1 u(k-d-1) + \cdots + b_n y(k-d-n) \tag{4}$$

where $y(k)$ and $u(k)$ denote the output (the piston velocity) and input (the voltage acting on the servovalve), respectively, and d is the delay time.

Firstly, the system order and the delay time should be determined. The delay time was approximately estimated to be 0.02sec in terms of the response curve shown in Fig. 3. Assuming this delay time, in order to find the proper model order, Akaike's Information Criterion (AIC) is adopted to compare the differences among various orders based on the estimated model. The comparison results for the different order models are given in Table 1.

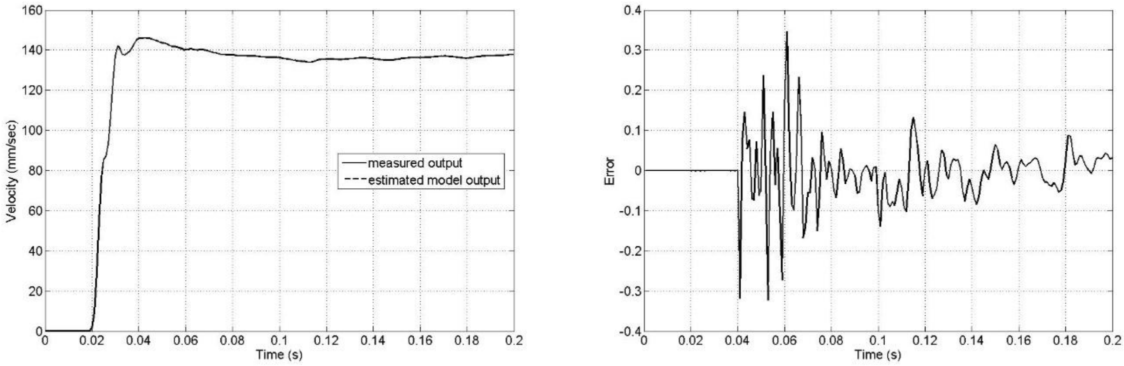
Table 1 Model order determination

Order	1	2	3	4	5
AIC	6.928	2.131	0.545	0.124	0.070

From the Table 1 results, it seems that the third order model is suitable for the electrohydraulic servo system because further increasing the model order could not result in significant improvement in the model predictive performance. By means of process models function in Identification Toolbox of MATLAB, the delay time is further determined to be 0.037sec. This delay time accounts for the system's hysteresis characteristics, especially the servovalve. Thereby, by using the least squares (LS) method, the model of the electrohydraulic servo system without load is estimated as the following form:

$$\begin{aligned} y(k) = 2.009\,y(k-1) - 1.293\,y(k-2) + 0.2719\,y(k-3) + 0.5545\,u(k-38) \\ + 0.5545\,u(k-39) + 0.5545\,u(k-40) \end{aligned} \tag{5}$$

Based on above modeling, it is more necessary and significant to validate the estimated model. A new set of measured response data with step input was applied to the estimated model. The comparison and error between the measured output and the estimated model output are shown in Fig. 4. It is found from Fig.4 that the prediction using the estimated model can reproduce the measured data to a great extent, which shows that the estimated model could correctly describe the electrohydraulic servo system.



(a) output (b) error

Fig. 4 Comparison of the measured output and the estimated model output

Other measured response data (with 200 sampling points) are also applied the estimated model. Residuals Sum of Squares (RSS) is shown in Table 2. The small RSS suggests that the prediction error is small.

Table 2 Residuals Sum of Squares (RSS)

No.	1	2	3	4	5	6
RSS	1.0073	1.0102	1.0603	0.9855	1.0896	1.0228

IV. MODELING WITH FRICTION LOAD

If friction acts in the electrohydraulic servo system, the system model will become more complicated than what was discussed in previous section. The assumption that load pressure is zero is not correct any more because the fiction load exits in the system. The relation between the spool displacement x_s, servovalve control flow Q_L and load pressure P_L could be presented in the form as below.

$$Q_L = C_d w x_s \sqrt{\frac{1}{\rho}(P_s - \frac{x_s}{|x_s|} P_L)} \tag{6}$$

where

C_d: orifice coefficient

w: spool valve area gradient

ρ: fluid density

P_s: supply pressure

When the cylinder drives the friction load, the Newton's second law is applied to friction and moving of the slip plane as following.

$$F = P_1 A_1 - P_2 A_2 = P_L A = M\dot{v} + F_f \tag{7}$$

where F is the cylinder output force; for the symmetrical doubt action cylinder the areas of two sides A_1 and A_2 are same as A; M is the mass of the slip plane; F_f is the friction force. The following block diagram could describe the relation in Eq (7).

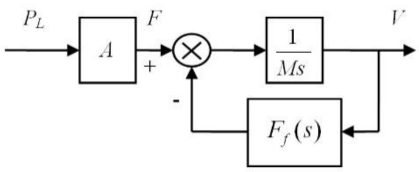


Fig. 5 A block diagram of the friction load

If the normal force is set as 30kgf and the step input signal is also set as 1 Volt, the response curve with the friction load was obtained, which is shown in Fig. 6.

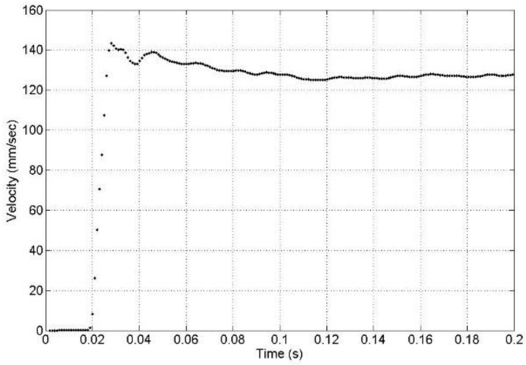


Fig. 6 The piston velocity response with friction load (normal force=30kgf)

Similarly, in light of AIC rule, the AIC values are given in Table 3, assuming that the delay time was 0.037sec. The values in Table 3 are greatly smaller than those in Table 1 because the delay time value estimated in the section 3 is adopted. And it appears that the third order is still the best choice. When the system operates with friction load, though the system characteristics changes to some extent the apparently measured feature does not change too much.

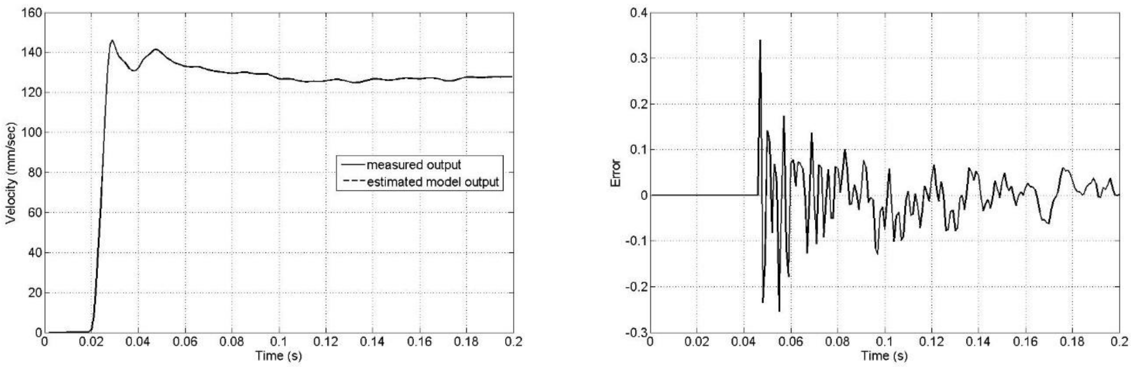
Table 3 model order determination

Order	1	2	3	4	5
AIC	0.0813	0.0201	0.0063	0.0039	0.0021

Based on the third order model, the delay time should be modified further as same as the method used in the section 3. Finally, the delay time is determined as 0.043sec, and is longer than the response without friction load because it needs time to build load pressure to resist the friction load. However, it is difficult directly to distinguish deference between Fig. 3 and Fig. 6. Until now the system model with friction load as below can be estimated by means of LS method.

$$
\begin{aligned}
y(k) = 2.415\,y(k-1) - 2.099\,y(k-2) + 0.6723\,y(k-3) + 0.491u(k-44) \\
+ 0.491u(k-45) + 0.491u(k-46)
\end{aligned}
\tag{8}
$$

Another set of measured data was applied to the above estimated model. The measured data and the predictive value are shown in Fig. 7.



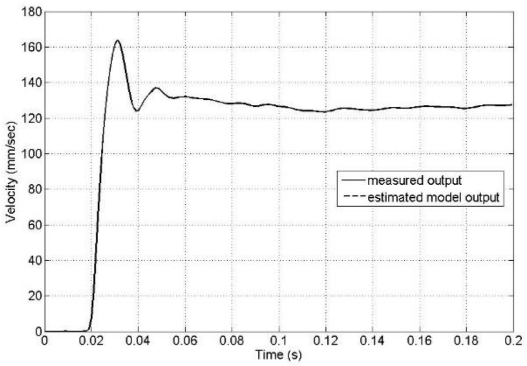
(a) output (b) error

Fig. 7 Comparison measured output and estimated model output (normal force=30kgf)

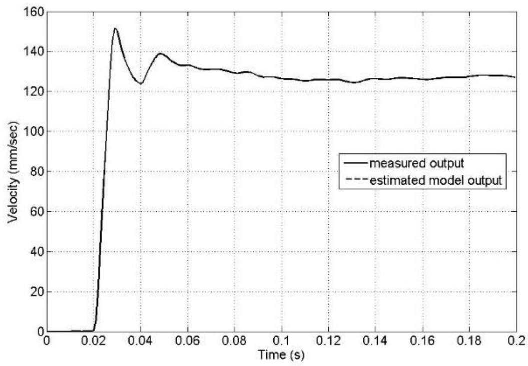
Residuals Sum of Squares of other sets of measured response data (normal force=30kgf) are shown in Table 4. In order to validate the model availability for different friction load, various normal forces were performed in the electrohydraulic servovalve system and the velocity responses under step input were measured as before. The measured data and the predictive value based on the estimated model are shown in Fig. 8.

Table 4 Residuals Sum of Squares (normal force=30kgf)

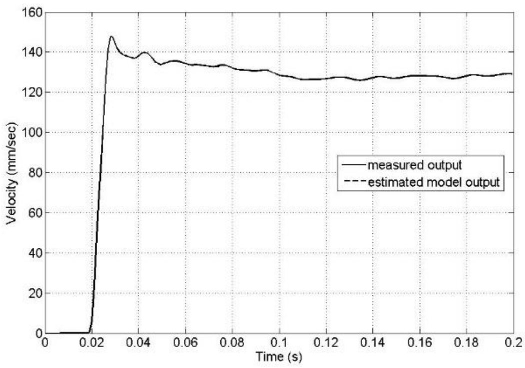
No	1	2	3	4	5	6
RSS	0.8274	0.7358	0.7649	0.9991	0.9790	0.7372



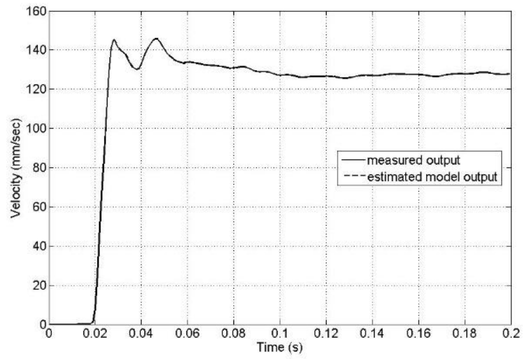
(a) normal force=10kgf RSS=0.8503



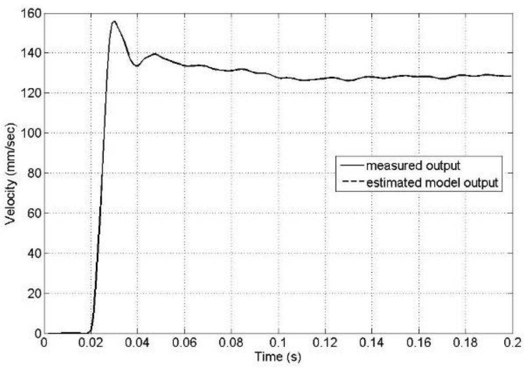
(b) normal force=15kgf RSS=1.0375



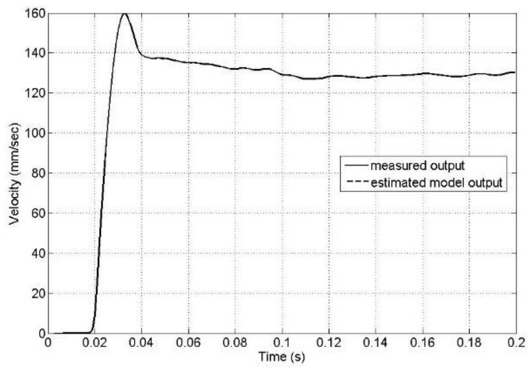
(c) normal force=20kgf RSS=0.7536



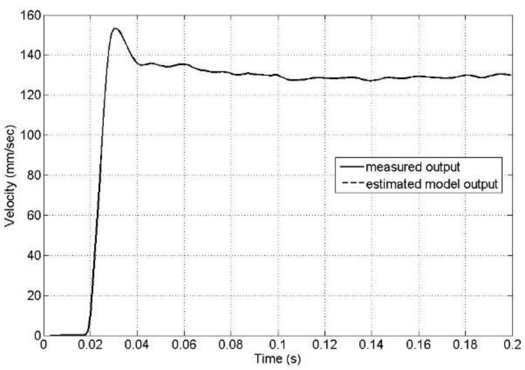
(d) normal force=25kgf RSS=0.8399



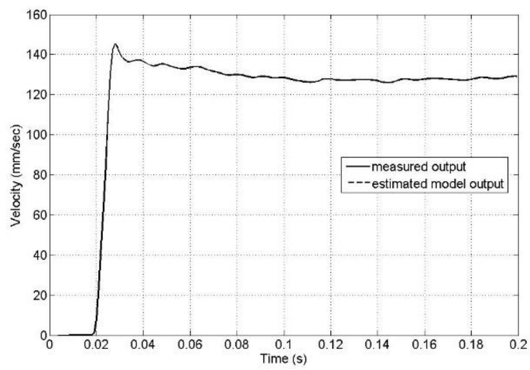
(e) normal force=30kgf RSS=0.8366



(f) normal force=35kgf RSS=0.7339



(g) normal force=40kgf RSS=0.7482



(h) normal force=45kgf RSS=0.5674

Fig. 8 Comparison measured output and estimated model output with various friction load

Fig. 8 shows that the predictive error is small, which suggests that the estimated model is correct. At the same time, overshoot exists in the responses under step stimulus, shown in Fig. 8. This phenomenon is because the friction load exits. As we know, the static friction is larger than the kinetic friction. Adequate load pressure should be attained to resist the static friction. Once moving occurs the friction goes down drastically, which is likely to produce overshoot. However, there should not be big velocity overshoot in that the flow control servovalve was utilized in the experiment.

V. CONCLUSION

In this research the cylinder velocity responses with the step voltage input show that friction is uncertain factor in the electrohydraulic servo system, which makes the system take on nonlinear characteristics. The electrohydraulic servo system theoretically is high order model however the third order model can rightly describe the system from the practically measured results. The system delay time was further affirmed. After the linear difference model was obtained by using LS method, it was applied to various friction loads and the predictive value is close to the measured one. The small predictive error shows the model availability.

The servovalve characteristics mainly determine that of the system. Therefore, in order to compensate the friction load in the system in future research, the servovalve model would be investigated more in terms of servovalve structure.

REFERENCE

[1]. H.E. Merritt, Hydraulic Control Systems, John Wiley & Sons, pp. 85, 212-217, 1967,

[2]. D.H. Kim, T.C. Tsao, “A Linear Electrohydraulic Servovalve Model for Valve Dynamics Sensitivity Analysis and Control System Design”, Dynamic Systems, Measurement, and Control, ASME, Vol.122, pp. 179-186, 2000

[3]. Y.Y. Chen, P.Y. Huang, J.Y. Yen, “Frequency-Domain Identification Algorithms for Servo Systems with Friction”, Control System Technology, IEEE, Vol.10, No.5, pp. 654-665, 2002

[4]. L. Ravanbod-Shiraziand, A. Besancon-Voda, “Friction Identification Using the Karnopp Model”, Applied to an Electropneumatic actuator, Systems and Control Engineering, Vol.217, pp.123-138, 2003

[5]. S. Tafazoli, C.W. de Sliva, P.D. Lawrence, “Tracking Control of an Electrohydraulic Manipulator in the Presence of Friction”, Control System Technology, IEEE, Vol.6, No.3, pp.401-411, 1998

[6]. K. Ziaei, N. Sepehri, “Modeling and Identification of Electrohydraulic servos”, Mechatronics, PERGAMON, No.10, pp.761-772, 2000

[7]. Z.G. Xie, G.L. Tao, H. Zhou, “Study on Cylinder at very low Velocities”, Proceedings of the 4th International Symposium on Fluid Power Transmission and Control., International Academic Publishers, pp.358-361, 2003

[8]. MOOG inc. controls division, Transfer Function for Moog Servovalves, Technical Bulletin

[9]. R. Pintelon, J. Schoukens, System Identification: a Frequency Domain Approach, IEEE Press, 2001