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COMPARISON BETWEEN TWO FRICTION MODEL PARAMETER ESTIMATION METHODS APPLIED TO CONTROL VALVES

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Abstract: Friction in the control valve is one of the major sources of oscillations and abnormal operations in process control loops. Models of control valves can be used to diagnose abnormal operations as well as to compensate for such undesirable effects. This work describes two methods used to estimate the parameters of a Karnopp friction model applied to control valves. The methods were tested using simulated data of three valves with different friction levels. The advantages of each one are emphasized.

Keywords: Parameter estimation, Control valves, Friction, Karnopp model.

1. INTRODUCTION

Abnormal situations in process plant operations are commonly encountered and can occur due to: poor controller tuning, oscillatory disturbances, poor process and control system design and control valve friction (Choudhury *et al.*, 2005).

Control valves are the most employed actuators in chemical engineering control applications. However, as they are the only moving part of the loop, they are prone to friction, which causes oscillations or steady-state errors in the stem position. Such undesirable occurrences affect the overall profitability of the process (Kaiyhan and Doyle, 2000).

Friction models can be used for detection and to deal with friction effects through model based compensation (Armstrong-Hélouvry *et al.*, 1994; Kaiyhan and Doyle, 2000).

Several friction models were proposed in the literature for different applications: from the models based on one (Stenman *et al.*, 2003) or two parameters (Choudhury *et al.*, 2005), to complex models with seven parameters (Armstrong-Hélouvry *et al.*, 1994). Among them, a good compromise between relative simplicity and correct representation of most friction effects is the model proposed by Karnopp (1985). Nevertheless, as stated by Ravanbod-Shirazi and Besançon-Voda (2003), there are still insufficient methods for parameter estimation of the Karnopp model.

In this paper, two promising methods for estimating the parameters of a Karnopp model, applied to a control valve, are compared. The data set used in the analysis was derived from simulations.

The main contributions of this paper are: [1] the adaptation of the method proposed by Ravanbod-Shirazi and Besançon-Voda (2003) for single action pneumatic actuators and [2] a comparison between this method and a novel analytic one proposed in Garcia (2007). The paper is organized as follows: the pneumatic actuator and the Karnopp friction model is presented in section 2. Two identification methods for control valves are briefly described in section 3. Both methods are used to identify the parameters of three valves with different friction levels on section 4. Finally, in section 5, conclusions are drawn.

2. THE VALVE MODEL

A dynamic model for the control valve is given by the force balance equation:

$$m \cdot \ddot{x}(t) = F_{ext}(t) - F_{spring}(t) + F_f(t) \qquad (1)$$

where:

m: mass of the valve moving parts.

$x(t)$: stem position.

$F_{ext} = S_a \cdot P(t)$: external force applied by the actuator; S_a is the diaphragm area and $P(t)$ is the air pressure.

$F_{spring} = k \cdot x(t)$: spring force; k is the spring constant.

F_f: friction force.

The Karnopp model includes static and moving friction, depending on the velocity of the moving parts. The expression for the moving friction is given by the first line of Eq. (2); in this case, the friction force is a static function of the stem velocity.

$$F_f = \begin{cases} -F_c \cdot \operatorname{sgn}\left[\dot{x}(t)\right] - F_v \cdot \dot{x}, & |\dot{x}| \geq DV \\ -\left(F_{ext} - k \cdot x\right), & \begin{cases} if \quad |\dot{x}| < DV \quad and \\ \quad |F_{ext} - k \cdot x| \leq F_S \end{cases} \\ -F_S \cdot \operatorname{sgn}\left(F_{ext} - k \cdot x\right), & \begin{cases} if \quad |\dot{x}| < DV \quad and \\ \quad |F_{ext} - k \cdot x| > F_S \end{cases} \end{cases} \tag{2}$$

where:

F_c: Coulomb friction coefficient.

F_v: viscous friction coefficient.

F_s: static friction coefficient.

DV: limit velocity.

If the magnitude of the stem velocity is smaller than the limit velocity DV, the model considers the stem velocity to be null. The second line in Eq. (2) is the case when the valve is stuck and the third one represents the situation at the instant of breakaway. In both situations, the friction force F_f is a saturated version of the external force.

3. PARAMETER ESTIMATION METHODS

In the work of Cheok et al. (1988), the parameters of a symmetrical Karnopp model are identified for a servomotor, using nonlinear optimization. Ravanbod-Shirazi and Besançon-Voda (2003) developed an identification method for an electro-pneumatic actuator where most of the parameters are estimated using a linear regression. This method, with slight modifications to include the spring force, is described in the following subsection.

3.1 The linear-regression based method

The identification procedure is divided into three steps. In the first one, a parameter vector is defined as:

$$\theta = \begin{bmatrix} m & F_v & F_c & k \end{bmatrix} \tag{3}$$

When the stem velocity is greater than DV, Eq. (1) can be rewritten as:

$$\begin{aligned} F_{ext}(t) &= m \cdot \ddot{x}(t) + F_c \cdot \operatorname{sgn}\left[\dot{x}(t)\right] \\ &\quad + F_v \cdot \dot{x}(t) + k \cdot x(t) \end{aligned} \tag{4}$$

Since Eq. (4) is linear with respect to the parameter vector θ, it can be estimated using the least squares:

$$\begin{aligned} \hat{\theta} &= \arg\min_{\theta} \sum_t \left[F_{ext}(t) - \hat{F}_{ext}(t)\right]^2 \\ &= \arg\min_{\theta} \sum_t \left[F_{ext}(t) - \varphi(t)\theta^T\right]^2 \end{aligned} \tag{5}$$

where the regression vector $\varphi(t)$ is:

$$\varphi(t) = \begin{bmatrix} \ddot{x}(t) & \operatorname{sgn}\left[\dot{x}(t)\right] & \dot{x}(t) & x(t) \end{bmatrix} \tag{6}$$

But the periods in which Eq. (4) is applicable are unknown, i.e. the limit velocity DV that characterizes the stem movement is not known a priori. To deal with this problem, a variable $\delta\upsilon(s)$ is defined, so that:

$$\delta v(s) = s \cdot \frac{|\dot{x}|_{\max}}{Z}, \quad s = 1, 2, \ldots, S \tag{7}$$

where: $Z >> 1$ and $S < Z$.

For each value of $\delta\upsilon(s)$, the regression vector $\varphi(t)$ and $F_{ext}(t)$ are chosen from the observed data, so that the condition $|\dot{x}| > \delta\upsilon(s)$ is fulfilled and then the parameter vector is estimated solving Eq. (5). The behavior of these estimations, as index s increases is: [1] For $\delta\upsilon(s) << DV$ the values of the estimated parameters vary significantly for different values of $\delta\upsilon(s)$, because data from periods in which Eq. (4) is not applicable are used in the estimations. [2] When $\delta\upsilon(s)$ approaches DV, the estimations come closer to their true value and do not change significantly, even for $\delta\upsilon(s)$ slightly greater than the limit velocity DV. This behavior was expected, because in this situation the regression model $\varphi(t) \cdot \hat{\theta}^T$ approximates $F_{ext}(t)$.

In the second step of the identification procedure, the limit velocity DV is estimated. A way of determining DV is applying the value of $\delta\upsilon(s)$, for which the estimations of the parameter vector do not change significantly. But as argued by Ravanbod-Shirazi and Besançon-Voda (2003), such approach can lead to overestimation in the presence of measurement noise.

An estimation of DV can be seen as the smallest velocity at which the estimated moving friction $\hat{F}_{mv}$ characterizes the estimated friction force $\hat{F}_f$, i.e., defining the error:

$$\varepsilon\left[\delta v(s)\right] = \left\{\hat{F}_f\left[\delta v(s)\right] - \hat{F}_{mv}\left[\delta v(s)\right]\right\}^2 \tag{8}$$

where:

$$\begin{aligned} \hat{F}_f\left[\delta v(s)\right] &= F_{cs} \cdot \operatorname{sgn}\left[\delta v(s)\right] + F_{vs}\delta v(s) \\ \hat{F}_{mv}\left[\delta v(s)\right] &= \hat{F}_c \cdot \operatorname{sgn}\left[\delta v(s)\right] + \hat{F}_v \delta v(s) \end{aligned}$$

The estimated limit velocity $\hat{DV}$ is considered to be the smallest value of $\delta\upsilon(s)$ for which the error calculated by Eq. (8) is close to zero. Notice that $\hat{F}_c$ and $\hat{F}_v$ have already been estimated and F_{cs} and F_{vs} are estimations obtained at the first step for each value of $\delta\upsilon(s)$.

In the last step, F_s can be estimated using a nonlinear optimization algorithm, defined as follows:

$$\hat{F}_s = \arg\min_{Fs} \left\{ \sum_t \left[\dot{x}(t) - \hat{\dot{x}}\left(t, \hat{\theta}, \hat{DV}, F_s\right)\right]^2 \right\} \tag{9}$$

To find $\hat{F}_s$, the Karnopp model is simulated using the parameters identified at previous steps and varying F_s, in order to minimize the prediction error of the stem velocity in Eq. (9).

However, to simulate the stick-slip phenomenon of a control valve with high friction, a simulation step of 10^{-6} s (Garcia, 2006) is required, which would demand excessive computational load. Furthermore, to avoid problems of local minima, F_s was estimated using the last line of Eq. (2), which corresponds to a situation where the stem is in imminence of moving. This procedure applies the same strategy used in the method proposed by Garcia (2007), which is described in the following subsection. This form of calculation was not described in the paper of Ravanbod-Shirazi and Besançon-Voda (2003), but it is an improvement suggested here.

3.2 Garcia's method

This method (Garcia, 2007) is totally based on the next force balance equation:

$$S_a \cdot P = \left[F_c + \left(F_s - F_c \right) \cdot \mathrm{e}^{-(\dot{x}/v_s)^2} \right] \cdot \mathrm{sgn}\left(\dot{x}\right) + k \cdot x + F_v \cdot \dot{x} + m \cdot \ddot{x} \tag{10}$$

where:
v_s = Stribeck velocity

It is presumed that the input signal (actuator pressure) is a triangular wave.

The procedure presented next is applicable to valves with stiction. For valves without stiction, see (Garcia, 2007).

3.2.1 Estimation of k

When the stem is moving with constant and positive velocity, if two points of the actuator pressure and of the stem position are taken, the results are:

$$S_a \cdot P_1 = k \cdot x_1 + F_c + F_v \cdot \dot{x}$$

$$S_a \cdot P_2 = k \cdot x_2 + F_c + F_v \cdot \dot{x}$$

Subtracting the first from the second expression:

$$S_a \cdot \left(P_2 - P_1 \right) = k \cdot \left(x_2 - x_1 \right) \tag{11}$$

As P_1, P_2, x_1 and x_2 are measured and S_a is available, it is possible to estimate k.

3.2.2 Estimation of F_s

It is supposed that the initial valve position is $x = 0$ and in imminence of moving. From Eq. (10), it results in:

$$S_a \cdot P = F_s \tag{12}$$

It is not obligatory to perform this calculation with $x = 0$. It just simplifies the calculation. It is possible to estimate F_S in any position that the valve is stuck. To analyze this case, see (Garcia, 2007).

3.2.3 Estimation of F_V and m

To calculate these parameters, it is necessary to analyze any valve slipping. In this case, Eq. (10) becomes:

$$S_a \cdot P = k \cdot x + F_c + F_v \cdot \dot{x} + m \cdot \ddot{x} \tag{13}$$

It is necessary to estimate $\dot{x}$ and $\ddot{x}$ in three consecutive position points. From Eq. (13) results in:

$$S_a \cdot P_1 = k \cdot x_1 + F_c + F_v \cdot \dot{x}_1 + m \cdot \ddot{x}_1 \tag{14}$$

$$S_a \cdot P_2 = k \cdot x_2 + F_c + F_v \cdot \dot{x}_2 + m \cdot \ddot{x}_2 \tag{15}$$

$$S_a \cdot P_3 = k \cdot x_3 + F_c + F_v \cdot \dot{x}_3 + m \cdot \ddot{x}_3 \tag{16}$$

Subtracting (15) from (14):

$$S_a \cdot \left(P_1 - P_2 \right) = k \cdot \left(x_1 - x_2 \right) + F_v \cdot \left(\dot{x}_1 - \dot{x}_2 \right) + m \cdot \left(\ddot{x}_1 - \ddot{x}_2 \right) \tag{17}$$

Subtracting (16) from (15):

$$S_a \cdot \left(P_2 - P_3 \right) = k \cdot \left(x_2 - x_3 \right) + F_v \cdot \left(\dot{x}_2 - \dot{x}_3 \right) + m \cdot \left(\ddot{x}_2 - \ddot{x}_3 \right) \tag{18}$$

Equations (17) and (18) represent a system where F_V and m are unknown. Solving it, results in F_V and m.

3.2.4 Estimation of F_C

It is assumed that the points are taken for the situation in which the valve movement is caused by the second or further triangular wave. It is necessary to collect one point, when the pressure is increasing and the velocity is positive and constant or when the pressure is decreasing and the velocity is negative and constant. The results are:

$$S_a \cdot P_i = k \cdot x_i + F_c + F_v \cdot \dot{x}_i \tag{19}$$

where the suffix "*i*" means increasing

$$S_a \cdot P_d = k \cdot x_d - F_c + F_v \cdot \dot{x}_d \tag{20}$$

where the suffix "*d*" means decreasing

As k and F_v have already been estimated, it is possible to apply equation (19) or (20) to calculate F_c.

4. RESULTS

The methods mentioned in the previous section are applied to estimate the parameters of three valves with increasing friction level: vendor, nominal and rough (Kaiyhan and Doyle, 2000). The valve characteristics used in simulations are:

$$m = 1.3608kg \qquad k = 52538 \, {}^{N}\!/_{m}$$

$$S_a = 0.06452m^2 \qquad DV = 1.524 \times 10^{-4} \, {}^{m}\!/_{s}$$

The friction parameters are shown in Table 1.

Table 1- Values used in simulations

	Vendor	Nominal	Rough
F_v [N·s/m]	612.944	612.944	1225.9
F_c [N]	44.4822	1423.4	2241.1
F_s [N]	44.4822	1708.1	2668.9

As in Garcia (2007), to identify the parameters of the friction models, the actuator pressure $P(t)$ was slowly cycled from 0 to 82737 Pa (0 to 12 psi), as depicted in Figure 1. The actuator pressure and the stem position are measured using a 1 ms sampling period.

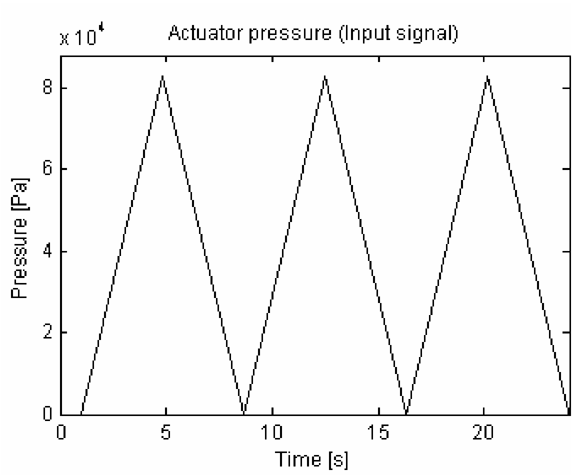

Fig. 1. Actuator pressure that is the input signal used in the parameter estimation procedures.

The values of S and Z, used in the first step of the linear regression based method, defined in Eq. (7) are, respectively, 1200 and 1500. The behavior of the estimated parameter vector for different values of $\delta\upsilon(s)$ is showed in Figure 2.

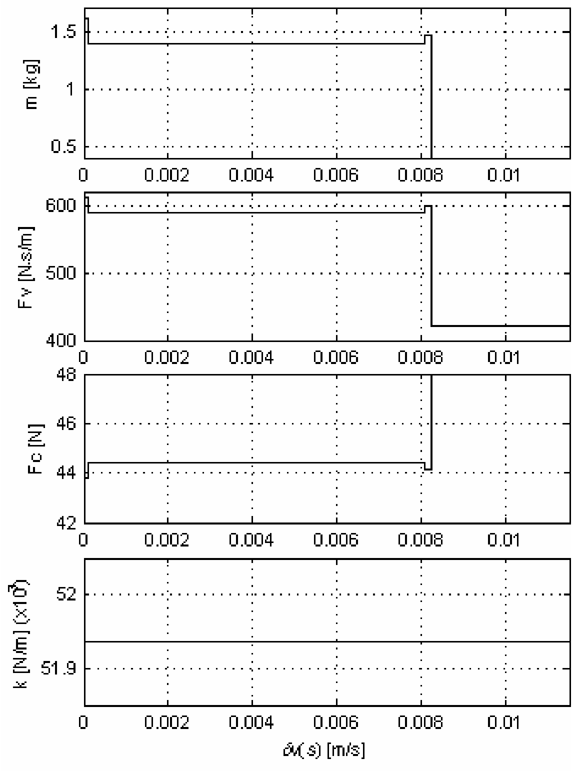

Fig. 2. Behavior of $\hat{\theta}$ for the vendor valve for different values of the variable $\delta\upsilon(s)$ defined in Eq. (7).

Notice that for velocities from $2\text{x}10^{-4}$ to 0.008 the estimations do not change considerably, hence the identified parameters: $\hat{m}, \hat{F}_v$, $\hat{F}_c$ and $\hat{k}$ are considered to be the ones obtained in this range.

The estimated parameters for the vendor valve are summarized in Table 2, where method [1] is the one proposed by Ravanbod-Shirazi and Besançon-Voda (2003) while method [2] is the one proposed by Garcia (2007).

Table 2- Results obtained for the vendor valve.

	Estimated by method [1]	|Error| (%)	Estimated by method [2]	|Error| (%)
m [kg]	1.3972	2.68	1.5038	10.51
k [N/m]	51937	1.144	52544	0.0114
DV [m/s]	1.016e-4	33.3	-	-
F_v [N·s/m]	589	3.91	623	1.65
F_c [N]	44.39	0.21	44.04	0.99
F_s [N]	44.39	0.202	44.04	0.989

Comparing the errors for each method, one can verify that method [1] was better to estimate the mass of the valve moving parts and the static and Coulomb friction coefficients. On the other hand, using the Garcia method, the errors of the viscous friction coefficient and the spring constant were lower.

To compare the estimated parameters from each method, their response for an actuator pressure excitation in the form of a sinusoidal wave (different from the input signal of the data set used for estimations) with 8 s period and varying from 0 to 70050 Pa is depicted in Figure 3, which shows that both models describe quite well the behavior of the vendor valve. In addition, notice that despite the large error in the estimation of the limit velocity DV, the effect in the response is negligible, as had been stated by Karnopp (1985).

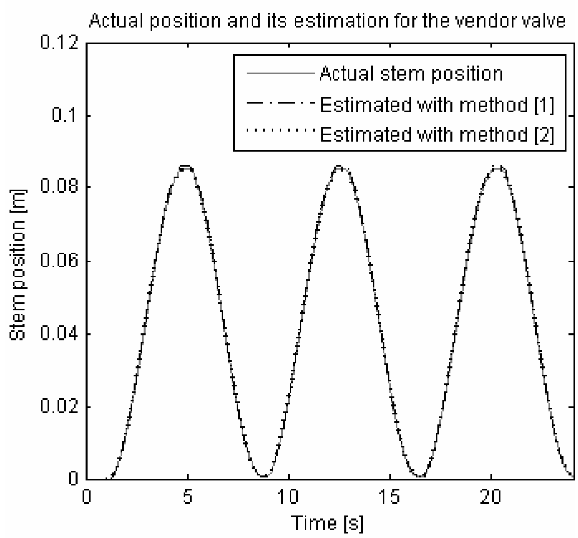

Fig. 3. Comparison of the actual stem position with the simulated one using parameters obtained by both methods.

Table 3 shows the estimated parameters for the nominal valve, which presents higher friction coefficients than the vendor valve. In this case, the estimation errors of the mass increased for both methods, with regard to the ones for the vendor valve. Another relevant aspect is the huge error (1656.7%) obtained from the estimation of the limit velocity using the linear regression based method.

Table 3- Results obtained for the nominal valve.

	Estimated by method [1]	|Error| (%)	Estimated by method [2]	|Error| (%)
m [kg]	1.068	21.52	1.0575	22.29
k [N/m]	51937	1.144	52542	0.0076
DV [m/s]	2.54e-2	1656.7	-	-
F_v [N·s/m]	555.9	9.31	565.5	7.73
F_c [N]	1408.4	1.06	1424.5	0.0773
F_s [N]	1684.7	1.37	1708.2	0.005

Except for the poor estimation of the limit velocity by method [1], both methods provided a good representation of the real parameters, with method [2] exhibiting lower errors, except for the mass m. This can be seen in Figure 4, which shows the identified model responses for an actuator pressure varying sinusoidally, with amplitude 70050 Pa and period 8 s.

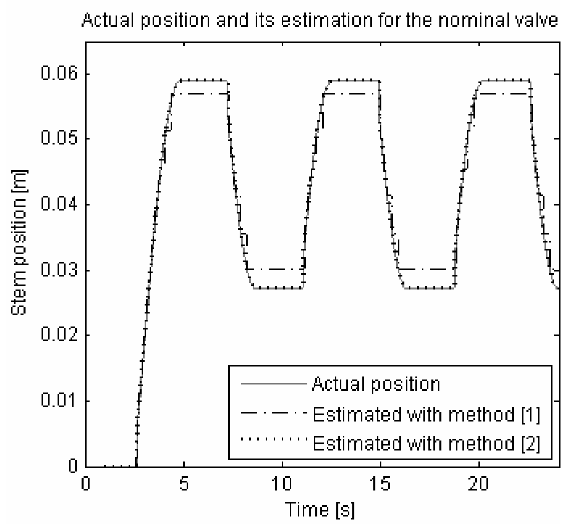

Fig. 4. Actual stem position compared to the position calculated by the simulation of the identified models using a sinusoidal excitation.

The last tested valve has a friction level higher than the others. The estimation of the parameter m for both methods is still worse than the ones obtained for the vendor and nominal valves. Conversely, the other parameters presented good estimations (Table 4).

Table 4- Results obtained for the rough valve.

	Estimated by method [1]	|Error| (%)	Estimated by method [2]	|Error| (%)
m [kg]	0.832	38.86	0.9528	29.98
k [N/m]	51937	1.144	52538	≈0
DV [m/s]	9.58e-3	61.86	-	-
F_v [N·s/m]	1163.7	5.07	1176.1	4.06
F_c [N]	2199.9	1.84	2225.2	0.0495
F_s [N]	2636.9	1.83	2668.5	0.65

Figure 5 shows the response of the identified models when excited by a sinusoidal wave, with amplitude 82665 Pa and period 8 s. For the rough valve, as well as for the other ones tested (vendor and nominal), both identification methods were able to estimate models that describe the actual valve behavior with adequate accuracy.

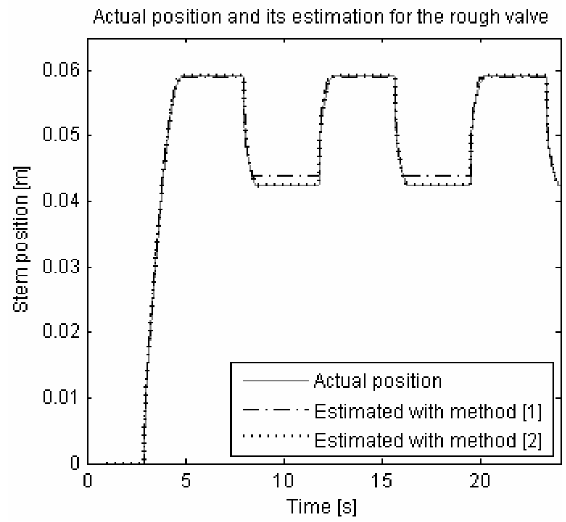

Fig. 5. Comparison of the calculated stem position generated by both methods for the rough valve with the actual stem position.

The amplitude of the actuator pressure $P(t)$ used in this simulation was increased to ensure stem movement even under the higher friction level presented by the rough valve. In terms of parameter estimation, when the friction level is increased, the mass of the moving parts is the most poorly estimated, for both identification methods.

The method proposed by Ravanbod-Shirazi and Besançon-Voda (2003) resulted in poor estimations for the limit velocity DV. Nevertheless, the results show that this fact did not cause significant deviations of the estimated model behavior from the actual one. For the nominal valve, for example, the error was 1656.7% but, in spite of that, the model response using the estimated parameters (Figure 4) was similar to the actual stem position. Hence, the inability of the analytical Garcia´s method in estimating the limit velocity is a negligible disadvantage. This point had already been mentioned by Karnopp (1985).

Another relevant point is that, regardless of the estimation method, when analyzing the errors in the estimated parameters, the most critical ones are: m and F_v. This result was expected, because such parameters depend on stem acceleration and velocity, which are calculated through the stem position. The acceleration $\ddot{x}(t)$ and the stem velocity $\dot{x}(t)$ used in both estimation methods were calculated using the backward difference approximation. In this case, accurate values were calculated because of the small sampling period used in the simulations.

5. CONCLUSIONS

Both methods presented similar results; however, according to the situation in which the estimation procedure is to be applied, some differences arise: the method [1] can be used with data obtained from the valve operating in closed-loop. Another advantage of this method is that, as it is implemented by means of computer software, where the user is only responsible for a few decisions, the entire procedure is automatic. On the other hand, the method proposed by Garcia (2007) is simpler to use and the parameters of the Karnopp friction model are calculated analytically through force balance equations, providing accurate results.

Another aspect concerning the linear regression based method is that identification techniques based on least squares algorithms provide well-known interesting statistical properties that lead to accurate estimations even when measurement noise is present. Furthermore, although in this work the actuator pressure was a triangular excitation (Figure 1), in order to compare the identification methods with the same input signal, the application of rich inputs could increase the reliability of the method [1] as stated by Ravanbod-Shirazi and Besançon-Voda (2003).

Although the main concepts of the estimation method proposed by Ravanbod-Shirazi and Besançon-Voda (2003) have been used in this work, the authors propose that the static friction F_s coefficient should be estimated using force balance equations to reduce an excessive computational load demand.

The authors are now working on model parameter estimation for real valves.

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