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FRICTION MODEL PARAMETER ESTIMATION FOR CONTROL VALVES

Claudio Garcia

Polytechnic School, University of São Paulo

Abstract: This paper presents a method to estimate offline the parameters of the friction model of Karnopp, applicable to a control valve. The method is simple and easy to use. It is presumed that the valve is submitted to a triangular shape input signal and that the stem position and the actuator pressure are measured. The results are fairly good. It is possible to estimate the three friction coefficients (viscous - F_v, Coulomb - F_c and static - F_s), the mass m of the moving parts of the valve and the coefficient k of the spring.

Keywords: friction, stiction, model, parameter estimation, control valves.

1. INTRODUCTION

This paper is concerned with friction identification in control valves. It is a relevant subject in the following areas:
a. friction model-based compensation;
b. quantification of stiction in control valves; and
c. simulation of control valves with friction parameters that are close to the real values.

Some models (Garcia, 2007) were studied to find the best one to describe the behavior of friction in control valves: Karnopp (Karnopp, 1985), Seven Parameters (Armstrong-Hélouvry *et al.*, 1994), Lugre (Canudas de Wit *et al.*, 1995), Stenman (Stenman *et al.*, 2003) and Choudhury (Choudhury *et al.*, 2005). In order of complexity, the Karnopp model demands more parameters than the models of Stenman and Choudhury, but not as many as Seven Parameters and Lugre models. Preliminary validation results of Karnopp's model performed by the author are very satisfactory when applied to describe the behavior of friction in real control valves. An important feature present in the model of Karnopp is its ability to deal with the situations where the moving parts of the valve are almost stopping. It is able to simulate a "true" zero velocity (Ravanbod-Shirazi; Besançon-Voda, 2003). So, the model of Karnopp was chosen to describe friction in control valves, as the model that has a satisfactory accuracy with not as many parameters. The purpose of this paper is to estimate offline the parameters of this model. This kind of estimation might be used in valve bench tests, as specified in (ISA, 2000a; ISA, 200b).

A proposal to estimate online the parameters of the Karnopp model has been published by (Ravanbod-Shirazi; Besançon-Voda, 2003), applicable not to a control valve but to an electro-pneumatic actuator. Nevertheless, the estimation of the static friction coefficient F_s by this method consumes a long time, due to the heavy computational load. The idea of the method proposed here is to be very simple and applicable to cases where stiction is present or not. To apply the method, it is just necessary to supply the stem position (model output) and the pressure in the actuator (model input) of a pneumatic (spring/diaphragm) valve. The data is obtained in a test in which the actuator pressure varies in a triangular form and the input and output signals are collected each 1 ms (1 kHz). All the data used in this paper was collected from simulations (Garcia, 2007).

The necessary data is presented in the next section. In section 3, the method is presented. In section 4 it is applied to two valves with stiction and in section 5, to a valve without stiction. In section 6 conclusions are drawn.

2. DATA ACQUISITION

In order to apply the proposed method, it is necessary to perform a test with the valve in which the actuator pressure is changed from 0 to 100% in ramp and back from 100% to 0% also in ramp. This kind of test, with the triangular wave input signal, is normally called valve signature and it usually starts with the valve completely open (or closed). It should be repeated at least two times. It is necessary to measure the actuator pressure and the stem position.

Figure 1 presents the actuator pressure, Figure 2 presents the stem position and Figure 3 presents a graph of actuator pressure versus stem position for a simulated control valve with high friction (rough valve - Kayihan; Doyle, 2000). The pressure is supposed to vary in the range from 0 to 12 psi (0-82737 Pa).

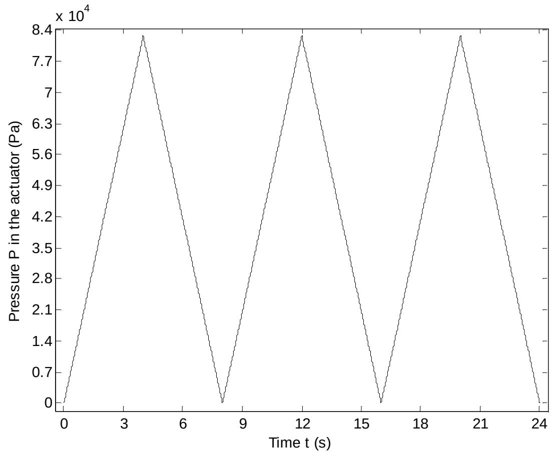

Fig. 1. Actuator pressure.

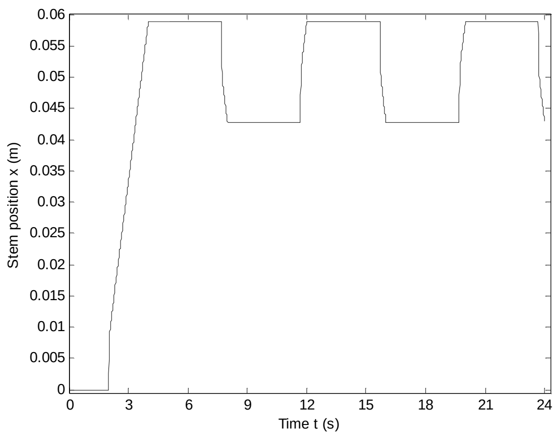

Fig. 2. Stem position – rough valve.

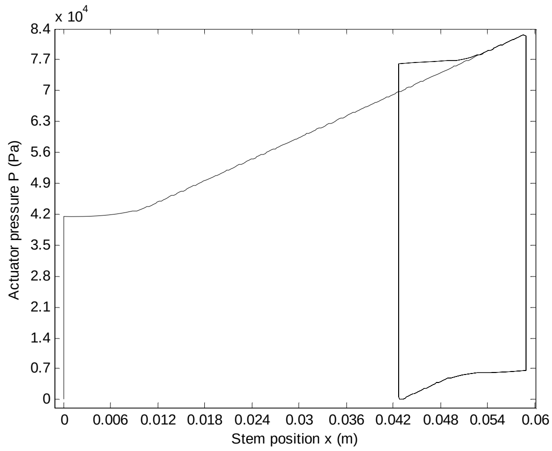

Fig. 3. Actuator pressure X stem position – rough valve.

3. METHOD DESCRIPTION

The considered model is:

$$m \cdot \ddot{x} = F_{pressure} - F_{spring} - F_{friction} - F_{fluid} - F_{seat}$$

where:

$m =$ mass of the valve moving parts (stem and plug)

$x =$ stem position

$F_{pressure} = S_a \cdot P =$ force applied by the actuator, being S_a the diaphragm area and P the air pressure

$F_{spring} = k \cdot x =$ spring force, being k its constant

$F_{friction} =$ friction force

$F_{fluid} = \alpha \cdot \Delta P =$ force due to the fluid pressure drop across the valve, being α the plug unbalance area and ΔP the pressure drop

$F_{seat} =$ extra force required for the valve to be forced into the seat

As it is assumed that the valve is submitted to a bench test (with no fluid), the force F_{fluid} exerted by the fluid is null. For simplicity, it is assumed that F_{seat} is also negligible.

The friction force can be calculated as follows:

$$F_{friction} = \left[F_c + (F_s - F_c) \cdot e^{-(\dot{x}/v_s)^2} \right] \cdot \mathrm{sgn}(\dot{x}) + F_v \cdot \dot{x}$$

where:

$F_c =$ Coulomb friction coefficient

$F_s =$ static friction coefficient

$v_s =$ Stribeck velocity

$F_v =$ viscous friction coefficient

The resulting model is:

$$S_a \cdot P = k \cdot x + \left[F_c + (F_s - F_c) \cdot e^{-(\dot{x}/v_s)^2} \right] \cdot \mathrm{sgn}(\dot{x}) + F_v \cdot \dot{x} + m \cdot \ddot{x} \qquad (1)$$

As it is shown next, the method is totally based on the force balance equation (1). For valves without stiction, the coefficient $F_s - F_c$ is negligible, so equation (1) becomes:

$$S_a \cdot P = k \cdot x + F_c \cdot \mathrm{sgn}(\dot{x}) + F_v \cdot \dot{x} + m \cdot \ddot{x} \qquad (1a)$$

It is presumed that the diaphragm area S_a is available, since it is a parameter that is part of the actuator specification.

3.1 Estimation of k

During a transition between closed and open, when the stem is moving with constant and positive velocity (null acceleration), 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:

$$S_a \cdot (P_2 - P_1) = k \cdot (x_2 - x_1) \qquad (2)$$

As P_1, P_2, x_1 and x_2 are measured, it is possible to estimate k. It is also possible to calculate k taking points when the velocity is constant and negative.

3.2 *Estimation of* F_s for valves with stiction and of F_c for valves without stiction

It is supposed that the initial valve position is $x = 0$ and in imminence of moving. From (1), it results in:

$$S_a \cdot P = F_s \tag{3}$$

As P is measured, it is possible to calculate F_s. For valves without stiction, based on equation (1a), equation (3) becomes:

$$S_a \cdot P = F_c \tag{3a}$$

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. In that case, equation (3) becomes:

$$S_a \cdot P = F_s + k \cdot x \tag{4}$$

if the valve slips with positive velocity or:

$$S_a \cdot P = -F_s + k \cdot x \tag{4a}$$

if the valve slips with negative velocity

As P and x are measured and k has already been estimated, it is possible to calculate F_s. The same reasoning applied to derive equations (4) and (4a) can be applied to F_c instead of F_s, for valves with no stiction.

3.3 *Estimation of* F_v and m for valves with stiction and of m for valves without stiction

To calculate these parameters, it is necessary to analyze any valve slipping. In this case, equation (1) becomes:

$$S_a \cdot P = k \cdot x + F_c + F_v \cdot \dot{x} + m \cdot \ddot{x} \tag{5}$$

The velocity $\dot{x}$ is calculated based on two consecutive position points and the acceleration $\ddot{x}$ is estimated considering three consecutive position points, as follows:

$$\dot{x}(k) = \frac{x(k) - x(k-1)}{\Delta t}$$

$$\ddot{x}(k) = \frac{[x(k) - x(k-1)] - [x(k-1) - x(k-2)]}{\Delta t^2}$$

The precision in the measurement of x is essential, since a fairly good approximation of $\dot{x}$ and $\ddot{x}$ depends on it. It is necessary to estimate $\dot{x}$ and $\ddot{x}$ in three different situations. From equation (5) results:

$$S_a \cdot P_1 = k \cdot x_1 + F_c + F_v \cdot \dot{x}_1 + m \cdot \ddot{x}_1 \tag{5a}$$

$$S_a \cdot P_2 = k \cdot x_2 + F_c + F_v \cdot \dot{x}_2 + m \cdot \ddot{x}_2 \tag{5b}$$

$$S_a \cdot P_3 = k \cdot x_3 + F_c + F_v \cdot \dot{x}_3 + m \cdot \ddot{x}_3 \tag{5c}$$

Subtracting (5b) from (5a):

$$S_a \cdot (P_1 - P_2) = k \cdot (x_1 - x_2) + F_v \cdot (\dot{x}_1 - \dot{x}_2) + \\ + m \cdot (\ddot{x}_1 - \ddot{x}_2) \tag{6a}$$

Subtracting (5c) from (5b):

$$S_a \cdot (P_2 - P_3) = k \cdot (x_2 - x_3) + F_v \cdot (\dot{x}_2 - \dot{x}_3) + \\ + m \cdot (\ddot{x}_2 - \ddot{x}_3) \tag{6b}$$

Equations (6a) and (6b) represent a system where F_v and m are unknown. Solving it, results in F_v and m. When the valve has no stiction, there is no slipping. As, in that case, F_v is estimated by equation (7), to calculate m it is necessary to take three consecutive position points of the signature curve as soon after as there is a stem movement when the valve is stuck. It is not necessary to collect three sets of points, as in equations (5a), (5b) and (5c), but just one and apply it to equation (5).

3.4 *Estimation of* F_c for valves with stiction and of F_v for valves without stiction

In this case, it is assumed that the points are taken for the situation in which the signature curve in Figure 3 is being generated 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:

$$S_a \cdot P_i = k \cdot x_i + F_c + F_v \cdot \dot{x}_i \tag{7}$$

where the suffix "i" means increasing

$$S_a \cdot P_d = k \cdot x_d - F_c + F_v \cdot \dot{x}_d \tag{7a}$$

where the suffix "d" means decreasing

As for valves with stiction, k and F_v have already been estimated, it is possible to apply equation (7) or (7a) to calculate F_c.

For valves without stiction, as k and F_c have already been estimated, it is possible to apply equation (7) or (7a) to calculate F_c.

4. APPLICATION TO VALVES WITH STICTION

In this section, two valves are considered: one with high friction (rough valve) and other with medium friction (nominal valve), as defined in (Kayihan; Doyle, 2000). The valves have $S_a = 100\ \text{in}^2 = 0.06452\ \text{m}^2$ and full stroke of 4 in = 0.1016 m.

4.1 *Rough valve*

By taking two ascending pressure values in figure 3, when $\dot{x}$ is constant and positive, the results are:

$$P_1 = 50056\ \text{Pa} \rightarrow x_1 = 0.01854\ \text{m}$$

$$P_2 = 70120\ \text{Pa} \rightarrow x_2 = 0.04318\ \text{m}$$

Substituting these values in equation (2) results in:

$$k = 52538\ \text{N/m}$$

The real value of k is the same, resulting in a null error.

According to Figure 3, the pressure just before the valve starts moving for the first time is $P = 41369\ \text{Pa}$. Substituting this value in equation (3) results in:

$$F_s = 2669.1\ \text{N}$$

The real value of F_s is 2668.9 N. The resulting error is 0.0075%.

As stated in section 3.2, it is not obligatory to calculate F_s based on the first valve slipping. Next F_s is calculated based on the second valve slipping, in which it occurs with negative velocity. The measured pressure just before the valve starts moving is $P = 6556.9\ \text{Pa}$ and the measured position is $x = 0.05887\ \text{m}$. Replacing these values in equation (4a), as k is available, results in:

$$F_s = 2669.9\ \text{N}$$

As the real value of F_s is 2668.9 N, the resulting error is 0.0375%.

Taking into account the first valve slipping and considering three sets of points:

$$P_1 = 41472\ \text{Pa} \rightarrow x_1(k) = 0.001170\ \text{m}$$
$$x_1(k-1) = 0.000844\ \text{m} \quad x_1(k-2) = 0.000521\ \text{m}$$

It results in:

$$\dot{x}_1 = 0.3260\ \text{m/s} \quad \text{and} \quad \ddot{x}_1 = 3\ \text{m/s}^2$$

The second set of points is:

$$P_2 = 41679\ \text{Pa} \rightarrow x_2(k) = 0.003898\ \text{m}$$
$$x_2(k-1) = 0.003672\ \text{m} \quad x_2(k-2) = 0.003436\ \text{m}$$

It results in:

$$\dot{x}_2 = 0.2260\ \text{m/s} \quad \text{and} \quad \ddot{x}_2 = -10\ \text{m/s}^2$$

The third set of points is:

$$P_3 = 41886\ \text{Pa} \rightarrow x_3(k) = 0.005734\ \text{m}$$
$$x_3(k-1) = 0.005581\ \text{m} \quad x_3(k-2) = 0.005581\ \text{m}$$

It results in:

$$\dot{x}_3 = 0.1530\ \text{m/s} \quad \text{and} \quad \ddot{x}_3 = -7\ \text{m/s}^2$$

Substituting the above values in equations (8a) and (8b) results in:

$$F_v = 1176.1\ \text{kg/s} \quad \text{and} \quad m = 0.9528\ \text{kg}$$

The expected values are $F_v = 1225.9\ \text{kg/s}$ (error = -4.06%) and $m = 1.3608\ \text{kg}$ (error = -29.98%).

The last step is to estimate F_c. By taking a point with increasing pressure, when the stem velocity is positive and constant, the results are as follows:

$$P_i = 82530\ \text{Pa}$$
$$x_\text{i}(k) = 0.05842\ \text{m} \ \text{and} \ x_\text{i}(k-20) = 0.05791\ \text{m}$$

In order to improve the precision in the estimation of the constant stem velocity, two different position points were considered, 0.02 s apart (20 samples) from each other. The resulting velocity is $\dot{x}_i = 0.0255\ \text{m/s}$. Substituting this value in equation (7) results in:

$$F_c = 2225.2\ \text{N}$$

By taking a point with decreasing pressure and negative constant velocity, the result is:

$$P_d = 206.8\ \text{Pa}$$
$$x_\text{d}(k) = 0.04318\ \text{m} \ \text{and} \ x_\text{d}(k-20) = 0.04369\ \text{m}$$

It results in $\dot{x}_d = -0.0255\ \text{m/s}$. Substituting this value in equation (7) results in the same value for F_c, that is:

$$F_c = 2225.2\ \text{N}$$

The correct value is $F_c = 2224.1\ \text{N}$ (error = 0.0495%).

In Figure 4 the curves of actuator pressure X stem position are plotted, considering both the original parameters and the estimated ones.

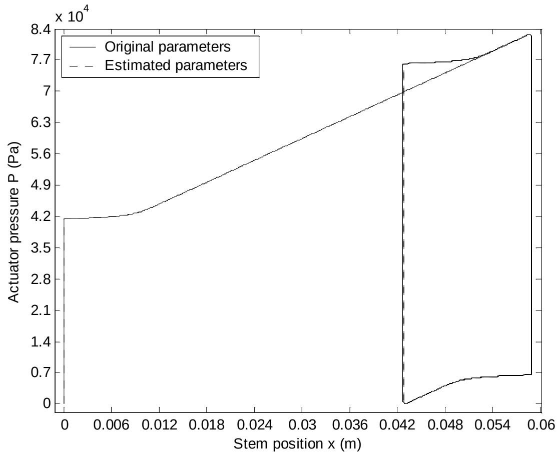

Fig. 4. Actuator pressure X stem position plotted with original and estimated parameters for rough valve.

The curves in Figure 4 indicate that the responses of the models are quite similar.

4.2 *Nominal valve*

Figure 5 shows the stem position and Figure 6 presents the actuator pressure X stem position of a valve with smaller friction (nominal valve).

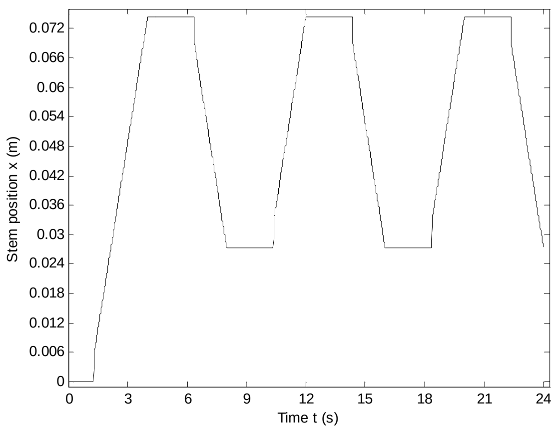

Fig. 5. Stem position – nominal valve.

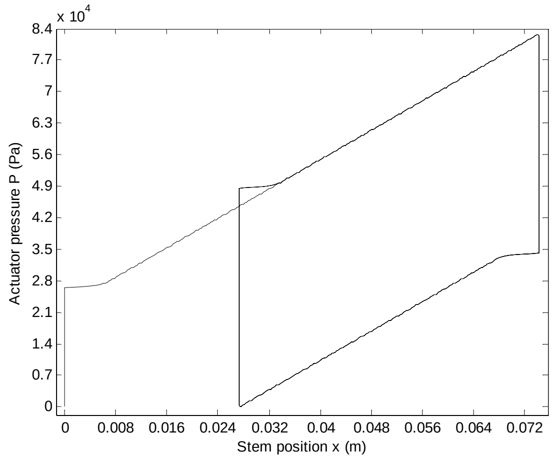

Fig. 6. Actuator pressure X stem position – nominal valve.

It results in the values shown in Table 1, in section 5.

5. APPLICATION TO A VALVE NOT AFFECTED BY STICTION

In this section, a valve with low friction (vendor valve) is considered (Kayihan; Doyle, 2000). The stem position is shown in Figure 7 and the actuator pressure X stem position in Figure 8.

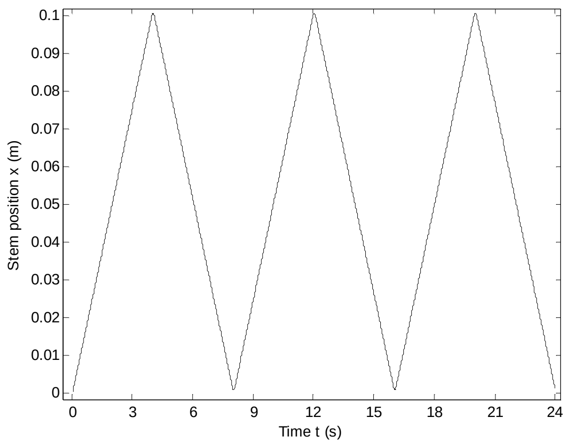

Fig. 7. Stem position – vendor valve.

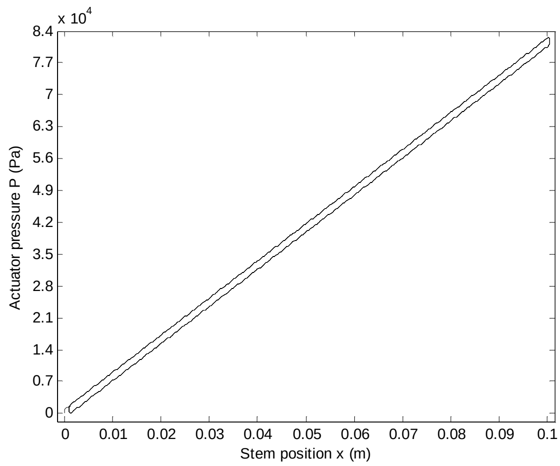

Fig. 8. Actuator pressure X stem position – vendor valve.

Taking two ascending pressure values in Figure 8, when $\dot{x}$ is constant and positive, results as follows:

$$P_1 = 2006.4 \text{ Pa} \;\rightarrow\; x_1 = 0.001321 \text{ m}$$

$$P_2 = 80007 \text{ Pa} \;\rightarrow\; x_2 = 0.09710 \text{ m}$$

Substituting these values in equation (2) results in:

$$k = 52544 \text{ N/m} \quad \text{(estimated)}$$

$$k = 52538 \text{ N/m} \quad \text{(real)} \quad \text{error} = 0.0114\%$$

According to Figure 8, the pressure just before the valve starts moving is $P = 682.6$ Pa. Substituting this value in equation (3a) results in:

$$F_c = 44.04 \text{ N} \quad \text{(estimated)}$$

$$F_c = 44.48 \text{ N} \quad \text{(real)} \quad \text{error} = -0.99\%$$

Considering the first movement of the stem:

$$P = 992.8 \text{ Pa} \;\rightarrow\; x(k) = 0.000138 \text{ m}$$

$$x(k-1) = 0.000120 \text{ m} \quad x(k-2) = 0.000103 \text{ m}$$

It results in:

$$\dot{x} = 0.0180 \text{ m/s} \quad \text{and} \quad \ddot{x} = 1 \text{ m/s}^2$$

Substituting these values in equation (7) results in:

$$m = 1.5038 \text{ kg} \quad \text{(estimated)}$$

$$m = 1.3608 \text{ kg} \quad \text{(real)} \quad \text{error} = 10.51\%$$

By taking a point with increasing pressure, when the stem velocity is positive and constant, results in:

$$P_i = 82530 \text{ Pa}$$

$$x_i(k) = 0.1002 \text{ m} \quad \text{and} \quad x_i(k-20) = 0.09969 \text{ m}$$

Applying these values in equation (7), results in:

$$F_v = 623.0 \text{ kg/s} \quad \text{(estimated)}$$

$$F_v = 612.9 \text{ kg/s} \quad \text{(real)} \quad \text{error} = 1.65\%$$

In Figure 9 the curves of actuator pressure X stem position are plotted, considering both the original parameters and the estimated ones.

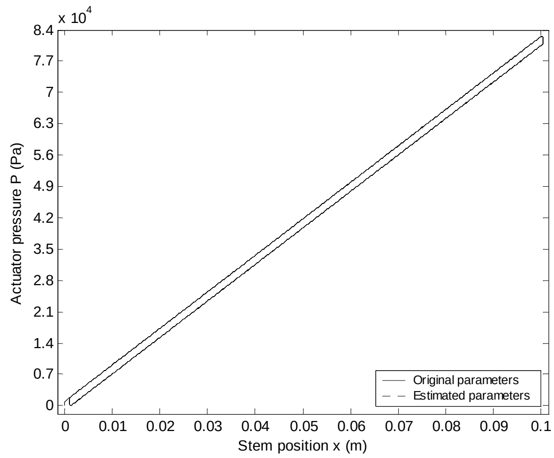

Fig. 9. Actuator pressure X stem position plotted with original and estimated parameters for vendor valve.

The responses of the models in Figure 4 are very similar.

In Table 1 the obtained results are summarized.

Table 1 – Results obtained by the proposed method

	Rough valve	Nominal valve	Vendor valve
m (kg)	0.9528 (estimated) 1.3608 (real) error = -29.98%	1.0575 (estim.) 1.3608 (real) error = -22.29%	1.5038 (estim.) 1.3608 (real) error = 10.51%
k (N/m)	52538 (estimated) 52538 (real) error = 0.0%	52542 (estim.) 52538 (real) error = 0.0076%	52544 (estim.) 52538 (real) error= 0.0114%
F_v (kg/s)	1176.1 (estimated) 1225.9 (real) error = -4.06%	565.5 (estim.) 612.9 (real) error = -7.73%	623.0 (estim.) 612.9 (real) error = 1.65%
F_c (N)	2225.2 (estimated) 2224.1 (real) error = 0.0495%	1424.5 (estim.) 1423.4 (real) error = 0.0773%	44.04 (estim.) 44.48 (real) error = -0.99%
F_s (N)	2669.1 (estimated) 2668.9 (real) error = 0.00749%	1708.2 (estim.) 1708.0 (real) error = 0.0117%	Negligible

As it can be inferred from Table 1, the estimation of k, F_c and F_s derives values that are very close to the real ones, with errors below 1%. In the estimation of F_v the errors are larger, oscillating between 1 to 8%. The worst results are obtained in the estimation of m, when the errors are in a range between 10 to 30%. These large errors are due to the fact that to calculate m, the proposed procedure employs the estimated value of the acceleration $\ddot{x}$, which is not very precisely calculated, due to the fact that it was assumed that the precision in the measurement of the position x was restricted to four significant digits.

6. CONCLUSIONS

The proposed method has derived values for k, F_c and F_s that are very satisfactory. Nevertheless, it is necessary to improve the results obtained for F_v and mainly for m. Despite these errors, the simulations presented in figures 4 and 9 show that the signature curves using the original and the estimated parameters are quite similar.

A great advantage of the proposed method is its simplicity. On the other hand, the method depends on a sufficiently high sampling rate in order to calculate the velocity and the acceleration of the stem with an adequate accuracy and to provide reliable values of the estimated parameters. Besides, as the slipping time is very short, it is necessary to be able to detect this movement.

As a continuation of this work, the idea is to apply this method to data obtained from tests with real valves. Another possibility is to adapt it to online estimation.

In another paper presented in this congress (Romano; Garcia, 2007), this method is compared to the one proposed in (Ravanbod-Shirazi; Besançon-Voda, 2003).

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