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SIMULATION AND EXPERIMENTAL STUDY OF THE PARTIAL EQUILIBRIUM OF AN ELECTROPNEUMATIC POSITIONING SYSTEM, CAUSE OF THE STICKING AND RESTARTING PHENOMENON

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ABSTRACT

The goal of this paper is to present the unsticking piston after stop stage of a fluid power system. This phenomenon is a real problem for the industrial development of pneumatic technology. Less apparent in hydraulic systems but also present, this phenomenon will be described, analysed and a method will be proposed to predict it. Simulation helps to explain certain experimental results obtained in positioning linear actuators.

KEY WORDS

Fluid Power, Friction, Unsticking, Equilibrium

NOMENCLATURE

$$C_{p_X^e}^e = \left. \frac{\partial q_J}{\partial p_X} \right|_e, \text{ } p_X \text{ partial derivative of the mass flow rate}$$

q_J around equilibrium state (kg/s/Pa)

F force (N)

$$G_{u_J}^e = \left. \frac{\partial q_J}{\partial u} \right|_e, \text{ } u \text{ partial derivative of the mass flow rate } q_J$$

around equilibrium state (kg/s/V)

k polytropic constant

k_r spring constant (N/m)

M total load (kg)

p pressure in the cylinder chamber (Pa)

Q_{m_X} mass flow rate provided from the servo-distributor to cylinder chamber X (kg/s)

r perfect gas constant (J/kg/K)
 S area of cylinder bore (m²)
 t time (s)
 T temperature (K)
 u servo-distributor input voltage (V)
 v velocity (m/s)
 V volume (m³)
 y position (m)
 δ variation near equilibrium set

Subscripts

C Coulomb friction
 DS dynamic stiction friction
 E exhaust
 f friction
 mes experimental measures
 N chamber N
 P chamber P
 S supply

sim simulated
 SK Stribeck friction
 SS static stiction friction
 t rod
 u unsticking
 v viscous friction

Superscripts

e equilibrium
 stop partial equilibrium at time stop

INTRODUCTION

In the past few years, new electropneumatic positioning devices have come on to the market. One of the most critical problems for their industrial integration originates from the existence of a steady state error due to dry friction. So naturally reducing this friction during the design of the product (with a new type of joint for example) can be a valid development. Unfortunately when position is controlled by position, velocity and acceleration feedback, even if there is no integrator in the control law, the consequence is an increase in the sticking and restarting effect, which is better known in mechanical processing as "stick-slip" [1], even though in this case the cause is different.

This paper presents a comparison between experimental and simulated results, obtained for piston displacements for which an adapted flow stage model of the servo-distributors has been used [2]. This paper, therefore, completes previous development carried out concerning piston unsticking in pneumatic systems by Sesmat et al [3].

1 GENERALITY ABOUT FRICTION

Friction effect on the industrial systems have been studied by many authors, [4 to 11] who have proposed some physical and mathematical models. Theoretically these models are not stationary but it is usual to make the assumption that only parameters evolve with time but the form of the model does not. Four kinds of friction are generally described: viscous, static, Stribeck and Coulomb friction (Figure 1).

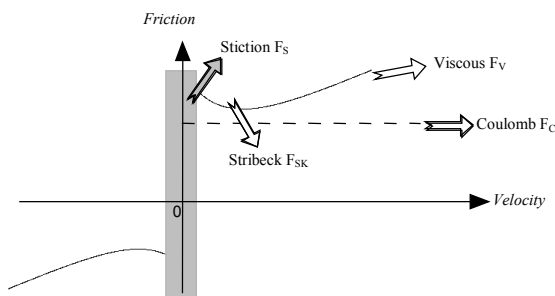


Figure 1 Different friction phenomena.

Modelling static friction is the most difficult phase, it concerns very low velocities when the joint is stick on to the cylinder wall due to a static stiction force F_S .

In 1902 Stribeck observed an exponential decrease of friction when the velocity is small and increasing. During this stage the piston moves along the cylinder wall, the joint loses its shape. This decrease in friction can reach 25% of stiction friction and tends towards Coulomb friction.

The main classic models describing all main friction phenomena of are compared in figure 2.

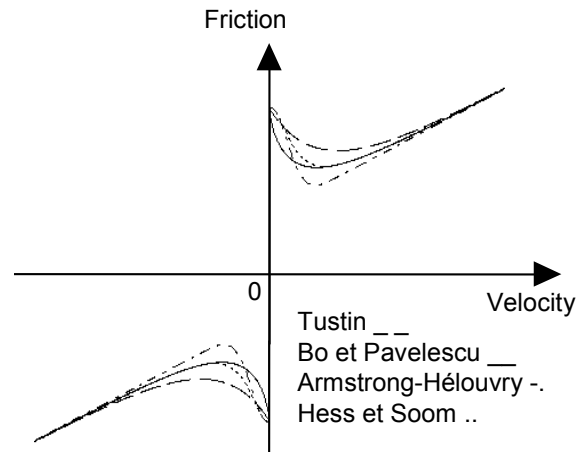


Figure 2 Comparison between the more usual friction models.

2 MEASURE AND IDENTIFICATION OF FRICTION

Generally friction forces are function of velocity. Figure 3 shows experimental results obtain in the electropneumatic system described in section 4.

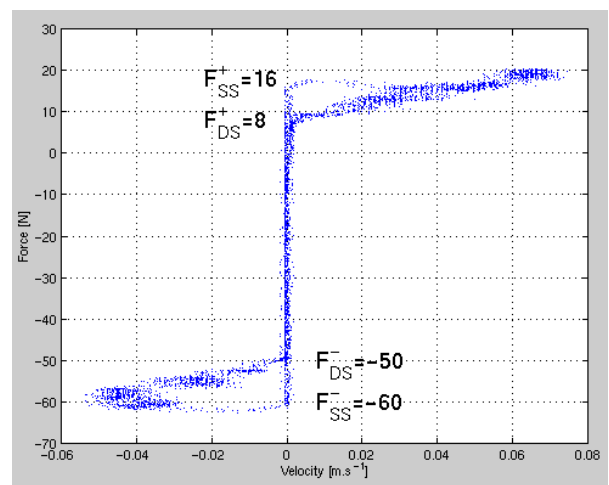


Figure 3 Experimental measure of friction force. The first thing to note is that the model is not symmetrical.

So the friction force is noted F^+ if velocity is positive and F^- in other case.

Secondly in electropneumatic actuators, the track surface quality (thus the piston position), the joint wear, the working conditions (temperature, pressure, quality of air) are all parameters which influence the friction value. It can be noticed that at low velocities the friction values are not constant. It is difficult to fix a Stribeck time constant. That is why the notions of dynamic F_{DS} and static F_{SS} stiction friction are defined as equal to the friction value at sticking and slipping time respectively.

3 SIMPLE CASE OF PISTON UNSTICKING: SIMPLE ACTING ACTUATOR

The model (Eq. 1) of an electropneumatic simple-acting actuator can be obtained using two physical laws: the first giving the pressure dynamics in the chamber with variable volume and the second being the fundamental mechanical relation. A pressure evolution law in a chamber with variable volume can be obtained with the following assumptions [12, 13]:

- the air is a perfect gas and its kinetic energy is negligible in the chamber,
- the process is polytropic characterised by coefficient k .
- the temperature variation in the chamber is negligible and so is considered equal to the supply temperature.

$$\begin{cases} \frac{dp}{dt} = \frac{krT_S}{V(y)} \left[q_m(u, p) - \frac{S}{rT_S} pv \right] \\ \frac{dv}{dt} = \frac{1}{M} [Sp - k_r y - F_f(v)] \\ \frac{dy}{dt} = v \end{cases} \quad (1)$$

The nullity of all time derivatives in the model (1) defines the equilibrium set. This definition is important and must not be confused with mechanical equilibrium. In fact this last notion corresponds to the nullity of the two last equations in the model (1). Physically this means that the piston has stopped but the pressure in the chamber continues to evolve. So the notion of partial equilibrium (noted with superscript stop) has to be defined as the case where the mechanical equilibrium is obtained but not the pneumatic equilibrium. The model (1) leads to the following conditions at the piston stop time:

$$\begin{cases} y = y^{stop} \\ v = v^{stop} = 0 \\ p = p^{stop} \\ u = u^{stop} \end{cases} \quad (2)$$

u^{stop} is the value of the control which corresponds to a pressure force equal to the sum of the friction force and spring force at time stop (Eq. 3). Due to the form of control law (acceleration feedback), this value is constant during all the partial equilibrium stage.

$$Sp^{stop} = k_r y^{stop} + F_f(t^{stop}) \quad (3)$$

with $F_{DS}^- \leq F_f(t^{stop}) \leq F_{DS}^+$

Then:

$$\frac{1}{S} (k_r y^{stop} + F_{DS}^-) \leq p^{stop} \leq \frac{1}{S} (k_r y^{stop} + F_{DS}^+) \quad (4)$$

A good servo-distributor must have a steep slope for the pressure gain characteristic at null mass flow rate and also the pressure force characteristic too. Then u^{stop} is small and figure 4 shows that the mass flow rate can be considered proportional to the pressure.

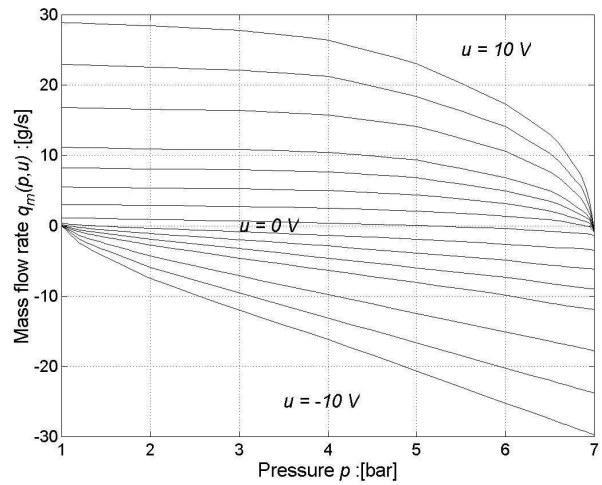


Figure 4 Mass flow rate characteristics series.

This assumption combined with the first equation of model (1) shows that the pressure evolution in the chamber is of first order. The pressure evolves and tends to a theoretical value of pressure $p(u^{stop})$ deduced from the pressure gain characteristic of the servo-distributor. Two cases are possible according to this pressure value. If relation (5) is true:

$$\lim_{t \rightarrow \infty} (Sp(t) - k_r y^{stop}) = [Sp(u^{stop}) - k_r y^{stop}] \in [F_{SS}^-, F_{SS}^+] \quad (5)$$

then it is certain that the partial equilibrium leads to a total equilibrium without the piston unsticking (see case \aleph in figure 5). But in the other case the piston slips after sticking (at $t=t_u$ in case \beth in figure 5). Another control value is calculated and the same analysis has to be carried out to predict whether or not there will be another stick-slip phase.

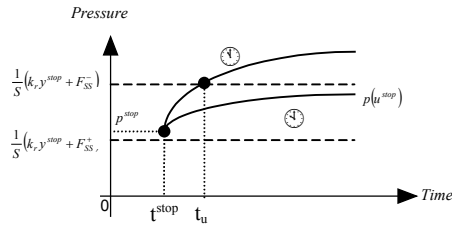


Figure 5 Possible pressure evolutions during partial equilibrium.

As opposed to the classical description of stick-slip in mechanical systems, a periodic cycle of alternating moving and stopping does not appear in all cases. It is not a question of limit cycle, the unsticking stage can appear only one time.

However in case of double-acting systems the phenomenon of piston unsticking has been noticed in both cases but never explained with experimental results. That is what this paper will explain in the next sections.

4 DESCRIPTION AND MODELLING OF AN ELECTROPNEUMATIC SYSTEM

The system under consideration (figure 6) is a linear double acting electropneumatic servodrive using a simple rod (32/20 mm) with a stroke of 500 mm controlled by two three-way servo-distributors. A potentiometer gives the position. Velocity is obtained by analogue derivation and acceleration by numerical derivation. A pressure sensor is implemented in each chamber.

With same assumptions as in section 2, considering that the two servo-distributors are identical and that there is no leakage between the two cylinder chambers, the tangent linearised model of the system is described by Eq. 7 [12].

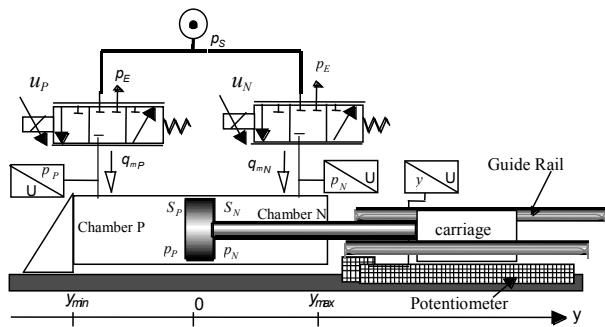


Figure 6 Electropneumatic system.

With variation near equilibrium set

$$\left\{ \begin{array}{l} \delta p_P = p_P - p_P^e, \delta p_N = p_N - p_N^e, \delta v = v - v^e, \\ \delta y = y - y^e, \delta u = u - u^e \end{array} \right\} \quad (6)$$

$$\frac{d}{dt} \begin{bmatrix} \delta p_P \\ \delta p_N \\ \delta v \\ \delta y \end{bmatrix} = \begin{bmatrix} -\frac{1}{\tau_P^e} & 0 & -\frac{k p_P^e S_P}{V_P(y^e)} & 0 \\ 0 & -\frac{1}{\tau_N^e} & \frac{k p_N^e S_N}{V_N(y^e)} & 0 \\ \frac{S_P}{M} & -\frac{S_N}{M} & -\frac{f_v}{M} & 0 \\ 0 & 0 & 1 & 0 \end{bmatrix} \begin{bmatrix} \delta p_P \\ \delta p_N \\ \delta v \\ \delta y \end{bmatrix} + \begin{bmatrix} \frac{k r T_S}{V_P(y^e)} G_{uP}^e \\ -\frac{k r T_S}{V_N(y^e)} G_{uN}^e \\ 0 \\ 0 \end{bmatrix} \delta u \quad (7)$$

With pressure time constants

$$\tau_P^e = \frac{V_P(y^e)}{k r T_S C_{PP}^e} \quad \text{and} \quad \tau_N^e = \frac{V_N(y^e)}{k r T_S C_{PN}^e} \quad (8)$$

5 EXPERIMENTAL RESULTS

With a classical partial state feedback in position, velocity and acceleration, for a displacement of 50 mm near the end stroke of the cylinder, figure 7 shows experimental results and occurrence of restart at $t=t_3$. Pressures and position are given by sensors, the pressure force is reconstructed as follow:

$$Force(t) = S_P p_P(t) - S_N p_N(t) - S_t p_E \quad (9)$$

This occurrence of piston unsticking has been explained in simulation by Sesmat et al [3] using an appropriate flow stage model of the servo-distributor. In this case of double-acting cylinder the pressure force evolution comes from the evolution of both pressures due to the parallel structure of the system. Then during all the time of the partial equilibrium stage, to assure total equilibrium, the evolution of the two pressures must verify Eq. 10:

$$\forall t \quad Force(t) \in [F_{SS}^-, F_{SS}^+] \quad (10)$$

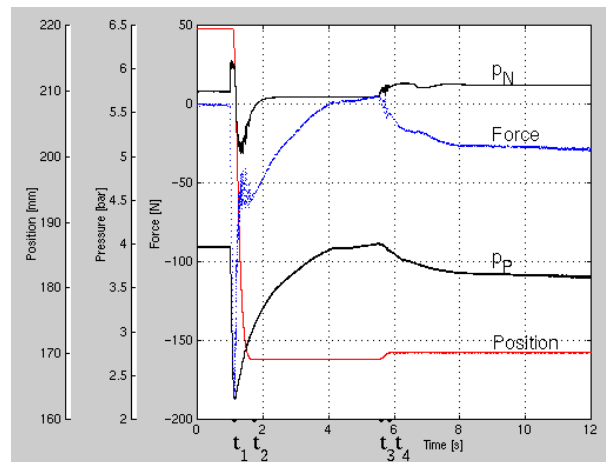


Figure 7 Experimental results: unsticking phenomenon

To understand the reason for this undesirable phenomenon the pressure evolution is presented in plane $(S_P p_P, S_N p_N + S_t p_E)$ in which stiction friction values are identify. By using information about position and direction of movement, we deduced:

$$\begin{cases} \text{at start time} & t = t_1 \quad F_f = F_{SS}^+ \\ \text{at first stop time} & t = t_2 \quad F_f = F_{DS}^+ \\ \text{at unsticking time} & t = t_3 \quad F_f = F_{SS}^- \\ \text{at second stop time} & t = t_4 \quad F_f = F_{DS}^- \end{cases} \quad (11)$$

Friction values (Eq. 11) are define by used Eq. 12 to plot the straight lines in figure 8:

$$S_N p_N + S_t p_E = S_P p_P + F_f \quad (12)$$

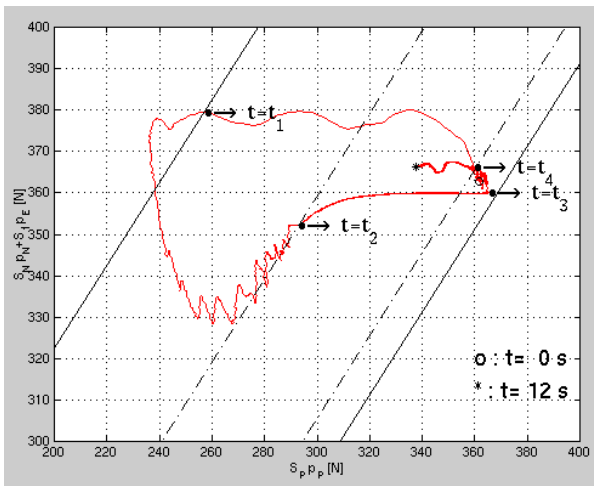


Figure 8 Analysis of unsticking piston and friction identification.

The last two figures show that between t_2 and t_3 the two pressure dynamics are very different (by a ratio of 8). The biggest chamber has the biggest time constant according to relation 8. In order to predict the unsticking time, the differential pressure equations can be resolved (Eq 13):

$$\begin{cases} p_P(t) = \left[p_P^{\text{stop}} + \frac{G_{uP}}{C_{PP}} (u^{\text{stop}} - u^e) \right] + \left[p_P^{\text{stop}} - p_P^e - \frac{G_{uP}}{C_{PP}} (u^{\text{stop}} - u^e) \right] e^{-\frac{t}{\tau_P^{\text{stop}}}} \\ p_N(t) = \left[p_N^{\text{stop}} - \frac{G_{uN}}{C_{PN}} (u^{\text{stop}} - u^e) \right] + \left[p_N^{\text{stop}} - p_N^e + \frac{G_{uN}}{C_{PN}} (u^{\text{stop}} - u^e) \right] e^{-\frac{t}{\tau_N^{\text{stop}}}} \end{cases} \quad (13)$$

So during the stop stage the force evolution is:

$$F(t) = \left[S_P p_P^{\text{stop}} - S_N p_N^{\text{stop}} - S_t p_E + \left(\frac{G_{uP}}{C_{PP}} + \frac{G_{uN}}{C_{PN}} \right) (u^{\text{stop}} - u^e) \right] + S_P \left[p_P^{\text{stop}} - p_P^e - \frac{G_{uP}}{C_{PP}} (u^{\text{stop}} - u^e) \right] e^{-\frac{t}{\tau_P^{\text{stop}}}} - S_N \left[p_N^{\text{stop}} - p_N^e + \frac{G_{uN}}{C_{PN}} (u^{\text{stop}} - u^e) \right] e^{-\frac{t}{\tau_N^{\text{stop}}}} \quad (14)$$

Using Eq. 17, the two inequation 10 can not be resolved

analytically. The numerical solution in this case is $t_u^{\text{sim}} = 3.80$ s. Figures 9 and 10 show comparisons between experimental measurements and simulation. The time at which the first stop occurs is taken as a reference. The experimental unsticking time $t_u^{\text{mes}} = t_3 - t_2 = 3.86$ s agrees with the simulation.

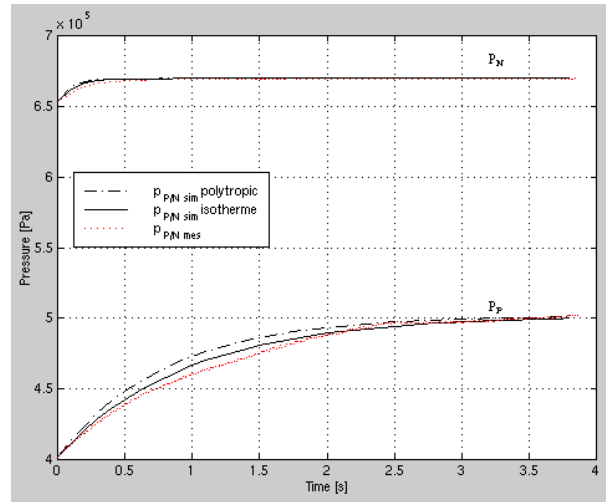


Figure 9 Simulated and experimental pressure evolutions after stop

By considering an isothermal evolution instead of a polytropic one, the simulation results are improved (figures 9 and 10).

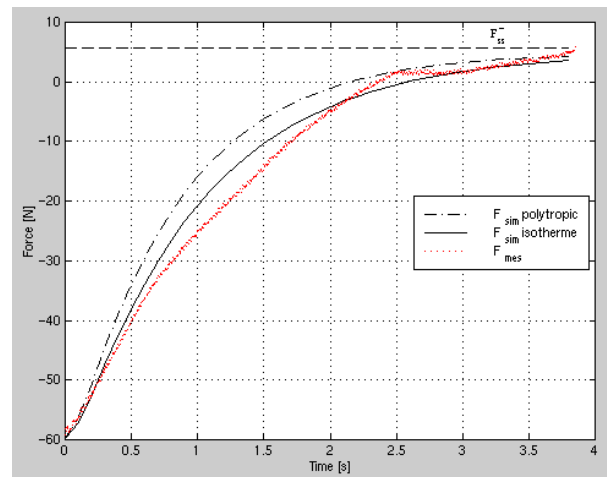


Figure 10 Simulated and experimental force evolution after stop

CONCLUSION

In this paper the phenomenon of piston unsticking has been studied. This is not caused by the use of an integrator in the positioning control law.

In the case of simple acting fluid power actuators the unsticking piston phenomenon depends only on the final value of the chamber pressure. This value can be predicted from pressure gain characteristic at null mass flow rate with the knowledge of the control value at time stop.

However for double acting actuators the combination of stiction friction and the parallel structure of the system can cause piston unsticking in one other case. During pressure evolution, if the pressure exits from the static stiction range, the piston moves again after sticking.

The difficulty of evaluating friction due principally to its variation with time, position and experimental conditions requires the development of the appropriate control law to reduce this undesirable effect. Some further work concerning the estimation of friction in real time [14] has to be carried out or robust control [15, 16] must be improved.

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REFERENCES

1. Armstrong-Hélouvry, B., Frictional Lag and Stick-Slip", IEEE Int. Conference on Robotics and Automation, Nice France, 1992, pp 1448-1453.
2. Sesmat, S., Scavarda, S., Static characteristics of three way servovalve. 12 Aachener Fluid. Kolloq., Aachen, 12-13 Marz, vol. 2, 1996, pp 643-652.
3. Sesmat, S., Scavarda, S., Study of the behaviour of an electropneumatic positioning system near the equilibrium state. 1 Int. Fluid. Kolloq. in Aachen, Aachen, 17-18 Marz, vol. 2, 1998, pp 321-334.
4. Tustin A. The Effect of Backlash and Speed-dependent Friction on the Stability of closed-cycle control systems. Jour. of the Institution of Electrical Engineers, Vol. 94, n°2A, 1947, pp 143-151.
5. Dahl, P. R. A Solid friction Model. TOR-158, The Aerospace Corporation, El Segundo, California, 1968, pp 3107-3118.
6. Bo, L. C., Pavelescu, D. The friction-speed relation and its influence on the critical velocity of the stick-slip motion. Wear, Vol 3, 1982, pp 277-289.
7. Hess, D.P., Soom ,A. Friction at a lubricated line contact operating at oscillating sliding velocities. Jour. of Tribology, Vol 112 (1), 1990, pp 147-152.
8. Armstrong-Hélouvry, B., Dupont, P., Canudas de Wit, C. A survey of analysis tools and compensation methods for the control of machines with friction. Automatica, 30, 1994, pp 1083-1138.
9. Canudas de Wit, C., Olsson, H., Aström, K.J., Lischinsky, P. A new model for control of systems with friction. IEEE Transactions on Automatic Control, Vol. 40, n°3, 1995, pp 419-425.
10. Bowns, D.E., Ballard, R.L. , Stiles, L. "The Effect of Seal Friction on the Dynamic Performance of Pneumatic Actuators"" 3rd Int. Fluid Power Symposium, BHRA, 1973, Paper C3, pp 29-48.
11. Belforte, G., D'Alfio, N., Raparelli, T. "Experimental Analysis of Friction Forces in Pneumatic Cylinders", Journ. of Fluid Control, vol 20, N°1, 1989, p 42-60.
12. Shearer, J.L. Study of pneumatic processes in the continuous control of motion with compressed air. Parts I and II. Trans. Am. Soc. Mech. Eng., Vol. 78, 1956, p.233-249.
13. Andersen, B.W. The analysis and design of pneumatic systems. New-York : John Wiley and Sons, 1967, 302p.
14. Tafazoli, S. Tracking control of an Electrohydraulic Manipulator in the presence of Friction. IEEE Trans. on Cont. Syst. Tech., Vol. 6, n°3, May 1998, pp 401-411.
15. Kang, M.S. Robust digital friction compensation. Cont. Eng. Practice, Vol 6 n°3, 1998, pp 359-367.
16. Cheng, C.C, Chen, C.Y. A PID approach to suppressing stick-slip in the positioning of transmission mechanisms. Cont. Eng. Pract., Vol 6 n°4, 1998, pp 359-367.