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STUDY OF THE FRICTION BEHAVIOR IN INDUSTRIAL PNEUMATIC ACTUATORS

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Abstract. *This paper presents a comparative study among friction behavior of several double-acting pneumatic actuators available to industrial use. A brief review covering the manner that pneumatic actuator manufacturers and current literature approach friction force and its usual models also is presented. After depicting friction test methodology and test apparatus, the comparison work is carried out through experimental tests to friction identification from steady state friction-velocity maps, that permit identify the main friction characteristics to each pneumatic cylinder in evaluation. They are Static friction, Coulumb friction, Stribeck friction and Viscous friction, that are important to design a precision pneumatic servo system. Experimental results to cylinders of several manufacturers are presented and show friction characteristics, that vary with actuator manufacturer. To know cylinder friction is an important step to friction compensation and to select the pneumatic actuator.The appropriate choice of actuator manufacturer also can contribute to the precision control success.*

Keywords*: Fluid power, pneumatic actuators, friction modelling, pneumatic automation, precision control.*

1. Introduction

This paper presents a comparative study among friction behavior of several double-acting pneumatic actuators available to industrial use. Pneumatic actuators are very common in industrial application because they have easy and simple maintenance, relatively low cost, self cooling properties, good power density (power/dimension rate), fast acting with high accelerations and installation flexibility. Also, compressed air is available in almost all industry plants. These characteristics become pneumatic actuators competitive in a large band of applications in motion control to materials and parts handling, packing machines, machine tools, robotics, food processing and process industry.

Otherwise, a pneumatic servo system has many disadvantages that have to be overcome by its control system. They have very low stiffness (caused by air compressibility), inherently non-linear behavior and low damping of the actuators systems, that cause control difficulties. The main non-linearities in pneumatic servo systems are the air flow-pressure relationship through valve orifice, the air compressibility and friction effects between contact surfaces in actuator seals. According to Vieira (1998) and Nouri *et al.* (2000), the most complex non-linearity in pneumatic position servo systems is the actuator friction force. It makes the position control more difficult because it can cause steady state position and trajectory tracking errors, limit cycles around the desired position (hunting) and stick-slip movements.

The knowledge of the friction force in pneumatic actuators is an important step to obtain the precise control and its appropriate design. The comparison work in this paper is carried out through experimental tests to friction identification and the analysis of catalog information available by pneumatic actuators manufacturers. In the sequence, experimental system and friction test methodology are depicted. Experimental results permit to obtain the characteristics to Static friction, Coulumb friction, Stribeck friction and viscous friction to each pneumatic cylinder in evaluation from experimental steady state friction-velocity maps.

There is a lack of information in catalogs and available literature covering friction characteristics in pneumatic actuators. For this, the authors intend to contribute in the characterization of pneumatic actuators from main Brazilian manufacturers to future selection to use in pneumatic servo systems with precision control. An additional contribution of this paper is the complete presentation of the friction test methodology to pneumatic cylinders and the corresponding test apparatus. A deeper knowledge of the pneumatic actuator friction and the development of mathematical models to represent this phenomenon in a suitable way will contribute to increase the use possibilities of pneumatic actuators in positioning tasks and in industrial robots. In this way, Valdiero *et al.* (2005) and Perondi (2002) present a complete friction model to pneumatic actuators.

2. Friction in pneumatic actuators

The most complex non-linearity in pneumatic position servo systems is the actuator friction force. It causes many control difficulties, such as position steady state and position trajectory tracking errors. Also, limit cycles around the desired position (hunting) and stick-slip movements are caused by friction effects. This section presents a short review of the manner that pneumatic actuator manufacturers and current literature approach friction force and its usual models.

Many books and catalogs supply information about friction in pneumatic actuator to current industrial use, but these information are insufficient to select an actuator and design the servo pneumatic drive. Catalogs of Brazilian pneumatic actuator manufacturers usually present friction force as a performance loss through of an efficiency factor m (Eq. 1).

$$F_{atr} = \mu.P.A \tag{1}$$

where

F_{atr} = friction force;

P = working pressure;

and

A = actuator area.

Festo (1996) recommends to use m = 0,10 and informs that calculated friction force is only an initial value because friction depends of many others factors as lubrication, work pressure, actuators seals, and others. Parker (1998) uses a similar approach where $m \approx 0{,}20$, where effective actuator force may be obtained from a special table. In this way, books recommend to use the efficiency factor m in the order of 0,05 to 0,10 (Bollmann, 1997) and 0,02 to 0,06 (Pinches and Callear, 1997). SMC (1996) also presents friction in this manner, but m is obtained from a graph in function of cylinder bore diameter and air supply pressure. Otherwise, other manufacturers don't present any information about friction in their actuators (Norgren, 1999 and Pró-Ar, 1997).

It is clear that these information above are not sufficient to model and design servo pneumatic drives. A more complete approach is presented by Belforte *et al.* (1989), where are used experimental results to identify experimental coefficients in Eq. (2) and to calculate actuator friction force. These coefficients are presented to a group of tested pneumatic cylinders and experiments have carried out with various constant supply pressures values.

$$F_F = F_A + (1 + K_1.v^{\alpha}).\left[K_2.\left|P_1 - P_2\right| + K_3.P_2\right] \tag{2}$$

where

F_F = actuator friction force;

F_A = static friction force with no counter pressure;

K_1, K_2 and K_3 = experimental coefficients;

α = experimental exponent;

v = actuator velocity;

P_1 = pressure in actuator chamber 1;

and

P_2 = pressure in actuator chamber 2.

Calculated value of F_F is bigger that really occurs in a pneumatic servo drive because the tests were carried out by Belforte *et al.* (1989) with values of actuator chamber pressure (P_1) and counter pressure (P_2) larger that really occurs if a pneumatic proportional directional valve was used to control the tested cylinders.

A recent form to represent friction in pneumatic actuators is through static friction-velocity maps in steady state (Fig. 1), obtained to a constant supply pressure, in a similar way that occurs in a servo pneumatic drive. These maps permit to obtain the four main static friction force coefficients, that are: Static Friction (F_S); Coulumb friction (F_C); viscous damping coefficient (B) and Stribeck velocity ($\dot{y}_s$). With these coefficients, Nouri *et al.* (2000) model friction force F_{atr} through Eq. (3).

$$F_{atr} = F_C + B.v + \left(F_S - F_C\right).e^{-\left(\frac{v}{\dot{y}_s}\right)^{\delta}} \tag{3}$$

In Eq. (3), δ is an arbitrary exponent. Dupont *et al.* (2000) uses $\delta = 2$.

Although analyzed pneumatic actuator manufacturer catalogs and books don't present friction-velocity maps neither manners to obtain these main friction parameters, their knowledge allows to carry out an actuator pre-selection, estimate the performance in servo positioning, choose the necessary mathematical model to friction and define the control strategy to pneumatic servo drive.

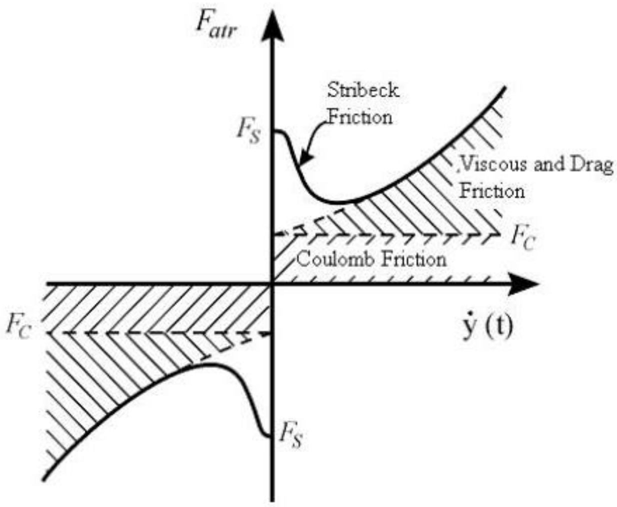

Figure 1. Friction force characteristics combined in steady state

The use of more accurate friction models to compensation in control system is necessary if more sophisticated control strategies are applied, if application needs precision positioning or if actuators with worse friction characteristics are employed. In this way, Perondi (2002) represents friction in pneumatic servo actuators by the LuGre model (Eq. 4), that is used also to friction compensation.

$$F_{atr} = \sigma_0.z + \sigma_1.\frac{dz}{dt} + \sigma_2.\dot{y} \tag{4}$$

where

σ_0 = stiffness coefficient of microscopic deformation of z;

z = average deflection of the asperities between surfaces, that is an internal state that can not be measured;

σ_1 = damping coefficient associated with dz/dt;

σ_2 = viscous friction coefficient (=B);

and

$\dot{y}$ = relative velocity between contact surfaces.

Perondi (2002) has faced many difficulties to identify σ_0 and σ_1 coefficients of Lugre model. Valdiero *et al*. (2005) presents a practical manner to identify the LuGre dynamic model parameters to pneumatic actuators from their static friction-velocity map, similar to Fig. 1.

3. Test rig

To overcome the lack of information about pneumatic actuators friction in manufacturer catalogs, and to know more accurately this non-linearity in industrial pneumatic cylinders, was configured an experimental test apparatus (Fig. 2) to obtain the static friction-velocity maps of industrial pneumatic actuators listed in Tab. 1, where actuator number one (N. 1) is a pneumatic rodless actuator and the others are single-rod double-acting cylinders. All tested cylinders can be used with non-lubricated compressed air.

Figure 2 is formed by one acquisition and control system mounted in a PC microcomputer and one pneumatic system, that is composed by one pneumatic actuator under test (2) and one proportional directional pneumatic valve (4). Sensors permit measure air system inlet pressure (1), the actuator position (3) and actuator chamber pressures (P_1 and P_2), (5) and (6). The acquisition and control system used is a dSPACE DS 1102 board. It is composed by 4 analog inputs (*ADCs*) and 4 analog outputs (*DACs*). Table 2 presents the main components of experimental system. All experiments in this paper were carried out with air supply pressure of 6,0 bar and DS 1102 board configured with a sample rate of 1 ms and acquisition rate of 10 ms. Temperature during these tests has been in the 20 ^oC to 23 ^oC range.

Actuators N. 6 and N. 7 have a great deal of built-in grease, that is necessary to lubrication for all actuator life. This fact can cause damage in very sensitive components as proportional directional pneumatic valves, that impedes to use these actuators in servo pneumatic drives. Just the same, friction-velocity maps were obtained to comparison with other tested actuators.

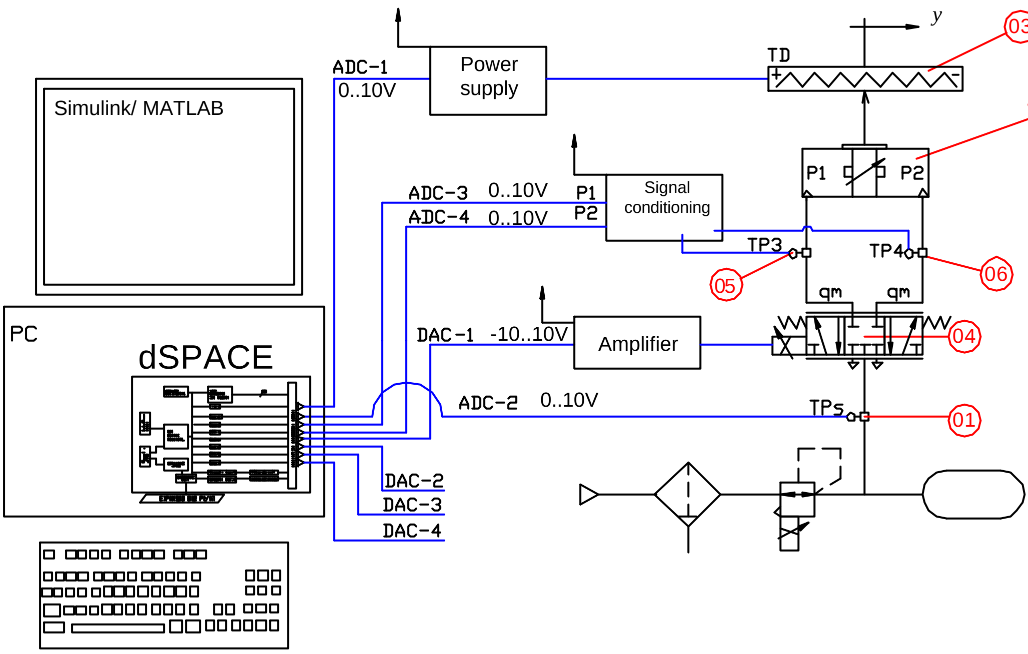

Figure 2. Experimental system

Table 1. Industrial pneumatic linear actuators tested.

Actuator Number	Manufacturer	Diameter (mm)	Stroke (mm)	Catalog Code
1	Rexroth	25	500	502 602 020 0
2	Norgren	32	100	RA/8032/M/100/C
3	Norgren	40	250	RA/8040/M/250/C
4	Norgren	50	400	RA/8050/M/400/C
5	Festo	32	100	DNGU-32-100-PPV-A
6	Pró-Ar	32	100	MM032.249.101x100
7	Pró-Ar	40	250	MM040.249.101x250
8	Dover	40	30	CEUPSW40D-B0030

Table 2. Main components from experimental test apparatus.

Component	Manufacturer	Catalog code	Main specifications
Proportional directional pneumatic valve	Festo	MPYE-5-1/8	5-port, 3-position valve flow rate = 700 l/min.
Pressure sensors	Gefran	TKG E 1 M 1D M	Scale pressure range = 0 to 10 bar
Position transducer	Festo	MLO-POT-500-TLF	Length = 514 mm
Compressed air reservoir	Pró-Ar	RA 080.500.1	Volume = 2,51.10^{-3} m^3

Because this context, the test rig was reconfigured to Fig. 3 to test actuators N. 6 and N. 7. Proportional directional valve was replaced by a common 5-port, 2 position directional valve (04) with double air pilot actuation. Extending velocity actuator control is carried out by a flow control valve (07), that regulates the air flow entering in cylinder port (meter-in). It reproduces operation conditions in a similar way that occurs with a proportional valve use. The pressure sensors (05 and 06) are placed to measure air pressure in cylinder chambers. Retracting tests are made after changing the position of flow control valve (07), to equivalent place before the pressure sensor (06). Actuator extension and retraction are controlled by 3-port, 2-position manual directional valves (08).

A brief comparison between these two test apparatus configurations permits to conclude that configuration of Fig. 3 is simpler and cheaper that Fig. 2, because tests are carried out without a proportional valve, that is an expensive and sensitive component and is made by a little number of manufacturers, generally foreign companies. These factors can complicate an acquisition and use of these proportional valves. Otherwise, tests with the proportional valve are simpler and quicker, because the valve opening is regulated through an electrical command sent by dSPACE board. In Fig. 3, to each test is necessary to adjust the flow control valve (07), because its opening is regulated manually. Also, to carry out retracting cylinder tests is necessary to change the flow control valve position, as indicated above.

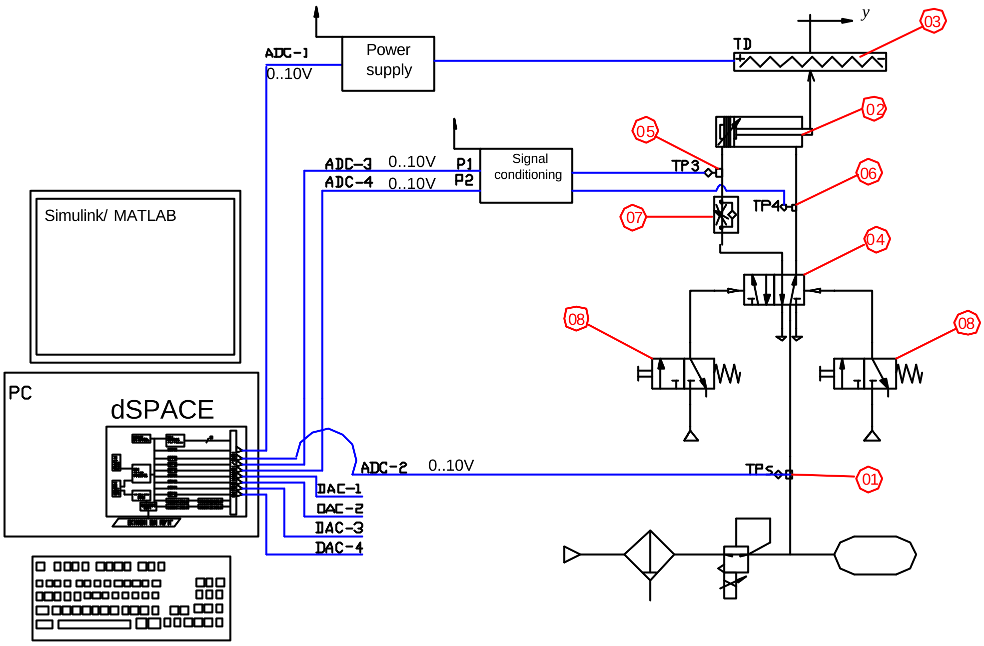

Figure 3. Reconfigured experimental system, without proportional directional valve

4. Experimental results

The static map that represents the value of friction force with corresponding steady state velocity is obtained from Load Dynamic Equation of the system, written by the application of the second law of Newton (Eq. 5).

$$M \cdot \frac{d^2y}{dt^2} + F_{atr} = A_1 \cdot P_1 - A_2 . P_2 \tag{5}$$

where

- y = actuator displacement;
- F_{atr} = actuator friction force;
- P_1 = pressure in actuator chamber 1;
- P_2 = pressure in actuator chamber 2;
- A_1 = actuator area of cylinder in chamber 1;
- A_2 = actuator area of cylinder in chamber 2;

and

- M = load mass.

The mathematical model to a pneumatic position servo system is completed by the Pneumatic Valve Flow Equation and the Continuity Equation, that may be obtained in Andrighetto *et al.* (2003).

The actuator friction force F_{atr} can be calculated by Eq. (5) if the acceleration is known (Belforte *et al.*, 1989). If the tests are carried out with a constant actuator velocity, acceleration values zero and friction force in steady state, $F_{atr,SS}$, is equivalent to the force produced in actuator by $A_1.P_1 - A_2.P_2$, according to Eq. (6), obtained from Eq. (5).

$$F_{atr,SS} = A_1 \cdot P_1 - A_2 . P_2 \tag{6}$$

During experiments realization, the control system maintains a constant valve opening (x_v). This does that pneumatic actuator moves with a constant velocity in a large part of its course. To each valve opening, actuator position y and chamber pressures P_1 and P_2 are measured. Friction force is calculated in accordance with Eq. (6) and depicted in a graph to each test. Velocity is calculated through adjustment procedure with Matlab software in a region where position is a straight line. Related pressure values, P_1 and P_2, are read in the same time interval for friction force calculation. To each actuator, many others similar experiments were carried out with different velocities, to plot Friction-velocity map to each pneumatic actuator under test. Figure 4 depicts results to actuator N.2.

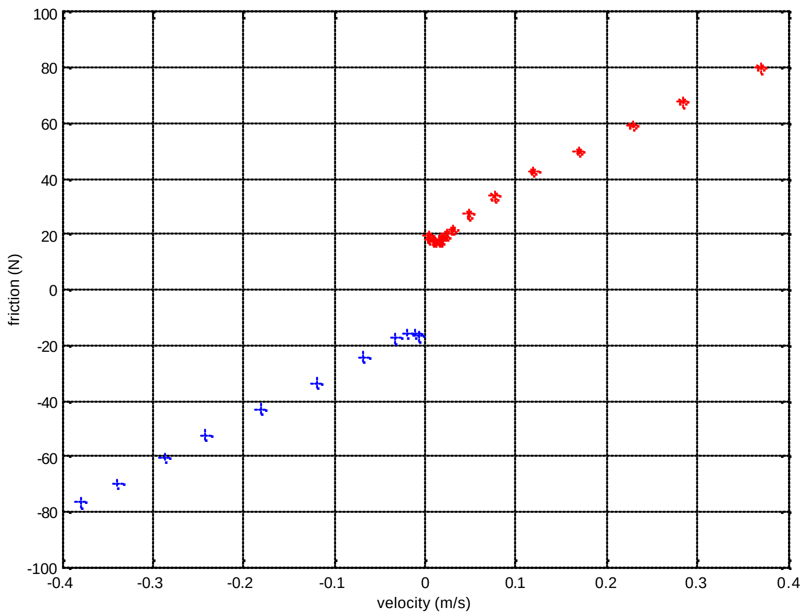

Figure 4. Friction-velocity map to Pneumatic Actuator N. 2 with experimental values.

From Friction-velocity maps obtained from experiments, the four main friction force static parameters F_S, F_C, B and $\dot{y}_s$ were calculated to each tested actuator. To determine viscous damping coefficient (B) and Coulumb friction (F_C), it is considered that, to velocities high enough, friction is almost a straight line, according to Fig. 4. Using Matlab software, values of these coefficients are easily determined and resulting straight line is also plotted in this graph (Fig. 5) to negative and positive velocities. This also can be seen through Eq. (3), where to higher velocities, the exponential term tends to zero and resulting equation is a straight line.

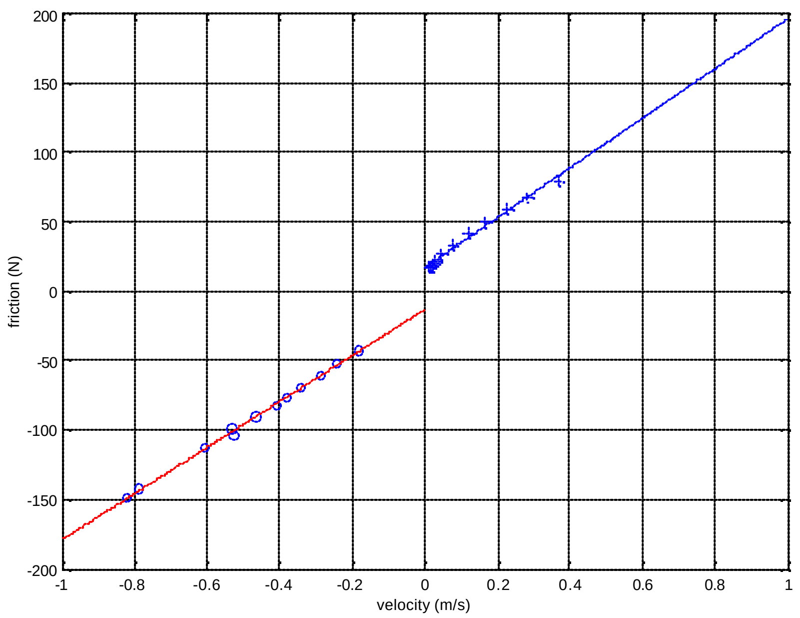

Figure 5. Determination of viscous damping coefficient (B) and Coulumb friction (F_C) to actuator N. 2.

Static Friction force (F_S) is determined by visual analysis in near to zero velocity region in each friction-velocity map, when is verified the tendency of friction curve and the possible point that it cuts vertical axis, that corresponds to F_S value. With determined values of F_S, F_C and B, Stribeck velocity ($\dot{y}_s$) is estimated by numerical optimization and curve adjustment procedures carried out Matlab software. Resulting curve is plotted in Fig. 6.

After repeating this procedure to each tested cylinder, Tab. 3 is obtained, with main friction force coefficients calculated to positive (extension) and negative (retraction) actuator velocities.

Friction static parameters from Tab. 3 permit to calculate friction force to each pneumatic actuator through Eq. (3). These results are plotted in Fig. 7, that is used to compare individual friction characteristics of tested cylinders.

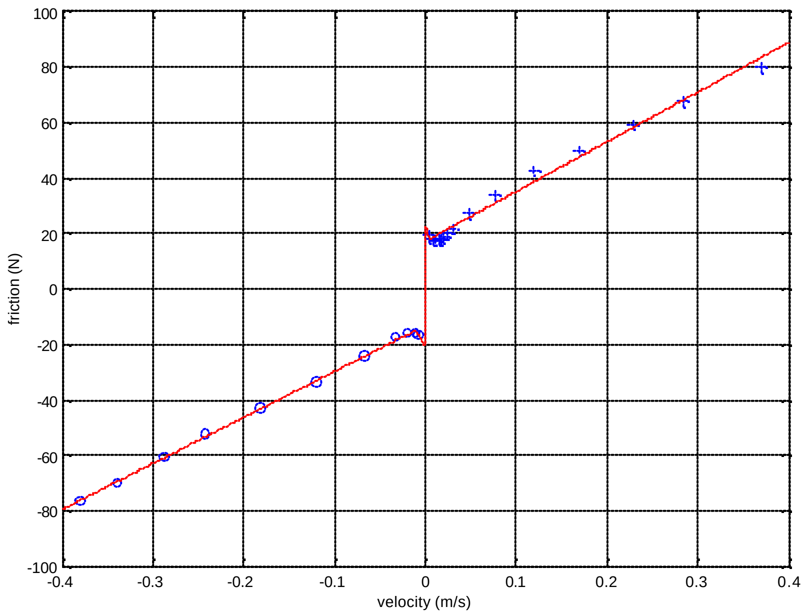

Figure 6. Modeled friction curve to actuator N. 2 and experimental friction points.

Table 3. Static friction parameters to tested pneumatic actuators.

Actuator Number	F_s ($\dot{y}>0$) (N)	F_s ($\dot{y}<0$) (N)	F_c ($\dot{y}>0$) (N)	F_c ($\dot{y}<0$) (N)	B ($\dot{y}>0$) N.s/m	B ($\dot{y}<0$) N.s/m	$\dot{y}_s$ ($\dot{y}>0$) mm/s	$\dot{y}_s$ ($\dot{y}<0$) mm/s
1	24	-31	23,10	-30,63	46,53	37,61	10,0	-10,0
2	22	-20	16,87	-13,47	178,85	165,59	3,42	- 6,16
3	5,5	-10	1,08	-6,63	201,35	247,59	10,4	-23,2
4	6	-11	1,59	-6,10	203,71	259,13	8,5	-8,9
5	10	-5	8,82	-3,71	43,89	40,13	10,0	-13,0
6	8	-6	6,42	-4,00	10,40	30,39	26,0	-25,9
7	10	-10	1,84	-3,99	74,69	78,04	26,0	-26,0
8	35	-32	20,36	-20,30	492,77	666,67	10,0	-10,0

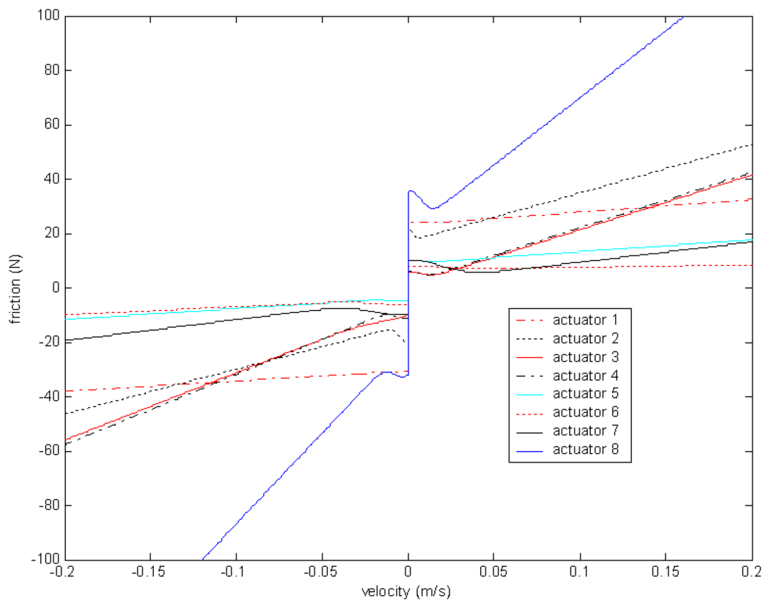

Figure 7. Friction map of tested actuators.

5. Analysis of the experimental tests

Experimental results depicted in Tab. 3 and Fig. 7 permit to conclude that a same actuator has different friction characteristics to extension and retraction movements. Also, friction varies with actuator manufacturer.

As a wanted characteristic, a minor difference between F_S and F_C friction parameters is better to obtain the precision control. In this way, actuator N. 1 has the smallest F_S - F_C value, although its viscous friction coefficient (B) is very small. This last characteristic conducts to low damping, that is negative to control. Actuator N. 1 is a rodlees actuator with special characteristics to precision control, even with its low B value.

Among the tested single-rod actuators, actuator N. 5 has the better relation between F_S and F_C and its B coefficient also is small. This actuator is the most adequate to precision control and its friction characteristics are similar to actuators N. 6 and N. 7, that have a great deal of grease. Because this reason, actuators N. 6 and N. 7 are not recommended to servo pneumatics, because their high risk to damage servo pneumatic valves. Actuator N. 8 has the biggest value to B coefficient, that conducts to larger damping.

The friction tests were fulfilled in a simpler way with experimental apparatus configured according to Fig. 2. The Fig. 3 configuration permits to carry out these tests without a proportional directional valve, although in a harder way.

6. Conclusions and future work

This work has shown friction characteristics of several tested pneumatic actuators from many different manufacturers through their main friction coefficients obtained from experimental friction-velocity maps. They can be used to model the friction behavior of pneumatic actuators and are important to select better actuators to application in servo pneumatics. The knowledge of the friction in pneumatic cylinder is an important step to friction compensation in the control system.

Manufacturers and the current literature approach friction in pneumatic actuators in an insufficient way to select and apply them in servo pneumatic positioning with precision control. This paper is a contribution to overcome this lack, thorough the comparison of the friction characteristics of several actuators and presentation of methods to identify friction in pneumatic cylinders.

As future works, the authors intend to verify the variation in friction maps caused by others air supply pressures and temperature changes. Extend these tests to actuators from other manufacturers will be other future step. A bigger challenge is to carry out experiments with a complete range of pneumatic cylinders to verify the possibility of identify practical rules, equations or experimental graphs to obtain main friction parameters from catalogue data without to do new tests.

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