Limited energy consumption in positioning control of an electropneumatic actuator

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ABSTRACT: This paper shows the possibility of using two three-way servo-distributors in place of one five-way for position control of an electropneumatic actuator. Two nonlinear control laws are developed and compared. The aim is to prove the advantage of a multi-input control law in terms of stability and energy consumption due to the flatness property.

NOTATION

a	piston acceleration [m/s^2]	T_c	period of the cycle (s)
b	viscous coefficient (N/m/s)	U	internal energy (J)
c	convection coefficient (W/K/m^2)	u	servo-distributor input voltage (V)
C_p	constant pressure specific heat (J/kg/K)	v	piston velocity (m/s)
		V	chamber volume (m^3)
CNL1	SISO Nonlinear Control	W	energy (J)
CNL2	MIMO Nonlinear Control	$\underline{x}$	state vector
F	force (N)	y	piston position (m)
h	mass enthalpy (J/kg)	γ	adiabatic coefficient
H	enthalpy flow (J/s)	$\Gamma(.)$	polynomial function of control [m^2]
j	jerk (m/s^3)		
k	polytropic constant	$\varphi(.)$	polynomial function [kg/s]
K	Feedback gain	$\psi(.)$	polynomial function [kg/s/m^2]
M	total moving load (kg)	ρ	characteristic number
n	system order		
p	pressure in the chamber (Pa)	Subscripts	
q_m	mass flow rate (kg/s)		
Q	heat transfer (J)	cyl	cylinder
r	perfect gas constant (J/kg/K)	E	exhaust
S	area of cylinder piston (m^2)	ext	external
t	time (s)	f	dry friction
T	temperature (K)	MECA	mechanical

N	chamber N	

N	chamber N
P	chamber P
PNEU	pneumatic
S	supply

Superscripts

d	desired
in	input: corresponds to a positive value
out	output: corresponds to a negative value

1   INTRODUCTION

The 1980s and 1990s have seen the origin and expansion of proportional pneumatic systems due to the development of new components such as servovalves and servo-distributors. Historically, the first pneumatic power modulator originated from hydraulic systems and it was a five-way device. Only one was needed for the control of a positioning system. The development of longer rodless cylinders is one of the principal reasons for the development of three-way electropneumatic modulators. In fact the integration of this component in each cylinder extremity reduces the congestion and the length of pneumatic pipes. Nowadays constructors usually use these two three-way modulators with the same air power supply and the same input controls but of opposite signs, which is equivalent to using one five-way servo-distributor.

The limitation of energy consumption is a very recent research theme, it has been developed with electric actuators (asynchronous motors) (**1**). In solutions envisaged for Fluid Power, Bachmann and Surgenor (**2**) proposed to modify the system structure. However in this paper the approach is different, since the structure of the system is not modified. The improvement in terms of fluid consumption is obtained without material cost: tanks, sensors, analogue cards; it is just a question of taking advantage of industrial actuator characteristics.

For positioning control the dimension of the equilibrium set of the system mathematical model is equal to one and so one input variable is sufficient to control the system output. It is attractive to use the second modulator to satisfy another criterion without deteriorating the position, velocity and acceleration tracking performance. As a consequence, the work presented in this paper uses the degree of freedom in the system to satisfy simultaneous tracking position, velocity and acceleration objectives and limitation of energy consumption. With two different control inputs the system is differentially flat and so the properties of flatness (**3**) are used to calculate a nonlinear tracking control law, which satisfies the double objective. Results of simulation are presented and compared with a classical monovariable nonlinear control law.

2   ELECTROPNEUMATIC ACTUATOR : MODEL FOR SIMULATION PURPOSE

The system under consideration (figure 1) 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.

It is assumed that the two servo-distributors are identical, without leakage and their dynamics are negligible, and that there is no leakage between the two cylinder chambers. Also that the air is a perfect gas and its kinetic energy is negligible in the chamber.
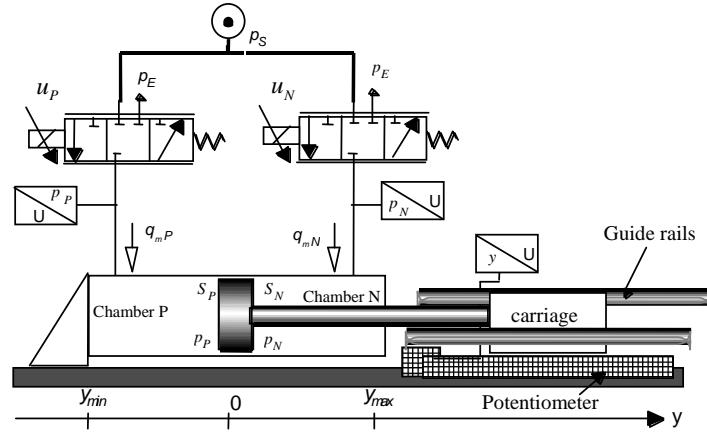
Figure 1 : Electropneumatic system

Using the first law of thermodynamics and the fundamental mechanical relation the model obtained is (**4**):

$$
\begin{cases}
\dfrac{dp_P}{dt} = \dfrac{1}{V_P(y)}\left[-\gamma p_P \dfrac{dV_P}{dt} + \gamma r\left(T_S q_m^{in}(u_P,p_P) - T_P q_m^{out}(u_P,p_P)\right) + (\gamma-1)cS_{cyl\,P}\left(T_{cyl} - T_P\right) \right] \\[2mm]
\dfrac{dT_P}{dt} = \dfrac{1}{V_P(y)}\left[-(\gamma-1)T_P \dfrac{dV_P}{dt} + \dfrac{rT_P}{p_P}(\gamma T_S - T_P)q_m^{in}(u_P,p_P) - \dfrac{rT_P^2}{p_P}(\gamma-1)q_m^{out}(u_P,p_P) + (\gamma-1)\dfrac{T_P}{p_P}cS_{cyl\,P}\left(T_{cyl} - T_P\right) \right] \\[2mm]
\dfrac{dp_N}{dt} = \dfrac{1}{V_N(y)}\left[-\gamma p_N \dfrac{dV_N}{dt} + \gamma r\left(T_S q_m^{in}(u_N,p_N) - T_N q_m^{out}(u_N,p_N)\right) + (\gamma-1)cS_{cyl\,N}\left(T_{cyl} - T_N\right) \right] \\[2mm]
\dfrac{dT_N}{dt} = \dfrac{1}{V_N(y)}\left[-(\gamma-1)T_N \dfrac{dV_N}{dt} + \dfrac{rT_N}{p_N}(\gamma T_S - T_N)q_m^{in}(u_N,p_N) - \dfrac{rT_N^2}{p_P}(\gamma-1)q_m^{out}(u_N,p_N) + (\gamma-1)\dfrac{T_N}{p_N}cS_{cyl\,N}\left(T_{cyl} - T_N\right) \right] \\[2mm]
\dfrac{dv}{dt} = \dfrac{1}{M}\left(S_P p_P - S_N p_N - bv - F_f - F_{ext} \right) \\[2mm]
\dfrac{dy}{dt} = v
\end{cases}
\tag{1}
$$

The static mass flow stage model $q_m(u,p)$ has been established by an experimental static global characterisation of the servo-distributor (**5**), the results are shown in figure 2.
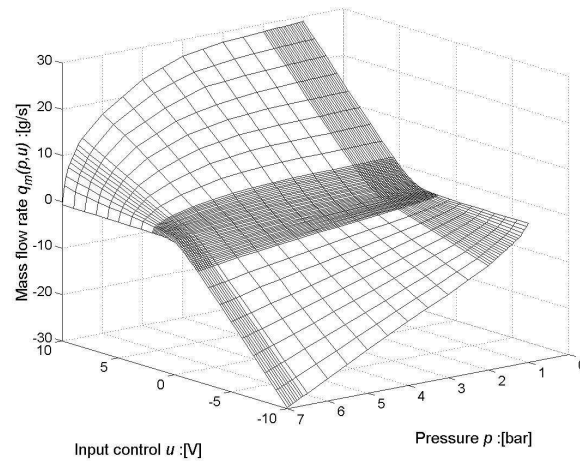

Figure 2 : Servo-distributor static characteristic

<u>Energy point of view:</u> In this part the electrical energy and the gas kinetic energy are neglected. For an open system, the first law of thermodynamics is given by the following equation:

$$\frac{dU}{dt} = h^{in}q_m^{in} - h^{out}q_m^{out} + \frac{\delta Q}{dt} + \frac{dW}{dt} \tag{2}$$

At the end of each position cycle the final values of pressure, temperature and volume in each chamber are equal to the initial values, so the compressibility energy and the internal energy variations are null. The energy given by the air supply W_{PNEU}^{in} is transformed into mechanical work W_{MECA}, into heat Q and is lost to exhaust W_{PNEU}^{out} as depicted in figure 3.

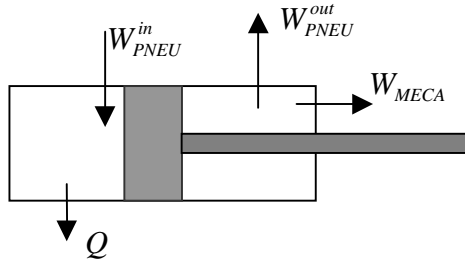$$W_{PNEU}^{in} = W_{PNEU}^{out} + W_{MECA} + Q \tag{3}$$

Figure 3 : Energy transfer

Enthalpy flows are defined as:
$$\begin{cases} H_P(t) = q_{mP}(t)C_pT_P \\ H_N(t) = q_{mN}(t)C_pT_N \end{cases}$$

- $W_{PNEU}^{in} = \int_0^{T_c} h^{in}q_m^{in}dt = \int_0^{T_c}\left[H_P^{in}(t) + H_N^{in}(t)\right]dt$

- $W_{PNEU}^{out} = \int_0^{T_c} h^{out}q_m^{out}dt = \int_0^{T_c}\left[H_P^{out}(t) + H_N^{out}(t)\right]dt$

- $W_{MECA} = \int_0^{T_c}\left[F(t).v(t)\right]dt$ with:

$F(t) = S_P p_P(t) - S_N p_N(t) - bv(t) - F_f - F_{ext}$

$Q = Q_P + Q_N$ with:

$$\begin{cases} Q_P = c\int_0^{T_c} S_{cylP}(t)(T_{ext} - T_P(t))dt \\ Q_N = c\int_0^{T_c} S_{cylN}(t)(T_{ext} - T_N(t))dt \end{cases}$$

3   CHOICE OF ADMISSIBLE TRAJECTORIES

The choice of position trajectory is a problem of sizing: the control input must not reach its maximum value during tracking control. Using nonlinear control theory, Richard and Scavarda (**6**) have shown that it is possible to calculate the theoretical control law associated with a given cylinder position trajectory. Sesmat et al (**7**) have proposed a procedure for verifying the servo-distributor size: using the reduced inverse model, the servo-distributor control area is calculated and its value must have a value smaller than a chosen fraction of the nominal area. With this constraint, the desired position trajectory in figure 4a is theoretically physically admissible by the system under consideration, without leading to control saturation. In this example, one cycle corresponds to one displacement in each direction around the central position of fifty percent of the stroke. Each displacement lasts 0.80 s and corresponds to a sixth order polynomial function due to position, velocity and acceleration initial and final conditions. Before each direction change, the piston stays in position for two seconds, the cycle period is $T_c=5.60$ s.

To reduce the energy consumption of the system, the mass flow rate delivered by the air supply in each chamber has to be reduced. The most important objective is not to increase the efficiency defined as $\eta = w_{MECA}/w_{PNEU}^{in}$ but to reduce the pneumatic energy W_{PNEU}^{in} necessary to assure the load displacement. So for a similar performance in terms of position, velocity and acceleration tracking, the aim is to control another output which is involved in the global pneumatic energy. The choice presented in this paper consists of controlling the pressure in one chamber. This idea comes from the following observation : during the part of the cycle

which corresponds to the rod output, the chamber P pressure successively increases, decreases and increases once more around a mean absolute pressure of 4.5 bar (figure 10). The same position trajectory could be obtained with many different pressure evolutions in each chamber. That is why the desired pressure trajectory in chamber P has been chosen with a smoother profile and a reduced absolute value of about 2 bar. The chosen chamber P reference pressure is a third order polynomial function (see figure 4b). Moreover, as explained in section 4.2 the pressure is a flat output which is appreciable for the system properties. When desired acceleration and pressure in chamber P are fixed, the pressure in chamber N is given by eq (4). The pressures in each chamber must evolve between the supply and exhaust pressures.

$$p_N(t) = \frac{1}{S_N}\left(S_P\, p_P^d(t) - bv^d(t) - F_f - F_{ext} - M\gamma^d(t)\right) \tag{4}$$

The figure 4 shows desired position, velocity, acceleration, jerk, pressure p_P and reconstructed pressure p_N and force during rod output. These trajectories during rod input are symmetrical.
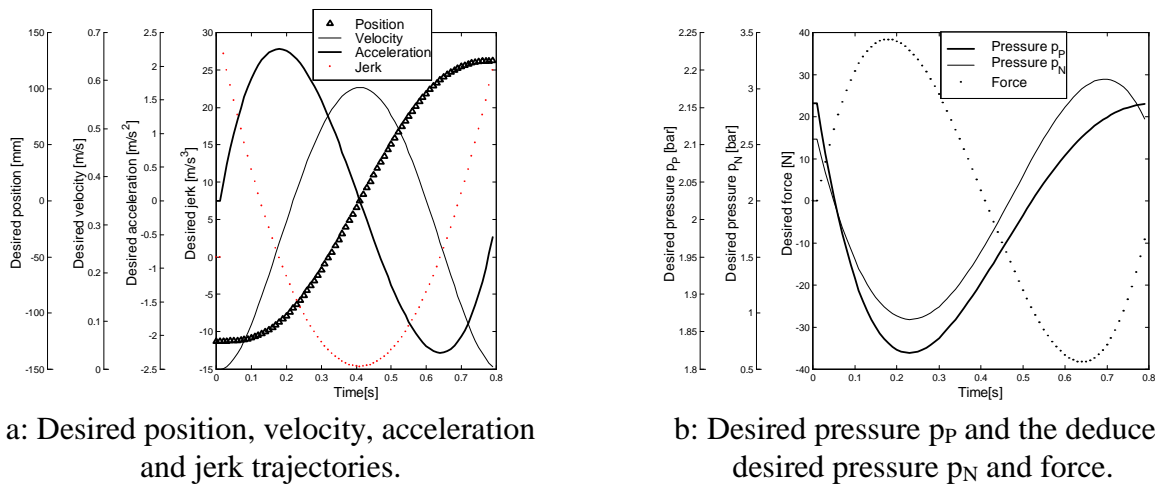

a: Desired position, velocity, acceleration
and jerk trajectories.

b: Desired pressure p_P and the deduced
desired pressure p_N and force.

Figure 4 : Example of desired trajectories

4 MODEL USED TO CALCULATE THE NONLINEAR CONTROL LAWS

For nonlinear control synthesis, the model must be a linear function of the control input, called affine in control theory. For this the static characteristic of the servo-distributor (figure 2) has been approximated to the following form by Belgharbi et al (**8**) :

$$q_m(u, p) = \varphi(p) + \psi(p, sgn(\Gamma(u))) \times \Gamma(u) \tag{5}$$

φ , ψ and Γ are polynomial functions of orders 5, 5 and 2 respectively.
$\Gamma(u)$ is a bijective function whose evolution is similar to the evolution of the equivalent section restriction that crosses the fluid as a function of spool position.
$\varphi(p)$ is a function whose evolution corresponds to the mass flow rate leakage.
$\psi(p, sgn(\Gamma(u)))$ is a function of the input control sign because the function is different for the inlet ($\Gamma(u) > 0$) and for the exhaust ($\Gamma(u) < 0$).

With the three following additional assumptions, the affine nonlinear model obtained is described by system 6.

- The process is supposed to be polytropic characterised by coefficient k
- The temperature variation in each chamber is negligible and so the temperature T_P and T_N are considered equal to supply temperature: $T_P = T_N = T_s$
- The importance of friction is a real problem in pneumatic positioning systems. It is difficult to identify because it depends on many parameters, such as position, pressure, temperature, past time, seal type… So in this first study the dry friction forces are neglected:

$$
\begin{cases}
\dfrac{dp_P}{dt} = \dfrac{krT_s}{V_P(y)}\left[\varphi(p_P) - \dfrac{S_P}{rT_s}p_P v\right] + \dfrac{krT_s}{V_P(y)}\psi\big(p_P, sgn(\Gamma(u_P))\big)\times\Gamma(u_P) \\[3mm]
\dfrac{dp_N}{dt} = \dfrac{krT_s}{V_N(y)}\left[\varphi(p_N) + \dfrac{S_N}{rT_s}p_N v\right] - \dfrac{krT_s}{V_N(y)}\psi\big(p_N, sgn(-\Gamma(u_N))\big)\times\Gamma(u_N) \\[3mm]
\dfrac{dv}{dt} = \dfrac{1}{M}\big[S_P p_P - S_N p_N - bv - F_{ext}\big] \\[3mm]
\dfrac{dy}{dt} = v
\end{cases}
\tag{6}
$$

4.1 S.I.S.O. system

In this section, the only objective is to minimise the tracking position errors. So only one control signal is used and sent with opposite signs to both servo-distributors : $u_P = -u_N = u$. Also the electropneumatic model (see equation 6) is in the nonlinear affine form $\dot{\underline{x}} = f(\underline{x}) + g(\underline{x})U$. A convenient way to linearise it for every position of the cylinder piston is to use the nonlinear linearising control law (**9**), which theoretically transforms the closed loop system into two parts. The first part consists of a cascade of ρ integrators and the second is an unobservable subsystem of dimension $(n-\rho)$ (ρ is the relative order of the output, n is the system order).

For the electropneumatic model (7) with $U = \Gamma(u)$

$$
\underline{x} = \begin{bmatrix} p_P \\ p_N \\ v \\ y \end{bmatrix} :\ f(\underline{x}) = \begin{pmatrix} \dfrac{krT_s}{V_P(y)}\left[\varphi(p_P) - \dfrac{S_P}{rT_s}p_P v\right] \\[3mm] \dfrac{krT_s}{V_N(y)}\left[\varphi(p_N) + \dfrac{S_N}{rT_s}p_N v\right] \\[3mm] \dfrac{1}{M}\big[S_P p_P - S_N p_N - bv - F_{ext}\big] \\[3mm] v \end{pmatrix} \text{ and } g(\underline{x}) = \begin{pmatrix} \dfrac{krT_s}{V_P(y)}\psi\big(p_P, sgn(U)\big) \\[3mm] -\dfrac{krT_s}{V_N(y)}\psi\big(p_N, sgn(-U)\big) \\[3mm] 0 \\[1mm] 0 \end{pmatrix}
\tag{7}
$$

The single output is $h(\underline{x}) = y$ By definition :

$$
\begin{cases}
L_f h \text{ is the Lie derivative of h along } f \text{ which is given by } L_f h(x) = \sum\limits_{i=1}^{n} \dfrac{\partial h}{\partial x_i}f_i(x) \\[3mm]
L_g h \text{ is the Lie derivative of h along } g \\[3mm]
\rho \text{ is the characteristic number associated to the output which is defined by } \rho = \min\left\{\rho \in N / L_g L_f^{\rho-1} \neq 0\right\}
\end{cases}
$$

A simple calculation shows that for the system under study the characteristic number is equal to three. The co-ordinates can be changed as $\begin{bmatrix} y & v & a & p_P \end{bmatrix} = \begin{bmatrix} h & L_f h & L_f^2 h & p_P \end{bmatrix}$. These relations define a local diffeomorphism around an equilibrium state. With the new co-ordinates, the closed-loop system becomes

$$\begin{cases} \dfrac{dy}{dt} = v \\[2mm] \dfrac{dv}{dt} = a \\[2mm] \dfrac{da}{dt} = w \\[2mm] \dfrac{dp_P}{dt} = \dfrac{krT_s}{V_P(y)}\left[\varphi(p_P) - \dfrac{S_P}{rT_S}p_P v + \psi(p_P, sgn(U))U\right] \end{cases} \qquad \text{where } w \text{ is the linearised system input.} \qquad (8)$$

In the single input output case, the linearising nonlinear control is given by:

$$U = \dfrac{1}{L_g L_f^2 h}\left(-L_f^3 h + w\right): \begin{cases} L_f^3 h = \dfrac{S_P}{M}\dfrac{krT_S}{V_P(y)}\left[\varphi(p_P) - \dfrac{S_P}{rT_s}p_P v\right] - \dfrac{S_N}{M}\dfrac{krT_s}{V_N(y)}\left[\varphi(p_N) + \dfrac{S_N}{rT_S}p_N v\right] \\[4mm] L_g L_f^2 h = \dfrac{S_P}{M}\dfrac{krT_S}{V_P(y)}\psi(p_P, sgn(U)) + \dfrac{S_N}{M}\dfrac{krT_s}{V_N(y)}\psi(p_N, sgn(-U)) \end{cases} \qquad (9)$$

It is important to note that $L_g L_f^2 h$ is always strictly positive. So U has the same sign as $\left(-L_f^3 h + w\right)$. By definition, w and $L_f^3 h$ are independent of U (relation 9). Consequently, the control law is not implicit. This remark explains the choice of the function φ independent of U in the mass flow rate expression of equation (5). The second feedback law which fixes the dynamic behaviour of the state tracking errors is:

$$w = j^d - K_y\left(y^d - y\right) - K_v\left(v^d - v\right) - K_a\left(a^d - a\right) \qquad (10)$$

The nonlinear control applied to the system is shown in figure 5 and may be written as:

$$u = \Gamma^{-1}\left(\dfrac{1}{L_g L_f^2 h}\left(-L_f^3 h + j^d - K_y\left(y^d - y\right) - K_v\left(v^d - v\right) - K_a\left(a^d - a\right)\right)\right) \qquad (11)$$
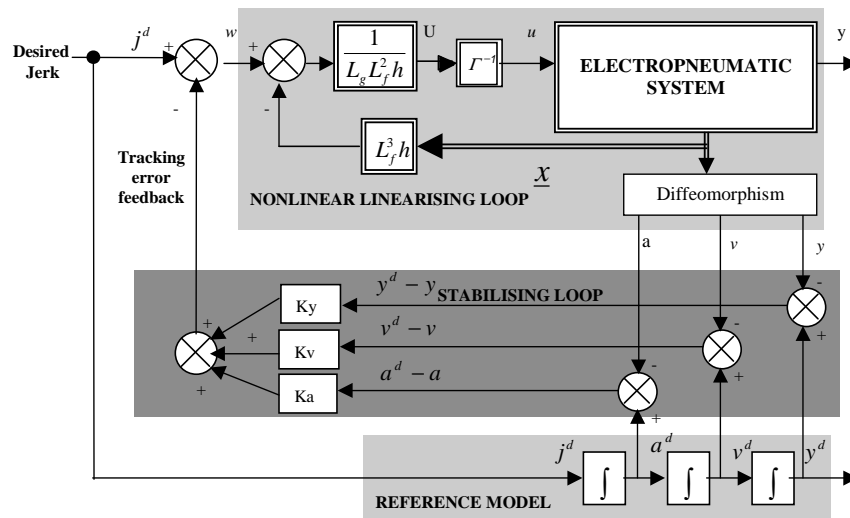

Figure 5 : Principle of nonlinear tracking control for the SISO system

The validity of the control law depends on the stability of the unobservable subsystem of dimension one. When the actuator stops (i.e. : $y = y_{stop}$, $v = 0$, $U = U_{stop}$), the asymptotic local stability of the equilibrium point of the internal dynamic can be proved. Moreover, the differential equation of p_P given by equation (8) has the form :

$$\frac{dp_P}{dt} = \frac{krT_s}{V_P(y_{stop})}\left[\varphi(p_P) + \psi(p_P, sgn(U_{stop}))U_{stop}\right] \tag{12}$$

Let us assume that $U_{stop}>0$. This first order nonlinear equation has only one equilibrium point p_P^e because the two functions φ and ψ decrease monotonously, $\psi(p_P)$ is always positive and $\varphi(p_P)$ has a zero. Then using the Lyapunov function $V = \frac{1}{2}(p_P - p_P^e)^2$ the asymptotic stability of the equilibrium point p_P^e can be easily proved. The same result is obtained for $U_{stop}<0$.

4.2 M.I.M.O. system

With two inputs u_P and u_N, the model given by equation (6) has the form $\underline{\dot{x}} = f(\underline{x}) + G(\underline{x})\underline{U}$ with the same vector f defined by equation (7) and :

$$G(\underline{x}) = (g_1(\underline{x})\ g_2(\underline{x})) = \begin{pmatrix} \frac{krT_s}{V_P(y)}\psi(p_P, sgn(U_P)) & 0 \\ 0 & -\frac{krT_s}{V_N(y)}\psi(p_N, sgn(U_N)) \\ 0 & 0 \\ 0 & 0 \end{pmatrix} \qquad \underline{U} = \begin{pmatrix} U_P \\ U_N \end{pmatrix} = \begin{pmatrix} \Gamma(u_P) \\ \Gamma(u_N) \end{pmatrix} \tag{13}$$

The two chosen outputs are the pressure in chamber P and the position:

$$h(\underline{x}) = \begin{pmatrix} h_1(\underline{x}) \\ h_2(\underline{x}) \end{pmatrix} = \begin{pmatrix} p_P \\ y \end{pmatrix} \tag{14}$$

The characteristic number associated to the pressure p_P and the position are one and three respectively, so the sum of them is equal to the dimension of the system. This is sufficient to affirm that the system is differentially flat (**3**). Due to the flatness definition , the inputs can be written as functions of outputs and a finite number (equal to the characteristic number) of their time derivatives. Two methods are possible for calculating the nonlinear linearising control law (**10**). The theory of Isidori (**9**) can be developed for a MIMO system that requires the calculation of these Lie derivatives: $\{L_f h_1, L_f^2 h_1, L_f h_2, L_f^2 h_2, L_f^3 h_2, L_{g_1} h_1, L_{g_1} L_f^2 h_2, L_{g_2} h_1, L_{g_2} L_f^2 h_2\}$.

Another way is to use the expressions of $\frac{dp_P^d}{dt}$ and j^d and to invert them which is a more direct way to obtain the nonlinear linearising control laws which decouple the system:

$$\begin{cases} u_P = \Gamma^{-1} \left[\dfrac{\dfrac{S_P}{rT_S} p_P^d v^d - \varphi(p_P^d)}{\psi(p_P^d, sgn(U_P))} + \dfrac{\dfrac{V_P(y^d)}{krT_S}}{\psi(p_P^d, sgn(U_P))} w_1 \right] \\[4em] u_N = \Gamma^{-1} \left[-\dfrac{\dfrac{S_N}{rT_S} p_N^d v^d + \varphi(p_N^d) + \dfrac{V_N(y^d)}{krT_S} \dfrac{b}{S_N} a^d}{\psi(p_N^d, sgn(U_N))} + \dfrac{\dfrac{V_N(y^d)}{krT_S}}{\psi(p_N^d, sgn(U_N))} \left(-\dfrac{S_P}{S_N} w_1 + \dfrac{M}{S_N} w_2 \right) \right] \end{cases}$$
(15)

With
$$p_N^d = \frac{1}{S_N}\left(S_P p_P^d - bv^d - F_{ext} - Ma^d \right)$$
(16)

With the two previous controls inputs u_P and u_N the nonlinear electropneumatic model is transformed into two linear models, one simple integrator and one triple integrator which can be stabilised by classic feedback laws:

$$\begin{cases} \dfrac{dp_P}{dt} = w_1 \\ \dfrac{dy}{dt} = v \\ \dfrac{dv}{dt} = a \\ \dfrac{da}{dt} = w_2 \end{cases} \quad \text{with} \quad \begin{cases} w_1 = \dfrac{dp_P^d}{dt} - K_{pP}\left(p_P - p_P^d\right) \\ w_2 = j^d - K_a\left(a - a^d\right) - K_v\left(v - v^d\right) - K_y\left(y - y^d\right) \end{cases}$$
(17)

In this case the global stability is obvious. This is one of the main advantages of flatness. The two inputs in relation (15) can be written in the following form appearing in figure 6.

$$\begin{cases} u_P = \Gamma^{-1}\left[\dfrac{1}{\psi(p_P^d)}\left(A_P(p_P^d, v^d, a^d) + B_{1P}(y^d) w_1 \right) \right] \\ u_N = \Gamma^{-1}\left[\dfrac{1}{\psi(p_P^d, v^d, a^d)}\left(A_N(p_P^d, v^d, a^d) + B_{1N}(y^d) w_1 + B_{2N}(y^d) w_2 \right) \right] \end{cases}$$
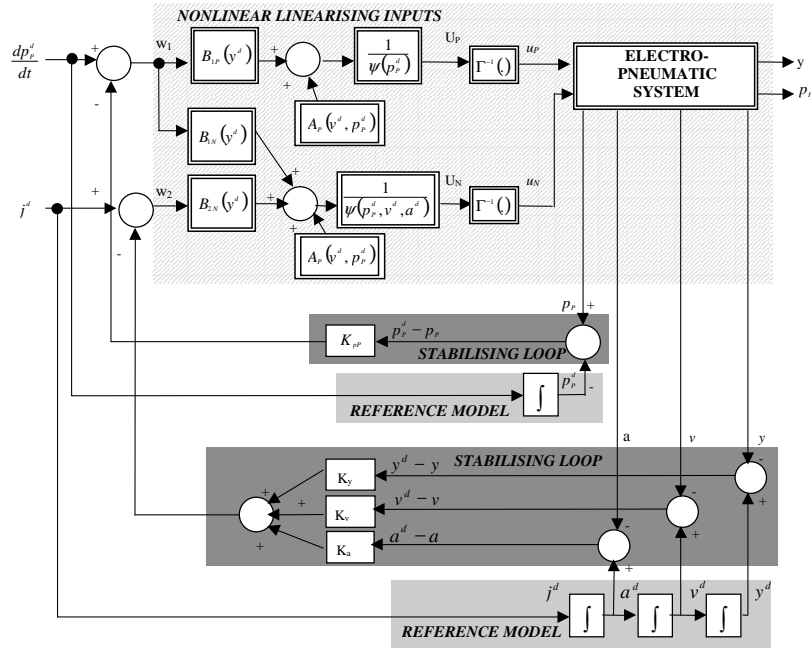(18)

Figure 6 : Principle of nonlinear tracking control for the MIMO system

5 SIMULATED RESULTS

The two nonlinear control laws have been implemented and tested using the model defined by relation 1 with industrial specialised software: AMESim. (**A**dvanced **M**odeling **E**nvironment for performing **Sim**ulations of engineering systems), see figure 7. One advantage of this software is due to its approach, which uses the bond graph language. The pneumatic energy port used in AMESim reproduces the power bond of the pseudo bond graph; the two couples of pseudo bond power variables are temperature and enthalpy mass flow rate or pressure and mass flow rate. The pneumatic library has been developed in the INSA laboratory (**11**) in collaboration with the Imagine company.
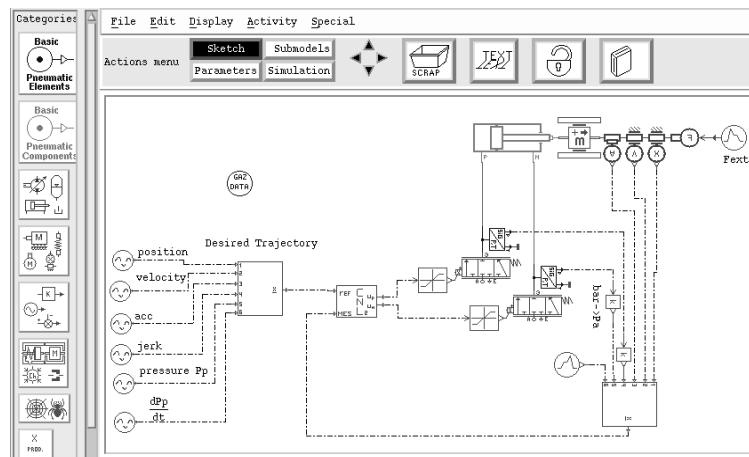

Figure 7 : Simulated model using AMESim software

The power evolution during one position cycle with the two different control laws is shown in figure 8. In both cases it is important to remark that the efficiency is very low. This characteristic of pneumatic systems has already been noticed by Bachmann and Surgenor (**2**) who considered adiabatic systems and obtained a partial efficiency of 9.4%. In the present case, considering heat exchanges, the efficiency is equal to 3.7% for the SISO system and 6.2% for the MIMO system. However, as explained in section 3, the most important value concerns the necessary pneumatic energy delivered by the air supply which corresponds to the input enthalpy flow integral. In this example (figure 8), the value is reduced from 537 J to 325 J for one cycle with the proposed new nonlinear control. This improvement of **52%** corresponds to real gain.
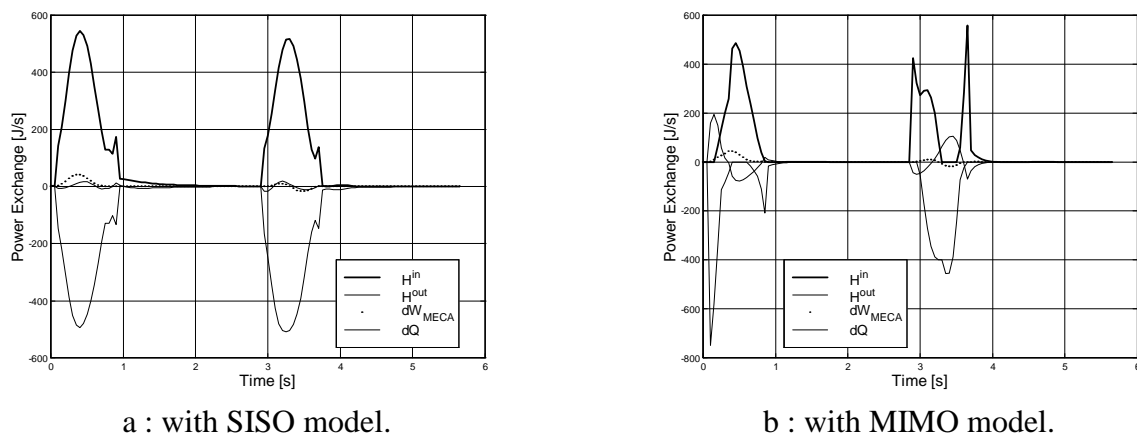

a : with SISO model.   b : with MIMO model.
Figure 8: Power variation during one cycle

This significant improvement is obtained without degradation of tracking results as shown in the figures 9 and 10. On the contrary, for this desired trajectory the simulated position and velocity tracking errors are both improved by about 50 % as summarised in table 1. This can be explained by smaller errors in the approximation of equation 5 for the working conditions of the second case (**8**).
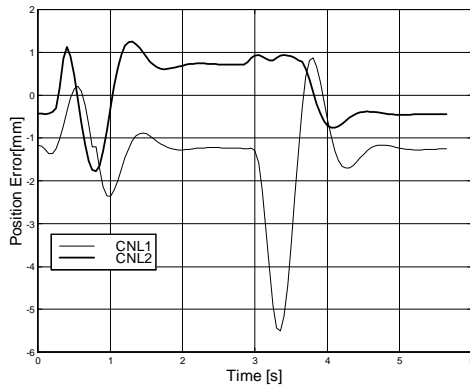

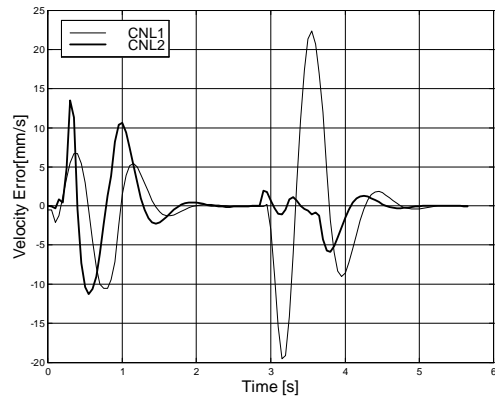**Figure 9 : Position errors with CNL1 and CNL2**

Figure 10 : Velocity errors with CNL1 and CNL2

With : $\overline{\varepsilon_y}$ and $\overline{\varepsilon_v}$: mean absolute errors along one cycle in position and velocity respectively.

ε_y^{max} and ε_y^{max} : optimal variations errors along one cycle in position and velocity respectively.

$\Delta\varepsilon_y^{max}$ and $\Delta\varepsilon_y^{max}$: maximum magnitude errors along one cycle in position and velocity respectively.

	Position Error			Velocity Error		
	$\overline{\varepsilon_y}$ [mm]	ε_y^{max} [mm]	$\Delta\varepsilon_y^{max}$ [mm]	$\overline{\varepsilon_v}$ [mm/s]	ε_y^{max} [mm/s]	$\Delta\varepsilon_y^{max}$ [mm/s]
CNL1	**1.43**	-5.51 to 0.87	6.37	**3.7**	-19.5 to 22.4	41.9
CNL2	**0.69**	-1.77 to 1.25	3.03	**1.9**	-11.3 to 13.5	24.7
	52 %	← _Improvement_ →		**48 %**		

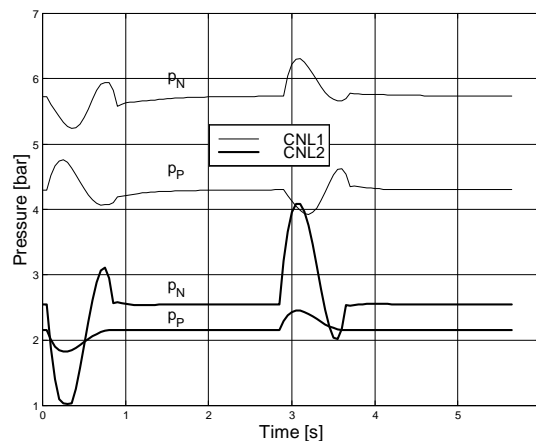**Table 1 : Tracking improvement**

Figure 11 : Pressure in chamber P and N with the two nonlinear controls

Figure 11 shows the pressure trajectories for the two different controls. The chosen pressure trajectory in chamber P has the consequence of producing a pressure in chamber N which

evolves between the supply and exhaust pressures. This is not the optimum solution, but for this case p_N is always lower than both CNL1 pressures and becomes near to p_E, so the energy is decreased. Moreover it is useful to notice than when the load stops at t=0.8s and t=3.6s the partial equilibrium due to the system parallel structure lasts a shorter time with the MIMO model. This means that the MIMO nonlinear control law may be a good solution to limit the sticking and restarting phenomenon (**12**).

6   CONCLUSIONS AND PERSPECTIVES

Since the early development of pneumatic actuators, the principle of control consists of filling one chamber with air and emptying the other. Energy is required to raise the pressure. Unfortunately nearly the same amount of energy is evacuated from the other chamber to the exhaust and less than 10 % is transformed into mechanical energy required for the displacement. So the energy consumption could be greatly reduced. For this two ways are possible, without change at the design of the system, it has been proved that if another output is controlled the efficiency can be increased and the consumption can be reduced. These first simulated results are encouraging and have to be continued in the following directions: the additional desired trajectory has to be optimised, the effect of friction and the system stiffness have to be studied, and the algorithm of the MIMO nonlinear control has to be implemented experimentally.

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