HYBRID FORCE CONTROL
WITH ON/OFF ELECTROPNEUMATIC STANDARD DISTRIBUTORS

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Abstract

This paper presents a new control method applied to the electro-pneumatic field. This strategy originates from the hybrid control theory recently developed for the control of asynchronous or synchronous electrical motors, e.g. Retif (2004). The interest of this strategy concerns the possibility of using standard on/off distributors instead of the usual servodistributors (components issued from proportional technology) for the force control of a pneumatic cylinder. Distributor components have less performance but are cheaper than a servodistributor. The aim is to obtain, with a distributor, the same performances as servodistributors on the global system.

Based on both cylinder and distributor models, the hybrid control presented here chooses the best state for each on/off distributor to reach the desired force value. Experimental results are presented and discussed.

Keywords: hybrid control, on/off distributor, valve, electro-pneumatic, experimental results, force control

1 Introduction

At present in the pneumatic field, when a system requires a wide range of force control, servodistributors are currently used, e.g. Ben-Dov (1995), Edge (1987), Richer (2000), Yamada, (2003), Yin (1998). On/off distributors are used when the system is simpler and only a small range of force values are required, e.g. Ham (2005). The work presented here consists of using on/off distributors in a system that requires a wide range of force values. This has been carried out in order to reduce the cost of complex systems, e.g. Legrand (2005), Van Varseveld (1997), because present day distributors are five to ten times cheaper than servodistributors.

Using on/off distributors for reaching a desired force produces a system with energy modulators (on/off distributors) and a continuous sub-process (cylinder). Such systems define a type of Hybrid Dynamic Process (HDP). Hybrid control, e.g. Retif (2004), Morel (2004), which is an efficient approach for controlling this kind of system. It is based on a state space model for both the energy modulator and the continuous sub-process.

For this model, a control vector, depending on the number of possible configurations for the energy modulator, is defined. A formal approach has been developed for choosing a control vector to track the reference state of interest in the state space. This control has been developed for the control of AC machines driven by inverters. Very good dynamic performance has been obtained (Retif, 2004). In this paper, the hybrid control is applied for the force control of the pneumatic cylinder where the chosen state variables of interest are the pressure in each chamber noted p_P and p_N. In fact, the control of both pressures leads to the control of the pneumatic force. Usually in the electrical field, from which the hybrid control algorithm is issued, the time commutation of the transistors can be neglected, e.g. Retif (2004), Morel (2004), with regards to the sample time (respectively equal to few microseconds and few hundred microseconds). The main difficulty in our context is that the on/off distributors used are quite slow. In the electropneumatic field, e.g. Burrows (1972), the bandwidth of cheap on/off components is very small (between 5 and 20 Hertz) and cannot be neglected when the control frequency is of the

same order. Some propositions were proposed in literature for increasing the dynamic performance of pneumatic servo-systems with digital valves. In this paper the aim is not to use PWM concept, e.g. Parnichkun (2001), for limited energy consumption, and not to increase the number of valves as in Belforte (2004), for limited cost. This problem is not present when the power modulator is a servodistributor and therefore other control algorithms are used, e.g. Noristugu (1988), Edge (1987), Brun (2002), Bouri (1996), Brun (2003), Liu (1993), because hybrid control is not appropriate with this type of component.

The study begins with the modelling of the continuous sub-process, then of the distributors, these are divided into two parts: static and dynamic models. The hybrid control algorithm is then developed. Finally, the first experimental results are presented and discussed and perspectives are announced for improving the performance of the system.

2 Electro-Pneumatic System Modelling

In order to determine the best distributor state for reaching the desired output value, the hybrid control requires both cylinder and distributor control models. This model must use physical parameters which can be given by industrial manufacturers or which can be obtained easily from simple experimental tests. In fact the following approach can be easily generalised to any other electropneumatic on/off distributor.

The pneumatic system consists of a pneumatic cylinder and two electropneumatic on/off distributors. The notation 'P', respectively 'N' on the schema in Fig. 1 is attached to elements whose positive actuation induces an increase in the exerted force, or respectively, a decrease.

Table 1 shows the nine different combinations of control vectors (C1 to C9) which can be applied to the distributors. The '0' value corresponds to a null voltage and the '1' value corresponds to 5 volts. All the states where $U_{Px} = U_{Nx} = 1$, are prohibited to avoid short-circuits in the electrical distributors. :

2.1 Actuator

The system is composed of a single rod double acting linear pneumatic cylinder (32/20). The rod of the actuator is attached to a fixed end. Considering that the stiffness is infinite, there are no variations in the volume of the cylinder chambers. In our case the piston position is set to obtain the same volume in both chambers of the actuator, noted *V* and equal to about 0.1 litre. The aim is to control the force exerted against the stop end. Considering that the dry friction forces can be neglected, the general principal of mechanics gives the expression of the force as a function of both pressures:

$$F = S_P p_P - S_N p_N - (S_P - S_N)p_E \qquad (1)$$

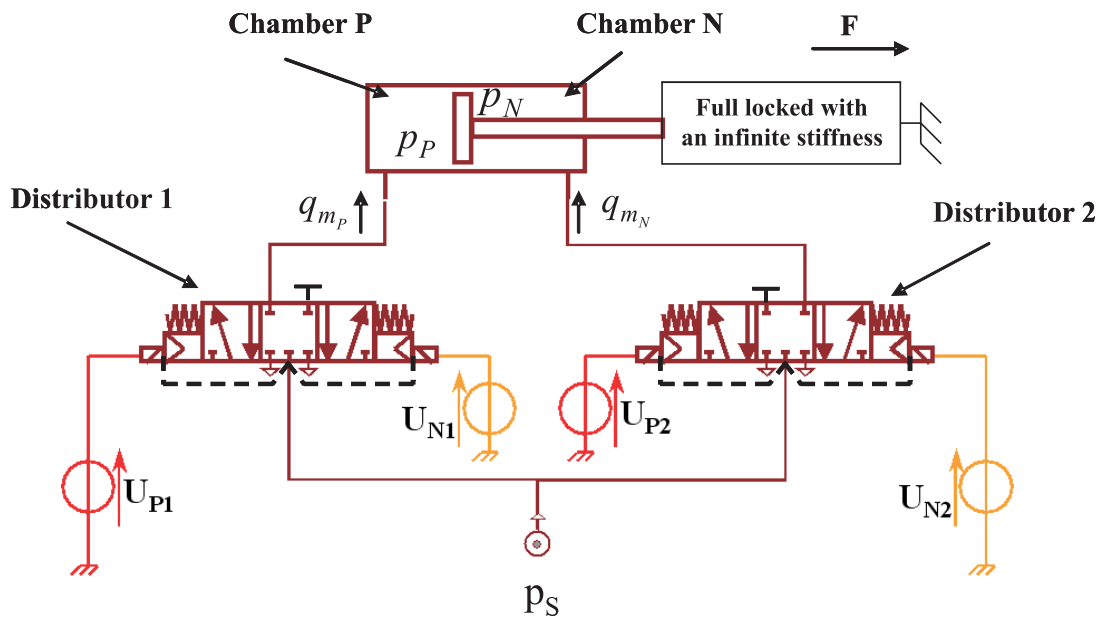So the force F can be controlled due to the control of the pressure in each chamber p_P and p_N.

Fig. 1: *Schema of the electro-pneumatic system*

Table 1: Applicable controls

	C1	C2	C3	C4	C5	C6	C7	C8	C9
U_{P1}	0	1	0	0	0	1	0	0	1
U_{N1}	0	0	1	0	0	0	1	1	0
U_{P2}	0	0	0	1	0	1	0	1	0
U_{N2}	0	0	0	0	1	0	1	0	1

With the following classical assumptions, e.g. Shearer (1956), Andersen (1967), the model used for synthesise control law can be described by the system (2).

- air is a perfect gas and its kinetic energy is negligible in the chamber,
- pressure and the temperature are homogeneous in each chamber,
- the evolution in each cylinder chamber is supposed to be polytrophic of index *n*,
- the temperature variation in each chamber is considered negligible with respect to the supply temperature,
- the mass flow rate leakages are neglected, supply and exhaust pressures are assumed to be constant.

$$\begin{cases} \dfrac{dp_P(t)}{dt} = \dfrac{nrT}{V} q_{m_P}\left(x_P\left(U_{P1}, U_{N1}, t\right), p_P(t)\right) \\ \dfrac{dp_N(t)}{dt} = \dfrac{nrT}{V} q_{m_N}\left(x_N\left(U_{P2}, U_{N2}, t\right), p_N(t)\right) \end{cases} \quad (2)$$

The main difficulty concerns the distributor mass flow rate modelling, q_m, described in the next part.

2.2 On/Off Distributor

The distributor used in this work is a 5 way / 3 position, centre position closed (from the Asco Joucomatic company) (Fig. 2a). Both components are supposed to be identical. There are two relays to move the spool to the left or to the right. These different positions allow the pressure to increase, decrease or stabilise in the chamber. According to classical hypothesis the establishment of flow is instantaneous, which means that the transient flows can be neglected, the modelling study of the distributor can be split into two parts: a static part giving the mass flow rate as a function of the distributor spool position x and of the output port pressure $p(q_m(x,p))$ and a dynamic part characterising the spool dynamic $(x(U_P, U_N, t))$. Figure 2b shows the system representation of the distributor model.
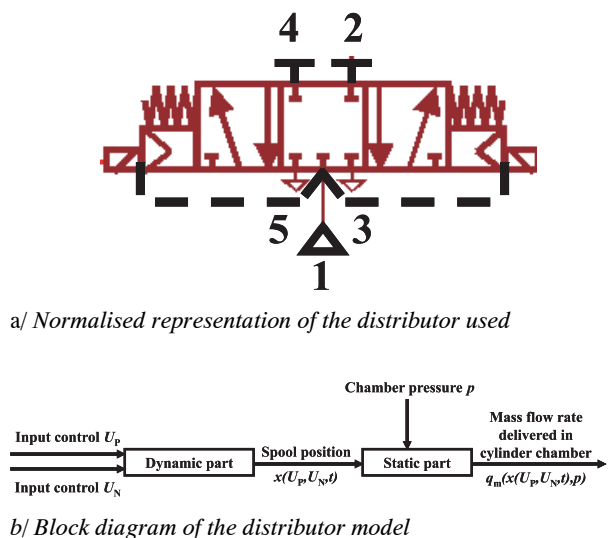

a/ Normalised representation of the distributor used

b/ Block diagram of the distributor model

Fig. 2: *Distributor*

2.3 Dynamic Part

The main drawback of on/off distributors compared to servodistributors is their high opening and closing time. In addition, for the components tested, the time is different for opening and closing, the closing time being about double the opening time. This difference is due to the force used to carry out the operation. The opening is carried out with the electro-magnetic force and the source pressure, while the closing is done by the mechanical force of the return spring. So, the component is asymmetrical.

This property has been compensated experimentally by applying a voltage, for a small time, on the opposite relay of the distributor in order to accelerate the spool to change position (go back to the centre or to the opposite side). The time of applying the voltage was specified for each distributor and tuned experimentally.

When connecting the distributor to a pneumatic chamber, it has been seen that the low dynamics is mainly due to the delay of 15 ms originating from high dry friction forces. Moving the spool then takes a few milliseconds. So, for a first control model, the displacement of the spool is assumed to be instantaneous and the dynamic part of the distributor model is just modelled as a delay (Fig. 3). The sampling period noted T_{st}, is then chosen as 20 milliseconds to guarantee that at each sampling time, the spool is either in an extreme position or in the central position. So the normalised position x of the spool, has only 3 different values: {-1 , 0 ,1}.

The consequence of the delay is that the variation of the pressure in a chamber for a control at time kT_{st} depends on the previous position of the distributor spool produced by the previous control at time $(k-1)T_{st}$. It means that the effective application of the control is delayed (Fig. 3). This problem has to be taken into account in the control algorithm.
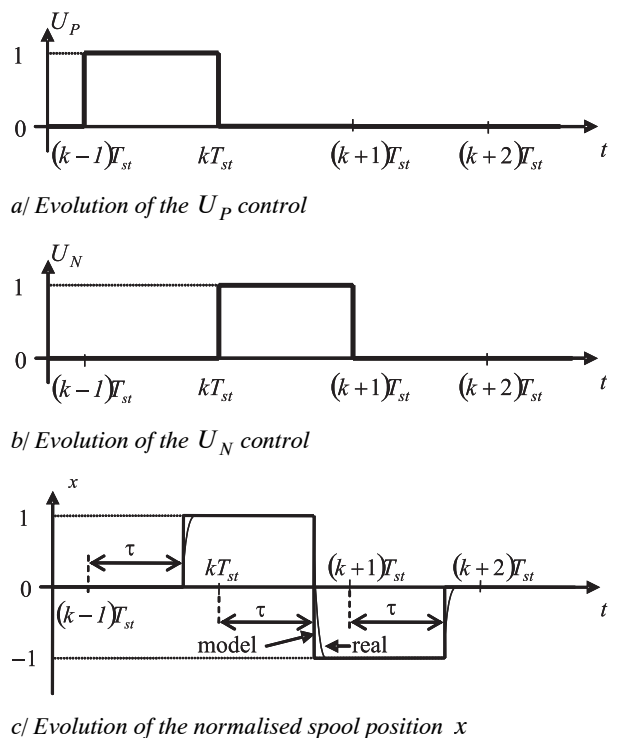

a/ Evolution of the U_P control

b/ Evolution of the U_N control

c/ Evolution of the normalised spool position x

Fig. 3: *Example of the evolution of the normalised spool position of the distributor*

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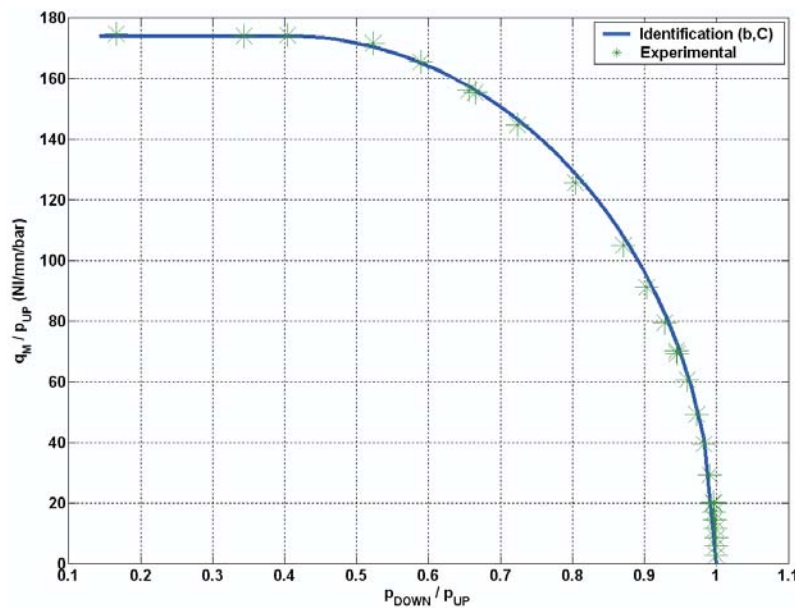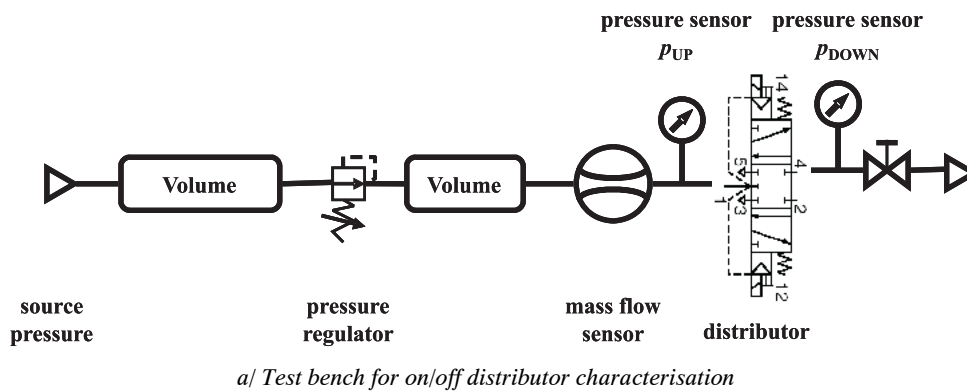
According to the modelling hypothesis, the following dynamic model is then proposed with τ being the time delay of the spool to change position when the control is applied to Eq. 3 :

With three possible states (Table 1) :

$$\begin{cases} U_P = 0 \\ U_N = 0 \end{cases} \quad or \quad \begin{cases} U_P = 1 \\ U_N = 0 \end{cases} \quad or \quad \begin{cases} U_P = 0 \\ U_N = 1 \end{cases}$$

$$x\left(U_P\left(kT_{st}\right), U_N\left(kT_{st}\right), t\right) = \begin{cases} U_P\left((k-1)T_{st}\right) - U_N\left((k-1)T_{st}\right) & \text{if} \quad kT_{st} \leq t < kT_{st} + \tau \\ \\ U_P\left(kT_{st}\right) - U_N\left(kT_{st}\right) & \text{if} \quad kT_{st} + \tau \leq t < (k+1)T_{st} \end{cases} \tag{3}$$

$$\text{if} \quad \frac{p_{DOWN}}{p_{UP}} \leq b \quad \text{(sonic flow)}, \quad q_m\left(p_{UP}, p_{DOWN}\right) = C\rho_0 p_{UP}\sqrt{\frac{T_0}{T_{UP}}}$$

$$\text{if} \quad b < \frac{p_{DOWN}}{p_{UP}} \leq 1 \quad \text{(subsonic flow)}, \quad q_m\left(p_{UP}, p_{DOWN}\right) = C\rho_0 p_{UP}\sqrt{\frac{T_0}{T_{UP}}}\sqrt{1 - \left(\frac{\dfrac{p_{DOWN}}{p_{UP}} - b}{1-b}\right)^2} \tag{4}$$

a/ Test bench for on/off distributor characterisation

b/ Experimental and identified mass flow rate distributor characteristics

Fig. 4: *Distributor characterisation*

2.4 Static Part

The static part of the on/off distributor model should consist of the expression of the mass flow rate through each useful restriction as a function of the chamber pressure p and of the normalised spool position x. As for the determination of the dynamic model, it has been considered that the movement of the spool is instantaneous, it is just necessary to know the mass flow rate characteristics of the on/off distributor for the full opening of the useful restrictions. According to Fig. 1 and 2, these restrictions are :

- restrictions 1 to 4 ($x_P = 1$) and 4 to 5 ($x_P = -1$) for distributor 1,
- restrictions 1 to 2 ($x_N = -1$) and 2 to 3 ($x_N = 1$) for distributor 2.

According to the ISO 6358 (1989) standard, a test bench was realised (see Fig. 4a) to measure the mass flow rate characteristics to determine the flow parameters b and C used by the standard approximation (4). This model gives the mass flow rate evolution as a function of the upstream pressure and temperature and of the downstream pressure defined along the flow in the restriction.

Figure 4b shows an example of the experimental characteristics obtained compared to those obtained from the identified b and C parameters according to the ISO 6358 standard (1989).

As the four characteristics are very similar, the same couple of parameters (b,C) according to the ISO 6358 standard have been determined for modelling every restriction :

Critical pressure ratio b = 0.4,
Sonic conductance C = 174 Nl/mn/bar

So the mass flow rates entering the 2 chambers can be expressed as a function of the spool position and of the chamber pressure by :

$$q_{m_P}(x_P, p_P) = \begin{cases} q_m(p_S, p_P) & \text{for} \quad x_P = 1 \\ 0 & \text{for} \quad x_P = 0 \\ -q_m(p_P, p_E) & \text{for} \quad x_P = -1 \end{cases} \quad (5)$$

$$q_{m_N}(x_N, p_N) = \begin{cases} -q_m(p_N, p_E) & \text{for} \quad x_N = 1 \\ 0 & \text{for} \quad x_N = 0 \\ q_m(p_S, p_N) & \text{for} \quad x_N = -1 \end{cases} \quad (6)$$

The q_m function in Eq. 5 and 6 is given by (4) in which the flow parameters b and C have the previous determined values and the upstream temperature is considered equal to the source temperature whatever the flow direction (according to hypotheses taken for the cylinder model section 2.1).

The Eq. 3 to 6 enable knowledge of each variable appearing in the system to be obtained (2) which together define the global control model of the system presented in Fig. 1.

3 Hybrid Control

3.1 Hybrid Control Principle

Hybrid control uses a hybrid model where the continuous state variables of the continuous sub-process depend on the energy modulator's configuration:

$$\underline{\dot{X}}(t) = f\left(\underline{X}(t), \underline{U}(t)\right)$$

$\underline{X} \in \Re^n$ and $\underline{U}(t)$ is a m-dimensional vector that is finite

$$U \in \{U_1, U_2, ..., U_N\}, N \geq 2$$

For a short sampling period T_{st}, the model can be written as(Euler method):

$$\underline{X}(k+1) = \underline{X}(k) + f\left(\underline{X}(t), \underline{U}(t)\right) \cdot T_{st}$$

Assuming that full state is available for measurement, at time kT, the state $\underline{X}(k)$ is known. Via the dynamic model of the system (1), the state at time $(k+1)T_{st}$, noted $\underline{X}_j(k+1)$, $j \in N$, for each possible value of the control set can be calculated. N directions are defined as $\underline{dj} = \underline{X}_j(k+1) - \underline{X}(k)$.

For a given reference state $\underline{X}^\#$, Hybrid Control consists of calculating the N possible direction of the continuous state vector evolution d, choosing a control configuration in order to track this reference state in the state space.

For the bi-dimensional example in Fig. 5, the desired value (target point) can be placed in the plane X_1 and X_2. To track this reference, at each sample time, hybrid control proceeds as follows:

- It acquires the state of the system at time kT_{st} ($X_1(kT_{st})$ and $X_2(kT_{st})$ see Fig. 5)
- Knowing the state of the system in the state space at time kT_{st}, this algorithm solves the system model equations, and calculates the different directions $\underline{dj}$, $1 \leq j \leq N$ of the continuous state vector evolution corresponding to the application of jth configuration of the energy modulator ($\underline{d1}$ to $\underline{d5}$ in Fig. 5)
- Knowing the target point, this algorithm selects the optimal configuration. For this many techniques are possible. For example, the configuration chosen can be the one that minimises the Euclidean distance between the different reachable points and the target point.

This algorithm chooses the shortest euclidean distance ($\underline{d4}$ in Fig. 5) and the corresponding control (U_4) is applied to the energy modulators for all the sampling period.
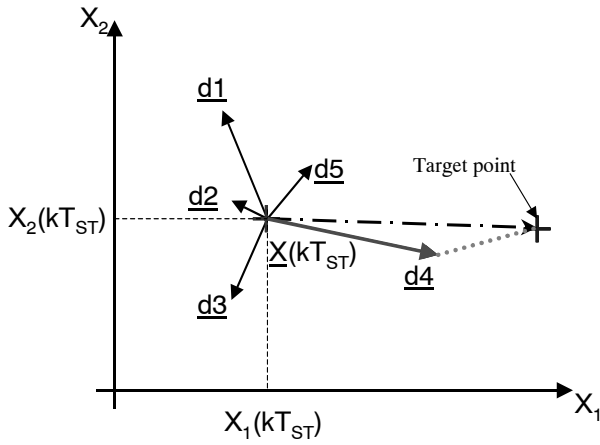
Fig. 5: *Principle of hybrid control*

•••••► **Direction corresponding to the 9 control distributors.**

——— **Line corresponding to the F order.**

•••••• **Physical domain limits.**

– – **Line perpendicular to the F order to determine the Euclidean distance.**

Fig. 7: *Example of control choice*

3.2 Application to the pneumatic system

For the system represented by Fig. 1, the pressures in both chambers P and N constitute the state space $\underline{X}(t) = \left[p_\text{P}, p_\text{n} \right]^t$. The state space is of dimension 2, so it can be represented by a (p_N, p_P) plane. Two distributors enable three different states (pressure admission, closed and pressure exhaust) to be established leading to nine different control vectors C1 to C9 (see table 1).

The objective is, knowing the pressure at the sampling instant kT_st, to estimate the evolution of the pressures at the next sampling instant, $(k+1)T_\text{st}$, in both chambers P and N for the nine controls (C1 to C9) and then to choose the best control for reaching the desired force.

At each sample time, the pressure in each chamber is measured. The hybrid control algorithm calculates the nine directions $\underline{d1}$ to $\underline{d9}$ reachable at the next sample time in the state space by integrating the pressure differential system (2) using equations (3) to (6). See the block diagram in Fig. 6.
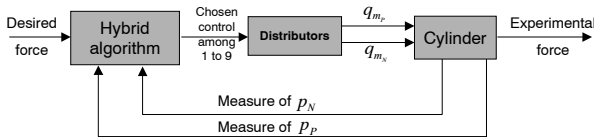

Fig. 6: *System structure*

The target is fixed according to equation (1). For a given desired pneumatic force *F*, the target is defined by a one-dimensional equation:

$$p_\text{P} = \left(\frac{S_\text{N}}{S_\text{P}} \right) p_\text{N} + \left(\frac{F}{S_\text{P}} + \frac{S_\text{P} - S_\text{N}}{S_\text{P}} p_\text{E} \right) \qquad (7)$$

So the target is a point on the straight line defined by Eq. 7 in the (p_N, p_P) plane. The algorithm calculates the nine possible directions ($\underline{d1}$ to $\underline{d9}$ in Fig. 7). Then it chooses the control configuration that corresponds to the lowest Euclidean distance in the physical domain (d6 for the desired force F1 and d4 for the desired force F2 in Fig. 7 because d6 leads out of the physical domain in this case).
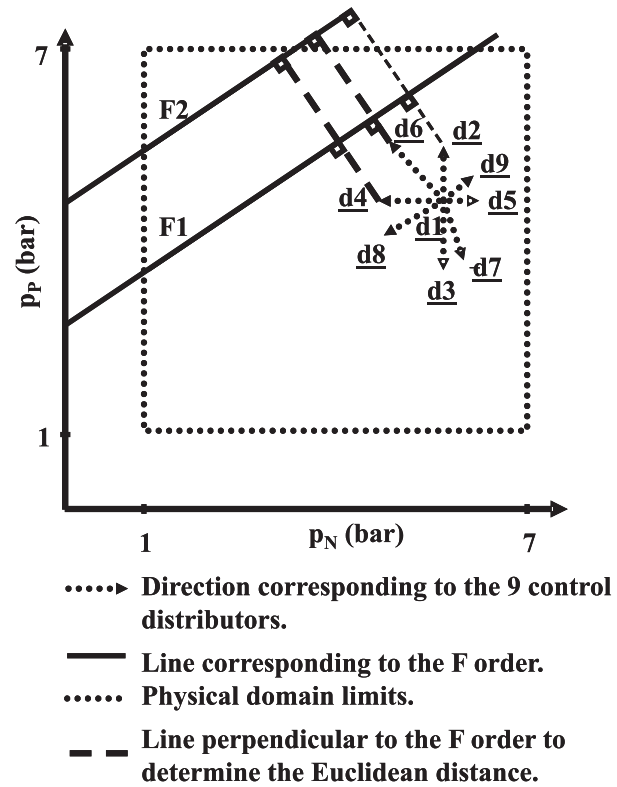
4 Experimental Result

The following results give an illustration of the overall control performance. The test bench is presented in Fig. 1. All the tests were realised with a DSP card from dSpace: DS1104. Programs implemented in DS1104 were Matlab/Simulink with C code. Value 1 given by table 1 means that the relay of the distributor is supplied at 5V and consumes 1.1W. Value 0 means 0V. The exhaust pressure value (1bar) and the supply pressure (7bar) limit the pressure.

### 4.1 Repetitive sequence of step responses

This part will compare two different results :
- In the first plots (Fig. 8), the desired force is 0N for 2.5 seconds and 230N for 2.5 seconds.
- In the second plots (Fig. 9), the sequence of force is 0N for 3 seconds, 50N for 0.5 seconds and 230N for 2 seconds.
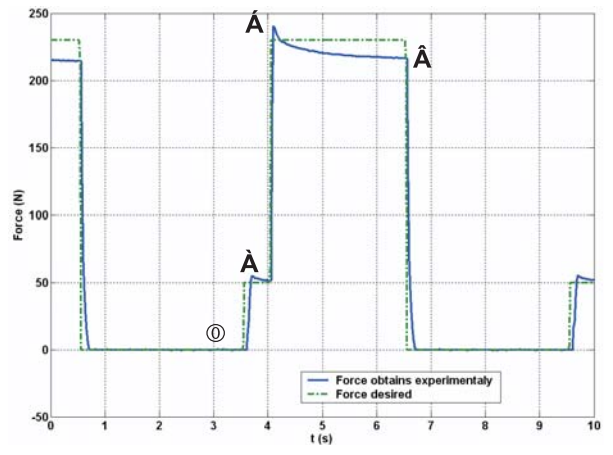
All the transitions are obtained step by step.

At the beginning of the process, the pressures in the chambers P and N are at 1 bar (the inferior left corner of the physical limit). At this stage, the force 50N is accessible with C9 control for T_st and then C1 to close the distributor and maintain the pressure in the chambers. At 50N, 230N is an accessible force with the C9 control. Then in order to obtain 0N, the chosen algorithm exhausts the chambers P and N with control C8 and maintains this value.
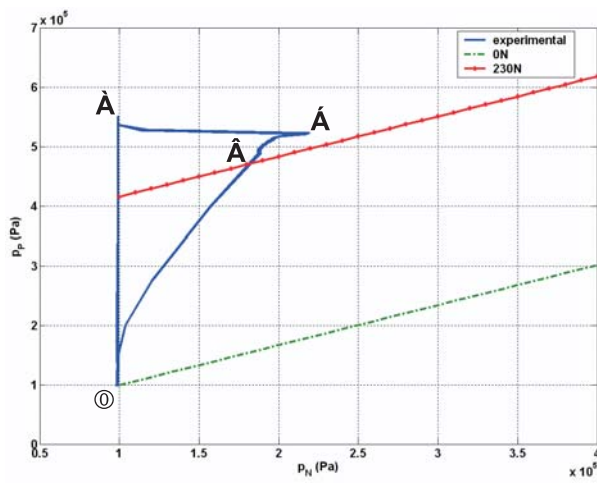
So the aim of the next two results is to show the difference in accessing 230N directly or passing by a step at 50N.
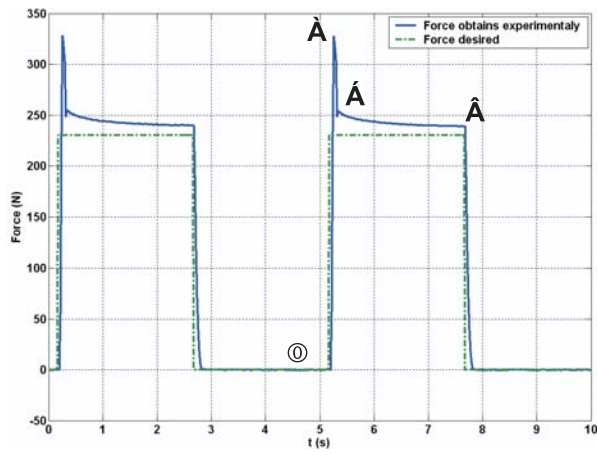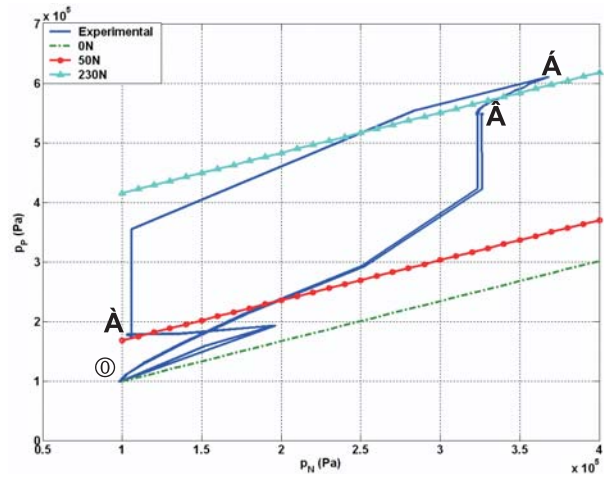
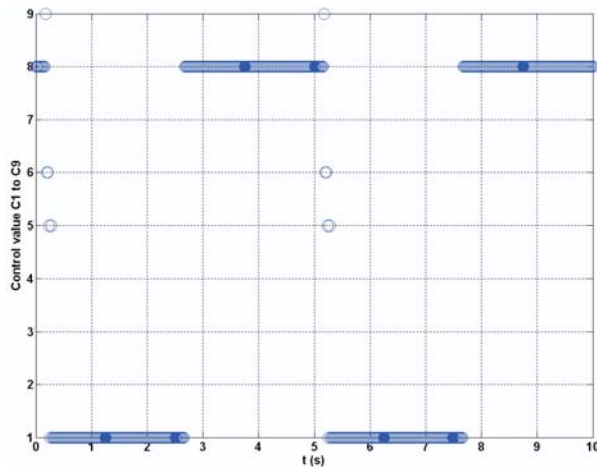*a/ Time evolution of the desired and experimental forces*
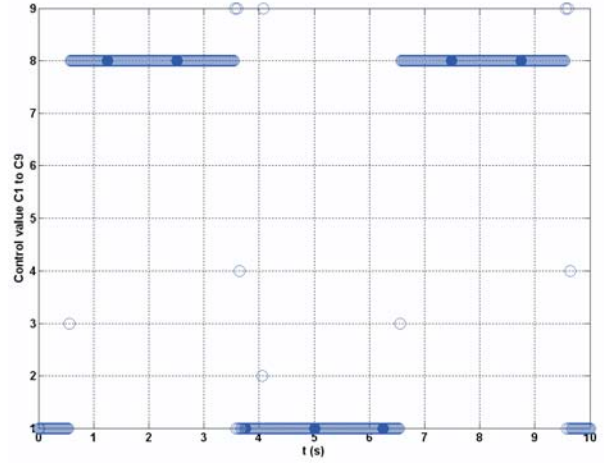

b/ Pressure evolution in the state plane (p_N, p_P)

c/ Time evolution of the controls

Fig. 8: *Desired force in one step*

a/ Time evolution of the force

b/ Evolution of the pressure in the plane (p_N, p_P)

c/ Time evolution of the controls

Fig. 9: *Desired force in two steps*

In the first case, at the beginning of the test, the experimental force was equal to 0 N (see point ⓪ in Fig. 8a), the pressures in the P and N chambers are 1 bar (see point ⓪ in Fig. 8b), the control vector is C8 ((see time around 5 seconds in Fig. 8c) which corresponds to emptying each chamber. When the desired force equals 230 N, the control algorithm increases the pressure in chamber

P (see point ① in Fig. 8b). A big overshoot (about 85 N) of desired force can be seen in Fig. 8a (see point ①). A new control vector is selected by the hybrid control algorithm to decrease the force value (see point ② in Fig. 8a) and converges to the line corresponding to the desired force in the (p_P, p_N) plane (see point ② in Fig. 8b). Between points ② and ③ the control vector is C1:

this value theoretically close the distributor and maintains the pressure in each chamber. It is important to note that experimentally the two pressures and therefore the force continue to evolve. This phenomenon can be due to the thermal exchanges taking place between the cylinder chambers and the environment. So the static stage is reached after about 1.5 seconds and the steady state error is about 10 N (see point ③ in Fig. 8a). At this stage all the control vectors will generate a greater steady state error than 10 N and so the control vector is kept at C1 until the desired force evolves.

The next test reaches 230 N too, but first passing by a stage at 50 N. At the beginning of the process, the experimental force equals 0 N as it did previously (see point ⓪ in Fig. 13). So the desired force equals 50 N (see point ① in Fig. 9a) by applying controls C9 and C4 (see Fig. 9c), C1 to close the distributor and maintain the pressures. When the desired force reaches 230 N (see point ② in Fig. 9a), the algorithm applies control C2 and C9 (see 4 second time value in Fig. 9c). Point ② to point ③ is due to thermal exchange (see the evolution of the chambers in Fig. 9b). The steady state error is about 10 N as seen previously. Finally the return to 0N is carried out by the C3 and C8 controls. Each application of a control can be observed by a break in the (p_P, p_N) plane.

The main difference between the previous two tests is in the size of the overshoot which is reduced from 85 N (Fig. 8a) to about 10 N (Fig. 9a) by using an intermediary stage for the desired force. It can be noted in Fig. 9a that the second case, using an intermediary step in the desired force, obviously requires more time to obtain the final objective of 230N and is more spread out on the (p_P, p_N) plane (Fig. 8b and 9b).
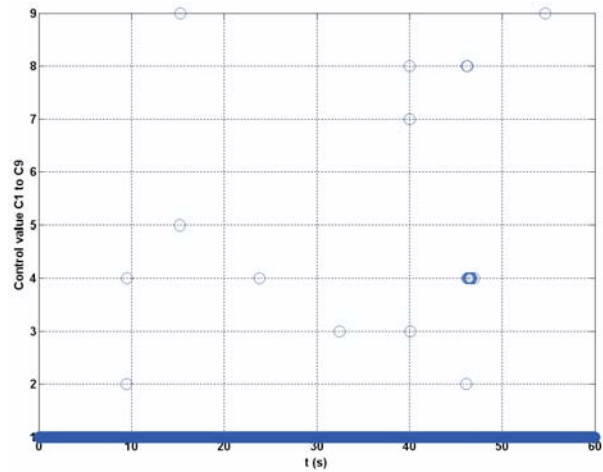
4.2 Random sequences of step responses

Figure 10a shows the experimental results for different tested forces. Figure 10b presents the corresponding control vector values.

It can to seen that the desired forces are quickly obtained and that the root mean square steady state error is small (equal to about 3 N). The mean absolute error is about 6 N. When a force is obtained, the system applies the C1 control to maintain the pressure in the chambers and there is no consumption of power. This new control strategy seems to be very interesting in terms of energy consumption (Brun 1999).
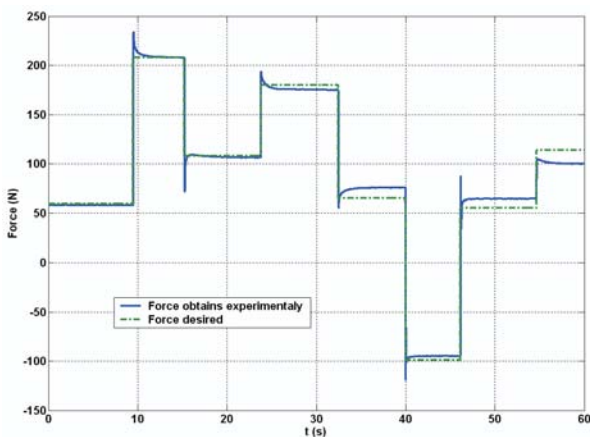
a/ Force time evolution

b/ Control time evolution

Fig. 10: *Random sequences of step response*

5 Conclusion

The contribution of this paper concerns:

- Firstly, the model synthesis and parameterisation of an on/off electropneumatic valve,
- secondly, the application of the hybrid method developed for electrical engine control, e.g. Retif (2004), of a system composed of a pneumatic cylinder and two on/off distributors designed for force control.
- Experimental results show that the principle of controlling the force by quantified components is realisable. Of course it needs to be perfected for static performances, but the results are satisfactory.

On/off control is a promising alternative for force control because of the low cost (compared to servodistributors) and the good performance obtained. Moreover there are no control parameters to tune.

It is possible to improve the system in three different ways :

- improvement of the control algorithm.
- add a closed loop by placing a force sensor at the end of the piston (to measure the force obtained) and to use a classical supplementary control loop (Doyle, 1992) as PID.
- to replace the distributors by ones which have a smaller sonic conductance C.

The first way is the cheapest of the three. The algorithm can be improved to determine several force step values accessible by only one control, in order to reach the desired force. This technique will enable the desired forces to be reached with only a small overshoot (see Fig. 9a compared to Fig. 8a). In addition, the smaller the step, the smaller the thermal exchange, but the longer the time needed to access the final desired force.

The second way requires adding a force sensor. At the moment the system has two sensors to measure the pressure in each chamber, and the force is recon-

structed using Eq. 1. At this stage dry friction is not taken into account. With a force sensor, it would be possible to determine the actual force and so decrease the steady state error.

The third way is a distributor-sizing problem. Accessible forces at time kT_{st} are quantified. So the smaller the sonic conductance is, the smaller the difference in the accessible force and the actual force, but the delay in reaching the desired force is greater. A compromise between dynamic and static performance must be carried out.

Another possible improvement is to apply a higher voltage to the distributor relay for a small time to accelerate the displacement of the spool (Van Ham et al, 2005, Lin 2005). This method would enable the time of applying a control (at the moment 20 ms) to be reduced and so decrease the pressure evolution in the chamber per control. This also would decrease the value of the accessible force in one control, and so decrease the transitional time due to the thermal exchange. Also it would reduce the time to reach the desired force.

Future work will focus on improving the algorithm and applying a higher voltage to the distributor relay to reduce the spool delay.

Nomenclature

b	critical pressure ratio	
C	sonic conductance	[Nl/mn/bar]
F	force	[N]
S	piston cylinder area	[m^2]
T	temperature	[K]
T_{st}	sampling period	[s]
U	boolean distributor input	[null]
V	pneumatic cylinder chamber volume	[m^3]
k	sampling period index	
n	polytropic constant	
p	pressure	[Pa]
q_m	mass flow rate	[kg/s]
r	perfect gas constant	[J/kg/K]
t	time	[s]
x	normalised position of the distributor spool	[null]
τ	spool time delay to move after the control has been applied	[s]
	specific mass	[kg/m^3]

Subscript and Superscript

UP	upstream
DOWN	downstream
P	relative to an increase of force
N	relative to a decrease of force
E	exhaust
S	supply
0	reference
1	distributor connected to chamber P
	distributor connected to chamber N

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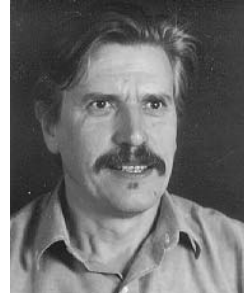
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