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A new experimental test bench for a high performance double electropneumatic actuator system

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***Abstract*— This paper presents design, modelization and control of a new electropneumatic test bench. This latter has been designed for many applications given that it allows high accurancy control and dynamic perturbation force. In fact, the main originality (with respect to previous test benchs) of this test bench is that it is composed by two actuators, the first one being controlled in position, the second one generating perturbation forces. This latter one allows to evaluate the performance of control laws with respect to dynamical forces.**

Keywords - **Feedback control, electropneumatic system, dynamic perturbation force.**

NOMENCLATURE

y, v, a	position, velocity and acceleration of the actuator $[m][m/s][m/s^2]$
y_d, v_d, a_d	desired position, velocity and acceleration $[m][m/s][m/s^2]$
p_X	pressure in the chamber X $[Pa]$
u_P, u_N	servodistributors voltages $[V]$
k	polytropic constant
V_X	chamber X volume $[m^3]$
b_v	viscous friction coefficient $[N/m/s]$
F_f	friction force $[N]$
M	total moving load mass $[kg]$
T_X	chamber X temperature $[K]$
r	perfect gaz constant $[J/kg/K]$
S	piston area $[m^2]$
q_m	mass flow rate provided from the servo--distributor $[kg/s]$
X	P or N
γ	adiabatic constant
T_r	temperature inside an upstream tank $[K]$
Q	thermal exchange$[J]$
λ	thermal exchange coefficient by conduction $[J/K/m^2/s]$
S_{cX}	total area inside X chamber $[m^2]$
F	Perturbation force $[N]$
T_{cX}	temperature of the X chamber wall $[K]$
$q_{mX_{in}}/q_{mX_{out}}$	mass flow rate brought inside/outside of a chamber $[kg/s]$

I. INTRODUCTION

Control of pneumatic actuators is a challenging problem, viewed their increasing popularity (law maintenance cost, lightweight and good force/weight ratio), in spite of their traditional drawbacks (friction, variation of the actuators dynamics due to large change of load and piston position along the cylinder stroke, nonlinearities, ...). The development of high-performance closed-loop linear/nonlinear controllers has shown the feasability of high-level positioning of pneumatic actuator without pertubation [7], [3], [14] or with linear pertubation [8], [9] (for example, with springs). The objective of the new experimental test bench consists in proposing a structure which allows generation of dynamical forces acting on the main actuator. The main actuators on previous experimental test bench [3], [2], [8] are coupled to spings [8] or moving troley with varying mass [3] which do not allow some drastic benchmark.

The new test bench presented here is designed in order to generate dynamic perturbations on the main actuator with an other electropneumatic actuator. This latter is totally independant of the main one and force control is used. Then, both controllers are necessary: a first acting on the main actuator in order to control position and, eventually, pressure. In fact, for a such application, only pneumatic actuator position can be controlled; however, in the case, it appears zero dynamics for which stability analysis is a hard task [3]. Then, pressures have an oscillating behaviour which induces hard constraint on the servodistributors. Given the experimental test bench structure (2 servodistributors for the main actuator), it is possible to control two variables, *i.e.* position and pressure in a chamber. As shown in [8], this multivariable control gives very good performances and improves pressure dynamics.

This paper only presents this new experimental test bench, which is original by its structure, and a first nonlinear controller based on input-output linearization. The paper is organized as follows. Section II describes the experimental test bench, the simulation and control models and the desired position trajectory. Section III describes the design of a nonlinear position controller and its implementation on the experimental test bench.

II. ELECTROPNEUMATIC SYSTEM

A. Description

The new electropneumatic system (see Figures 1-2 - The experimental test bench has been built by Sitia Co.) is composed by two actuators. The first one, named the "main" one (left hand side), is a double acting electropneumatic actuator controlled by two servodistributors (Figure 2) and is composed by two chambers denoted P and N. Piston diameter is 80 mm and rod diameter 25 mm. With a source pressure equal to 7 bar, the maximum force developed by the actuator is 2720 N. The air mass flow rates q_m entering in the chambers are modulated by two three-way servodistributors

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Servotronic (Asco-Joucomatic Co.) controlled by a micro-controller. The pneumatic jack horizontally moves a load carriage of mass M. This carriage is coupled to the second electropneumatic actuator, the so-called "perturbation" one. As previously mentioned, the goal of this latter is to produce a dynamical load force on the main actuator. The actuator has the same mechanical characteristics than the main one, but the air mass flow rate is modulated by a single five-way PVM064 Schneider servodistributor.

In the sequel of the paper, only the control of the "main" actuator position is considered; note that the force control of the "perturbation" actuator is currently made by an analogic PID controller developped by the test bench constructor[1]. In conclusion, the aim of this test bench is to evaluate performances of position controller with respect to unknow dynamical perturbation force.

Fig. 1. Photo of the electropneumatic system - On the left hand side is the "main" actuator whose its position is controlled. On the right hand side is the "perturbation" actuator whose the load force is controlled.

The experimental test bench is simulated with a fluid power systems dedicated software AMESim (LMS SA Co.), and the control law is developped under Matlab/Simulink (The Mathworks Co.). It implies a cosimulation program (Figure 3 and 4) through links between AMESim and Matlab/Simulink. In Figure 4, the block "AMESim Model" makes the link between Matlab simulation and Amesim simulator described by Figure 3. A consequence of the cosimulation is that two models are used: a "simulation" model simulated by AMESim, and a "control" model simulated by Matlab/Simulink. It can be summarized as follows

- the "simulation" model takes into account physical phenomena as temperature variations, experimentals values of mass flow rate delivred by each servodistributors, dynamics of servodistributors, dry friction..., and is developed under Amesim. The perturbation force is viewed as an input.
- the "control" model is simpler than the previous (for example, mass flow rates models are written as pressures polynomials [12]) and issued in order to design the nonlinear position controller under Matlab/Simulink. The perturbation force is supposed unknown.

[1]A future research axis will be to develop also a nonlinear force controller of the "perturbation" actuator.

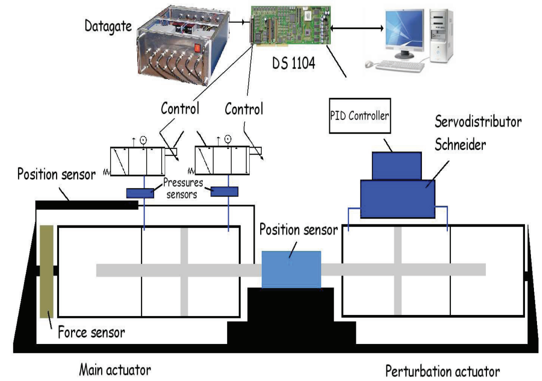

Fig. 2. Scheme of electropneumatic system - This figure displays the mechanical and software structures. The software structure is based on a dSpace board on which the position controller of the "main" actuator is implemented. The mechanical structure is composed by two actuators, the "main" one (left hand side) and the "perturbation" one (right hand side).

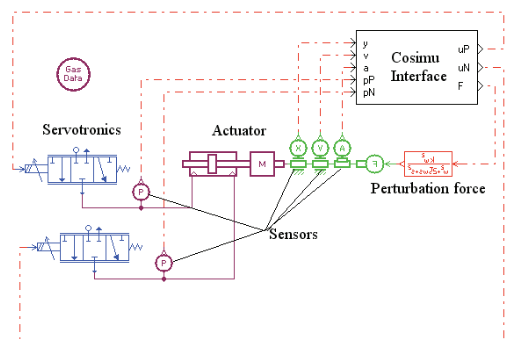

Fig. 3. AMESim model for cosimulation

Matlab/Simulink allows to use a DS1104 board (dSpace Co.) on which the control law is implemented. In the sequel, the experimental results have been obtained with a 1 ms sample time.

B. Simulation model

A standard pneumatic actuator is equipped by a pneumatic damper in order to protect the piston: this protection avoids high clashes between the piston and the external structure of the actuator. The damper is composed by a restriction which limits the exhaust mass flow rate. In order to obtain maximum performance, this restriction has been deleted. It implies that, in a first step, the control law has to be evaluated on cosimulation. The cosimulation is using the

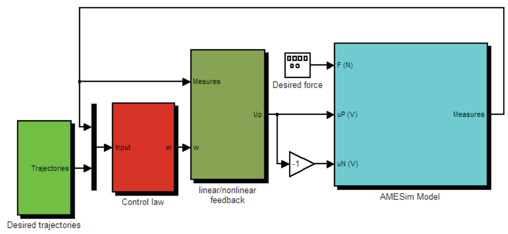

Fig. 4. MATLAB/Simulink Control law for cosimulation

"simulation" model developped with AMESim software, this model trying to be as close as possible to the physical behaviour.

Servodistributor model. The servodistributor model is composed in two parts, a dynamic part and a static one:

- Dynamic part is modelized by a second order transfer function identified from experimental measure

$$\mathcal{F}(s) = \frac{\omega_{ns}^2}{s^2 + 2 \cdot \zeta_s \cdot \omega_{ns} s + \omega_{ns}^2} \tag{1}$$

  with $\omega_{ns} = 246\ rad \cdot s^{-1}$ and $\zeta_s = 0.707$.
- Static part is modelized by an experimental table in which mass flow rate is given in function of ratio pressure (upstream/downstream) and control voltage [12].

Pneumatic chamber variable volume model. Each chamber of the pneumatic actuator is considered as a variable volume, in which the air mass evolves with time. State the classical following assumptions [13]:

- **A1.** Air is perfect gas and its kinetic inconsequential. ■
- **A2.** The pressure and the temperature are homogeneous in each chamber. ■
- **A3.** The mass flow is pseudo-stationary. ■

The first dynamic principle applied to the air mass and the thermodynamic evolution of air in each chamber read as (with $X = P$ or N) [13]

$$\begin{aligned}
\frac{\mathrm{d}p_X}{\mathrm{d}t} &= -\gamma \frac{p_X}{V_X}\frac{\mathrm{d}V_X}{\mathrm{d}t} + \frac{\gamma r T_r}{V_X} q_{mX_{in}} \\
&\quad - \frac{\gamma r T_X}{V_X} q_{mX_{out}} + \frac{(\gamma - 1)}{V_X}\frac{\delta Q_X}{\mathrm{d}t} \\
\frac{\mathrm{d}T_X}{\mathrm{d}t} &= -(\gamma - 1)\frac{T_X}{V_X}\frac{\mathrm{d}V_X}{\mathrm{d}t} + \frac{r T_X}{p_X V_X}(\gamma T_r \\
&\quad - T_X) q_{mX_{in}} - \frac{r T_X^2}{p_X V_X}(\gamma - 1) q_{mX_{out}} \\
&\quad + (\gamma - 1)\frac{T_X}{p_X V_X}\frac{\delta Q_X}{\mathrm{d}t}
\end{aligned} \tag{2}$$

with γ the adiabatic constant, T_r the temperature inside the upstream tank, $q_{mX_{in}}$ the mass flow rate brought inside the X chamber, and $q_{mX_{out}}$ the mass flow rate brought outside the X chamber. Q_X, the thermal exchange with the X chamber wall, is described by assumption A4.

- **A4.** The thermal exchange is due only by conduction described by

$$\frac{\delta Q_X}{\mathrm{d}t} = \lambda S_{cX} \left(T_{cX} - T_X \right) \tag{3}$$

  with λ the thermal exchange coefficient by conduction, S_{cX} the total area inside a X chamber, and T_{cX} the temperature of the X chamber wall. ■

Mechanical model. The second Newton law gives

$$\begin{aligned}
\frac{\mathrm{d}v}{\mathrm{d}t} &= \frac{1}{M}\left[S\left(p_P - p_N\right) - F_f - b_v v - F \right] \\
\frac{\mathrm{d}y}{\mathrm{d}t} &= v
\end{aligned} \tag{4}$$

with friction force F_f including stiction, Coulomb and Stribeck phenomena.

Samplers and saturation. Samplers are added in AMESim's model in order to take into account samplers of acquisition card; sample time is $1\ ms$ which is very smaller than the natural frequency of this electropneumatic system. So it is not necessary to discretize the model all the control law are synthesize in continuous time. Saturation signal control are added, *i.e.* $|u_{sat}| = 10\ V$.

C. Control model

This model is developed in order to design the control law in order to obtain a simplest version which allows the design of control law. The following hypotheses are added

- **A5.** The process is polytropic and characterized by coefficient k (with $1 < k < \gamma$).
- **A6.** The leakage between system and atmosphere are neglected
- **A7.** Furthermore, the temperature variations in each chamber are inconsequential with respect to the supply temperature, *i.e.* $T_P = T_N = T$. ■

Then, pressures dynamics reads as

$$\frac{\mathrm{d}p_X}{\mathrm{d}t} = -k\frac{p_X}{V_X}\frac{\mathrm{d}V_X}{\mathrm{d}t} + \frac{krT}{V_X}\left(q_{mX_{in}} - q_{mX_{out}}\right) \quad (5)$$

- **A8.** The leakages between the two chamber and between servodistribuor and jack are negligible. ■
- **A9.**Supply and exhaust pressure are supposed constant.

By defining $q_m(u_X, p_X) := q_{mX_{in}} - q_{mX_{out}}$, one gets

$$\begin{aligned} \frac{\mathrm{d}p_P}{\mathrm{d}t} &= -k\frac{p_P}{V_P(y)}\frac{\mathrm{d}V_P(y)}{\mathrm{d}t} + \frac{krT}{V_P}q_m(u_P, p_P) \\ \frac{\mathrm{d}p_N}{\mathrm{d}t} &= -k\frac{p_N}{V_N(y)}\frac{\mathrm{d}V_N(y)}{\mathrm{d}t} + \frac{krT}{V_N}q_m(u_N, p_N) \end{aligned} \quad (6)$$

- **A10.** All dry frictions forces are neglected. ■

  In fact, the viscous friction forces have been identified on real system: it has been established that the carriage presents such frictions $b_v v$ with $b_v = 30$.
- **A11.** There is no control signal saturation. ■
- **A12.** Dynamic part of servodistributor is neglected. ■
- **A13.** Static part of servodistributor depends on pressures and control value

  $$q_m(u_X, p_X) = \varphi(p_X) + \psi(p_X, \mathrm{sign}(u_X))\, u_X$$

  with φ and ψ 5^{th}-order polynomials with respect to p_X [12] and issued from experimental measures. ■
- **A14.** Only the position of the actuator is controlled, which means that the problem is a single input-single output (SISO). It implies that $u_P = -u_N = u$. ■

With $V_P(y) = V_0 + S \cdot y$ and $V_N(y) = V_0 - S \cdot y$ (V_0 being equal to the half of the cylinder volume), the model used for the design of controller is a nonlinear system and reads as

$$\begin{aligned} \dot{p}_P &= \frac{krT}{V_P(y)}[\varphi_P + \psi_P \cdot u - \frac{S}{rT}p_P v] \\ \dot{p}_N &= \frac{krT}{V_N(y)}[\varphi_N - \psi_N \cdot u + \frac{S}{rT}p_N v] \\ \dot{v} &= \frac{1}{M}[Sp_P - Sp_N - b_v v - F] \\ \dot{y} &= v \end{aligned} \quad (7)$$

with F the unknow perturbation force, $\varphi_P = \varphi(p_P)$, $\varphi_N = \varphi(p_N)$,

$$\begin{aligned} \psi_P &= \psi(p_P, \mathrm{sign}(u)), \\ \psi_N &= \psi(p_N, \mathrm{sign}(-u)). \end{aligned}$$

It is obvious that the system (7) reads as a nonlinear system affine in control input u such that

$$\dot{x} = f(x) + g(x)u \quad (8)$$

with $x = [p_P \;\; p_N \;\; v \;\; y]^T$,

$$f(x) = \begin{bmatrix} \frac{krT}{V_P(y)}[\varphi_P - \frac{S}{rT}p_P v] \\ \frac{krT}{V_N(y)}[\varphi_N + \frac{S}{rT}p_N v] \\ \frac{1}{M}[Sp_P - Sp_N - b_v v - F] \\ v \end{bmatrix},$$

$$g(x) = \left[\frac{krT}{V_P(y)}\psi_P \;\; \frac{krT}{V_N(y)}\psi_N \;\; 0 \;\; 0\right]^T$$

D. Desired trajectory and perturbation force

The desired position trajectory is displayed by Figure 5 (dotted line) and consists into square signal with magnitude equal to 50 mm. The solid line of Figure 5 displays the perturbation force acting on the "main" actuator and generated by the "perturbation" one. This load perturbation is generated thanks to a force PID controller. This control law (designed by SITIA Co. which builds the experimental test bench) has been tuned such that the behaviour of the closed loop system is close to a second order transfer function with a damping ratio equal to 0.53 and a natural pulsation equal to 16 $rad.s^{-1}$.

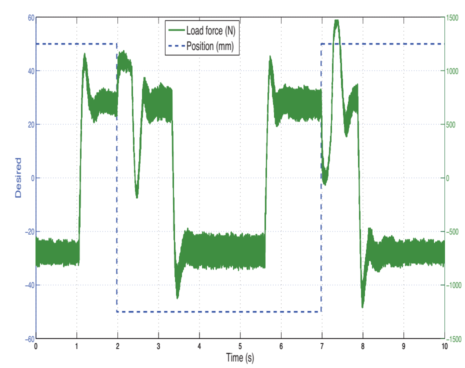

Fig. 5. **Dotted line.** Desired position (mm) of the actuator versus time ($sec.$). **Solid line.** Load force ($N.m$) versus time ($sec.$).

III. Application of a nonlinear controller

A. Controller design

In this current paper, a nonlinear control with state feedback is presented, this controller being based on an input-output linearization scheme [6], [5]. As mentioned previously, the objective consists in designing position controller. Note that this kind of controller method has already been used on electropneumatic systems [11], [3], but never on the current kind of experimental test bench and also never versus such load force dynamics.
By following the classical way for input-output linearization

[6], [5], define z the state coordinates transformation as, with $h(x) = y$ the system output,[2]

$$z = \begin{bmatrix} z_1 \\ z_2 \\ z_3 \\ z_4 \end{bmatrix} = \begin{bmatrix} h(x) \\ L_f h(x) \\ L_f^2 h(x) \\ p_P \end{bmatrix} = \begin{bmatrix} y \\ v \\ a \\ p_P \end{bmatrix} = \phi(x) \qquad (9)$$

System (8) is equivalent, via the coordinates transformation (9), to

$$\begin{aligned} \dot{z}_1 &= z_2 \\ \dot{z}_2 &= z_3 \\ \dot{z}_3 &= L_f^3 h\left(\phi^{-1}(z)\right) + L_g L_f^2 h\left(\phi^{-1}(z)\right) u \\ \dot{z}_4 &= \dot{p}_P\left(\phi^{-1}(z)\right) \end{aligned} \qquad (10)$$

with

$$\begin{aligned} L_g L_f^2 h(x) &= \frac{krTS}{M}\left(\frac{\psi_P}{V_P(y)} + \frac{\psi_P}{V_N(y)}\right) \\ L_f^3 h(x) &= \frac{krTS}{M}\left(\frac{S\varphi_P}{V_P(y)} - \frac{S\varphi_N}{V_N(y)}\right. \\ &\quad \left. - \frac{b}{M^2}\left(Sp_P - Sp_N - bv - \hat{F}\right)\right) \end{aligned} \qquad (11)$$

with $\hat{F}$ the estimated perturbation force derived from (by using a derivation filter[3] for the computation of the estimated velocity $\hat{v}$ and acceleration $\hat{a}$)

$$\hat{F} = M\hat{a} + b_v\hat{v} - S\left(p_P - p_N\right)$$

It appears that z_4-dynamics is the so-called zero-dynamics. In case of electropneumatic actuators, the proof of zero dynamics stability is a very hard task by a formal way, and has been achieved [11] only numerically for a given structure of actuators (close to the current, but without dynamical perurbation). Then, a position control law reads as

$$u = \frac{1}{L_g L_f^2 h}\left(-L_f^3 h + w\right) \qquad (12)$$

Note that, under the physical domain $\mathcal{X}$ defined as

$$\mathcal{X} = \left\{x \in I\!R^4 \,/\, 1\text{bar} \le p_P \le 7\text{bar}, 1\text{bar} \le p_N \le 7\text{bar}, \right. \\ \left. -72mm \le y \le 72mm\right\},$$

the function $L_g L_f^2 h$ is invertible. Under the previous control law, and supposing that the estimated values $\hat{v}$ and $\hat{a}$ are perfect, it yields that $\hat{F} = F$, the system reading as $y^{(3)} = w$. Then, the control law w is a linear state feedback such that

$$w = K_y(y - y_d) + K_v\hat{v} + K_a\hat{a} \qquad (13)$$

[2]Given $a(x)$ a real-valued function and $b(x)$ a vector field, both defined on $I\!R^n$, the derivative of $a(\cdot)$ along $b(\cdot)$ is written as $L_b a$ and is defined as $L_b a = \frac{\partial a}{\partial x} b(x)$ [5].

[3]The derivation filter is designed such that the measured position is differentiated with numerical derivative block based on a first order transfert function. This transfert function has a single zero equal to zero, and one pole chosen with respect to the derivation order. In the current case, pole is -10 for position, whereas -1000 for acceleration.

with gains K_y, K_v and K_a computed by Ackerman's approach such that poles placement allows 4.6% overshoot [4]. Furthermore, these gains are calculated in order to have the maximum of bandwidth compare to actuator position.

B. Experimental results

The control law is implemented on DS1104 Board. The control law and the measurement are made under the sample time $1\ ms$. The source pressure equals $7\ bar$. Note that for the control law synthesis, the force F is not considered given that it is viewed as an unknown perturbation. As previously mentioned, the double numerical derivation of position measurement y gives velocity $\hat{v}$ and acceleration $\hat{a}$.

The actuator position (Figure 6) converges to the desired trajectory without pressure saturation (Figure 7 - Top), the static error being equal to around $0.3\ mm$. Note the effect of perturbation force on the actuator position which is relativey limited given that the transient maximal static error value equals $4.5\ mm$. In fact, in the steady beaviour, the pressure difference between the two chambers is enough to compensate the external force. When the pertubation is changing, the controller allows to sufficiently compensate this force change.

The control inputs u_P and u_N (Figure 7 - Bottom) present some saturations during the large transients (of desired trajectories and perturbation force).

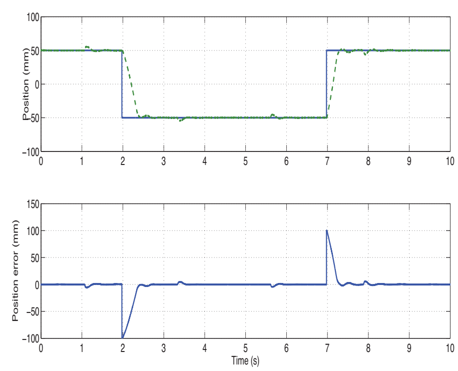

Fig. 6. **Top.** Desired and current positions (mm) versus time (sec). **Bottom.** Positions errors (mm) versus time (sec).

IV. Conclusion

The objective of this paper has been, firstly, to present a new experimental test bench for performances evaluation of electropneumatic actuators with respect to dynamical perturbation forces and, secondly, to propose a first position control law. This latter has shown that the experimentl test bench is functional and has opened axis for future research

- Mutivariable control. As the test bench is equipped by two servodistributors, it yields that it is possible to simultaneously control position and pressure [8]. For

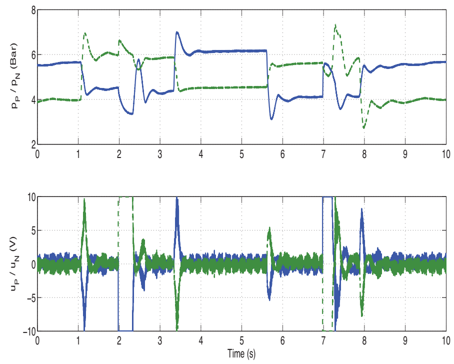

Fig. 7. **Top.** Pressures in P (solid line) and N (dotted line) chambers (bar) versus time (sec). **Bottom.** Control inputs u_P (solid line) and u_N (dotted line) (V) versus time (sec).

this system, the main advantage of multivariable control is that there is no zero dynamics. Furthermore, as one of the pressure is controlled, it is possible to act on the actuator accuracy and rigidity in case of perturbation.

- Observation. In order to reduce the sensors number (cost reduction, improvementof the operational safety, ...), the objective will be to design pressure(s) observer.

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