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HYBRID CONTROL FOR SWITCHING BETWEEN POSITION AND FORCE TRACKING

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ABSTRACT

This paper is the logical result of previous researches about advanced control in electropneumatic systems. After different works in feedback control of electropneumatic actuators in position tracking [1] or in force tracking [2], this study led about a method to switch between these two different focuses. This question is an important problem in many industrial systems, for example in the case of vehicle active suspension or for pneumatic or hydraulic pliers, fingers and spot welding, fatigue test. Indeed, some dangerous damages could occur when the aim is to move a load near a structure and to apply a force against this structure. The major difficulty is to know when and how to switch between the position control and the force control (this is here the notion of *hybrid* control). A classical occurrence is due to a static position error when the control algorithm switches from position feedback to force feedback. The result leads very often to an undesirable impact to the structure.

KEYWORDS

Force / position tracking, hybrid control, pneumatic

NOMENCLATURE

f_v	viscous friction coefficient	[N/m/s]
F_{ext}	external force	[N]
k	polytropic constant	
M	total moving load mass	[kg]
$p_{P/N}$	pressure in chamber P or N	[Pa]
r	perfect gas constant	[J/kg/K]
$S_{P/N}$	piston areas	[m^2]
$\tau_{P/N}$	time constant	[s]
S_{stop}	structure area where force is applied	[m^2]
T_S	ambient temperatures	[K]
u	servo-distributor voltage	[V]
$V_{P/N}$	volume of chamber P or N	[m^3]
y,v,a	position, velocity, acceleration	[m], [m/s], [m/s^2]
j	jerk (acceleration derivation)	[m/s^3]
$q_{m\,P/N}$	mass flow rates provided from the servo-distributors P or N to the cylinder chambers P or N	[kg/s]
$C^{e}_{p_X J}$	$= -\left.\frac{\partial q_J}{\partial p_X}\right\|_e$, partial derivative of the mass flow rate according to *p* around equilibrium set	[kg/s/Pa]
G^{e}_{uJ}	$= \left.\frac{\partial q_J}{\partial u}\right\|_e$, partial derivative of the mass flow rate according to *u* around equilibrium set	[kg/s/V]
Gu_P	static gain	[V/Pa]
K_y, K_v, K_a	position, velocity, acceleration feedback gains	[V/m], [V/m/s], [V/m/s^2]
K_P, K_N,	pressure in chamber P/N feedback gains	[V/Pa]
δ	variation near the equilibrium set	
e	equilibrium	
d	desired	

INTRODUCTION

In many industrial applications, different objectives lead to different control algorithm and switching algorithm permit to control successively: position, velocity, force or stiffness for example. Lot of research works are issue from robotic problems where the position control is required for the stiffness, and the force control is for the compliance [3]. Some recent results propose to switch from position control to force control using force sensor useful to detect an obstacle [4]. The approach of this paper is different because the switch occurrences do not depend of an external signal, but is programmed in-line for a desired tracking objective. An example of industrial application concerns mechanical fatigue test.

The two different control algorithms presented in this paper concerns linear state feedback in position for tracking objectives and for force tracking. The desired trajectories of position and force are issued of an industrial benchmark. A methodology to switch between the two special objectives without damages on the process is presented. Some experimental results with a rigid structure are showed. The conclusion concerns the first results obtained with the proposed method, some extension will be discussed.

1. EXPERIMENTAL DEVICE

The considered experimental process is an in-line electropneumatic servodrive using a simple rod double acting linear pneumatic cylinder (Fig. 1). Two three-way proportional servo-distributors modulate the mass flow rates into the cylinder chambers. The rod of the actuator is connected to one side of the carriage, and the users have the possibility to choose the structure and the position where the carriage applies a force. Different configurations have been tested with different materials and so with different stiffness characteristics. In this paper the structure is assumed to be rigid.

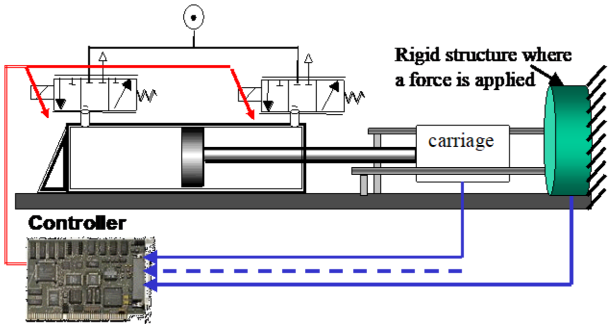

Fig. 1 *Position / Force tracking control experimental system*

2. MODELLING

For synthesis control laws, two kinds of model are established in this study. The first one concern the model for position tracking obtained after linearization and a model reduction. The second useful for force tracking is obtained after a linearization around a reference trajectory. Both of them will be used to synthesise control laws in section 4.

(1) Model for position tracking

In this case the carriage do not touch the structure where the force will be applied (see Fig 1.) and it is assumed that the external force (noted F_{ext} in Eq. 1) is only due to atmospheric pressure. The electropneumatic system model can be obtained using three physical laws:

- the mass flow rate through a restriction,
- the pressure behaviour in a chamber with a variable volume,
- and the fundamental mechanical equation.

In our case, the natural pulsation of the Servotronic Joucomatic Servo-distributor and the actuator are respectively about 200 Hz and 2,4 Hz. Using the singular perturbation theory, the faster dynamic can be neglected.

The pressure evolution law in a chamber with variable volume is obtained assuming the following assumptions [5] :

- air is a perfect gas and its kinetic energy is negligible,
- the pressure and the temperature are homogeneous in each chamber,
- the process is polytropic and characterised by coefficient *k*,
- dry frictions are neglected.

Moreover, the electropneumatic system model is obtained by combining all the previous relations and assuming that the temperature variation is negligible with respect to average and equal to the supply temperature. The following relation gives the model of the considered system, where the control signals applied to both servo-distributors have opposite signs:

$$\begin{cases} \dfrac{dp_P}{dt} = \dfrac{krT_s}{V_P(y)}\left[q_{m\,P}(u, p_P) - \dfrac{S_P}{rT_s} p_P v\right] \\ \dfrac{dp_N}{dt} = \dfrac{krT_s}{V_N(y)}\left[q_{m\,N}(-u, p_N) + \dfrac{S_N}{rT_s} p_N v\right] \\ \dfrac{dv}{dt} = \dfrac{1}{M}\left[S_P p_P - S_N p_N - f_v v - F_{ext}\right] \\ \dfrac{dy}{dt} = v \end{cases} \qquad (1)$$

A procedure to obtain the linearized model is given in [6] and the following model (see Eq. 2) is obtained:

$$\frac{d}{dt}\begin{bmatrix}\delta p_P \\ \delta p_N \\ \delta v \\ \delta y\end{bmatrix} = \begin{bmatrix} -\dfrac{1}{\tau_P{}^e} & 0 & -\dfrac{kp_P{}^e S_P}{V_P(y^e)} & 0 \\ 0 & -\dfrac{1}{\tau_N{}^e} & \dfrac{kp_N{}^e S_N}{V_N(y^e)} & 0 \\ \dfrac{S_P}{M} & -\dfrac{S_N}{M} & -\dfrac{f_v}{M} & 0 \\ 0 & 0 & 1 & 0 \end{bmatrix}\begin{bmatrix}\delta p_P \\ \delta p_N \\ \delta v \\ \delta y\end{bmatrix} + \begin{bmatrix} \dfrac{krT_S}{V_P(y^e)} G_{uP}{}^e \\ -\dfrac{krT_S}{V_N(y^e)} G_{uN}{}^e \\ 0 \\ 0 \end{bmatrix}\delta u \qquad (2)$$

With time constants:

$$\tau_P{}^e = \frac{V_P(y^e)}{krT_S C_{p_P P}{}^e} \quad \text{and} \quad \tau_N{}^e = \frac{V_N(y^e)}{krT_S C_{p_N N}{}^e}$$

where $$\left\{ \begin{array}{l} C_{p_P P}{}^{e} = -\left.\frac{\partial q_m\left(u^e, p_P\right)}{\partial p_P}\right|_e, G_{uP}{}^{e} = \left.\frac{\partial q_m\left(u, p_P{}^{e}\right)}{\partial u}\right|_e \\ C_{p_N N}{}^{e} = -\left.\frac{\partial q_m\left(-u^e, p_N\right)}{\partial p_N}\right|_e, G_{uN}{}^{e} = \left.\frac{\partial q_m\left(u, p_N{}^{e}\right)}{\partial u}\right|_e \end{array} \right\}$$

are deducted from the global static characteristics of the mass flow rates.

In the pneumatic field, the conventional position control law consists of position, velocity and acceleration feedback. To obtain a third order model with position, velocity and acceleration as state variables, a classical method consist in to replace each time constant of each chamber by an average time constant τ_m^e (geometric mean). Then the third order model is given by Eq. (3).

$$\frac{d}{dt}\begin{bmatrix} y \\ v \\ a \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ 0 & -\omega_{ol}{}^{2} & -2\zeta_{ol}\omega_{ol} \end{bmatrix}\begin{bmatrix} y \\ v \\ a \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ b \end{bmatrix} u \qquad (3)$$

With $$\left\{ \begin{array}{l} \zeta_{ol} = \frac{1}{2\omega_{ol}}\left(\frac{1}{\tau_m^e} + \frac{f_v}{M}\right) \\ b = \frac{krT_S}{M}\left[\frac{S_P G_{uP}^e}{V_P\left(y^e\right)} + \frac{S_N G_{uN}^e}{V_N\left(y^e\right)}\right] \end{array} \right. \qquad (4)$$

Where $$\left\{ \begin{array}{l} \omega_{ol} = \sqrt{\omega_{cyl}{}^{2} + \frac{f_v}{\tau_m^e M}} \\ \omega_{cyl} = \sqrt{\frac{k}{M}\left(\frac{S_P{}^{2} p_P^e}{V_P\left(y^e\right)} + \frac{S_N{}^{2} p_N^e}{V_N\left(y^e\right)}\right)} \end{array} \right. \qquad (5)$$

(2) Model for force tracking

Considering that the structure where the force is applied is rigid and so has an infinite stiffness, the model for force tracking issued from Eq. (1) is given by Eq. (6) assuming that the carriage does not move:

$$\left\{ \begin{array}{l} \frac{dp_P}{dt} = \frac{krT_s}{V_P(y)} q_{m\,P}\left(u, p_P\right) \\ \frac{dp_N}{dt} = \frac{krT_s}{V_N(y)} q_{m\,N}\left(-u, p_N\right) \end{array} \right. \qquad (6)$$

The force output is estimated using Eq. (7), where p_P and p_N are measured and where the area of the rigid structure where the force is applied is noted S_{stop}.

$$F = S_P p_P - S_N p_N - \left(S_P - S_N - S_{stop}\right) p_E \qquad (7)$$

An approach for the tracking control often used in robotics is the linearization around a reference trajectory of the nonlinear system (Eq. 6). This method has been applied in electropneumatic process [2, 6] and his briefly explain here. It consists in an off-line calculation of the reference trajectory of state and the input then in on-line regulation of the error trajectory. Let us consider a reference trajectory model:

$$\frac{d\underline{x}_d}{dt} = \underline{f}(\underline{x}_d, u_d) \qquad (8)$$

A trajectory error model can be defined:

$$\frac{d\delta\underline{x}_d}{dt} = \underline{\tilde{f}}(\delta\underline{x}, \delta u, \underline{x}_d, u_d) \qquad (9)$$

with $\delta\underline{x} = \underline{x} - \underline{x}_d$, $\delta u = u - u_d$

and $\underline{\tilde{f}}(\delta\underline{x}, \delta u, \underline{x}_d, u_d) = \underline{f}(\underline{x}, u) - \underline{f}(\underline{x}_d, u_d)$

The linearization at first order around the reference trajectory may be written and the model (6) can be transformed in model given by Eq. (10).

$$\frac{d\delta\underline{x}}{dt} = A(\underline{x}_d)\delta\underline{x} + B(\underline{x}_d)\delta u \qquad (10)$$

This method has been applied to electropneumatic system [6]. For this, after a determination of the reference trajectory, the development in Taylor's series limited to the first order calculated around the reference trajectory gives the following linearized tangent model of the trajectory error model (Eq. 11). The time constants characterising the chamber pressure evolutions (τ_P and τ_N) appear as function of the reference trajectory.

$$\frac{d}{dt}\begin{bmatrix} \delta p_P \\ \delta p_N \end{bmatrix} = \begin{bmatrix} -\frac{1}{\tau_P\left(F_d\right)} & 0 \\ 0 & -\frac{1}{\tau_N\left(F_d\right)} \end{bmatrix}\begin{bmatrix} \delta p_P \\ \delta p_N \end{bmatrix} + \begin{bmatrix} \frac{Gu_P\left(F_d\right)}{\tau_P} \\ -\frac{Gu_N\left(F_d\right)}{\tau_N} \end{bmatrix}\delta u \qquad (11)$$

For more details the parameters variations of the linearized model with respect to F_d given by Eq. (12) can be founded in [6].

$$Gu_P(F^r) = \frac{G_{u_P}(F_d)}{C_{p_P P}(F_d)} \qquad Gu_N(F^r) = \frac{G_{u_N}(F_d)}{C_{p_N N}(F_d)} \qquad (12)$$

3. TWO KINDS OF TRACKING TRAJECTORIES

Two different trajectories are presented in this section. The industrial benchmark defined a cycle as follow: the carriage must moving from left to right (see Fig. 1) and when it arrives against the rigid structure a constant force must be applied during a fixed time and after this the carriage must go back to the right side. This cycle must be effected one thousand times for fatigue tests.

To respect these focuses better results have been obtained with tracking methodology. That is why the carriage displacement is not fixed by a step response but by the following desired trajectory (see Fig. 2) :

- Concerning position tracking, the displacement magnitude is equal about 90 % of the total stroke (from –227 mm to +227 mm) and is duration is about 1.1s (from 0.7s to 1.8s on Fig. 2). The velocity is constant during about 70% of the total displacement and its maximum value is 0.48 m/s. So acceleration is limited to 5.80 m/s^2. The trajectory is symmetric during acceleration and deceleration stage.
- When the carriage is close to the rigid structure the **switch from position control to force control** occurs, it depends from the desired position. The best results obtained experimentally concerns desired position equal or higher to +226.5 mm.
- After 0.4 s the desired force increases linearly during 0.5 s with a total magnitude equal to 250 N. This force is maintain constant during 0.5 s and then decreases linearly until to be equal to the atmospheric pressure force. The force trajectory is symmetrical;
- The **switch from force control to position control** occurs after that the desired force is equal to the atmospheric force and is synchronised with the desired position. Once the force reach the atmospheric pressure, the best experimental results are obtained if the position control switch on when the desired position is equal or lower than +226.5 mm.
- The desired position for carriage backward displacement is symmetric to the forward displacement described in first point.

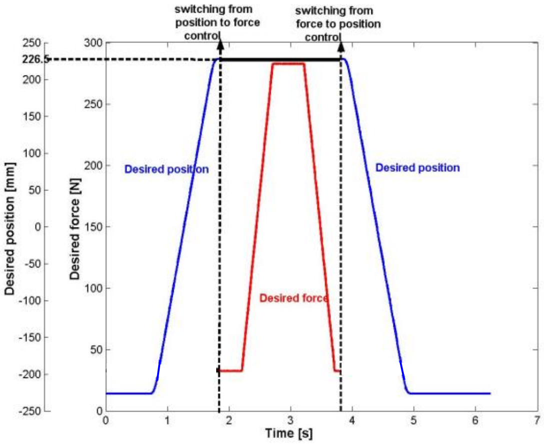

Fig. 2 *Choice of desired trajectories*

4. CONTROL

(1) Control in position

The model given by Eq. (2) has a characteristic number associated to the output y equal to three. Then taking a triple integrator as reference model and using a first loop it is theoretically possible to obtain a perfect tracking result. If that is the case, using state feedback control law given by Eq. (13), the model (2) is transform in a triple integrator given by Eq. (14) (canonical Brunovsky form).

$$u = -\underline{K}_1 \underline{z} + u_c \tag{13}$$

with $u_c = \frac{1}{b} j_c$ and $\underline{K}_1 = \begin{bmatrix} K_{y1} \\ K_{v1} \\ K_{a1} \end{bmatrix} = \begin{bmatrix} 0 \\ -\frac{\omega_{ol}^2}{b} \\ -\frac{2\zeta_{ol}\omega_{ol}}{b} \end{bmatrix}$

$$\underline{\dot{z}} = I_d \underline{z} + B_J j_c \tag{14}$$

where $I_d = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ 0 & 0 & 0 \end{bmatrix} \quad B_J = \begin{bmatrix} 0 \\ 0 \\ 1 \end{bmatrix} \quad \underline{z} = \begin{bmatrix} y \\ v \\ a \end{bmatrix}$

Here j_c is the difference between the desired jerk (acceleration derivation) and a second feedback (Eq. 15) useful and necessary to stabilise the system by fixing the dynamic of error cancellation.

$$j_c = j_d - K_{y2} \times (y_d - y) - K_{v2} \times (v_d - v) - K_{a2} \times (a_d - a) \quad (15)$$

The control law for position tracking is resumed by Fig. 3.

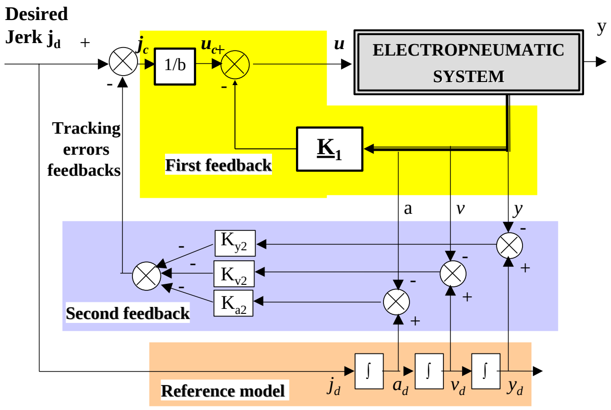

Fig. 3 *Principle of position tracking control*

(2) Control in force

The system (11) clearly shows a parallel structure. So a loss of controllability and observability appears when the parallel dynamics are equal. However, the output is always controllable [6]. The practical method of going through this singular point is to keep the control input constant. In our case the system is always asymptotically stable and so the force output will not go off its reference trajectory. For the state feedback the choice of the eigenvalues in closed loop fixes dynamic of the error trajectory with a control law on the form given by Eq. (16).

$$u = u_d - K_P(p_P - p_{Pd}) - K_N(p_N - p_{Nd}) \quad (16)$$

Because the parameters of linearized model are functions of the reference force the state feedback gains are also functions of the reference force. More details can be founded in [2, 6].

(3) Hybrid Control

In this first study concerning hybrid control an elementary strategy has been tested as explain in section 3. It concerns switching from position control to force control depending on the desired position (see figure 4).. The instance of switching is fixed only by the desired position. When the carriage arrive close to the structure the strategy of control change from position tracking to force tracking. When the desired output leads a separation between carriage and rigid structure the position tracking succeed to force feedback.

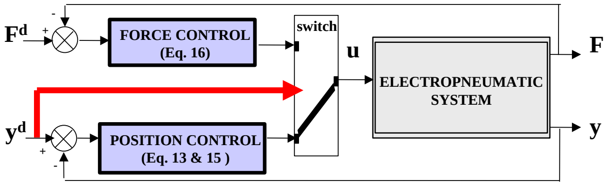

Fig. 4 *Principle of tracking control*

5. EXPERIMENTAL RESULTS

The experimental results are achieved after a sufficient overheating of electronical, mechanical and pneumatic part. Figure 5 presents the quality of tracking performances where desired trajectories and measured one are similar. It is important to note that no problems are visible during the switching phases. Indeed when the carriage arrive near the structure where the force is applied no bang occurs and the control is not saturated (see figure 7). At the second switch, the transition is smooth too.

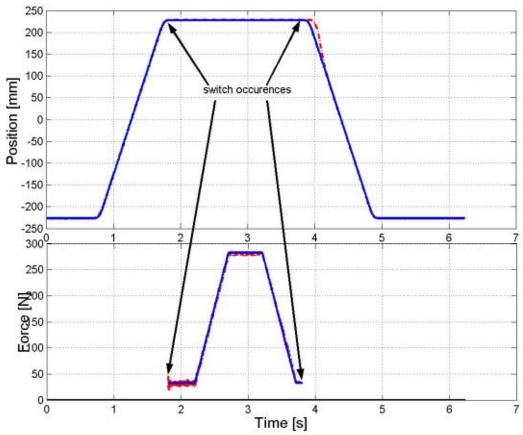

Fig. 5 *Tracking results in position and force*

The mean absolute error in position is equal to 1.6 millimetres all along the displacement. The results are better in static stage where the static error is equal to + 0.4 millimetres on one extremity and equal to –0.6 millimetres on the other one. This difference is due to the friction that depends on the position. This small static error and the smooth desired force are the principal reasons of success of switching.

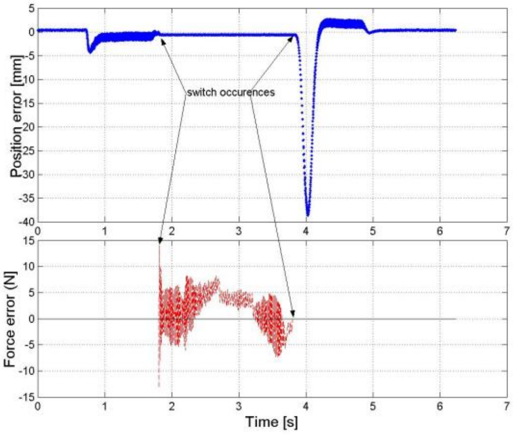

Fig. 6 *Error tracking results in position and force*

The large position error at time equal to 4 s equal to 38 millimetres (see figure 6) is not due to hybrid control but to the dry friction. Indeed the carriage stick on the guiding rails until the pressure in the two chambers creates a force superior to frictions. Concerning force error the mean absolute error is equal to 3.5 N (less than 2% of the total desired force) and the maximum error is near 14 N. The importance of force oscillation are due to pressure sensors. Indeed for these tests no force sensor is used but the force is estimated using pressure measured in each chamber. So this estimation coupled with high gains (K_P and K_N) in control law explain these little oscillations.

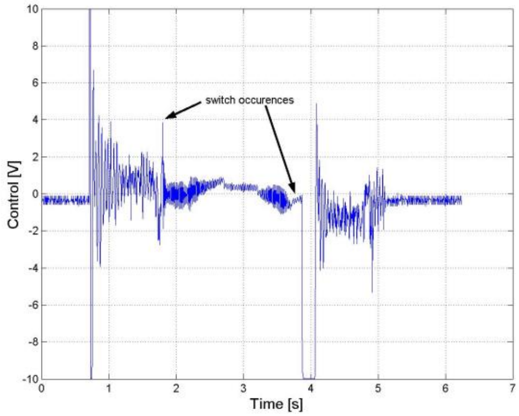

Fig. 7 *Control signal*

The results obtained do not lead to a signal control harmful to the servovalve component. Indeed as show Fig. 7 the input signal does not evolves with high magnitude and does not creates high excitation.

The quality of these experimental results at switching occurrences are essentially due the quality of tracking trajectory in term of position and force. No good results have been obtained with classical control law, which do not use tracking algorithm. The advantage of this method concerns the simplicity of switching occurrence.

CONCLUSION AND PERSPECTIVES

Thanks to previous work lead in the laboratory, this paper used trajectory tracking in position [1] and trajectory tracking in force [2] to synthesised hybrid controller capable to switch between each control law without damage on the test bench. The simplicity of proposed method used only one parameter of tuning (indeed the desired output fix the switch time). This technique seems attractive for industrial applications especially for fatigue tests. An important reason of the success of switching concerns the good quality tracking. In aim to obtain better results and improved the quality of tracking, some nonlinear control law [1, 8] and robust one are today study.

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