FPNI Ph.D. Symposium on Fluid Power

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A Park - like transform for fluid power systems : application to pneumatic stiffness control

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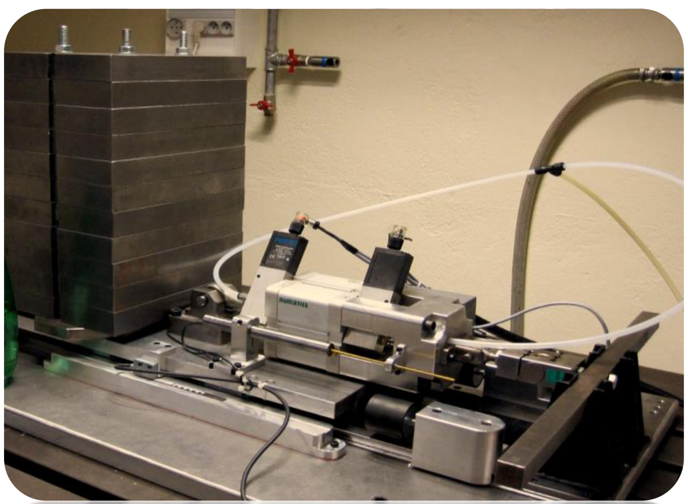
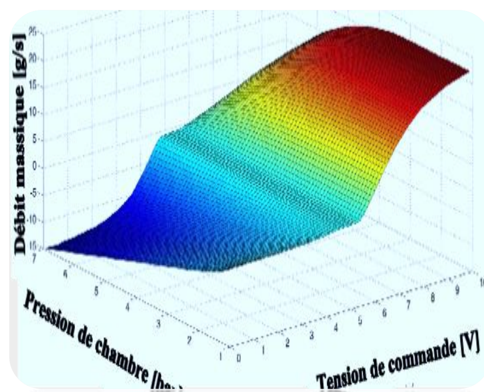
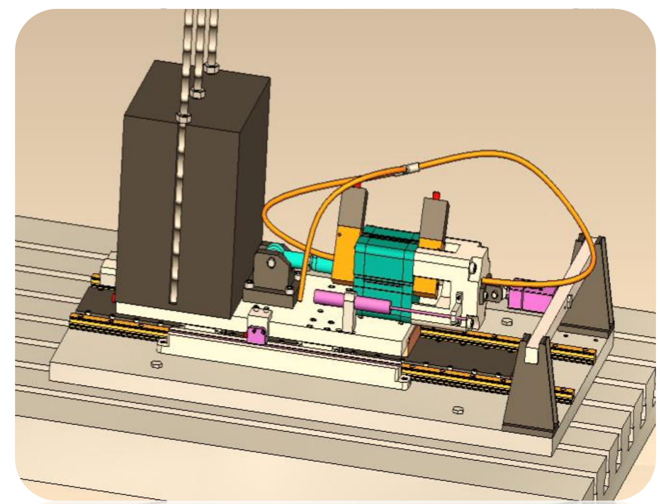
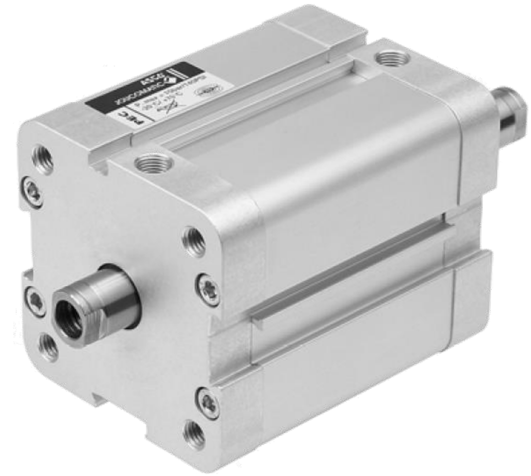






Context & problematics

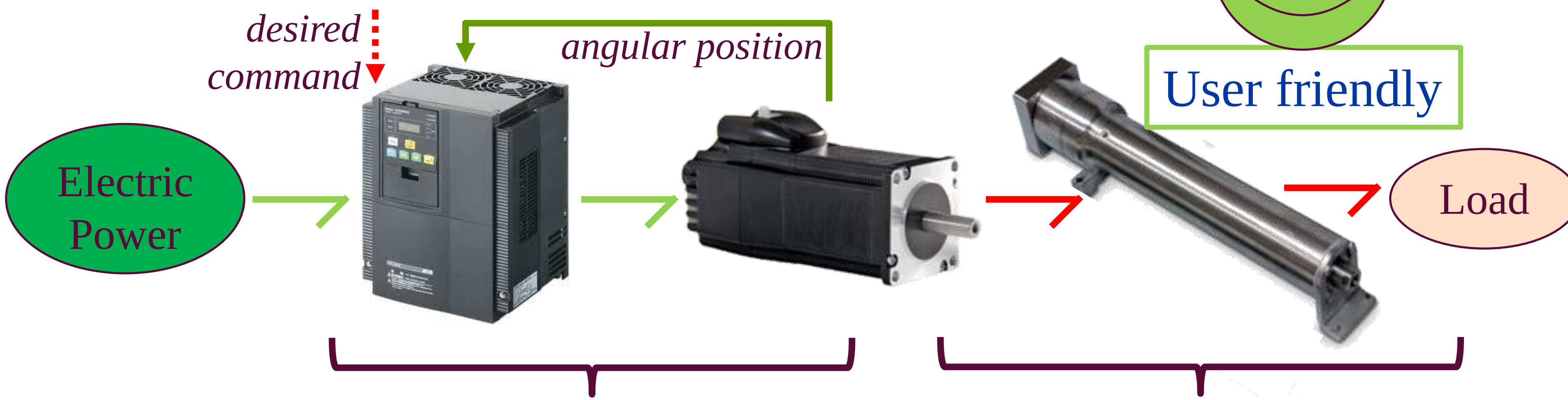
- Modeling and control of Fluid Power systems is usually considered as a difficult task :
  - **multiphysic :** mechanics, fluid dynamics, thermodynamics, electrical eng., …
  - **highly non linear behavior**



more difficult to control than electrical drives ?

NO

❑ Electric drives



High bandwidth

Low bandwidth

- lot of sensors (current, voltage)
- complex control algorithm embedded (large computation capacity)
- but everything is included !!!

- high reflected inertia
- complex friction phenomena
- can be considered part of the load (disturbance)

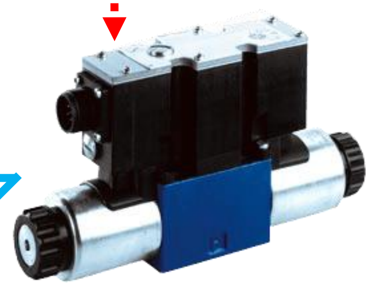
The whole package is on-the-shelves !!!

Differences electric and fluid power drives (2/3)

❑ Hydraulic drives

command

Hydraulic Power



Load

Low bandwidth

High bandwidth

- ~ no sensor (mech/elec feedback)
- each design is different
- safety component have to be added

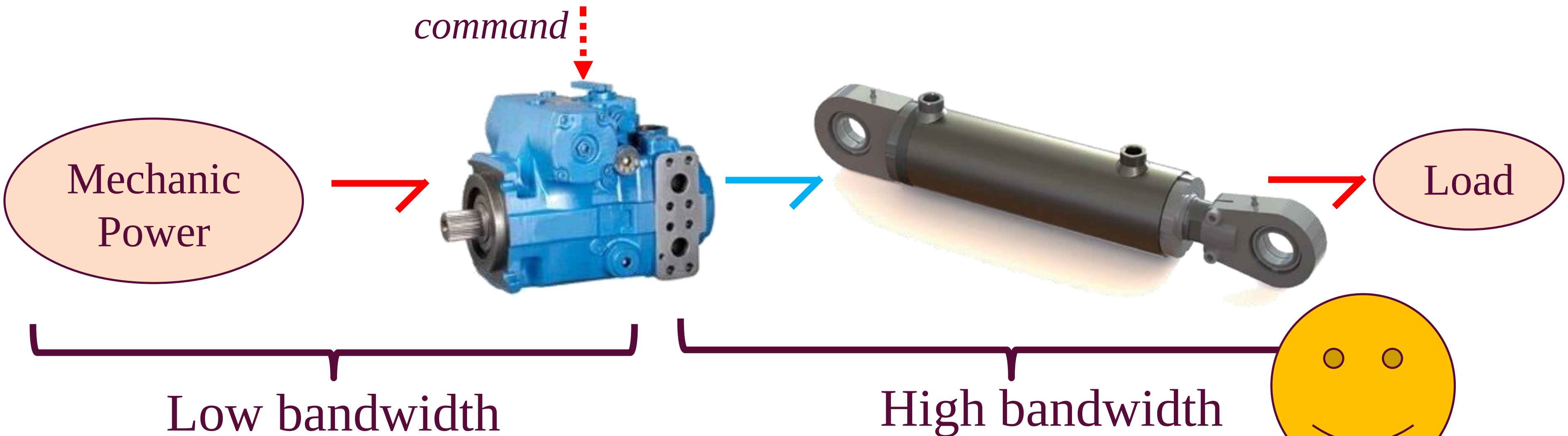
- low inertia
- nearly a integrator

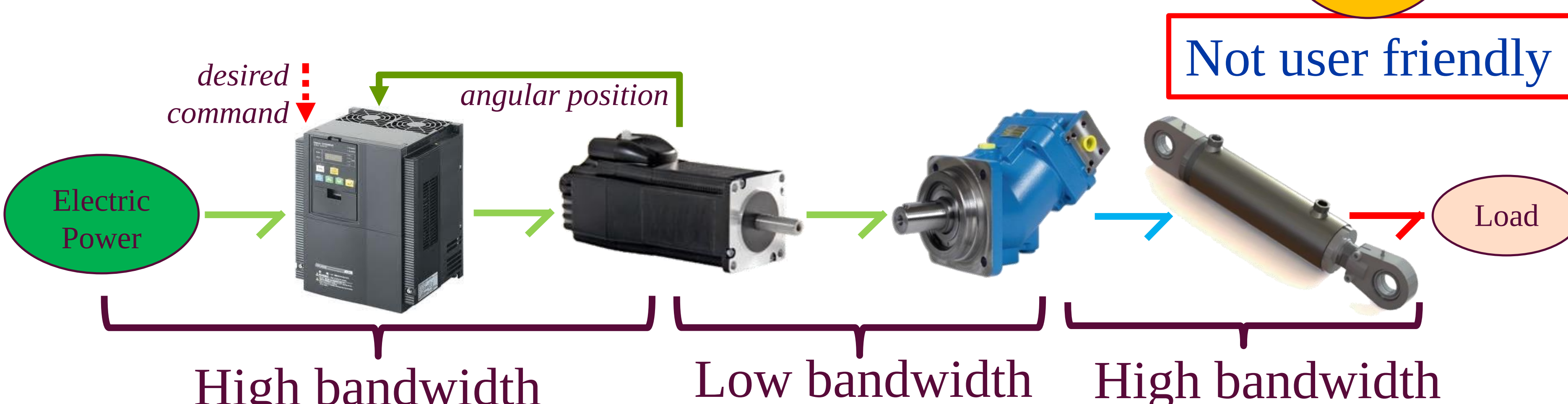
Not user friendly

Help yourself !!!!

Differences electric and fluid power drives (2/3)

❑ Other architectures of hydraulic drives



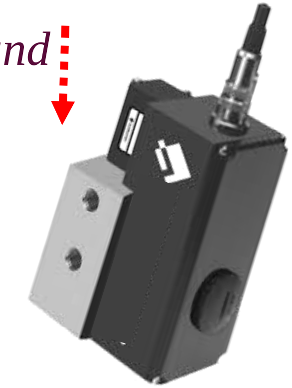


Differences electric and fluid power drives (3/3)

❑ Pneumatic drives

command

Pneumatic Power



Load

High bandwidth

Low bandwidth

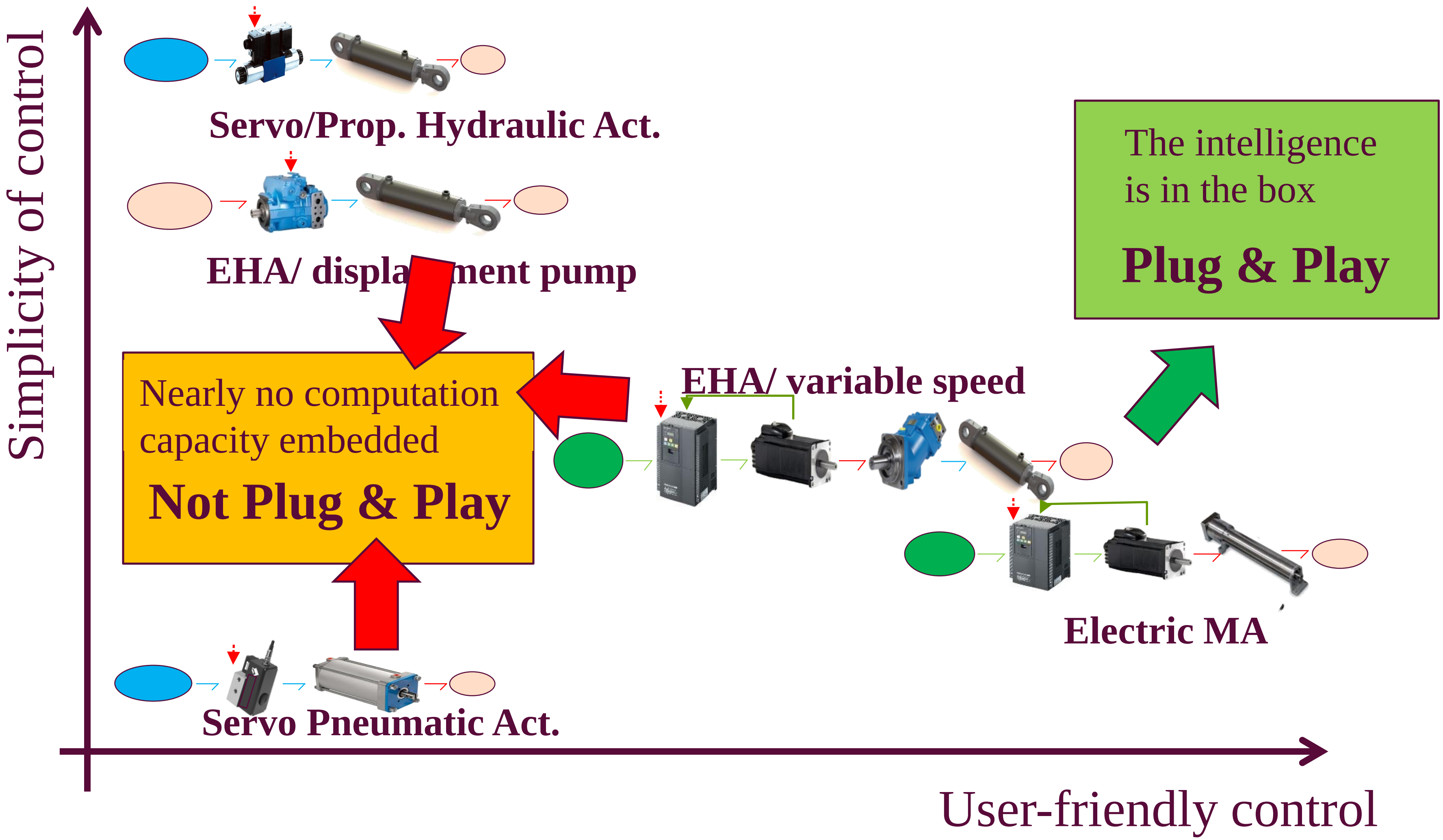
- ~ no sensor (mech/elec feedback)
- low efficiency
- each design is different
- safety component to be added

- low inertia
- low pressure dynamic

Not user friendly

Help yourself !!!!

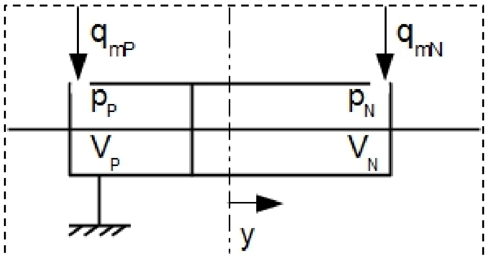
Why electrical drive are so user-friendly ?



How to make the control easier ?

- The physical variables are not always the best for control purposes :

  - On mechanical side



$$\begin{cases} \dfrac{dv}{dt} = \dfrac{S.(p_P - p_N) - F_{frot}(v) - F_{ext}}{M} \\ \dfrac{dy}{dt} = v \end{cases}$$

$F_{pneu} = S.(p_P - p_N)$

$\int$

Speed

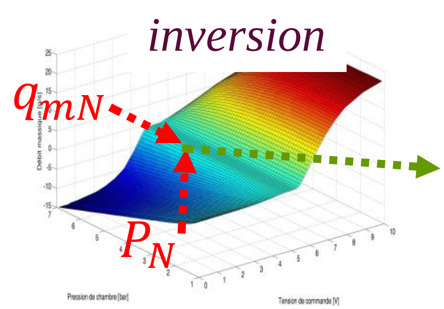
$\int$

Displacement

  - On fluid side (pneumatic)

$$\begin{cases} \dfrac{dp_P}{dt} = \dfrac{k.r.T}{V_P}.(q_{mP} - \dfrac{S}{r.T}.p_P.v) \\ \dfrac{dp_N}{dt} = \dfrac{k.r.T}{V_N}.(q_{mN} + \dfrac{S}{r.T}.p_N.v) \end{cases}$$

2 commands : $\begin{cases} q_{mP(u_P, P_P)} \\ q_{mN(u_N, P_N)} \end{cases}$



$$\begin{cases} u_N = \psi^{-1}(q_{mN}, P_N) \\ u_P = \psi^{-1}(q_{mP}, P_p) \end{cases}$$

The AT transform (1/2)

2 different phenomena

- The differential pressurization : $\frac{d\Delta_p}{dt} = \frac{dp_P}{dt} - \frac{dp_N}{dt}$ ➡ Force
- The symmetric pressurization : $\frac{dp_T}{dt} = \frac{1}{2}.\left(\frac{dp_P}{dt} + \frac{dp_N}{dt}\right)$ ➡ do not modify the force

Let us consider the following coordinate transformation:

$$\begin{bmatrix} q_{mA} \\ q_{mT} \end{bmatrix} = \Lambda(y).\begin{bmatrix} q_{mP} \\ q_{mN} \end{bmatrix}$$

$$\begin{cases} q_{mP} \\ q_{mN} \end{cases} \Rightarrow \begin{cases} q_{mA} = \text{active flow} \\ q_{mT} = \text{pressurization flow} \end{cases}$$

with (*)

$$\Lambda(y) = V_0.\begin{bmatrix} \frac{1}{V_P(y)} & -\frac{1}{V_N(y)} \\ \frac{1}{V_P(y)} & \frac{1}{V_N(y)} \end{bmatrix}$$

** Note : this transformation can easily be extended to non-symmetric cylinder*

The AT transform (2/2)

- Make the really interesting variables appear in the equations

$$\begin{cases} \dfrac{dp_P}{dt} = \dfrac{k.r.T}{V_P}.(q_{mP} - \dfrac{S}{r.T}.p_P.v) \\ \dfrac{dp_N}{dt} = \dfrac{k.r.T}{V_N}.(q_{mN} + \dfrac{S}{r.T}.p_N.v) \end{cases} + \begin{bmatrix} q_{mA} \\ q_{mT} \end{bmatrix} = \Lambda(y).\begin{bmatrix} q_{mP} \\ q_{mN} \end{bmatrix}$$

$$\begin{cases} F_{pneu} = \mathrm{S}.(P_P - P_N) = \mathrm{S}.\Delta p \\ P_T = \dfrac{P_N + P_P}{2} \end{cases}$$

$$\frac{dp_T}{dt} = -\frac{k.S.v}{2}.\left(\frac{p_P}{V_P} - \frac{p_N}{V_N}\right) + \frac{k.r.T}{2.V_0}.q_{mT}$$

$$\frac{d\Delta p}{dt} = -k.S.v.\left(\frac{p_P}{V_P} + \frac{p_N}{V_N}\right) + \frac{k.r.T}{V_0}.q_{mA}$$

The AT transform (2/2)

- Make the really interesting variables appear in the equations

$$
\begin{cases}
\dfrac{dy}{dt} = v \\[2ex]
\dfrac{dv}{dt} = \dfrac{-b.v - F_{sec}(v) + F_{pneu}}{M} \\[2ex]
\dfrac{dF_{pneu}}{dt} = \dfrac{A_1.v.y.F_{pneu} - A_2.v.p_T}{V_P.V_N} + B_1.q_{mA} \\[2ex]
\dfrac{dp_T}{dt} = \dfrac{-A_3.v.F_{pneu} + A_4.v.y.p_T}{V_P.V_N} + B_2.q_{mT}
\end{cases}
$$

Decoupling **Force generation** *from* **Pressurization**

The AT transform (2/2)

- Make the really interesting variables appear in the equations

$$
\begin{cases}
\dfrac{dy}{dt} = v \\[2ex]
\dfrac{dv}{dt} = \dfrac{-b.v - F_{sec}(v) + F_{pneu}}{M} \\[2ex]
\dfrac{dF_{pneu}}{dt} = \dfrac{A_1.v.y.F_{pneu} - A_2.v.p_T}{V_P.V_N} + B_1.q_{m\,A} \\[2ex]
\dfrac{dp_T}{dt} = \dfrac{-A_3.v.F_{pneu} + A_4.v.y.p_T}{V_P.V_N} + B_2.q_{mT}
\end{cases}
$$

$F_{pneu} = S.(p_P - p_N)$

$\int$ Speed

$\int$ Displacement

Active flow

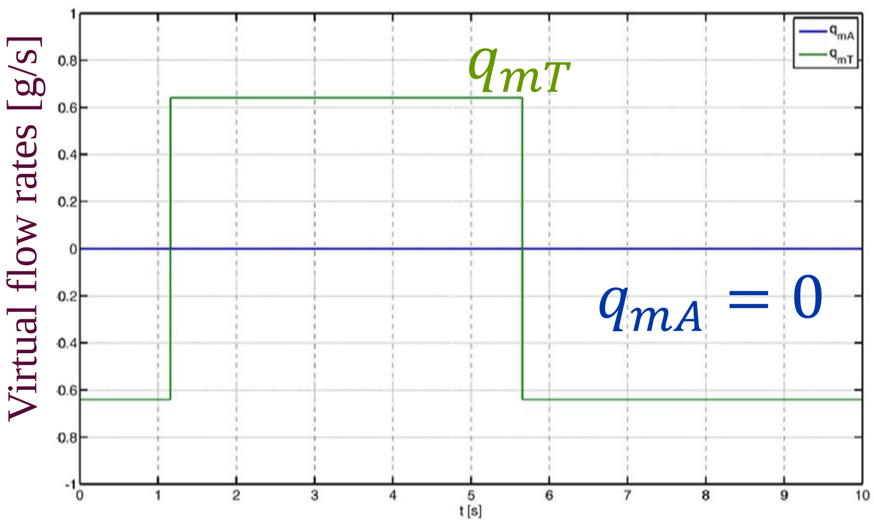
Pressurization flow

P_T ~ Actuator Stiffness

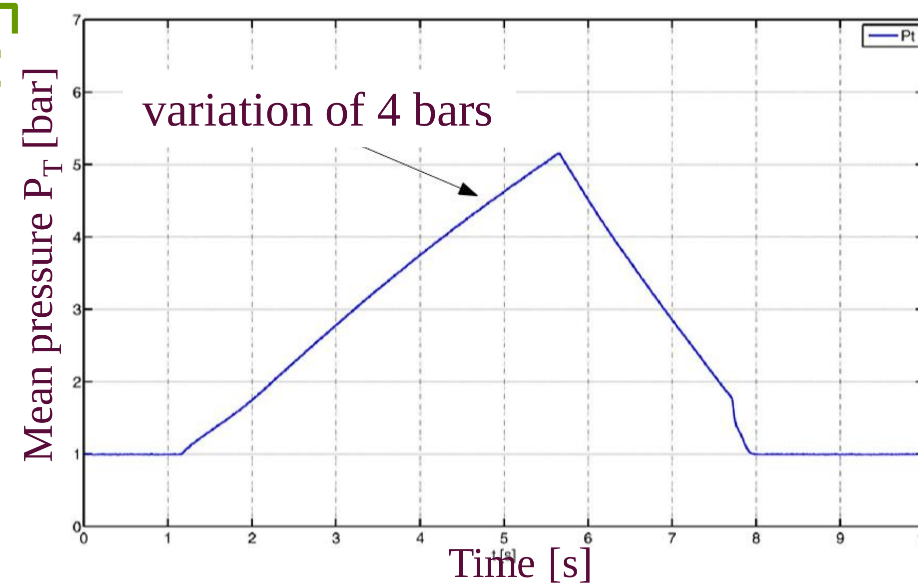
Experimental validation

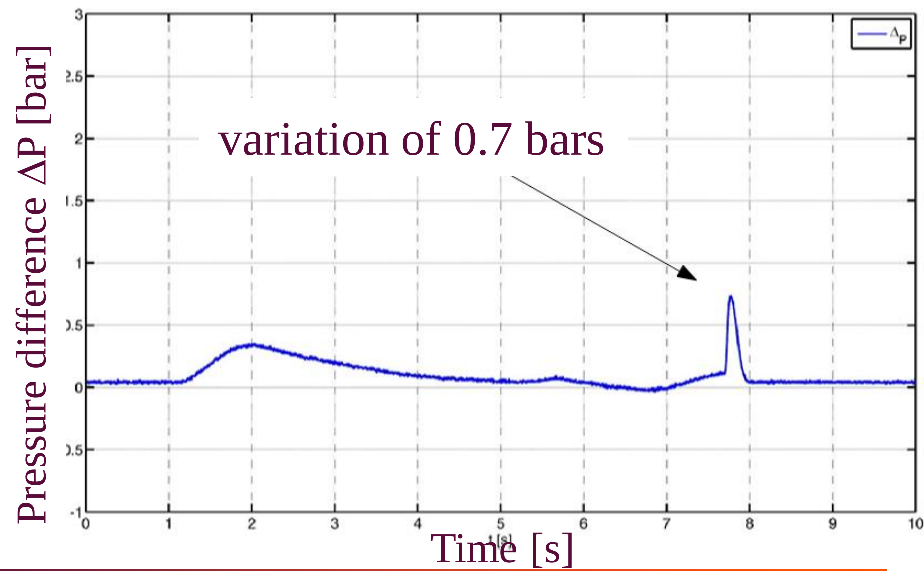
❑ Open-loop : variation on q_{mT}

$$\frac{dp_T}{dt} = -\frac{k.S.v}{2}.\left(\frac{p_P}{V_P} - \frac{p_N}{V_N}\right) + \frac{k.r.T}{2.V_0}.q_{mT}$$



$$\frac{d\Delta p}{dt} = -k.S.v.\left(\frac{p_P}{V_P} + \frac{p_N}{V_N}\right) + \frac{k.r.T}{V_0}.q_{mA}$$

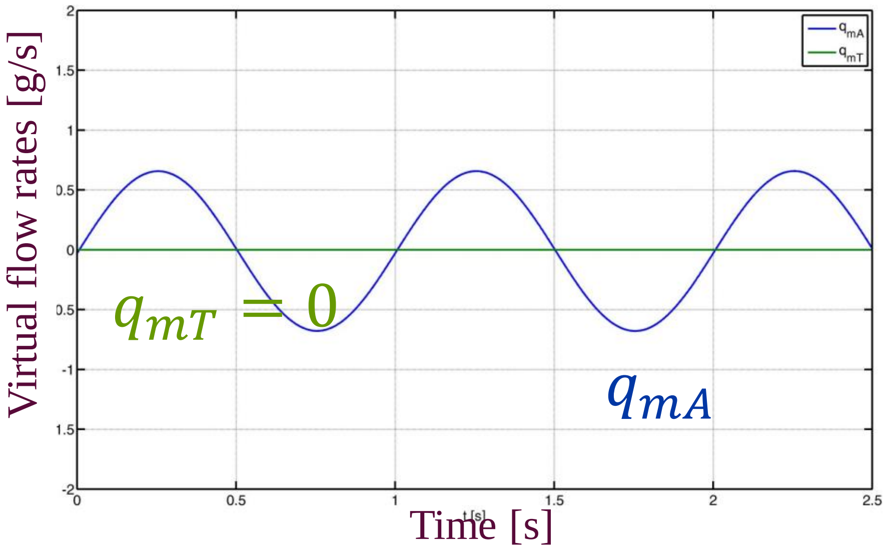




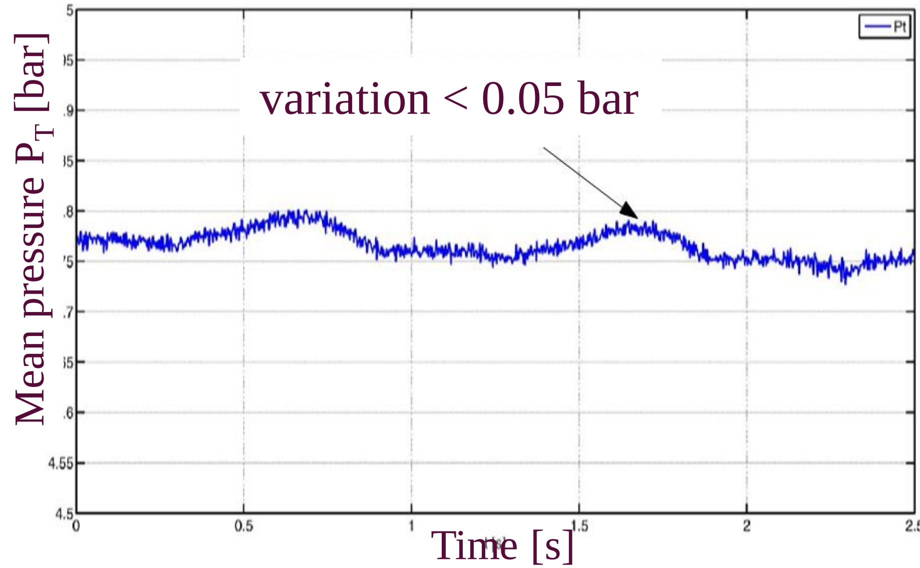
Experimental validation

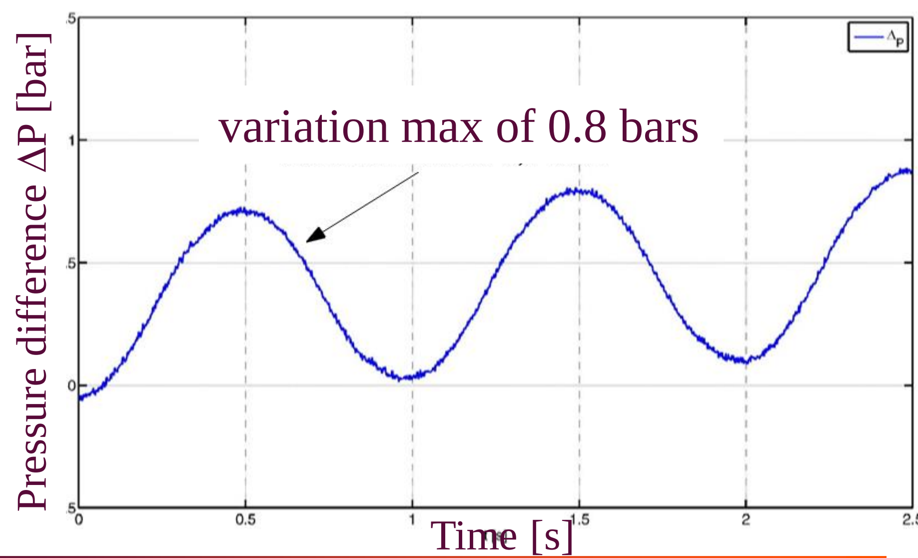
❑ Open-loop : variation on $\boxed{q_{mA}}$

$$\frac{dp_T}{dt} = -\frac{k.S.v}{2}.\left(\frac{p_P}{V_P} - \frac{p_N}{V_N}\right) + \frac{k.r.T}{2.V_0}.q_{mT}$$



$$\frac{d\Delta p}{dt} = -k.S.v.\left(\frac{p_P}{V_P} + \frac{p_N}{V_N}\right) + \frac{k.r.T}{V_0}.q_{mA}$$





AT Transform vs. Park Transform

AT Transform

- change of variables for control purpose:

$$\begin{bmatrix} q_{mA} \\ q_{mT} \end{bmatrix} = \Lambda(y) . \begin{bmatrix} q_{mP} \\ q_{mN} \end{bmatrix}$$

- original flows of power modulator change in :
  - ✓ *virtual active flow qmA*

    Force control

Park Transform

- change of variables for control purpose:

$$\begin{bmatrix} V_d \\ V_q \end{bmatrix} = \sqrt{\frac{2}{3}} . \begin{bmatrix} \cos(\theta) & \cos(\theta - \frac{2.\pi}{3}) & \cos(\theta + \frac{2.\pi}{3}) \\ -\sin(\theta) & -\sin(\theta - \frac{2.\pi}{3}) & -\sin(\theta + \frac{2.\pi}{3}) \end{bmatrix} \begin{bmatrix} V_a \\ V_b \\ V_c \end{bmatrix}$$

- original voltages of power modulator change in :
  - ✓ *virtual voltage Vd*

    Torque control

Displacement and force generation

- ✓ *virtual presurization flow qmT*

  Mean pressure control

- ✓ *virtual voltage Vq*

  Magnetic flux control

Actuator stiffness

Applications of the AT Transform

- Trajectory control (Y-P_T control)
- Energy saving (Y-P_{Topti} control)
- Displacement / Stiffness (Y-K control)
- Position oberver (at 0 speed)
- Mono-distributor

Application of the AT Transform

❑ Active flow

q_{mA}

Pressure difference

$F_{pneu} = S.(p_P - p_N)$

$\int$

Speed

$\int$

Displacement

❑ Pressurization flow

q_{mT}

Pressurization control (Y-P_T control)

Energy saving (Y-P_{Topti} control)

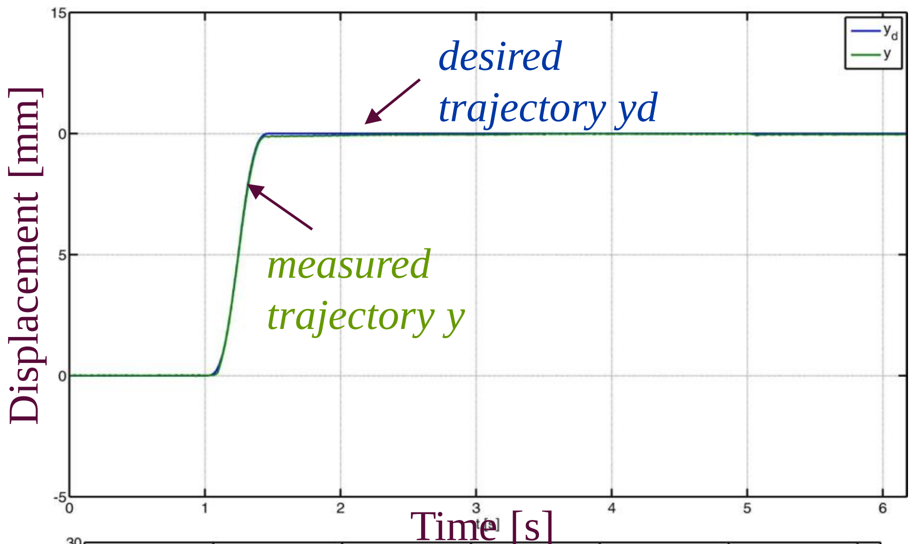
Stiffness (Y-K control)

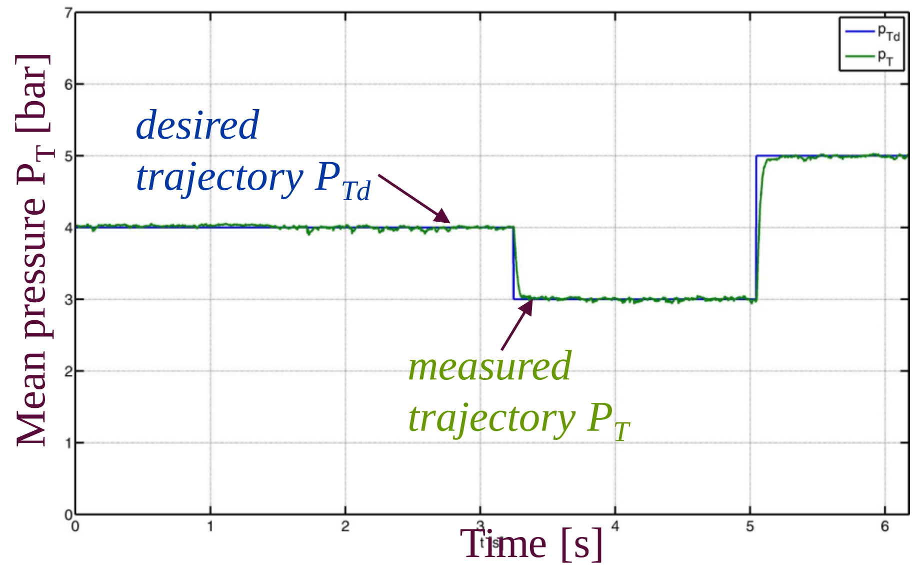
Mono-distributor

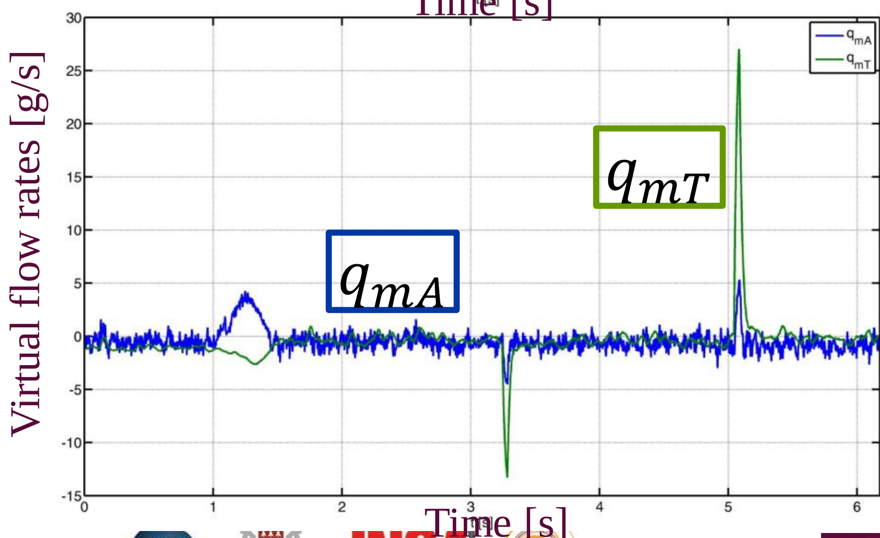
Position oberver (at 0 speed)

Trajectory control (Y-P_T control)

❑ Control synthesis : Backstepping





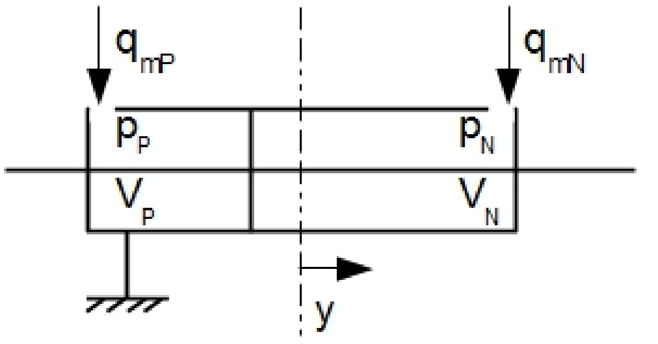


Same results to what was achieved with other control techniques

Energy saving (Y-P_{Topti} control)

❑ Control synthesis :

- q_{mA} is imposed (y trajectory)
- minimize $\boxed{C(t) = |q_{mP}(t)| + |q_{mN}(t)|}$



with $q_{mA} = V_0 . \left(\frac{q_{mP}}{V_P} - \frac{q_{mN}}{V_N} \right)$

If V_N smaller than V_P, less flow is required to produce q_{mA} if q_{mN} is used
If V_P smaller than V_N, less flow is required to produce q_{mA} if q_{mP} is used

$$\begin{cases} q_{mP} = 0 \; ; \; q_{mN} = -\frac{V_N}{V_0} . q_{mA} \text{ if } y > 0, \\ q_{mP} = \frac{V_P}{V_0} . q_{mA} \; ; \; q_{mN} = 0 \text{ if } y < 0, \\ q_{mP} = -q_{mN} = \frac{q_{mA}}{2} \text{ if } y = 0 \end{cases}$$

as $q_{mT} = V_0 . \left(\frac{q_{mP}}{V_P} + \frac{q_{mN}}{V_N} \right)$

Optimal control is obtained for :

$$q_{mT} = -q_{mA} . sgn(y)$$

Stiffness (Y-K control)

Stiffness control:

- reject or not disturbances in open-loop (do not required fast closed loop)
- variable compliance

Control synthesis :

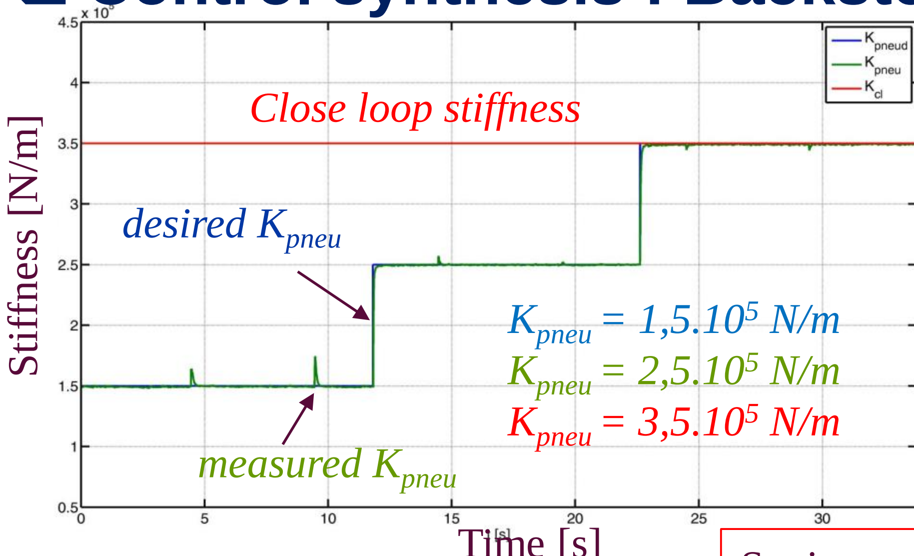
- q_{mA} is imposed (y trajectory)
- Actuator stiffness $K_{pneu} = \left(\frac{p_P}{V_P(y)} + \frac{p_N}{V_N(y)} \right) .k.S^2$

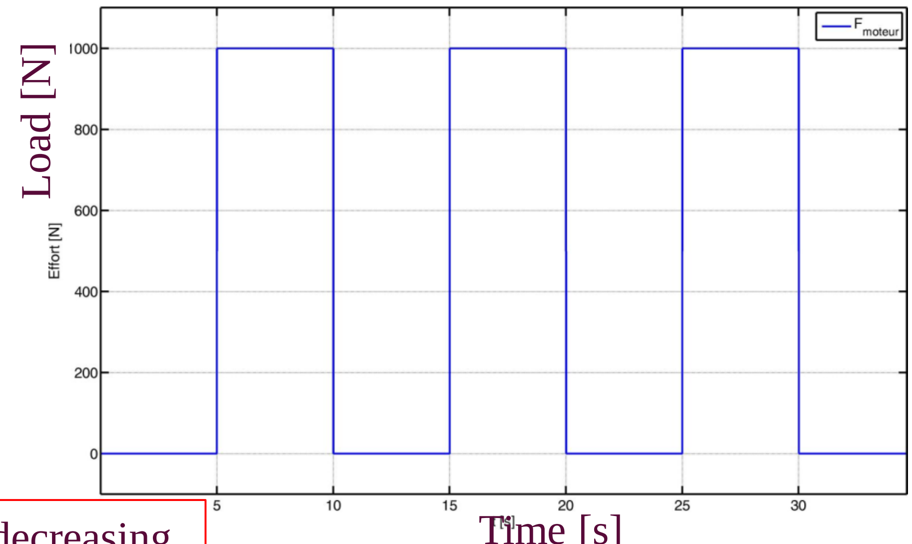
$$\left\{ \begin{array}{l} \frac{dy}{dt} = v \\ \\ \frac{dv}{dt} = \frac{-b.v - F_{sec}(v) + F_{pneu}}{M} \\ \\ \frac{dF_{pneu}}{dt} = -K_{pneu}.v + B_1.q_{m\,A} \\ \\ \frac{dK_{pneu}}{dt} = \frac{A_5.K_{pneu}.y.v - A_6.F_{pneu}.v - B_4.y.q_{m\,A} + B_5.q_{m\,T}}{V_N(y).V_P(y)} \end{array} \right.$$

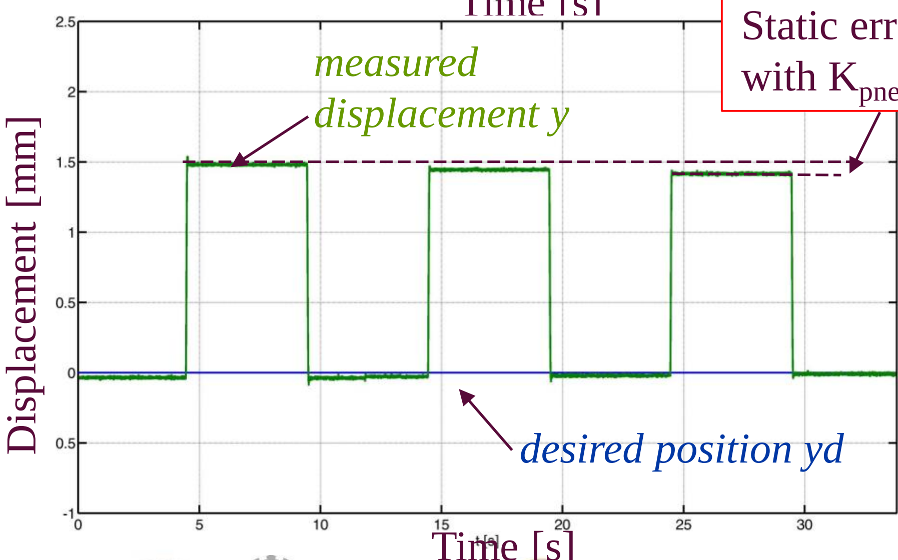
natural behavior of the actuator to come back to its equilibrium point

• **Stiffness (Y-K control)**

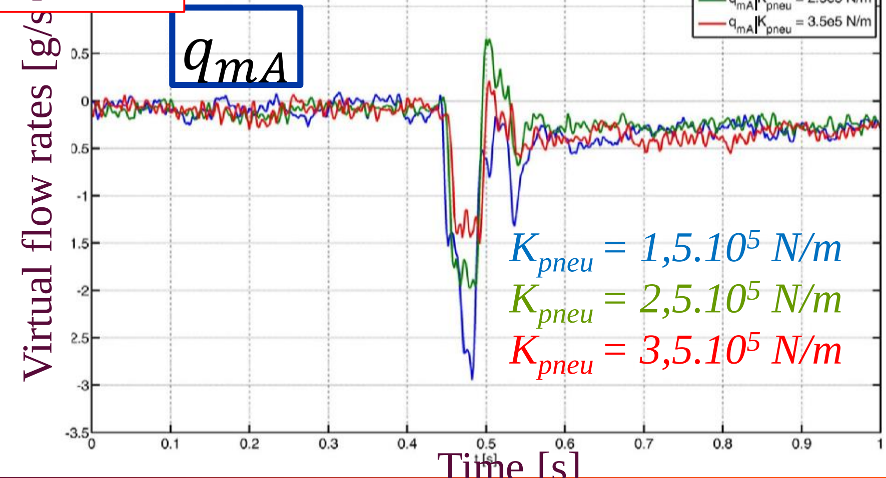
❑ Control synthesis : Backstepping







Static error decreasing with K_{pneu} at impact



Position observer at standstill

Position observer at standstill (v=0) :

$$\begin{cases} \frac{dy}{dt} = 0 \\ \frac{dv}{dt} = \frac{-F_{sec}(v=0) + F_{pneu}}{M} \\ \frac{dF_{pneu}}{dt} = B_1.q_{mA} \\ \frac{dp_T}{dt} = B_2.q_{mT} \end{cases}$$

y, q_{mA}, q_{mT} not known

$$\begin{cases} \frac{dy}{dt} = 0 \\ \frac{dv}{dt} = \frac{-F_{sec}(v=0) + F_{pneu}}{M} \\ \frac{dF_{pneu}}{dt} = B_1.[\hat{q_{mA}} + \bar{y}.\frac{S.V_0.\hat{q_{mT}} - S^2.y.\hat{q_{mA}}}{V_P.V_N}] \\ \frac{dp_T}{dt} = B_2.[\hat{q_{mT}} + \bar{y}.\frac{S.V_0.\hat{q_{mA}} - S^2.y.\hat{q_{mT}}}{V_P.V_N}] \end{cases}$$

position estimation error

estimated system inputs

From pressure sensors only :

- ✓ F_{pneu} and P_T can be calculated,
- ✓ the full state can be obtained by differentiation:

$$y_m = \begin{bmatrix} F_{pneu} \\ p_T \\ \frac{dF_{pneu}}{dt} \\ \frac{dp_T}{dt} \end{bmatrix}$$

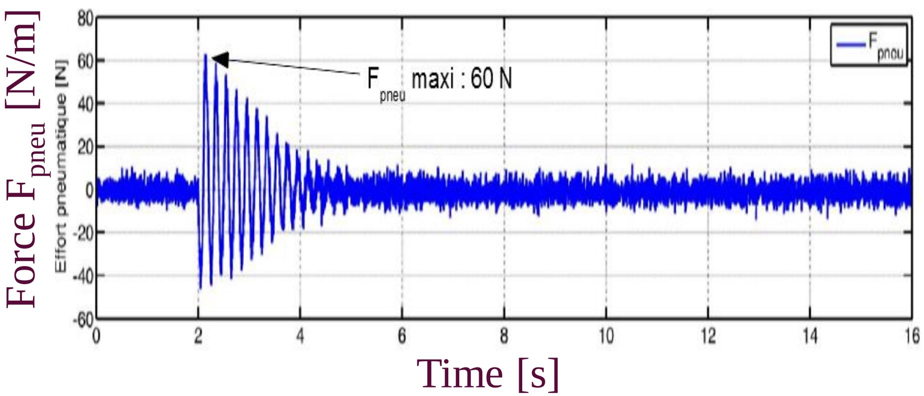
Procedure

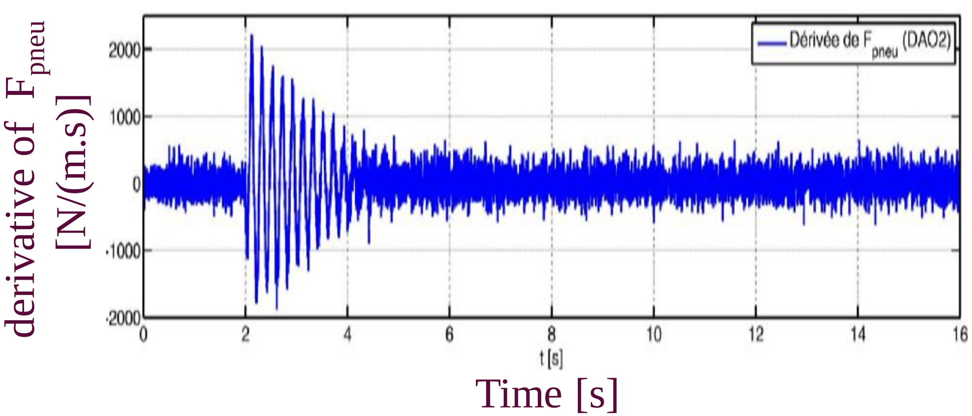
- **change slightly q_{mT} with $q_{mA} = 0$**
  - theoritically F_{pneu} do not change (or < dry friction)
  - the piston do not move (v=0)

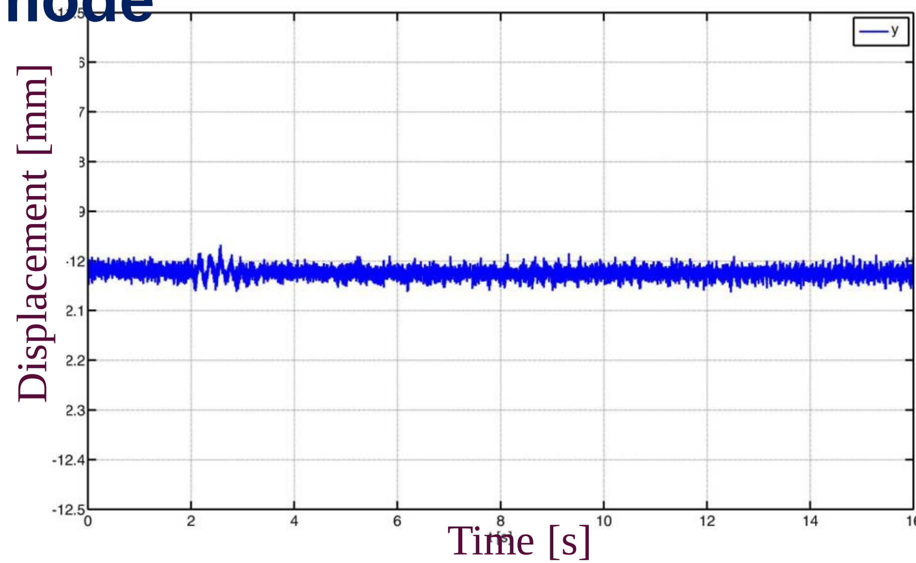
Position observer at standstill

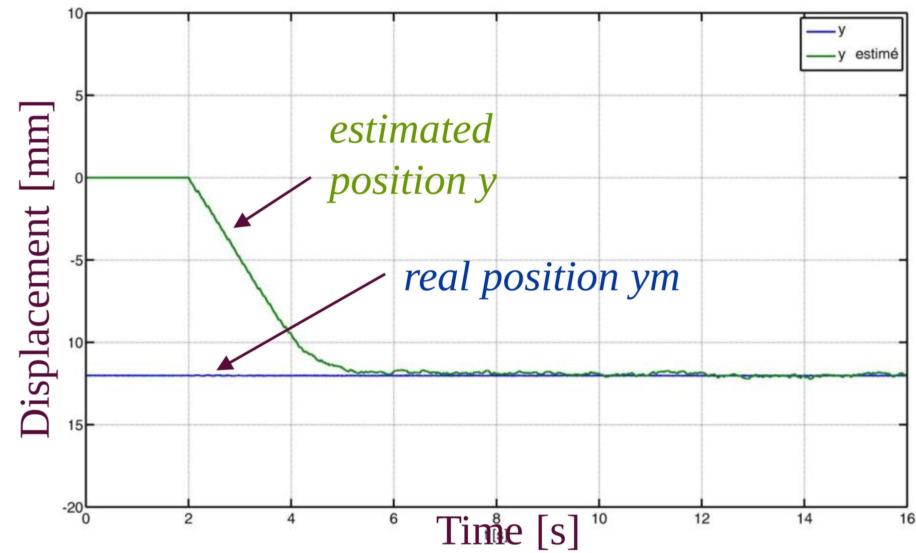
❑ Observer synthesis : Sliding mode

- ○ 5 Hz Sinusoisal trajectory on P_T









Conclusion

❑ Active flow

$\boxed{q_{mA}}$

Pressure difference

$F_{pneu} = S.(p_P - p_N)$

$\int$

Speed

$\int$

Displacement

❑ Pressurization flow

$\boxed{q_{mT}}$

Pressurization control (Y-P_T control)

Energy saving (Y-P_{Topti} control)

Stiffness (Y-K control)

Mono-distributor

Position oberver (at standstill)

What else ?

Thanks a lot for your kind attention…

References :

1. Frédéric Abry, Xavier Brun, Sylvie Sesmat, Eric Bideaux. Non-linear position control of a pneumatic actuator with closed-loop stiffness and damping tuning, ECC, Jul 2013, Zürich, Switzerland. pp.1089 - 1094, 2013

2. Frédéric Abry, Xavier Brun, Michaël Di Loreto, Sylvie Sesmat, Eric Bideaux. Piston position estimation for an electro-pneumatic actuator at standstill, Control Engineering Practice, Elsevier, 2015, 41 (8), pp.176-185