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Robust PID Controller for a Pneumatic Actuator

Michael G. Skarpetis[1,a], Fotis N. Koumboulis[1] , George Panagiotakis[1] and Nikolaos D. Kouvakas[1]

[1] *Department of Automation Eng., Technological Educational Institute of Sterea Ellada, 34400 Psahna Evias, Chalkida, Greece*

Abstract. In this paper the position control pneumatic actuator using a robust PID controller is presented. The parameters of the PID controller are computed using a Hurwitz invariability technique enriched with a Simulated Annealing Algorithm. The nonlinear model involves uncertain parameters due to linearization of the servo valve, variations of the initial volume of the cylinder and variation of the external load. The problem is proven to be solvable and the controller parameters are chosen to provide a suboptimal solution for tracking error minimization. Simulation results are presented for the nonlinear model.

1 Introduction

The major aspects for controlling pneumatic actuators are the uncertainties appearing in the linear and nonlinear model and the nature of fluid power. Many control techniques such as H–infinity, quantitative feedback theory (QFT) Robust control techniques appear to be effective for controlling the position and the velocity of industrial pneumatic actuators (see f.e. [1]-[12]). These techniques can work properly to linear and nonlinear models and produce accurate system performance despite plant uncertainty as well as plant variations and disturbances.

In this paper a robust PID controller is proposed in order to produce accurate system performance despite plant uncertainty and plant variations and disturbances. The problem is solved using a Hurwitz invariability technique [13], [14] and a Simulated Annealing Technique [11]. First, the nonlinear model of the pneumatic actuator is linearized around equilibrium point. To the linear uncertain system a robust dynamic tracking controller is designed, combining the results in [9], [13] and [14]. The effectiveness of the PID controller is illustrated through simulations for the nonlinear model of the plant and for various values of the model uncertain parameters.

2. Pneumatic servo actuator model

The motion of a hydraulic cylinder controlled by a servo valve linearized around an initial spool valve position (see Figure 1) can be expressed by the following set of nonlinear differential equations [3], [4]:

$$\dot{P}_a(t) = \frac{aRTK_vP_s}{2(A_py_p(t)+V_0)}u(t) - \frac{aP_a(t)A_p}{(A_py_p(t)+V_0)}\dot{y}_p(t) \quad (1)$$

$$\dot{P}_b(t) = \frac{-aRTK_vP_s}{2(V_0-A_py_p(t))}u(t) + \frac{aP_b(t)A_p}{(V_0-A_py_p(t))}\dot{y}_p(t) \quad (2)$$

$$\ddot{y}_p(t) = \frac{A_p}{M}P_a(t) - \frac{A_p}{M}P_b(t) - \frac{1}{M}F_L(t) - \frac{1}{M}\dot{y}_p(t) \quad (3)$$

where $P_a(t)$ and $P_b(t)$ are the pressures in chamber a and b of the pneumatic cylinder, $y_p(t)$ is the position of the piston, $u(t)$ is the electro valve input signal, $F_L(t)$ is the disturbance force, M is the load mass, A_p is the piston area, V_0 is the air volume when the piston is in the mid position, R is the gas constant, a is the specific heat ratio, T is the temperature of the air source, K_v is the valve coefficient and P_s is the supplied pressure.

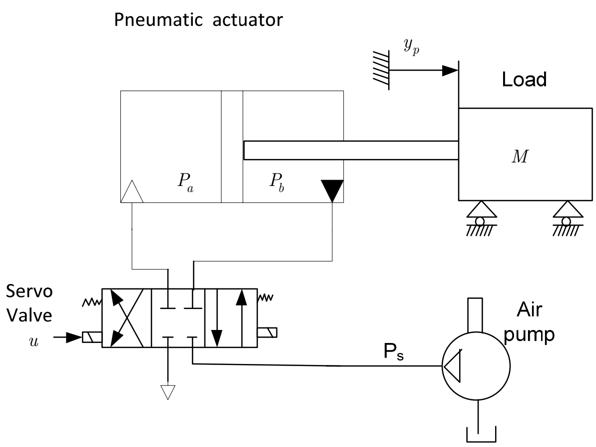


Figure 1. Pneumatic actuator

The nonlinear system in (1)-(3) can equivalently be re-written as

[a] Corresponding author: mskarpetis@teiste.gr

$$\dot{x}(t) = f\big(x(t)\big) + g\big(x(t)\big)u(t) + \frac{1}{M}F_L(t) \qquad (4)$$

where $x(t) = \begin{bmatrix} P_a(t) & P_b(t) & \dot{y}_p(t) & y_p(t) \end{bmatrix}^T$,

$$f\big(x(t)\big) = \begin{bmatrix} -\dfrac{aP_a(t)A_p}{(A_p y_p(t) + V_0)}\dot{y}_p(t) \\ \dfrac{aP_b(t)A_p}{(-A_p y_p(t) + V_0)}\dot{y}_p(t) \\ \dfrac{A_p}{M}P_a(t) - \dfrac{A_p}{M}P_b(t) - \dfrac{1}{M}\dot{y}_p(t) \end{bmatrix}$$

$$g\big(x(t)\big) = \begin{bmatrix} \dfrac{aRTK_vP_s}{2(A_p y_p(t) + V_0)} \\ \dfrac{-aRTK_vP_s}{2(-A_p y_p(t) + V_0)} \\ 0 \end{bmatrix}$$

The nonlinear system (4) is linearized around the equilibrium point $\left(P_{a,0}, P_{b,0},\ \dot{y}_{p,0}, y_{p,0}, u_0\right)$ yielding the following state space linear uncertain model:

$$\begin{aligned} \Delta\dot{x}(t) &= A(q)\Delta x(t) + B(q)\Delta u(t) + DF_L(t) \\ \Delta y_h(t) &= C\Delta x(t) \end{aligned} \qquad (5)$$

where

$$\Delta x(t) = \begin{bmatrix} \Delta P_a(t) \\ \Delta P_b(t) \\ \Delta\dot{y}_p(t) \\ \Delta y_p(t) \end{bmatrix}$$

is the state vector of the linear system, and where $\Delta P_a(t) = P_a(t) - P_{a,0}$, $\Delta P_b(t) = P_b(t) - P_{b,0}$, $\Delta\dot{y}_p(t) = \dot{y}_p(t) - \dot{y}_{p,0}$, $\Delta y_p(t) = y_p(t) - y_{p,0}$ and $\Delta u(t) = u(t) - u_0$ denote small perturbations of the state variables and the input variable around the equilibrium point $\left(P_{a,0}, P_{b,0},\ \dot{y}_{p,0}, y_{p,0}, u_0\right)$.

The system matrices are

$$A(q) = \begin{bmatrix} -\dfrac{a\dot{y}_{p,0}A_p}{q_1 + A_p y_{p,0}} & 0 & -\dfrac{aA_pP_{a,0}}{q_1 + A_p y_{p,0}} & \dfrac{aA_p(2\dot{y}_{p,0}A_pP_{a0} - Ru_0q_4P_sT_b)}{2(V_0 + A_p y_{p,0})^2} \\ 0 & \dfrac{a\dot{y}_{p,0}A_p}{q_1 - A_p y_{p,0}} & \dfrac{aA_pP_{b,0}}{q_1 - A_p y_{p,0}} & \dfrac{aA_p(2\dot{y}_{p,0}A_pP_{b,0} - Rq_4u_0P_sT_b)}{2(V_0 - A_p y_{p,0})^2} \\ \dfrac{A_p}{M} & -\dfrac{A_p}{M} & -\dfrac{q_2}{q_3} & 0 \\ 0 & 0 & 1 & 0 \end{bmatrix},$$

$$B(q) = \begin{bmatrix} \dfrac{aRq_4P_sT_b}{2(q_1 + A_p y_{p,0})} \\ \dfrac{aRq_4P_sT_b}{2A_p y_{p,0} - 2q_1} \\ 0 \\ 0 \end{bmatrix},\ D = \begin{bmatrix} 0 \\ 0 \\ \dfrac{-1}{M} \\ 0 \end{bmatrix},$$

$$C = \begin{bmatrix} 0 & 0 & 0 & 1 \end{bmatrix}$$

The vector

$$q = \begin{bmatrix} q_1 & q_2 & q_3 & q_4 \end{bmatrix} = \begin{bmatrix} V_0 & f & M & K_v \end{bmatrix} \in \mathbb{Q}$$

is the uncertainty vector involving variations of the initial air volume, variations of the viscous damping coefficient, mass variations and valve gain variations. $\mathbb{Q}$ denotes the domain of uncertainty. The nominal values of the parameters of the system are shown in Table 1 and the expected range of variations of the uncertain parameters is shown in Table 2.

Table 1. Nominal Values for the System Parameters

Symbol	Definition	Nominal Values
A_p	Piston Area	$0{,}005m^2$
R	Ideal gas constant	$287\left(J\ /\ Kg * K\right)$
P_s	Supply pressure	$4 * 10^5\,Pa$
T_b	Air temperature	$293{,}15K^o$
a	Specific heat ratio	1.4

Table 2. Expected Range of Variations of the Uncertain Parameters

Symbol	**Definition**	**Min. Values**	**Nom. Values**	**Max. Values**
V_0	Initial air volume (m^3)	$1.5{*}10^{-4}$	$2.5{*}10^{-4}$	$4{*}10^{-4}$
f	Viscous damping coeff. $\left(\dfrac{N\sec}{m}\right)$	50	60	80
M	Load Mass (Kg)	0.1	1	2

K_v	Valve gain $\left(\frac{Kg}{\sec * V}\right)$	$3.2 * 10^{-3}$	$3.4 * 10^{-3}$	

3. Robust PID controller

To the open loop system (5) apply the PID controller (see Figure 2):

$$\Delta u(t) = K_p e(t) + K_i \int_0^t e(\tau) d\tau + K_d \frac{de(t)}{dt} \qquad (6)$$

where $e(t) = r_{ref}(t) - \Delta y_p(t)$ and r_{ref} is the external command (or reference signal). The resulting closed loop characteristic uncertain polynomial for $\left(P_{a,0} = \frac{P_s}{2}, P_{b,0} = \frac{P_s}{2}, \; \dot{y}_{p,0} = 0, y_{p,0} = 0, u_0\right)$ is computed to be:

$$p_c(s, q, K_i, K_p, K_d) = s^4 + f_3(q)s^3 + f_2(q, K_d)s^2 + f_1(q, K_p)s + f_0(q, K_i) \qquad (7)$$

where

$$f_3(q) = \frac{q_2}{q_3},$$

$$f_2(q, K_D) = \frac{aA_p P_s (A_p + RK_d q_4 T_b)}{q_3 q_1},$$

$$f_1(q, K_p) = \frac{aRA_p K_p q_4 P_s T_b}{q_3 q_1},$$

$$f_0(q, K_i) = \frac{aRA_p K_i q_4 P_s T_b}{q_3 q_1}.$$

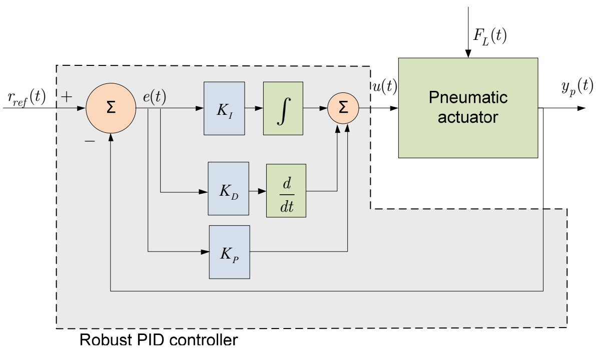
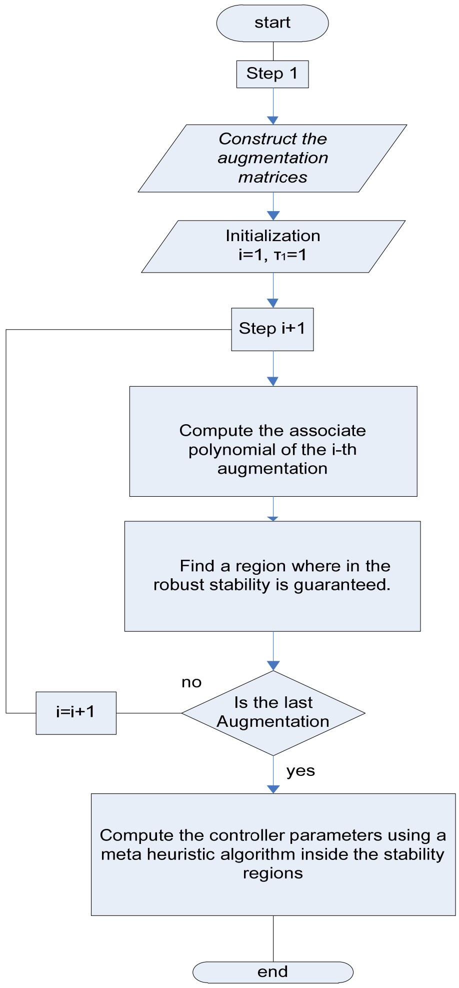


Figure 2. Robust PID control scheme

3.1.1 Finite step algorithm

The parameters of the PID controller will be computed using the following algorithm which has been presented in [11]-[13]:



Applying the algorithm to the closed loop characteristic uncertain polynomial (7) the following values are computed

$$K_p = 0.3, \; K_i = 0.05, \; K_d = 0.5$$

3.1.2 Simulation results

Using the data values of Table 1 and 2, the aforementioned PID parameters, the external command of magnitude 0.01 and the external disturbance 0.01Cos[$3t$], closed loop simulation results for the performance of the actuator, for all the range of uncertainties are illustrated in Figures 3 - 7.

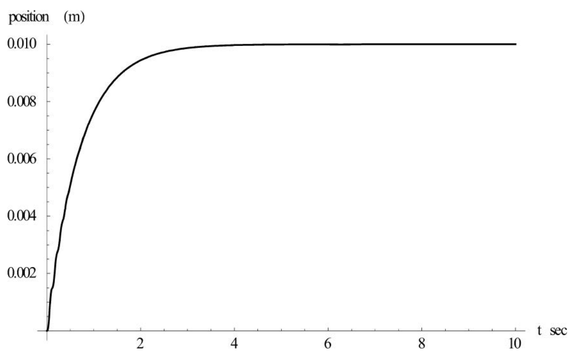


Figure 3. Actuator position

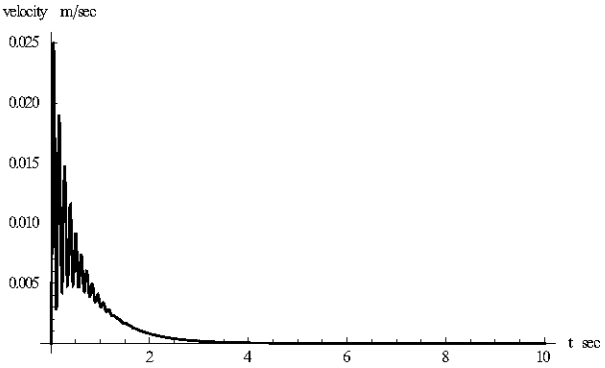


Figure 4. Actuator velocity

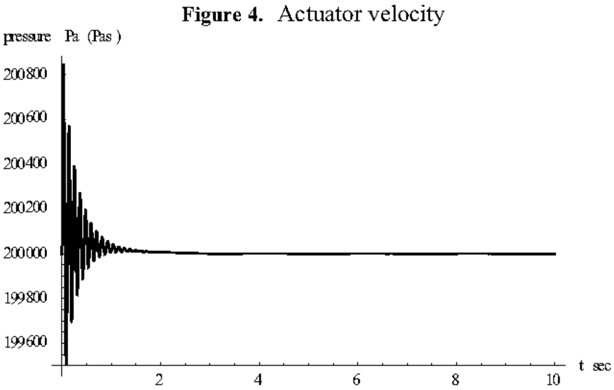


Figure 5. Actuator differential pressure chamber A

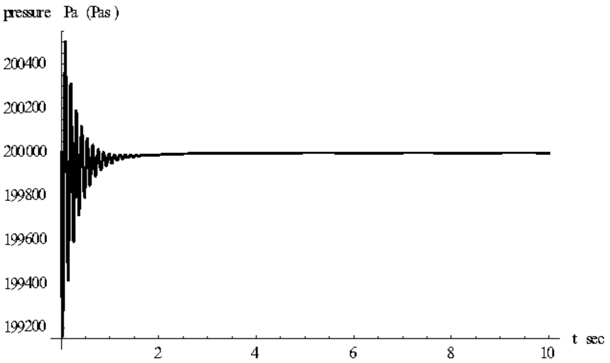


Figure 6. Actuator differential pressure chamber B

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Conclusion

The problem of controlling the position of a pneumatic actuator has been studied using a robust arbitrary tracking controller. The performance of the controller to external disturbance and uncertainties has been tested to the nonlinear model of the actuator. The present results can easily be applied to real time environment with appropriate sensor and DSP cards.

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