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ScienceDirect
IFAC-PapersOnLine 49-21 (2016) 559–566

Soft Landing and Disturbance Rejection for Pneumatic Drives with Partial Position Information ⋆

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Abstract: Pneumatic drives are used in a wide range of industrial applications. Most of the pneumatic drive applications are simple point-to-point movements, where the motion characteristics is typically set up once by manual tuning. Changes in the operating conditions demand a new manual adjustment and thus additional costs. This work aims at developing a conrol strategy for pneumatic drives to save manual tuning effort and to minimuze the overall system costs. For this cheap position sensors that operate only near the end stops in combination with energy efficient switching valves are used to ensure a smooth movement of the drive and soft landing at the end stops. To pass through the region with no position information, a two-degrees-of-freedom control strategy is employed to account for model uncertainties and disturbances. Inside the position measurement region, compliance control ensures soft landing. The presented strategy is validated by a series of measurements on an experimental test bench.

Keywords: Pneumatic systems; disturbance rejection; impedance control; feedforward control.

1. INTRODUCTION

Pneumatic drives are often used in manufacturing industry, see, e.g., Saidur et al. (2010); Doll et al. (2011). The low investment costs and the high achievable power density makes pneumatic drives particularly suitable for simple handling tasks such as point-to-point movements, see, e.g., Shen et al. (2006); Hildebrandt et al. (2010). There are basically two approaches to perform an point-to-point movement with a pneumatic drive. On the one hand, a servo-pneumatic controller can be used. This approach requires a rather expensive position measuring system but allows to perform a smooth transition of the end-effector, see, e.g., Ilchmann et al. (2006); Richer and Hurmuzlu (2000); Hodgson et al. (2012, 2015). Most of the concepts presented in literature require continuous position sensors over the full stroke length to ensure a high control performance. On the other hand, a simple control strategy may be used, which empties one chamber and fills the other one. This simple switching strategy requires an additional end-of-stroke damper to absorb the resulting high impact energy at the end stops or throttle valves to limit the piston velocity. In the latter approach, the adjustment of the end-of-stroke dampers or the throttle valves turns out to be problematic. Typically, the characteristics of the end-of-stroke damper, i.e., the throttle valve cross sections, have to be manually tuned depending on the supply pressure level and the moving mass. Hence, if the working conditions of a pneumatic drive in a production line change, the damping element or the throttle have to be readjusted, which might require pausing the production. In large pneumatic systems, the supply pressure level varies depending on the distance to the next service unit. This can also lead to the necessity of a costly readjustment of the dampers or the throttle valves. Another common problem in industrial applications are supply pressure drops, which result from several pneumatic loads using the same pressure supply. These pressure drops can lead to lower velocities of the pneumatic piston due to the manually adjusted throttle valves. In this case, the movement may no longer fulfil the timing requirements of the production line.

To overcome these drawbacks, a control strategy is proposed that mimics an end-of-stroke damper. The idea is to place short position sensors close to the desired end positions and to use compliance control to emulate a mass-spring-damper behaviour at the end stops. In order to pass through the range with no position information, a combined position feedforward and pressure feedback control strategy is used.

The actuation of pneumatic drives is classically performed with a costly 5-port/3-way proportional valve, see, e.g., Hildebrandt et al. (2010); Riachy and Ghanes (2014); Toedtheide et al. (2016). In this approach, since only a single input is available, the end-effector position can be controlled but the chamber pressures cannot be influenced separately. Furthermore, due to the construction of proportional valves, they exhibit leakage flows, which reduce the overall energy efficiency, see, e.g., Krichel et al. (2012); Doll et al. (2011). The usage of two pneumatic half-bridges equipped with two cheap 2-port/2-way switching valves each allows to control the end-effector position and the sum pressure of the pneumatic drive. Moreover, it allows to minimize the leakage flows and to ensure cost savings, see, e.g., Saidur et al. (2010); Murrenhoff (2006); Belforte et al. (2004); Ye et al. (1992); van Varseveld and Bone (1997); Schindele et al. (2012); Shen et al. (2006). Hence, this approach is also adopted in this work.

⋆ The authors thank Festo AG & Co. KG for funding this project.

10.1016/j.ifacol.2016.10.661

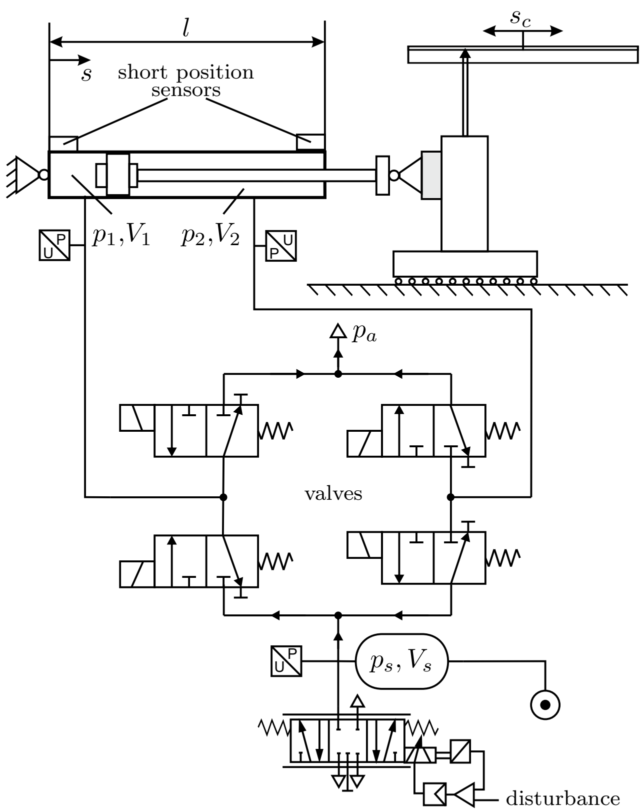

Fig. 1. Schematic of the pneumatic linear drive containing the four fast switching valves, the position sensors, and the proportional valve to realize supply pressure drops.

The paper is organized as follows: Section 2 describes the experimental setup. In Section 3, a mathematical model of the system at hand is presented. Section 4 is devoted to the controller design and Section 5 gives measurement results from the implementation of the proposed control strategy on a test bench. Some conclusions are drawn in Section 6.

2. EXPERIMENTAL SETUP

Fig. 1 shows a schematic of the system under consideration, which consists of a differential cylinder equipped with two short position sensors mounted at the stroke ends with a measurement range of about 50 mm. Low cost pressure sensors are located at the piston inlets and the outlet of the supply pressure chamber. An additional position measuring system provides full position information over the entire stroke to validate the proposed approach. Four fast switching valves, arranged in a full-bridge, control the motion of the piston and the sum pressure. To verify the robustness of the presented control strategy, an additional proportional valve is installed to simulate disturbances in the supply pressure.

3. MATHEMATICAL MODEL

The pneumatic drive can be described by the following set of differential equations, see, e.g., Andersen (2001),

$$\ddot{s} = \frac{1}{m}\big(F_f(\dot{s}) + F_p(p_1,p_2) + F_a\big) \tag{1a}$$

$$\dot{p}_i = \frac{\kappa}{V_i(s)}\left((-1)^i A_i \dot{s} p_i + R\theta_g \dot{m}_i\right),\ i \in \{1,2\}, \tag{1b}$$

with piston position s and chamber pressures p_1 and p_2. In (1a), m denotes the overall moving mass of the system, $F_p(p_1,p_2) = p_1 A_1 - p_2 A_2$ is the pressure force with effective piston areas A_1 and A_2, and $F_a = p_a(A_2 - A_1)$ is the pressure force offset due to the (constant) ambient pressure p_a. Moreover, viscous and Coulomb friction is assumed and modelled by $F_f(\dot{s}) = -c\tanh(\dot{s}/\varepsilon) - d\dot{s}$ with coefficients $\varepsilon \ll 1$, $c > 0$, and $d > 0$. The differential equations for the chamber pressures (1b) contain the chamber volumes $V_1(s) = A_1 s + V_{1,0}$ and $V_2(s) = A_2(l-s) + V_{2,0}$, with dead volumes $V_{1,0}$ and $V_{2,0}$ and maximal stroke length l, the specific gas constant R, the (constant) gas temperature θ_g, and the specific heat ratio κ. Since the valve dynamics are reasonably fast compared to the temperature and pressure dynamics, the instantaneous switching of the valves is assumed in the following. Furthermore, assuming an adiabatic lossless flow, the mass flows $\dot{m}_i$ can be described, according to ISO 6358 (2012), by

$$\dot{m}_1 = C_{1s}\Gamma_{1s}(p_1) - C_{a1}\Gamma_{a1}(p_1) \tag{2a}$$

$$\dot{m}_2 = C_{2s}\Gamma_{2s}(p_2) - C_{a2}\Gamma_{a2}(p_2), \tag{2b}$$

with pneumatic conductances $C_{ij} = \{0, C_{\max}\}$ and

$$\Gamma_{ij}(p_i) = \rho_0 p_j \Psi\left(p_i/p_j\right)\ , \tag{3}$$

where $\rho_0 = 1.1845\,\mathrm{kg/m^3}$ denotes the technical density and p_s is the supply pressure. In (3),

$$\Psi\left(\Pi_{ij}\right) = \begin{cases} \sqrt{1 - \left(\frac{\Pi_{ij} - \Pi_c}{1 - \Pi_c}\right)^2} & \text{for} \quad \Pi_{ij} \geq \Pi_c \\ 1 & \text{for} \quad \Pi_{ij} < \Pi_c \end{cases} \tag{4}$$

represents the flow-through function with pressure ratios $\Pi_{ij} = p_i/p_j$ and critical pressure ratio $\Pi_c \geq 0$. In the application at hand, the conductances C_{ij} are pulse-width modulated (pwm). For $k = 0,1,\ldots$ they read as

$$C_{ij} = \begin{cases} C_{\max} & \text{for} \quad \left(k + \frac{1-\chi_{ij}}{2}\right)T < t \leq \left(k + \frac{1+\chi_{ij}}{2}\right)T \\ 0 & \text{else}\ , \end{cases} \tag{5}$$

where $\chi_{ij} \in [0,1]$ are the duty ratios and T is the fixed modulation period. The pulse-width modulation results in modulated state variables. In the following, an average model is derived from (1). For this, the mean value $\bar{\xi}$ of a variable ξ over a modulation period T is introduced in the form

$$\bar{\xi} = \frac{1}{T}\int_{t-T}^{t} \xi(\tau)\,\mathrm{d}\tau. \tag{6}$$

Because the modulation period T can be chosen sufficiently small, only small variations $\Delta\xi$ of the variables ξ are considered within a modulation period, i.e., $\xi = \bar{\xi} + \mathcal{O}(\Delta\xi)$ with Landau symbol $\mathcal{O}(\cdot)$. Moreover, for small position and pressure variations, the functions $\Gamma_{ij}(p_i)$ according to (3) and the chamber volumes $V_i(s)$ may be written as $\Gamma_{ij}(p_i) = \Gamma_{ij}(\bar{p}_i) + \mathcal{O}(\Delta p_i)$ and $V_i(s) = V_i(\bar{s}) + \mathcal{O}(\Delta s)$. This allows us to infer an average model from (1) in the form

$$\ddot{\bar{s}} = \frac{1}{m}\Big(F_f(\dot{\bar{s}}) + F_p(\bar{p}_1,\bar{p}_2) + F_a\Big) \tag{7a}$$

$$\dot{\bar{p}}_i = \frac{\kappa}{V_i(\bar{s})}\left((-1)^i A_i \dot{\bar{s}} \bar{p}_i + R\theta_g \dot{\bar{m}}_i\right),\ i \in \{1,2\}, \tag{7b}$$