Mechatronics xxx (xxxx) xxx–xxx

Contents lists available at ScienceDirect

Mechatronics

journal homepage: www.elsevier.com/locate/mechatronics

Control and estimation strategies for pneumatic drives with partial position information

Andreas Pfeffer[*,a], Tobias Glück[b], Florian Schausberger[a], Andreas Kugi[a]

[a] Automation and Control Institute, TU Wien, Vienna, Austria
[b] Complex Dynamical Systems, Austrian Institute of Technology, Vienna, Austria

ARTICLE INFO

Keywords:
Pneumatic systems
Disturbance rejection
Parameter estimation
Feedforward control
Trajectory planning

ABSTRACT

Flexibility in production demands for flexible components which automatically adjust to new operating conditions. Pneumatic drives are often used in various industrial applications, e.g., for point-to-point movements. The motion characteristics is typically set up by manual tuning. Therefore, changes in the production lines typically require costly manual readjustments. This can be avoided by using a robust, but expensive servo-control with pressure sensors and a full-stroke position sensor. In this paper, we propose a cost-efficient and flexible alternative by combining classical pressure sensors with cheap partial position sensors at the stroke ends to estimate the parameters of the mechanical system of the drive. This allows to readjust the motion characteristics in real time. Another costly issue is the limitation of the lifetime of the drive, when varying loads lead to large impacts at the stroke ends. To increase the lifetime, a novel non-overshooting trajectory planning algorithm is presented in this work. The overall control concept is implemented on two lab test benches and the experimental results prove its excellent performance and robustness with respect to changing operating conditions.

1. Introduction

In today's manufacturing plants, many tasks are performed with pneumatics, see, e.g., [1]. One of the standard applications are simple endpoint-to-endpoint movements. Pneumatic drives are ideally suited to such tasks because they feature low investment costs and a high power density, see, e.g., [2–4]. In general, two approaches are used to control the movement of pneumatic drives.

In the majority of cases, the pneumatic drives are equipped with simple open-loop switching strategies and throttle valves without any further sensors. The lack of measurement information makes this approach cheap, but quite inflexible in terms of operating conditions. The typical fields of application are endpoint-to-endpoint movements, where mechanical end-stops are employed to limit the movement of the drive. A speed limitation is realized by manual tuning of the throttle valves, while built-in damping elements (for example elastomer's, pneumatic dampers,...) in the pneumatic drive reduce the impact energy at the mechanical end-stops. These damping measures are intended to ensure a so called soft landing of the piston and thus increase the lifetime of the drive. Beside the extra effort of including such dampers in the construction of the drive, some of these technologies require manual adjustments and thus cause additional costs when bringing the system into service. Often these pneumatic drives are oversized and do not operate at high efficiency, see, e.g., [5], which of course entails higher running costs. To sum it up, the main drawbacks of these simple pneumatic drives are the inflexibility due to the lack of sensors and the typically higher operational costs. The initial cost savings may finally result in high follow-up costs because also every change in the environment of the drive demands for new adjustments, e.g., of the throttle valves.

The second state-of-the-art approach is classical servo-control, see, e.g., [6–13], which allows to control the position of the piston along a desired trajectory. Clearly, this requires full-stroke position sensors. An overview of different types of servo-control strategies can be found in [14,15]. Nowadays, often modern non-linear control strategies like exact linearization, see, e.g., [15,16], sliding mode control, see, e.g., [3,7,9,10,17,18], or immersion and invariance concepts [19] are applied. Typically an expensive, e.g., 5-port/3-way, proportional valve is used to control the drive, which leads to a single-input-single-output system for position control, see, e.g., [4,7,18,20,21]. A pure feedback position controller can then be used for the drive. But, even if measurable, the chamber pressures cannot be controlled, since the only control input is used for the position control. This in combination with the typical leakage flows of such proportional valves results in a lower

* Corresponding author.
E-mail addresses: pfeffer@acin.tuwien.ac.at (A. Pfeffer), Tobias.Glueck@ait.ac.at (T. Glück), kugi@acin.tuwien.ac.at (A. Kugi).

http://dx.doi.org/10.1016/j.mechatronics.2017.09.012
Received 18 November 2016; Received in revised form 21 July 2017; Accepted 20 September 2017

efficiency due to high chamber pressures, see, e.g., [2,22]. An attractive alternative is given by two pneumatic half-bridges equipped with two, e.g., 2-port/2-way switching valves each. Beside the cost savings due to the elimination of the leakage flows, this approach allows to separately control the chamber pressure or the actuator force and the piston position, see, e.g., [1,3,23–28]. The combination of a full-stroke position sensor with pressure sensors and two half bridges is the most flexible approach, but also the one with the highest equipment costs. In literature, several works can be found which reduce the equipment costs of servo-controlled drives, e.g., by omitting the chamber pressure sensors, see, e.g., [29]. Alternatively, an observer can be designed to estimate both chamber pressures, see, e.g., [29,30]. Note that for all these approaches still a full-stroke position sensor is needed. Typically this is cost effective for short-stroke drives where cheap position measurement systems are available. For drives with long strokes, the price of a full-stroke position sensor is much higher compared to the costs of two pressure sensors.

Nowadays, the major challenge of pneumatic drives is to reduce the overall running costs, see, e.g., [1,22]. The main cost drivers are air consumption and maintenance. The air consumption can be curbed by avoiding leakage flows and by reducing at least the sum of the chamber pressures, while the maintenance costs can be kept small by a flexible and robust control concept.

Moreover in industrial applications, pneumatic drives are often faced with varying or not exactly known operating conditions. A typical example is the supply pressure. Of course, the supply pressure level at the compressor power station can be chosen within certain limits and usually different service units with mechanical pressure controllers are used to regulate the supply pressure level in a manufacturing plant. But due to the costs of the service units, they typically supply a large number of pneumatic components. This in combination with different pipe diameters and pipe lengths to the valves and to the actuators can lead to varying supply pressure levels and sometimes even to significant pressure drops at the pneumatic components. For standard applications, the varying supply pressure may thus lead to unintended changes in the clock speed of the production line or may result in harmful impacts of the piston on the stroke ends. In [31] a novel robust approach for endpoint-to-endpoint movements is presented, which tackles the systematic handling of such pressure drops and represents the basis for the current work. For this approach, the drive is equipped with leakage free switching valves and a sensor concept comprising pressure sensors and two cheap non-contacting position sensors with short measurement range placed near each stroke end. Compared to a combination of a proportional valve with only a low cost full-stroke position sensor for a classical servo control, the costs of the utilized valves and sensors are about 20 percent smaller. When the servo control also comprises pressure sensors, the cost advantage of the presented approach rises up to more than 30 percent. The presented control approach allows to suppress supply pressure drops even during the movement of the piston and to reduce the overall air consumption compared to the classical approaches. In a nutshell, the movement is realized by using chamber pressure trajectory control. The desired pressure trajectories are planned by utilizing the differential flatness property of the system. This also allows to systematically compute a feedforward controller for the valves in real time, which offers the possibility to account for the measurable supply pressure drops. An additional feedback controller is used to ensure robustness with respect to model uncertainties and disturbances. Apart from the supply pressure also the real moving mass of the drive is not always exactly known. Moreover, it is well known that the friction of the system may exhibit large variations during operation depending on the application. For most model-based control applications, the knowledge of the exact mass is important to achieve a high control performance and accuracy. In literature, different estimation strategies for pneumatic drives can be found, see, e.g., [19,29]. The concept of Rapp et al. [19] is based on an adaptive immersion and invariance approach, see, e.g., [32], where the friction force is estimated online. The limitation is that the mass has to be known exactly for accurate estimations. In the latter approach from Keller and Isermann [29], a least-squares estimation is proposed. The drawback of this approach is that the Coulomb friction parameter must be known in advance for online estimation of the mass and of the viscous friction parameter.

In the following, an extension of Pfeffer et al. [31] to overcome the challenges concerning varying parameters is presented. To estimate the mass and the friction of the system simultaneously an online parameter estimation algorithm is derived. The algorithm presented in this work uses two identical estimators in parallel to combine fast convergence with high robustness with respect to noise. The two estimators are based on recursive least-squares algorithms, see, e.g., [33]. Investigations concerning the influence of the mounting orientation of the drive and of the sensor noise lead to a tailored estimation concept. This parameter estimator offers new possibilities for the control strategy presented in [31]. First, the knowledge of the actual parameters allows to update the flatness-based feedforward controller of the endpoint-to-endpoint motion planning. Second, instead of the robust but not very precise impedance control presented in [31] a combination of feedforward and position control can be used in the region where position information is available at the stroke ends. For this, a tailored non-overshooting trajectory planning algorithm is developed.

The paper is structured as follows: In Section 2, the experimental setup is introduced and its mathematical model is derived. Based on this model, the overall control strategy is presented in Section 3 and the parameter estimation algorithm is given in Section 4. Finally in Section 5, measurements are shown to validate the feasibility of the proposed concepts. The paper closes with some short conclusions.

2. Experimental setup and mathematical model

In the following, the experimental setup is described and the corresponding mathematical model is derived.

2.1. Experimental setup

A schematic of the system under investigation is shown in Fig. 1. A pneumatic drive consisting of a differential cylinder and four fast 2-port/2-way switching valves is considered. Two short position sensors are mounted near each stroke end. For the estimation strategy as well as for the position control concept, three pressure sensors for the supply pressure and the two chamber pressures are required. In addition, a full-stroke position measurement sensor is mounted for validation purposes only.

2.2. Mathematical model

An average model of the pneumatic drive with PWM (pulse-width modulation)-controlled switching valves is presented in [31]. For the reason of completeness, it will be shortly repeated in the following. Let $\bar{\xi}$ denote an average value of the variable ξ over a modulation period T, i.e.,

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

Then the average model of the PWM-controlled pneumatic drive reads as [31]

$$\ddot{\bar{s}} = \frac{1}{m}(F_f(\dot{\bar{s}}) + F_p(\bar{p}_1, \bar{p}_2) + F_g + F_a) \tag{2a}$$

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

Coulomb and viscous friction is modelled by $F_f(\dot{\bar{s}}) = -c\tanh(\dot{\bar{s}}/\varepsilon) - d\dot{\bar{s}}$ with the Coulomb friction coefficient $c > 0$, the parameter $0 < \varepsilon \ll 1$, and the viscous friction coefficient d. Moreover, $F_p(\bar{p}_1, \bar{p}_2) = \bar{p}_1 A_1 - \bar{p}_2 A_2$ denotes the pressure force, with the effective