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Improving dynamic performances of PWM-driven servo-pneumatic systems via a novel pneumatic circuit

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ABSTRACT

In this paper, the effect of pneumatic circuit design on the input–output behavior of PWM-driven servo-pneumatic systems is investigated and their control performances are improved using linear controllers instead of complex and costly nonlinear ones. Generally, servo-pneumatic systems are well known for their nonlinear behavior. However, PWM-driven servo-pneumatic systems have the advantage of flexibility in the design of pneumatic circuits which affects the input–output linearity of the whole system. A simple pneumatic circuit with only one fast switching valve is designed which leads to a quasi-linear input–output relation. The quasi-linear behavior of the proposed circuit is verified both experimentally and by simulations. Closed loop position control experiments are then carried out using linear P- and PD-controllers. Since the output position is noisy and cannot be directly differentiated, a Kalman filter is designed to estimate the velocity of the cylinder. Highly improved tracking performances are obtained using these linear controllers, compared to previous works with nonlinear controllers.

1. Introduction

Pneumatic actuators are widely used in industry since they have many desirable properties. They are reliable, clean, low cost, fast acting and self-cooling actuators and have good power to weight ratios. They are also compliant and can be directly connected to their payload which makes them a natural choice for certain robotic applications with interaction between a human and a robot.

However, the major drawback of pneumatic systems is their nonlinear behavior mainly due to air compressibility, cylinder friction and the nonlinear discontinuous regime of air flow through the valve. These nonlinearities make it difficult to achieve a proper control for pneumatic actuators. Although some of the advanced nonlinear control techniques have been applied to servo-pneumatic systems, these techniques are usually complicated and may cause new problems like chattering in the realization of the control system [1].

Servo-pneumatic systems are usually realized by the continuously acting servo valves. In order to decrease the costs and complexity of the system, fast switching valves driven by pulse width modulated (PWM) inputs, are used instead of servo valves [2–5]. Using on–off switching valves with inherently nonlinear discrete behavior seems to increase the nonlinearity of the system. However, this type of servo-pneumatic system has a so far ignored advantage over those using servo valves; the flexibility in the pneumatic circuit design. Different numbers and types of fast switching valves can be used together to form the pneumatic circuit. The effect of pneumatic circuit on the input–output linearity of the whole system was shown by Taghizadeh et al. [6].

Regarding the control aspects of PWM-driven servo-pneumatic systems, some papers used PID based controllers with empirical tuning [7], but others were mainly concerned with modeling and/or controller design. Some efforts for analytical modeling of pneumatic actuators, valves and friction can be found in [8–13]. Model identification was also studied in some papers like [14] in which a linear digital model was identified and a digital PID controller was designed. Different valve pulsing schemes were also investigated in this reference which will be discussed in the next paragraph. Shih and Hwang [15] and Shih and Ma [16] used a fuzzy PWM method in position control of pneumatic cylinders. Shih and Ma also used the sliding mode control with the modified differential PWM method to control the position of a pneumatic rodless cylinder [17]. A linear averaged model and a linear robust controller based on loop shaping method were introduced in [18]. This paper was followed by a nonlinear average modeling and a sliding mode controller design in [19–21].

Most of the previous papers used a pneumatic circuit comprising two 3–2 fast switching valves each at one side of a pneumatic

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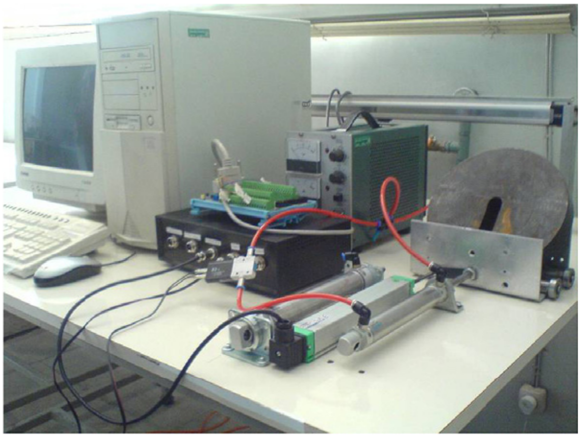

Fig. 1. Photograph of the setup.

cylinder. Other works were implemented by pneumatic circuits comprising four [22,23] or even eight [24] normally closed 2–2 way fast switching valves. A major problem faced by all these circuits is that at least two valves (one at each side) should be driven simultaneously. In fact, the controller output should be shared between the valves. A good strategy for sharing the control signal and allocating an appropriate duty cycle to each valve was introduced by Varseveld and Bone [14]. They used the typical two-valve pneumatic circuit and tried to improve the input–output linearity of the whole system by changing the valve pulsing schemes. They obtained a quasi-linear input–output behavior; however the proposed pulsing scheme was more complicated than the previous ones. Four-valve circuits are even more complicated and costly; however due to using normally closed 2–2 valves, they are more appropriate for positioning the piston in desired points.

In this paper, a very simple and low cost pneumatic circuit is proposed and the effect of circuit on the input–output linearity of the system is investigated. In what follows, in Section 2, the experimental setup is illustrated. In Section 3, the proposed pneumatic circuit comprising only one 3–2 fast switching valve is indicated and the open loop behavior of the system is obtained experimentally. In Section 4, dynamical equations of the proposed system are investigated and the quasi-linear open loop response of the system is verified by simulation results. In Section 5, dead bands in the open loop response, which are caused by the valve delay times, are omitted by applying a simple mapping algorithm. In Section 6, a Kalman filter is designed to estimate and use the velocity of the cylinder instead of direct differentiation of the measured position. In Section 7, closed loop experiments are carried out using linear P- and PD-controllers and the obtained results are compared with previous works, indicating high improvements. Finally, Section 8 provides some concluding remarks.

2. Experimental setup

The actuator under control is a double-acting single-rod cylinder from FESTO (DSNU-25-125-PPV) with 125 mm stroke and 25 mm bore diameter. FESTO fast response 3–2 way switching valve (MHE2-MS1H-3/2G-M7 or MHE4-MS1H-3/2G-1/4K) is used to control the air flow. In order to add a 10 kg mass in some of the tests, a carriage is installed to the end of the piston rod. Experiments are conducted under a pressure of 5 bars. A photograph of the experimental setup is shown in Fig. 1.

A computer is applied to control the switching solenoid valve and to exchange experimental data via an ADVANTECH interface board (PCI 1710 HG). A linear potentiometer from GEFRAN with a resolution of 0.05 mm is used to measure the displacement of piston which is fed back to the computer through the I/O board. A program is designed in SIMULINK and coded by real-time windows target toolbox in which reference values are generated, data of the sensor are read and the controller output signal is calculated. This signal is then transferred through an analog output of the I/O card toward the electronic circuit of Fig. 2. The electronic circuit converts the control output of the computer into amplified PWM pulse to drive the pneumatic valve.

3. Design of the pneumatic circuit

Different pneumatic circuits comprising one, two or four PWM-driven fast switching valves were experimentally investigated by the authors and their open loop behavior between the valve input voltage and the velocity of the cylinder were obtained. Among those, the pneumatic circuit of Fig. 3 showed the best linearity in its input–output behavior. This pneumatic circuit consists of only one 3–2 fast switching valve. The full diameter side of the piston is connected to the valve, and the annulus side is connected to the pressure source via a three-port pressure regulator. In this circuit, when the input voltage is zero, the valve stays in its normal position and the cylinder retracts with the highest speed. A bias voltage is needed to keep the cylinder stopped.

Results of the open loop experiments, between the input voltage and the mean velocity of the cylinder are shown in Fig. 4. This figure indicates a quasi-linear behavior in the region approximately between 1.2 to 3.4 V. The nonlinear dead bands below 1.2 V and above 3.4 V are caused by the valve delay times and the working range of the PWM electronic circuit. These dead bands will be removed by a mapping algorithm in Section 5. From Fig. 4, the bias voltage corresponding to zero velocity is about 2.7 V.

In addition to linearity, the applied pneumatic circuit has several other advantages. Using one fast switching valve decreases the total cost of the system, and removes the problem of allocating different duty cycles to simultaneously driven valves (in pneumatic circuits with two or more valves). It could be also stated that having one PWM-driven valve reduces the induced vibration, occupies one output channel of the I/O card and requires one PWM generating electronic circuit.

4. Modeling and simulation

In this section, the dynamical equations of servo-pneumatic systems are discussed and the effect of pneumatic circuit on these equations is investigated. The dynamics of the proposed system is then simulated to obtain the open loop relation between the duty cycle of the PWM input pulse and the velocity of the cylinder. Simulation results verify the experimental data of Fig. 4, indicating a quasi-linear input–output behavior.

Nonlinear dynamic modeling of servo-pneumatic systems has been investigated in many previous works [10]. These systems are well known for their nonlinear behavior which is mainly caused by the Coulomb friction in the cylinder, compressibility of air, and the nonlinear flow of air through valves. In the state space realization, the equations can be summarized as follows:

$$\begin{cases} \dot{x} = v \\ \dot{v} = \dfrac{1}{M}(A_a P_a - A_b P_b - bv - F_f) \\ \dot{P}_a = -\dfrac{kP_a}{(l/2+x)}v + \dfrac{kRT_s}{A_a(l/2+x)} C_d C_o w X_a f\left(\dfrac{P_a}{P_s} or \dfrac{P_e}{P_a}\right) \\ \dot{P}_b = +\dfrac{kP_b}{(l/2-x)}v + \dfrac{kRT_s}{A_b(l/2-x)} C_d C_o w X_b f\left(\dfrac{P_e}{P_b} or \dfrac{P_b}{P_s}\right) \end{cases} \quad (1)$$

in which the state variables x, v, P_a and P_b represent the position and velocity of the piston, and the pressures in the cylinder