

POSITION CONTROL FOR A RODLESS PNEUMATIC CYLINDER WITH FOUR ON – OFF VALVES

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ABSTRACT

This paper examines the position control ability of a rodless pneumatic cylinder using on – off valves. A pneumatic system including a rodless pneumatic cylinder and 04 on – off valves is firstly proposed. The system is then modeled by dynamic equations with consideration of the valve characteristics and of friction in the rodless pneumatic cylinder. Logic Control Method is applied to control the cylinder position. Effects of the external load and the transient times to the control ability of the Logic Control Method are considered. Simulation results show that the Logic Control Method gives good control performances under different operating conditions.

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1. Introduction

Pneumatic systems are widely applied in many industrial applications because they are cheap, lightweight, clean, easy to assemble and create a good force/weight ratio. However, it is very difficult to achieve high-precision position control using pneumatic cylinders due to the compressive properties of the air, the nonlinearity of the servo valve or proportional valve and nonlinear friction properties existing between the contact surfaces in the pneumatic cylinders (Nguyen Van Lai et al., 2019).

Electro-pneumatic control valves are used as interfaces to electronic controls to allow the infinitely variable electrical remote adjustment of fluid flow of the valve output driving a pneumatic actuator. There are two types of electro-pneumatic valves used in the realizing control of fluid flow of the pneumatic actuator. These are

servovalves and on-off valves (Elif Erzan Topcu et al., 2006).

The servovalves are used to achieve high linear control accuracy in pneumatic actuators, but they have complex structures and they are very costly. On the other hand, owing to low cost, high flow rate gain, small size and simple structure, the fast-switching on-off valves that are used to control the pneumatic actuators have received considerable attention (Robert B. V. et al., 1997, Angelo Gentile et al., 2002). They are also known (Belforte G. et al., 2004), (Massimo Sorli and Stefano Pastorelli, 2000) as digital valves. The on-off switching valves are inherently nonlinear discrete electro-pneumatic converters. To obtain similar linear characteristics of the servovalves with simple on-off switching valves, pulse width modulation technique is used. To reach a large modulated linear area, the valve switching times

must be very rapid during the short cycle time. The limits of the modulation depend on the valve switching times and the modulation frequency. Solving this problem allows pneumatics to conquer new areas applications, successfully competing with hydraulics and electromechanics. So the solution is, in particular, pneumatic positioning systems rodless pneumatic cylinder using discrete valves (Fig. 1). It consists of the following parts:

- Rodless pneumatic cylinder (made from special profiled pipe with an opening through the entire length of the sleeve. Through this opening the piston is directly connected to the outer carriage);
- Four on-off valves, of which two valves regulate the flow air in the cavity of the pneumatic cylinder, and the other two discharge air into atmosphere. All valves have the same nominal flow;
- A potentiometer which is used as feedback, with which measures the displacement of the pneumatic cylinder piston.

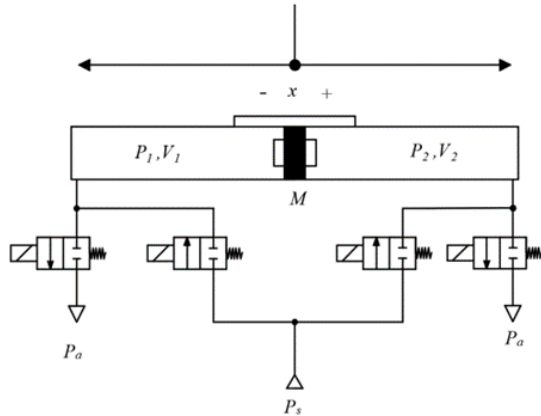


Figure 1. Pneumatic diagram of the position control system

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2. Mathematical model of the pneumatic system

This section develops mathematical equations of the system. In order to obtain the air flow dynamics in a cylinder, the following assumptions are used (Nguyen Van Lai et al., 2019):

- The used air is an ideal gas and its kinetic energy is negligible in the chamber.
- The leakages of the cylinder are negligible.
- The temperature variation in cylinder chambers is negligible with respect to the supply temperature.
- The pressure and the temperature in the cylinder chamber are homogeneous.
- The evolution of the gas in each chamber is polytropic.
- The supply and exhaust pressures are constant.

The equation of motion of the piston of a rodless pneumatic cylinder has the form:

$$M\ddot{x} + \beta\dot{x} + F_{fr} = (p_1 - p_2)S \quad (1)$$

In which: M – the total mass of the piston and payload, (kg); x – the displacement of the piston (m); $\dot{x}$ – the position derivatives of a time/velocity of piston, (m/s); $\ddot{x}$ – the acceleration of piston, (m/s²); β – the coefficient of viscous friction; p_1 and p_2 – the pressure in the chambers of pneumatic cylinder, (Pa); S – the cross-sectional area of the piston (m²); F_{fr} – the friction force (N).

For a gas volume V with a density ρ at a temperature T and pressure p , the equation of state for an ideal gas is:

$$p = \rho RT \quad (2)$$

In which: p – the absolute pressure of gas, (Pa); ρ – the density of gas at particular condition, (kg/m³); T – the temperature of gas, (K); R – the universal gas constant (J/(kg.K)).

Using the continuity equation, one can find an expression for the mass air flow:

$$\dot{m}_{in} - \dot{m}_{out} = \dot{\rho}V + \rho\dot{V} \quad (3)$$

In which: $\dot{m}_{in}$, $\dot{m}_{out}$ – the mass flows of air entering and exiting the chamber, respectively, (kg/s); $\dot{\rho}$ – the density derivatives of a time, (kg/(m³.s)); ρ – the density of gas at particular condition, (kg/m³); V – the volume of the control volume, (m³); $\dot{V}$ – the volume derivatives of a time, (m³/s).

The law of conservation of energy can be written in the following form:

$$q_{in} - q_{out} = \dot{u} - \gamma C_v (\dot{m}_{in} T_{in} - \dot{m}_{out} T_{out}) + \dot{W} \quad (4)$$

In which: q_{in} and q_{out} – the amount of heat supplied from the environment and released into the environment, (J); C_v – the specific heat

capacity of the gas at constant volume, (J/(kg.k)); γ – the adiabatic index; $W = pV$ is work on account gas expansion (J); u – the change in internal energy (J); T_{in} and T_{out} – the temperature of air entering and exiting the chamber, respectively, (K); $\dot{m}_{in}$, $\dot{m}_{out}$ – the mass flows of air entering and exiting the chamber, respectively, (kg/s).

On the other side, the change in internal energy is equal to the form:

$$\dot{u} = \frac{d}{dt}(C_v m T) = \frac{1}{\gamma - 1} (V \dot{p} + p \dot{V}) \quad (5)$$

In which: u – the change in internal energy derivatives of a time, (J/s); C_v – the specific heat capacity of the gas at constant volume, (J/(kg.k)); γ – the adiabatic index; m – the mass of the air, (kg); T – the temperature of gas, (K); V – the volume of the control volume, (m³); $\dot{V}$ – the volume derivatives of a time, (m³/s).

The specific heat capacity of the gas at constant volume:

$$C_v = \frac{R}{\gamma - 1} \quad (6)$$

In which: C_v – the specific heat capacity of the gas at constant volume, (J/(kg.k)); R – the universal gas constant (J/(kg.K)); γ – the adiabatic index.

Joint consideration of equations (4) and (5) and taking into account the nature of the thermodynamic process we obtain:

$$\dot{p} = \frac{RT}{V} (\alpha_{in} \dot{m}_{in} - \alpha_{out} \dot{m}_{out}) - \alpha \frac{p \dot{V}}{V} \quad (7)$$

In which: α_{in} and α_{out} take a value from 1 to $\gamma = 1.4$ depending on the nature thermodynamic processes. Since the process of filling the cavity air is close to adiabatic, and the process of air outflow into the atmosphere is to isothermal, then α_{in} is taken equal to γ , and α_{out} equal to 1. Process gas compression/expansion due to piston movement is best described when $\alpha = 1, 2$. If we assume that the zero carriage offset corresponds to the middle pneumatic cylinder, then the volumes of the cavities can be represented as:

$$V_i = V_{0i} + S(0.5L \pm x) \quad (8)$$

In which: $i = 1, 2$ - indices of the left and right chambers; V_{0i} - dead volumes at the ends pneumatic cylinder, (m³); L - the length of the pneumatic cylinder, (m); x - piston position/ the

displacement of the piston (m); S – the cross-sectional area of the piston (m²).

Eventually expression (7) will be written in the form:

$$\dot{p} = \frac{RT}{V_{0i} + S(0.5L \pm x)} (\gamma \dot{m}_{in} - \dot{m}_{out}) - \alpha \frac{pS}{V_{0i} + S(0.5L \pm x)} \dot{x} \quad (9)$$

In which: $\dot{p}$ – the pressure derivatives of a time, (Pa/s); p – the pressure of the gas, (Pa); T – the temperature of gas, (K); R – the universal gas constant (J/(kg.K)); $\dot{m}_{in}$, $\dot{m}_{out}$ – the mass flows of air entering and exiting the chamber, respectively, (kg/s); γ – the adiabatic index; V_{0i} - dead volumes at the ends pneumatic cylinder, (m³); L - the length of the pneumatic cylinder, (m); x - piston position/ the displacement of the piston (m) $\dot{x}$ – the velocity of piston, (m/s); S – the cross-sectional area of the piston (m²).

Next, consider the expression for determining the mass air flow through a valve with a cross section A_v . Since the pressure difference across valves is quite large, the flow must be considered as compressible and turbulent. If the pressure ratio before and after the valve is greater critical, then the mass flow depends linearly on the pressure up to valves. Otherwise, the dependence is non-linear. Expression for mass air flow has the following form:

$$\begin{cases} \dot{m}_v = C_f A_v C_1 \frac{p_u}{\sqrt{T}} \\ \text{if } \frac{p_d}{p_u} \leq \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}} \\ \dot{m}_v = C_f A_v C_1 \frac{p_u}{\sqrt{T}} \left(\frac{p_d}{p_u} \right)^{1/\gamma} \sqrt{1 - \left(\frac{p_d}{p_u} \right)^{(\gamma - 1)/\gamma}} \\ \text{if } \frac{p_d}{p_u} > \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}} \end{cases} \quad (10)$$

Where C_f is the dimensionless flow coefficient; p_u - pressure to the hole; p_d - pressure after the valve. Constant values C_1 and C_2 determined by expressions:

$$C_1 = \sqrt{\frac{\gamma}{R} \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma+1}{\gamma-1}}} \quad (11)$$

$$C_2 = \sqrt{\frac{2\gamma}{R(\gamma-1)}}$$

Considering that for the process of filling the chamber $p_u = p_s$ và $p_d = p_i$ and for air release process $p_u = p_i$ và $p_d = p_a$, where p_s – pressure supply, p_a – atmospheric pressure, p_i – pressure in the cavities of the pneumatic cylinder, we can rewrite equation (9) in the form:

$$\begin{cases} \dot{m}_{in} = C_f A_{in} \frac{p_s}{\sqrt{T}} \dot{m}_r(p_s, p_i) \\ \dot{m}_{out} = C_f A_{out} \frac{p_i}{\sqrt{T}} \dot{m}_r(p_i, p_a) \end{cases} \quad (12)$$

Where:

$$\begin{cases} \dot{m}_r(p_u, p_d) = C_1 & \text{if } \frac{p_d}{p_u} \leq \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma}{\gamma-1}} \\ \dot{m}_r = C_2 \left(\frac{p_d}{p_u} \right)^{1/\gamma} \sqrt{1 - \left(\frac{p_d}{p_u} \right)^{(\gamma-1)/\gamma}} & \\ \text{if } \frac{p_d}{p_u} > \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma}{\gamma-1}} & \end{cases} \quad (13)$$

Substituting these expressions into equation (9), we finally get:

$$\begin{aligned} \dot{p}_1 &= \frac{C_f R \sqrt{T}}{V_{01} + S(0.5L + x)} [A - B] \\ -\alpha \frac{p_1 S}{V_{01} + S(0.5L + x)} \dot{x} & \end{aligned} \quad (14)$$

$$A = \gamma p_s A_{in} u_{1in} \dot{m}_r(p_s, p_1)$$

$$B = p_1 A_{out} u_{1out} \dot{m}_r(p_1, p_a)$$

$$\begin{aligned} \dot{p}_2 &= \frac{C_f R \sqrt{T}}{V_{02} + S(0.5L - x)} [C - D] \\ -\alpha \frac{p_2 S}{V_{02} + S(0.5L - x)} \dot{x} & \end{aligned} \quad (15)$$

$$C = \gamma p_s A_{in} u_{2in} \dot{m}_r(p_s, p_2)$$

$$D = p_2 A_{out} u_{2out} \dot{m}_r(p_2, p_a)$$

In equations (14) and (15), four quantities were introduced $u_{1,2(in, out)}$ which are control actions and can take values equal to 0 (corresponding to a closed valve) or 1 (corresponding to an open valve). When considering a system, valve dynamics are not taken into account.

The considered equations completely describe the behavior of the system position control of a rodless pneumatic cylinder.

3. Controlling algorithm

In the considered position control system based on the rodless pneumatic cylinder using control valves is required to provide speed about 1m/s and position accuracy $\varepsilon = 0.1\text{mm}$. Were considered two possible control logics: "passive" and "active", allowing implement sliding mode and provide the required speed and position control accuracy.

When equating $\sigma(x - x_0) = 0$ will take place the following passive control logic algorithm:

If $\sigma(x - x_0) > 0$ and $x - x_0 < -\varepsilon$, then $u_{1in} = u_{2out} = 1$ and $u_{2in} = u_{1out} = 0$. Thus, air enters the left chamber and is discharged from the right;

If $\sigma(x - x_0) > 0$ and $x - x_0 > +\varepsilon$, then $u_{1in} = u_{2out} = 0$ and $u_{2in} = u_{1out} = 1$. Thus, air enters the right chamber and is discharged from the left;

In all other cases $u_{1in} = u_{1out} = u_{2in} = u_{2out} = 1$.

The active control logic algorithm is as follows:

If $\sigma < -\varepsilon$, then $u_{1in} = u_{2out} = 1$ and $u_{2in} = u_{1out} = 0$. Thus, air enters the left chamber and is discharged from the right;

If $\sigma > +\varepsilon$, then $u_{1in} = u_{2out} = 0$ and $u_{2in} = u_{1out} = 1$. Thus, air enters the right chamber and is discharged from the left;

In all other cases $u_{1in} = u_{1out} = u_{2in} = u_{2out} = 1$.

Table 1: System parameters

Parameter (unit)	Value	Parameter (unit)	Value
M , (kg)	4; 8; 12	S , (m ²)	4.9×10^{-4}
p_a , (Pa)	1×10^5	A_v , (m ²)	7.1×10^{-5}
R , (J/(kgK))	287	L , (m)	0.3
T , (K)	295	V_{01} , (m ³)	4.9×10^{-7}
γ	1.4	V_{02} , (m ³)	4.9×10^{-7}

4. Results and discussion

To review the results of mathematical modeling of the pneumatic system, transient graphs were built and phase portraits of the position control process for various masses.

On Fig.2 shows transient graphs and phase portraits results of mathematical modeling of position control from position $x = 0$ m to position $x = 0.25$ m for three different masses. Controlling the system carried out with a resolution of 2 ms.

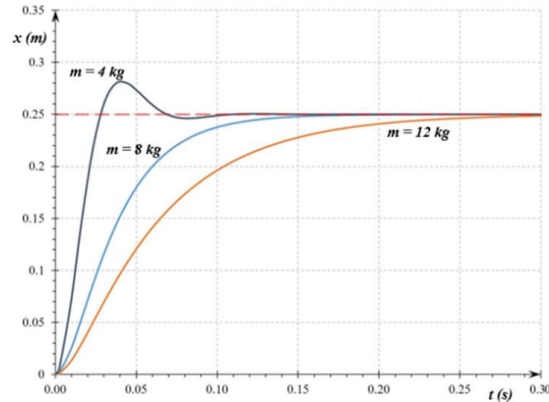


Figure 2. Position transients for different masses according to "passive" algorithm

As can be seen from the graphs of transient processes, despite the spread in mass, for all options, the position control error did not exceed 0.1 mm. In this case, the use of "passive" logic is more beneficial, since for "passive" logic is characterized by a shorter positioning time than with use of "active" logic. From Fig.2 and Fig.3 shows that the best result of all masses obtained at a nominal mass of 8 kg. Position control time at this mass was 0.15 s.

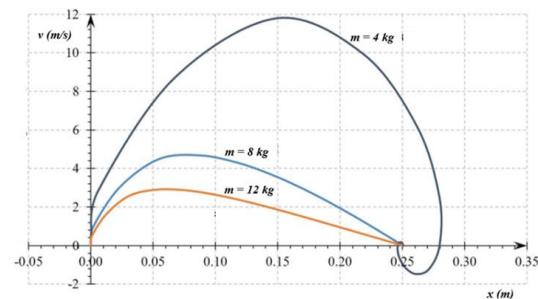


Figure 3. Phase portraits of the position control process for various masses according to the "passive" algorithm

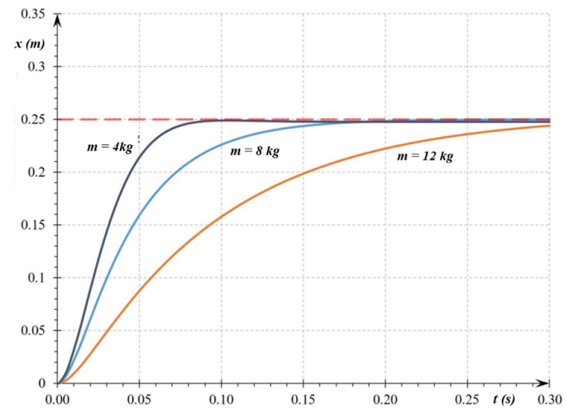


Figure 4. Position control transients for different masses according to "active" algorithm

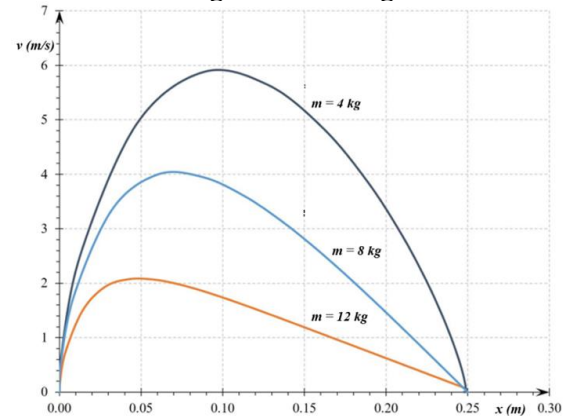


Figure 5. Phase portraits of the position control process for various masses according to the "active" algorithm

A similar indicator for "active" logic at the same nominal weight, as can be seen from Fig. 4 and 5 is 0.15 s. In addition, the application "passive" logic is more preferable than "active" logic in terms of valve actuation. So, with the same nominal weight of 8 kg each pneumatic system valve switched 46 times with "passive" logic and 175 times with "active" logic.

5. Conclusion

Highlight the main contributory outcomes of the paper. Propose future research or suggest the application of research results.

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Contribution of authors

The manuscript is written by Son Tung Nguyen. The modeling and simulating is carried out by Lai Nguyen Van. Both of us have tried to collect the precious references. Kim Binh Doan has commented and edited the original manuscript.

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