Precise Control of the Piston Position and the Pressure of Cylinder Chambers for the Pneumatically Actuated Leg

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Abstract: - In the pneumatically actuated leg control system, precise control of the piston position and the pressure of cylinder chambers is a tricky problem. In this paper, a new control scheme is introduced, in which the gas internal energy of the cylinder's two chambers is used as the set point for PD controller. The proposed control scheme is essential to control the piston position and the chamber pressure simultaneously. A pneumatic position servo model is developed by Simulink tools in the form of an S-function. The maximum steady-state error of 0.19 mm for the piston position and 0.001 MPa for the chamber pressure have been verified in simulations. The test rig of the vertical hopping is constructed, and the experimental results of the pneumatic position servo control on the pneumatically actuated leg show that the chamber pressure error is less than 0.002 MPa, the piston positioning error is less than 0.31 mm and there is no overshooting. The experimental results also show that the established mathematical model and the simulation results are correct. Consequently, the practical experiments of vertical hopping are successful, which validates the proposed method further.

Key-Words: - piston position, the chamber pressure, the gas internal energy, the pneumatically actuated leg, vertical hopping

1 Introduction

As an important driving element, the pneumatically actuated leg has been widely used in the hopping robots due to its good buffering effects [1-2]. Pneumatically actuated leg was made springy by trapping air in the upper chamber of the leg cylinder, with both solenoid valves closed. Delivery of pressurized air to the top chamber of the cylinder drives the piston and rod assembly downward, providing a vertical thrust for hopping [1]. Motivated by the need of motion control of a pneumatic hopping robot using on/off solenoid valves, Jian Wu et al studied the pneumatic motion control problem and recast the problem as an energy regulation problem [3]. Kuldip Gopal Naik et al proposed a method that by adjusting the pressure inside the pneumatic cylinder to achieve a constant jumping height in shortest possible time for the hopping robot [4]. John Leavitt et al developed a feedback linearizing inner torque control system for the pneumatic actuator. The pneumatic actuator is capable of delivering high power levels [5].

In order to achieve the satisfied hopping performance for the pneumatically actuated leg, the piston position as well as the pressure of upper and lower chamber have to be controlled precisely [1,6].

In recent years, for improving the control performance of the pneumatic servo system, some control strategies based on new theories and methods have been studied for the purpose of minimizing the influence of air compressibility and friction [7]. ZHU Jianmin et al proposed a new method of grey relational compensation control [8]. By using the proposed method, the average error of piston position is less than 0.1mm. As to deal with the robust control problems against parameter uncertainty, noise and load disturbance, WANG Zuwen et al employed quantitative feedback theory, the error of piston position is less than 0.05mm [9]. To enhance the stability, reliability and working performance of the pneumatic servo system, Lee L W et al incorporated the H∞ tracking technique into the conventional adaptive sliding-mode control method [10]. Wanayuth Sanngoen et al developed a control algorithm based on precompensator nuero-fuzzy for a typical pneumatic servo system [11]. Its experimental results illustrated that the precompensated intelligent framework can improve the overall the performance. Navneet Gulati et al presented a Lyapunov-based observer for a pneumatic servo system, which can control the system accurately [12]. However, for most of the mentioned mechanisms, the control objective is piston position, and the chambers' pressure is not considered. In this research, we design a combination control mechanisms by considering the piston position and the chambers' pressure. A new PD controller is studied for pneumatic servo systems by using the gas internal energy of two chambers as the set point of the PD controller. It can then be so designed that both the control accuracy for piston position and chambers' pressure can be guaranteed. Compared with existing pneumatic servo systems, although the output steady-state error cannot be the minimum, it is able to control the piston position and chambers' pressure simultaneously, and the control accuracy can meet the requirement of the vertical hopping for the pneumatically actuated leg.

This paper is organized as following. In Section 2 we developed the model of the system, which including not only the model of the pneumatic servo systems, but also the model of the proposed control scheme. The simulation and practical application of experimental results are shown in Section 3 and 4. Finally, in Section 5 we draw some conclusions of this work.

2 Model of the System

In this section, a mathematical model is proposed for the pneumatically actuated leg. For further analysis, some assumptions are made as the following:

(a) The charge-and-exhaust of the cylinder is in quasi-static process, and follows the ideal gas law.

(b) The leakages between the system and the outside and between the drive cavity and back-pressure chamber are negligible.

(c) There is no heat exchange between the system and the outside.

(d) The thermodynamic process is an adiabatic process.

2.1 Model of Pneumatically Actuated Leg

First, the pneumatically actuated leg is shown in Fig. 1, which shows the configuration of the pneumatically actuated leg.

According to the Newton's second law, the dynamics of the piston rod relative to the cylinder can be described by the following equation.

$$\frac{\mathrm{d}^2 z}{\mathrm{d}t^2} = \begin{cases} \frac{1}{m}\left(p_2 A - p_1 A - F_\mathrm{f} - mg\right) & (0 < z < s) \cup \left(z = 0 \cap \frac{\mathrm{d}^2 z}{\mathrm{d}t^2} \geq 0\right) \cup \left(z = s \cap \frac{\mathrm{d}^2 z}{\mathrm{d}t^2} \leq 0\right) \\ 0 & \left(z = 0 \cap \frac{\mathrm{d}^2 z}{\mathrm{d}t^2} < 0\right) \cup \left(z = s \cap \frac{\mathrm{d}^2 z}{\mathrm{d}t^2} > 0\right) \end{cases} \tag{1}$$

where s represents the stroke, m is the mass of the piston rod, and z means the position of piston, p_1 and p_2 are represent the relative pressure of the upper and lower chamber respectively.

For simplicity in the analysis, we assume that friction is composed by static friction, coulomb friction and viscous friction, where, both static friction and coulomb friction are constant. The viscous friction is supposed as a linear function of the velocity of the piston, and so the friction can be calculated by the Eq. (2).

$$F_f = \begin{cases} F_s & v = 0 \\ F_c + B_v v & v > 0 \\ -F_c + B_v v & v < 0 \end{cases} \tag{2}$$

where F_s represent the static friction force, F_c is the coulomb friction force and B_v means the viscous frictional coefficient.

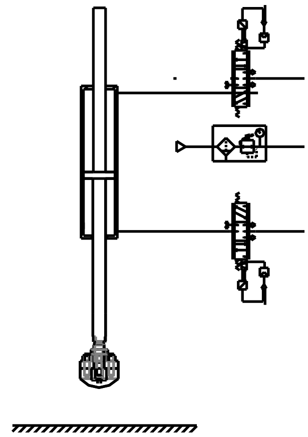


Fig. 1. Schematic diagram of the pneumatically actuated leg

A gas pressure equation of upper chamber of cylinder is derived as

$$\frac{\mathrm{d}p_{1\mathrm{t}}}{\mathrm{d}t}=\begin{cases}\dfrac{kRT_{\mathrm{s}}q_{\mathrm{m1in}}}{A\left(s-z+z_{10}\right)}-\dfrac{kp_{1\mathrm{t}}}{z+z_{10}}\dfrac{\mathrm{d}z}{\mathrm{d}t} & A_{\mathrm{e1}}\ge 0\\ \dfrac{kp_{1\mathrm{t}}}{s-z+z_{10}}\dfrac{\mathrm{d}z}{\mathrm{d}t}-\dfrac{kRT_1}{A\left(s-z+z_{10}\right)}q_{\mathrm{m1out}}\left(\dfrac{p_{1\mathrm{t}}}{p_{\mathrm{st}}}\right)^{\frac{k-1}{k}} & A_{\mathrm{e1}}<0\end{cases} \tag{3}$$

where k, R, q_{m1in}, q_{m1out}, A, z_{10}, z_{20}, $p_{1\mathrm{t}}$ and $p_{2\mathrm{t}}$ represent the specific heat constant, the universal gas constant, the cylinder air temperature, the input mass flow of the proportional valve, the output mass flow of the proportional valve, the effective area of piston, the converted piston displacement of the upper closed dead volume, the converted piston displacement of the lower closed dead volume, the absolute pressure of the upper and lower chamber respectively.

The lower chamber is similar, which do not repeat in this paper.

In the case the air flow through the valve is considered as a sonic one, mass flow q_{m} is given as

$$q_{\mathrm{m}}=C_{\mathrm{d}}\left|A_{ei}\right|p_{\mathrm{u}}\frac{\Phi}{\sqrt{T_{\mathrm{u}}}}$$

$$\Phi=\begin{cases}\sqrt{\dfrac{k}{R}\left(\dfrac{2}{k+1}\right)^{\frac{k+1}{k-1}}} & \dfrac{p_{\mathrm{d}}}{p_{\mathrm{u}}}\le b\\ \sqrt{\dfrac{2k}{R\left(k-1\right)}\left[\left(\dfrac{p_{\mathrm{d}}}{p_{\mathrm{u}}}\right)^{\frac{2}{k}}-\left(\dfrac{p_{\mathrm{d}}}{p_{\mathrm{u}}}\right)^{\frac{k+1}{k}}\right]} & b<\dfrac{p_{\mathrm{d}}}{p_{\mathrm{u}}}\le 1\end{cases} \tag{4}$$

where b, p_{u}, p_{d}, T_{u} and C_{d} represent the critical pressure ratio, absolute pressure of up-stream, absolute pressure of down-stream, absolute temperature of the upstream air, and the discharge coefficient respectively.

Absolute temperature of the two chambers' gas can be calculated by the following formula.

$$T_i=T_s\left(\frac{p_{it}}{p_{st}}\right)^{\frac{k-1}{k}} \tag{5}$$

Kinetic equation of the proportional valve can be expressed approximately by using the first-order differential equations.

$$\frac{\mathrm{d}x_{\mathrm{v}i}}{\mathrm{d}t}=\frac{K_{\mathrm{v}}\left(u_i-u_0\right)-x_{\mathrm{v}i}}{\tau_{\mathrm{v}}} \tag{6}$$

where τ_{v}, K_{v}, u_i and u_0 represent the first-order time constant, the spool displacement increment of unit voltage, the i-th (i=1,2.) control voltage signal of the proportional valve and the reference voltage signal respectively.

Under the assumption that the rated voltage range of the proportional valve is 0~10 V, the control signal can be modeled as

$$u_i=\begin{cases}10 & u_i\ge 10\\ u_i & 0<u_i<10\\ 0 & u_i\le 0\end{cases} \tag{7}$$

The intake and exhaust valve port area of the proportional valve is proportional to the spool displacement, which can be expressed as

$$A_{\mathrm{e}i}=K_{\mathrm{va}}x_{\mathrm{v}i} \tag{8}$$

where $A_{\mathrm{e}i}$ represents the i-th (i=1,2.) servo valve orifice equivalent area, and K_{va} represents equivalent area of unit orifice spool displacement.

The parameters of the model above are obtained by actual measurements of the test rig and consulting product manual: m=1.277 Kg, u_0=5 V, A=1649.34 mm^2, s=0.25 m, F_{c}=10 N, B_v=100 N s/m, z_{10}=z_{20}=0.004m, K_v=0.8mm/V, T_{s}=293.15K, p_{s}=0.52 MPa, R=287.1 J/kg K, k=1.4, b=0.23, τ_{v}=4.8×10-3 s, ω=12.5664 mm^2/mm.

2.2 Equations of Control Algorithm

This paper proposes an algorithm to control the air energy of chambers to control the piston position and the pressure simultaneously. Assuming that the thermodynamic process is quasi-static process, and thus gas internal energy of the chambers can be expressed as below [13].

$$\begin{cases}E_1=\dfrac{p_1V_1}{k-1}=\dfrac{p_1A(s-z+z_{10})}{k-1}\\ E_2=\dfrac{p_2V_2}{k-1}=\dfrac{p_2A(z+z_{20})}{k-1}\end{cases} \tag{9}$$

where E_1, E_2, V_1 and V_2 represent internal energy of the upper chamber's gas, internal energy of the lower chamber's gas, volume of the upper chamber and volume of the lower chamber respectively.

And then the desired control values of the gas internal energy in chambers can be expressed as

$$\begin{cases} E_{1\mathrm{d}} = \dfrac{p_{1\mathrm{d}} V_{1\mathrm{d}}}{k-1} = \dfrac{p_{1\mathrm{d}} A(s - z_{\mathrm{d}} + z_{10})}{k-1} \\ E_{2\mathrm{d}} = \dfrac{p_{2\mathrm{d}} V_2}{k-1} = \dfrac{p_{2\mathrm{d}} A(z_{\mathrm{d}} + z_{20})}{k-1} \end{cases} \tag{10}$$

where $E_{1\mathrm{d}}$, $E_{2\mathrm{d}}$, $p_{1\mathrm{d}}$, $p_{2\mathrm{d}}$ and z_{d} represent desired value of the gas internal energy of upper chamber, desired value of the gas internal energy of lower chamber, desired pressure of upper chamber, desired pressure of lower chamber and desired piston position respectively.

The system error equations can be defined as

$$\Delta E_1 = E_{1\mathrm{d}} - E_1 \qquad \Delta E_2 = E_{2\mathrm{d}} - E_2 \tag{11}$$

Based on Eq. (10) and Eq. (11), the control law is obtained as

$$u_1 = PD_1 = K_{\mathrm{p}} \Delta E_1 + K_{\mathrm{d}} \frac{\mathrm{d}\Delta E_1}{\mathrm{d}t} \tag{12}$$

$$u_2 = PD_2 = K_{\mathrm{p}} \Delta E_2 + K_{\mathrm{d}} \frac{\mathrm{d}\Delta E_2}{\mathrm{d}t} \tag{13}$$

where K_{p} and K_{d} are the proportional and the differential gains of the PD controller.

In order to reduce the impact of the friction on the positioning control of the piston, the proposed control law needs to be modified. We hope that the velocity direction never changed during the control progress. Thereby, the set point of the upper or lower chamber's gas internal energy has to be revised according to the direction of piston velocity. Taking the effects of friction and gravity into account, according to Eq. (1), when the piston moves upward, infinitely close to the desired position, and tends to be the equilibrium state. The following equation is established.

$$\Delta p = p_2 - p_1 = \sum F \big/ A = \frac{mg + F_{\mathrm{c}}}{A} \tag{14}$$

Conversely, when the piston moves downward, infinitely close to the desired position, and tends to be the equilibrium state, the following equation is given as:

$$\Delta p = p_2 - p_1 = \sum F \big/ A = \frac{mg - F_{\mathrm{c}}}{A} \tag{15}$$

where $\sum F$ means the composition of the gravity and friction and Δp represents the difference between the two chambers' pressure.

Based on the Eq. (9), the offset value δ_{E} can be calculated by the following equation.

$$\delta_{\mathrm{E}} = \frac{\Delta p \times V}{k-1} \tag{16}$$

where V represents the set point of the compensated chamber's volume.

By using the compensated value as the set point of the PD controller, the influence of friction on the piston positioning can be reduced. However, in practical situations, friction is difficult to express analytically, the compensation of the gas internal energy need fine tuning in the experiment.

The selection of K_{p} and K_{d} has to satisfy the following conditions: (a) an occurrence of an overshoot during the movement of the piston should be avoid. (b) Both the accuracy and rapidity of the system have to be ensured by selecting appropriate PD parameters.

3 Simulations

To evaluate the performance of the proposed control scheme, the following simulations are performed.

Firstly, we carry out a simulation to examine the effectiveness of the proposed control scheme at ideal situation. Then, the influences of the friction and supply pressure fluctuations on the control accuracy are analyzed. Finally, a simulation for practical situation is studied.

The initial parameters are determined as following: the upper and lower chamber pressures are 0.23 MPa and 0.24366 MPa respectively, and the piston position is 0.03 m. The desired parameters are chosen as following: the upper and lower chamber pressures are 0.3 MPa and 0.31366 MPa respectively, and the piston position is 0.09 m.

3.1 Simulation for the ideal system

In the following, the simulation will be done under the ideal condition, without any fluctuations of friction and supply pressure.

In this simulation, we use the compensated gas internal energy as the set point for the PD controller. According to the restrict conditions for choosing K_{p} and K_{d} proposed in section 2.2, the control parameters of PD controller are set up as K_{p}=0.0458 and K_{d}=0.0001.

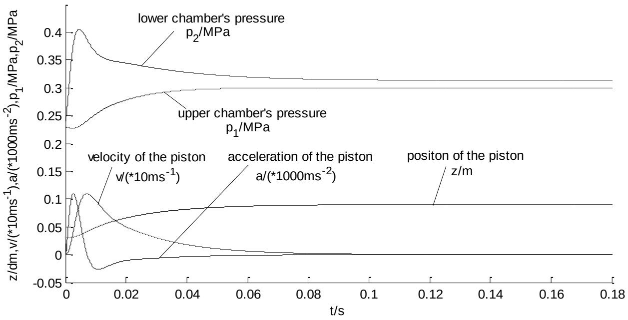


Fig. 2. Simulation curves of dynamic parameters of ideal system

Simulation results are shown in Fig. 2. In order to reflect the pattern of the curves more clearly, the speed data and the acceleration data are reduced by 10 times and 1000 times overall respectively, and the acceleration data are reduced by 1000 times overall. Fig. 2 illustrates that the steady error of the piston position and the pressure of the chambers are zero respectively. It is clear that the proposed control is effective for the above-mentioned plant.

3.2 The influence of friction on the control accuracy

However, in a practical system, the nonlinear friction factor of the relative movement between the coupling member and the cylinder seal packing is difficult to express analytically. Therefore, it's impossible to accurately predict the friction. In this paper, the friction fluctuation is supposed to be within 20%.

Fig. 3 and 4 show the effect of friction fluctuation on the piston position and chambers' pressure. Assumingly, the error is positive when there is overshooting, and on the contrary, the error is negative.

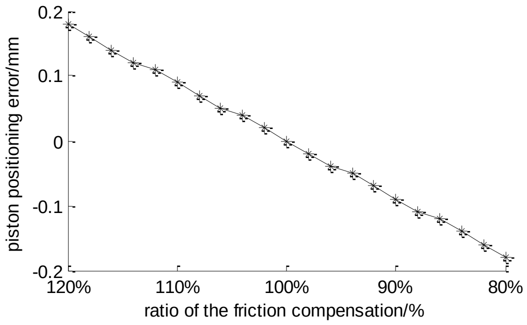


Fig. 3. Simulation curves of piston positioning error to friction

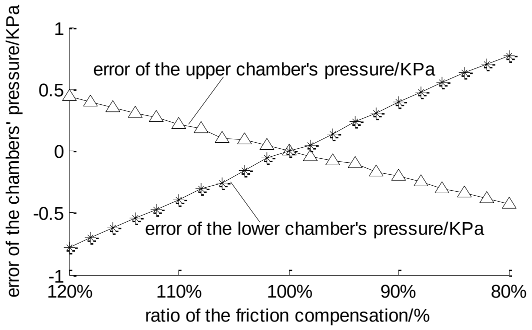


Fig. 4. Simulation curves of chamber pressure error to friction

It can be seen that piston positioning and the chambers' pressure are affected by the friction. Following these results, we can find that ±20% fluctuation of the friction can cause maximum positioning error of 0.18 mm; the steady error of the upper and lower chamber pressure is less than 0.001 MPa.

3.3 The influence of the supply pressure fluctuation on the control accuracy

In practical system, the intermittent work of the air compressor could easily lead to supply pressure fluctuations. In our simulation, the supply pressure fluctuation is set to be within ±20%. Figs. 5 and 6 show the errors of piston position and chambers' pressure.

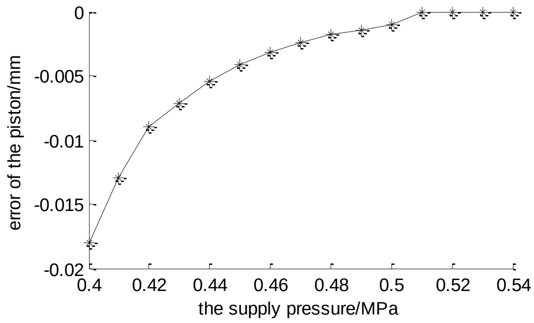


Fig. 5. Simulation curves of piston positioning error to the supply pressure fluctuation

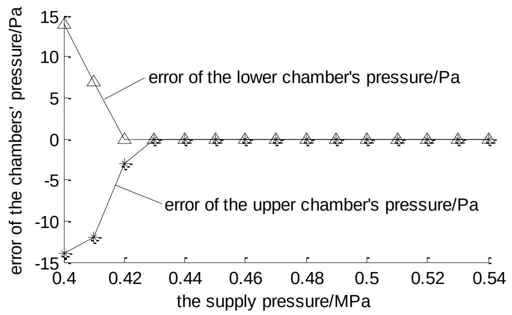


Fig. 6. Simulation curves of chamber pressure error to the supply pressure fluctuation

As can be seen from Fig. 5, when the supply pressure is set to 0.52 MPa and the supply pressure fluctuation is within ±20%, the piston positioning error is less than 0.02 mm. If the supply pressure is higher than 0.51 MPa, then the error is so small that it can be ignored. Fig. 6 shows the influence of the supply pressure fluctuation on the chambers' pressure. From Fig. 6, we can see that the error is not obvious, and while the supply pressure is higher than 0.43 MPa, the influence of the supply pressure is negligible

3.4 Simulation for the practical system

In the practical system, the affection of the friction and supply pressure fluctuations cannot be ignored. For simplicity in the analysis, we assume that both the fluctuation of the friction and the supply pressure are within ±20%. Fig. 7 shows the curves of dynamic parameters in practical system.

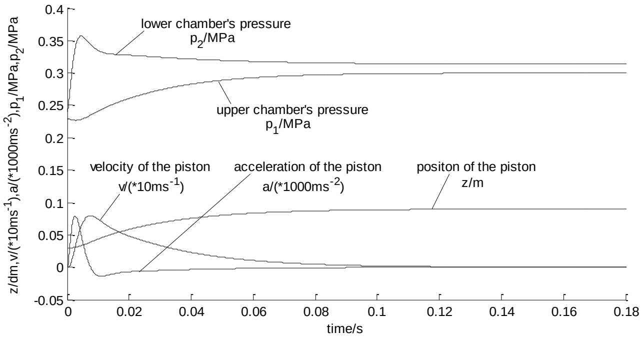
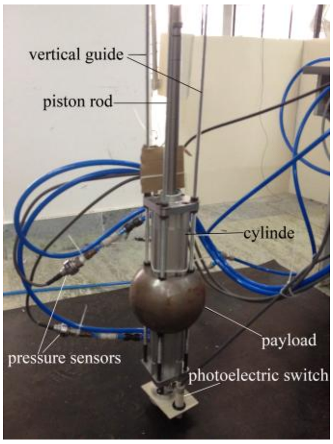


Fig. 7. Simulation curves of dynamic parameters affected by the supply pressure and the friction fluctuation

Compared with Fig. 2, in Fig. 7, the peak pressure of the lower chamber reduced by about 0.03 MPa. Both the acceleration and the velocity of the piston are reduced. The maximum positioning steady error is 0.19 mm. The steady error of the upper and lower chamber pressure is less than 0.001 MPa.

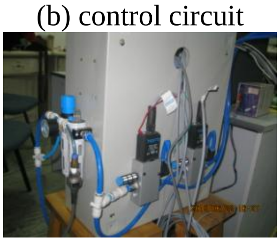
4 Experiments

In order to prove the applicability in practical pneumatic servo systems and continuous hopping, an application of the proposed control scheme is carried out in this study.



(a) pneumatically actuated leg vertical hopping test rig

(b) control circuit



(c) flow-type proportional valves and the FR.L

Fig. 8. Photographs of the pneumatically actuated leg vertical hopping test rig

Fig. 8 shows the configuration of the vertical hopping test rig of the pneumatically actuated leg. The test rig is primarily composed of the vertical guide frame, the pneumatic cylinder, two flow-type proportional valves, an Air Source Treatment FR.L, a 32-bit DSP processor, two pressure sensors and a photoelectric switch.

The pneumatic cylinder (DNCI-50-250-P-A-S2: FESTO Corporation) is a double rod and double acting cylinder, which is set vertically. The vertical guide frame constrains the pneumatically actuated leg's movement into the vertical direction. The DSP (TMS320F2812: TI Corporation) integrates AD and DA converters. The pressures of two chambers are measured by two pressure sensors (PSE520-01: SMC Corporation) respectively. Photoelectric switch (E2K-X8ME1: OMRON Corporation) is used to detect the signal of touchdown. The control signal calculated in DSP drives the two flow-type proportional valves. Two proportional valves (MPYE-5-1/4-010-B: FESTO Corporation) receive the voltage signals independently and control the flow to the upper and lower chamber respectively. A payload is attached to the cylinder to increase the mass ratio of the cylinder to the piston rod.

4.1 Experiment for the pneumatic servo system

In the experiment, the initial state of the plant is assumed as follows: p_1=0.23 MPa, p_2=0.24366 MPa, and z=0.03 m. The desired state of the plant is chosen as follows: p_{1d}=0.3 MPa, p_{2d}=0.31366 MPa, and z=0.09 m.

Fig. 9 shows the results of the pneumatic servo experiment. As shown in Fig. 9, without overshooting, the steady error of the positioning is no larger than 0.31mm, and the steady error of the pressure is no larger than 0.002MPa. It can be seen that the experimental results of the dynamics parameters are in good agreement with that of the simulation results. Therefore, the proposed control scheme is sufficiently effective to be used to the continuous and vertical hopping experiments which is conducted in the following.

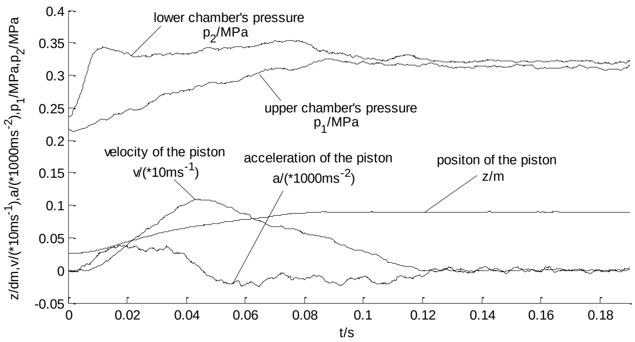


Fig. 9. Experimental curves of dynamic parameters

4.2 Experiment for the continuous and vertical hopping

Finally, the performance of the continuous and vertical hopping of the pneumatically actuated leg is examined with the proposed control scheme. The proportional and the differential gains of the PD controller which is used in the experiments is the same as the simulation. The experimental result of the vertical hopping is shown in Fig. 10.

It is clear from Fig. 10 that both the piston position and the chambers' pressure can be controlled rapidly and precisely during the flight phase. The positioning error can be controlled within 0.31 mm, and the pressure error can be controlled within 0.002 MPa. Stable and continuous hopping of the pneumatically actuated leg is achieved, which validate the proposed control scheme further.

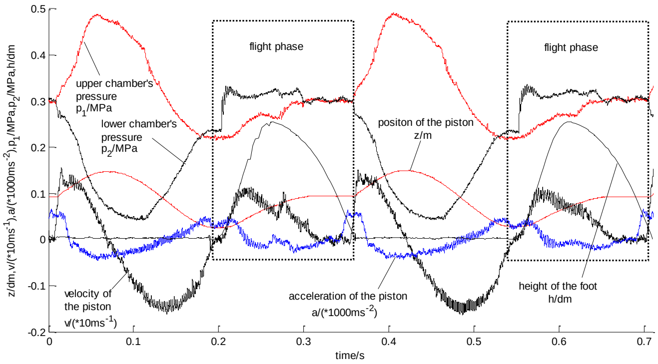


Fig. 10. Experimental curves of dynamic parameters of the pneumatically actuated leg for continuous hopping

5 Conclusions

Because of the influence of the gas compressibility and pneumatic system's inherent strong non-line feature, it's difficult to control the piston position and chambers' pressure independently and achieve ideal result of control precision. In this research, we have studied the pneumatic process in terms of gas internal energy; we have found out that the two chambers' gas internal energy can determine the position of the piston and the two chambers' pressure. Then a PD controller which uses the gas internal energy of cylinder's chambers as set points has been proposed. The main contribution of this paper is that it has found a control scheme for the pneumatically actuated leg which can control the piston position and chambers' pressure simultaneously and rapidly. Base on the proposed control scheme, the pneumatically actuated leg can achieve stable and continuous hopping.

During the continuous hopping, the steady error of the positioning can be kept less than 0.31 mm, and there is no overshooting, and the steady error of the pressure is no larger than 0.002 MPa, which are enough accurate for stable and continuous hopping of the pneumatically actuated leg.

The proposed method not only controls the piston position and the chambers' pressure simultaneously and precisely, but also guarantees the real-time for the hopping of the pneumatically actuated leg. Moreover, the controller's parameters can be adjusted easily, which is convenient for project realization.

Although we cannot achieve arbitrary precision for the piston position and chambers' pressure with the proposed control scheme, the experimental results still suggest that this control scheme is useful for the pneumatically actuated leg, because the control accuracy can meet the requirement for stable and continuous hopping.

Further research will be carried out to realize the height-controllable hopping in vertical direction for the pneumatically actuated leg. In addition, it is necessary for the one-legged robot actuated by the pneumatically actuated leg to balance itself in three dimensions.

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