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Simulation and experimental study of electro-pneumatic valve used in air-powered engine

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Abstract: To evaluate the performance of newly designed electro-pneumatic valves (EPVs) for the air-powered engine (APE) and study laws of parameters affecting them, a simulation model was established based on the thermodynamics and mechanics theories. Experiments were set up to determine the instantaneous effective orifice area of solenoid valve by the constant volume discharge method. The simulation model was also validated by comparing the measured displacement curve with the simulated displacement curve of the valve in the pressure of 0.16 and 0.49 MPa. Simulation and experimental results showed that maximum working frequency of the designed EPV could reach 30 Hz corresponding to 2000 r/min of engine rotating speed. Based on simulation results, impacts of temperature and pressure of control air on delay time, full opening/closing time and seating velocity of EPV were analyzed. The simulation model could also act as EPV simulation prototype in designing the air exchange control system of APE.

Key words: Pneumatic, Variable valve actuation (VVA), Air-powered engine (APE), Simulation

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INTRODUCTION

Researching air-powered engine (APE) (Yu *et al.*, 2002; Chen *et al.*, 2005) and its hybrid power (Higelin *et al.*, 2004) is becoming hot due to its zero emission and high efficiency characters. Practicality and optimization studies on APE show that it is critical for APE to realize valve timing and duration changeable in different engine speeds and loads (Cai *et al.*, 2004; Liu and Yu, 2008). One solution is adopting variable valve actuation (VVA).

Research on different types of variable valve actuators for internal combustion engines has been conducted for many years, including electromagnetic (Sugimoto *et al.*, 2004), electro-hydraulic (Lenz *et al.*, 1989), and electro-pneumatic actuators (Trajkovic *et al.*, 2006). Because of the contradiction between fast response and large flow rate, electromagnetic actuators were not fit for their application in APE. Electro-pneumatic actuators and electro-hydraulic ones have the same operating principals except their different working media. Compared with hydraulic ones, pneumatic actuators have the advantages of fast response, low cost and insensitivity to working fluid (Watson and Wakeman, 2005). However, the control system of pneumatics is somehow difficult for the air's compressibility and highly nonlinear features. Since APE uses high-pressure air as its energy, the electro-pneumatic valve (EPV) can offer a better alternative to electro-hydraulic ones.

A typical EPV mechanism used in the engine includes a pneumatic actuator, an engine valve and valve springs (Jia e*t al.*, 2007). Using the two-stage pneumatic on/off valves for gas exchange could simplify the structure of APE because they need not have the traditional engine valves and valve springs. Unfortunately, it is difficult to purchase them in the market due to the requirement of response time, flow

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rate and structure. Researchers have developed several types of large flow rate and high speed two-stage on/off valves (Zhou *et al.*, 1998; Ruan *et al.*, 2002; Jia *et al.*, 2003). However, the switch processes of these valves are affected by their inlet gas pressure and the fluctuated intake pressure of APE during their operating period would make it more difficult to control the valve timing and duration.

An EPV used in APE was made in Zhejiang University, Hangzhou, and its operating scheme was introduced in this paper. We first focused on modeling the opening and closing process of EPV. Then one EPV test bench was built up to verify the availability of this designed EPV and the validity of our proposed simulation model. A detailed study of EPV working characters and parameters sensitivity was carried out on the basis of simulation model.

SYSTEM DYNAMICS

As shown in the schematic representation of the EPV system (Fig.1), EPV is composed of a piston, a pneumatic cylinder, standard three-way solenoid valves, piston rings and springs. The main function of springs is to decrease the impact of piston on cylinder at both ends. Both solenoid valves are selected based on their large nominal diameter of 4 mm and high switching frequency of 120 Hz, and then equipped oppositely on the two cylinder ends. The two solenoid valves are almost the same except that one is normally open and the other is normally closed. When both of them are de-energized, solenoid valve A is open and solenoid valve B is closed. High-pressure control air would rush into the chamber A and push the piston to the right end which would cut off compressed air into engine cylinder. When there is electricity applied to both solenoid valves, the piston of EPV would move

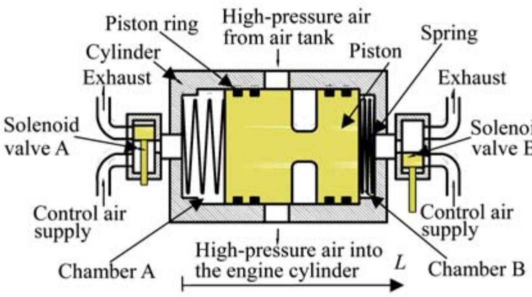

Fig.1 Schematic representation of EPV

to left. Compressed air would pass through the groove in the middle of valve piston and enter into the APE cylinder. Thus, the valve timing and duration of APE could be controlled by adjusting the time of electrical pulse applied to the solenoid valves.

OPERATING PROCESS MODEL OF EPV

Dynamic model of valve piston

Let the positive direction point to right, and based on Newtonian mechanics, the kinetic equation of the valve piston could be written as:

$$p_1 A - p_2 A + K(L_{\max} - 2L) - F_f - \beta \frac{dL}{dt} = M \frac{d^2 L}{dt^2}, \quad (1)$$

where p_1, p_2 are absolute pressures in cylinder chamber A and chamber B, respectively; A is the surface area of valve piston; K is the stiffness of each spring; $L_{\max}$ means the max displacement of piston; L behalves the instantaneous displacement of piston; M is the total mass of piston and piston rings; t is the time, and F_f is the Coulomb friction force; β is the viscous friction coefficient and its value can be obtained using genetic algorithms (GAs) (Daw *et al.*, 2003). F_f can be expressed as (Richer and Hurmuzlu, 2000)

$$F_f = \begin{cases} F_s, & \text{if } \dfrac{dL}{dt} = 0, \\ F_d \operatorname{sign}\left(\dfrac{dL}{dt}\right), & \text{if } \dfrac{dL}{dt} \neq 0, \end{cases} \quad (2)$$

where F_s and F_d are the static and dynamic friction force, respectively.

Model of gas state within cylinder chambers

To simplify the analysis process, it is assumed that: the high-pressure air within the chamber is ideal gas; the pressure and temperature within each chamber are homogeneous; the kinetic energy of gas is negligible and there is no leakage in the whole process. Regarding the chamber as a thermodynamic system (control volume), modeling the chamber consists of three equations (Liu *et al.*, 2005):

$$pV = mRT, \quad (3)$$

$$\frac{\mathrm{d}m}{\mathrm{d}t}=\frac{\mathrm{d}m_{\mathrm{in}}}{\mathrm{d}t}+\frac{\mathrm{d}m_{\mathrm{out}}}{\mathrm{d}t}, \tag{4}$$

$$\frac{\mathrm{d}T}{\mathrm{d}t}=\frac{1}{mC_{\mathrm{V}}}\left(\frac{\mathrm{d}Q_{\mathrm{w}}}{\mathrm{d}t}-p\frac{\mathrm{d}V}{\mathrm{d}t}+h_{\mathrm{in}}\frac{\mathrm{d}m_{\mathrm{in}}}{\mathrm{d}t}+h\frac{\mathrm{d}m_{\mathrm{out}}}{\mathrm{d}t}-u\frac{\mathrm{d}m}{\mathrm{d}t}\right), \tag{5}$$

where V is the control volume; m is the mass of air enclosed by the chamber; p is the pressure; T is the gas temperature; R is the gas constant; h and u are the specific enthalpy and the specific internal energy of air within the chamber, respectively; Q_{w} is the heat exchange between the air within the chamber and environment; m_{in}, m_{out}, and h_{in} are the mass of air flow into the chamber, the mass of air flow out the chamber, and the specific enthalpy of air flow into the chamber, respectively; C_{V} is the specific heat at constant volume.

Because the time consuming on opening and closing EPV is very short, heat exchange Q_{w} could be omitted and the process was modeled as adiabatic:

$$\frac{\mathrm{d}Q_{\mathrm{w}}}{\mathrm{d}t}=0. \tag{6}$$

The instantaneous mass flow rates during the charging and discharging processes were calculated using the following equation:

$$\frac{\mathrm{d}m_{\mathrm{in/out}}}{\mathrm{d}t}=\mu_{\mathrm{in/out}}S_{\mathrm{in/out}}\psi_{\mathrm{in/out}}\sqrt{p_{\mathrm{I}}\rho_{\mathrm{I}}}, \tag{7}$$

where variables with subscripts "in" or "out" indicate the charging or discharging parameters; p_{I} and ρ_{I} are the pressure and density of air on the upstream of solenoid valve; μ is the coefficient of charge; S is the effective orifice area of solenoid valve, and ψ is the stream function associated with the pressure different between upstream and downstream valves. ψ can be expressed as

$$\psi=\begin{cases}\sqrt{\dfrac{2k}{k-1}\left[\left(\dfrac{p_{\mathrm{II}}}{p_{\mathrm{I}}}\right)^{2/k}-\left(\dfrac{p_{\mathrm{II}}}{p_{\mathrm{I}}}\right)^{(k+1)/k}\right]}, & \dfrac{p_{\mathrm{II}}}{p_{\mathrm{I}}}>\left(\dfrac{2}{k+1}\right)^{k/(k-1)},\\ \left(\dfrac{2}{k+1}\right)^{1/(k-1)}\sqrt{\dfrac{2k}{k+1}}, & \dfrac{p_{\mathrm{II}}}{p_{\mathrm{I}}}\le\left(\dfrac{2}{k+1}\right)^{k/(k-1)},\end{cases} \tag{8}$$

where p_{II} is the downstream pressure, and k is the specific heat ratio.

Determination on effective orifice area of solenoid valve

As a critical component of EPV, the solenoid valve acted before the valve piston and should run in higher frequency than EPV, and thus the diameter of solenoid valve was made very small. All these made it difficult to determine the curve of effective orifice area with time. Topcu *et al.*(2006) established mathematical models of the spool valve by applying electromagnetic and dynamics theories. However, this method could not be applied to model solenoid valves of EPV for lots of parameters are unknown. And measuring the valve's instantaneous displacement directly is still difficult for limited space. The method adopted in this article was to measure the instantaneous air pressure in a constant volume chamber when it was discharged through the solenoid valve to atmosphere as shown in Fig.2. Fig.3 shows the measured instantaneous pressure p_{i} of air in the constant volume chamber during the switching of solenoid valve with initial relative pressure p_{ini} of 0.4, 0.6, 0.8 MPa, respectively. Since the switching process of solenoid valves is very short, the discharging process can be regarded as adiabatic. Thus we can get

$$\frac{T_{\mathrm{i}}}{T_{\mathrm{ini}}}=\left(\frac{p_{\mathrm{i}}}{p_{\mathrm{ini}}}\right)^{(k-1)/k}, \tag{9}$$

where T_{i}, p_{i} are the instantaneous temperature and pressure of air in the constant volume chamber, and T_{ini}, p_{ini} are the initial values.

Combining Eq.(7) with Eq.(9) and the ideal gas equation, the effective orifice area of solenoid valve can be written as

$$\mu_{\mathrm{in/out}}S_{\mathrm{in/out}}=-\frac{\mathrm{d}p_{\mathrm{i}}}{\mathrm{d}t}\frac{p_{\mathrm{i}}^{\frac{1-3k}{2k}}\,p_{\mathrm{ini}}^{\frac{k-1}{2k}}V_{\mathrm{c}}}{k\psi_{\mathrm{in/out}}\sqrt{RT_{\mathrm{ini}}}}, \tag{10}$$

where V_{c} is the volume of discharging chamber. Fig.4 shows that calculation results are approximately the same for different initial pressures. This means that the method to acquire the effective orifice area of solenoid valve is effective.

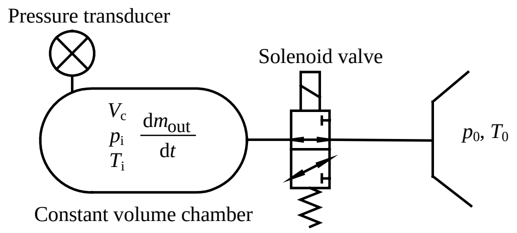

Fig.2 Experimental setup for the discharge method

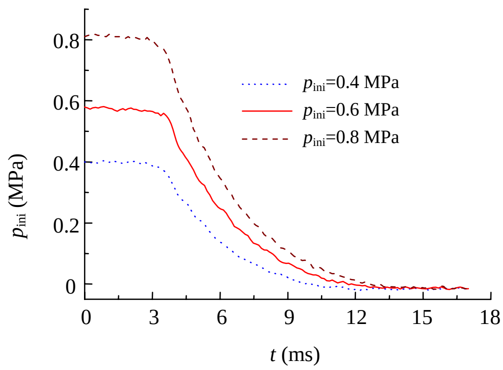

Fig.3 Measured instantaneous pressures of discharging chamber

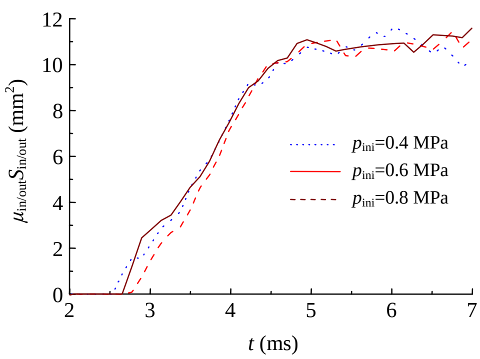

Fig.4 Instantaneous effective orifice area curves of solenoid valve

SIMULATION SETTING AND EXPRIMENTAL VALIDATION

The complete mathematical model for EPV consists of piston dynamic Eq.(1) and gas state Eq.(3), mass conservation Eq.(4) and energy conservation Eq.(5) of two chambers. The step-fix 4-order Runge-Kutta method was used to calculate these differential equation sets, where the calculation step was 0.01 ms. According to preliminary experiments, the friction force F_f is much smaller (less than 1.1 N) as compared with the others, thus negligible in simulation calculation. The main parameters are listed in Table 1. The options of GA to calculate the value of β are presented in Table 2 and the evaluation function is expressed as

$$f_{eval} = \frac{1}{n-1}\sqrt{\sum_{i=1}^{n}(L_{si} - L_{ei})^2}, \qquad (11)$$

where L_{si} and L_{ei} are the simulation and experimental instantaneous displacement of piston, respectively, n is the sample size. The calculation result of β was 111.08 N·s/m.

Table 1 EPV parameters used in model

Parameter	Value	Parameter	Value
M (kg)	0.1145	L_{max} (mm)	9
Clearance volume (mm^3)	11382	T_0 (K)	287
A (mm^2)	1134.1	K (N/m)	1828

Table 2 Options of GA used to calculate the value of β

Parameter	Description
Range of search space	100~120 N·s/m
Population size	20 strings
Encoding method	Floating point encoding
Maximum generation	80 generations

Experiments were conducted to demonstrate the simulation accuracy and identify the value of β. Fig.5 shows the experimental setup. The length of electrical pulse applied to the solenoid valves was 50 ms. Three control air pressures or upstream pressures of solenoid valves were used: 0.16, 0.40, and 0.49 MPa (relative pressure), and the intake air pressure of EPV was set the same as the control air pressure. The experimental result obtained under the pressure of 0.40 MPa was used to calculate the value of β, and the

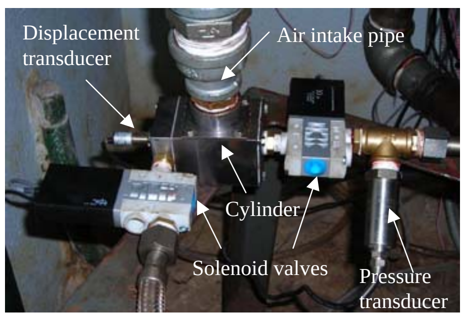

Fig.5 Experimental setup of EPV

others were used to validate the simulation model. Fig.6 shows both simulation and experimental results for the instantaneous displacements of piston. There is a close agreement between the simulation and experiment curves, with very good amplitude match and a maximum time error less than 1.2 ms, which confirms that the model presented in this paper could describe the operating process of EPV and has enough accuracy to be used for the control and dynamic performances analysis.

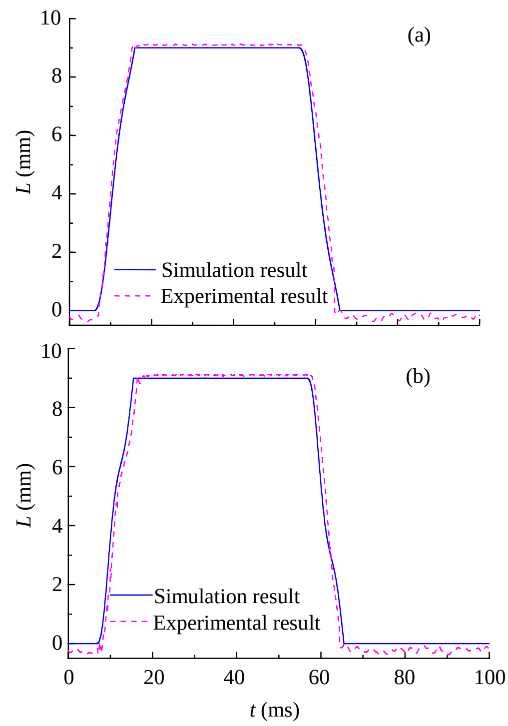

Fig.6 Instantaneous displacement curves of piston. (a) p_{in}=0.16 MPa; (b) p_{in}=0.49 MPa

SIMULATION ANALYSIS AND DISCUSSION

Based on the model proposed, impacts of two parameters, pressure p_{in} and temperature T_0 of control air or upstream high-pressure air of solenoid valves on EPV performances were studied by simulation. The EPV performances were defined by delay time t_d, full opening/closing time t_f and seating velocity v_e (Fig.7). Since the structure of EPV is symmetry, its opening and closing processes are almost the same. The following discussions focus on the opening process.

In simulation, T_0 ranged from 250 to 322 K with the step of 8 K and p_{in} ranged from 0.1 to 1.9 MPa (relative pressure) with the step of 0.2 MPa. Fig.8 shows their influence on t_d. It can be seen that t_d increases with increasing of p_{in} and T_0. But their effect magnitudes are different. When the pressure is lower than 0.7 MPa, increasing p_{in} could cause rapid increase of t_d. When the control air pressure is higher than 0.7 MPa, the change of p_{in} has little influence on the delay time t_d since more air would enter the chamber in higher pressure and it would take more time to discharge the air in the chamber in the following act. Delay time t_d almost increases linearly with T_0 on different levels of p_{in}, but its maximum increasing value of 0.41 ms is very small. That is to say, small change of the control air temperature has very small effect on the delay time of EPV and these effects could be negligent. When there is a large difference of T_0, it is easy to correct the advance and delay angles of APE because of the approximately linear relationship between T_0 and t_d.

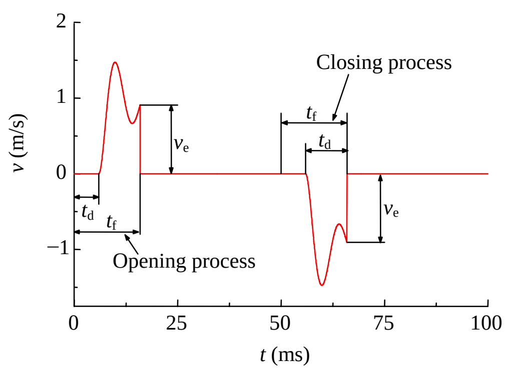

Fig.7 Parameters for defining EPV's performances

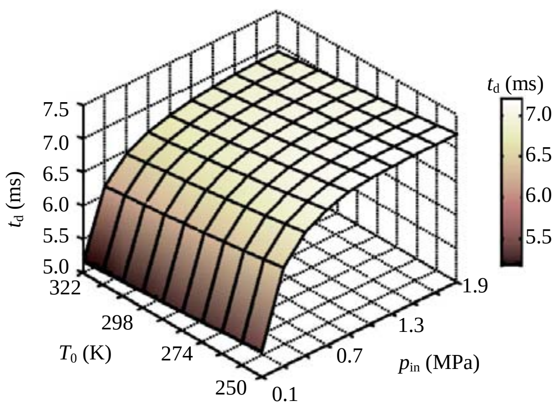

Fig.8 Influence map of temperature and pressure of control air on delay time

Fig.9 shows the piston velocity trace in different control air pressures. It can be seen that the piston velocity appears to fluctuate with the increase of air pressure. This phenomenon is related to the limited effective orifice area of solenoid valves. It can only meet the inspiration requirement in low pressure. When the control air pressure is high, the fast moving valve piston would press the air ahead of it and vacuum the chamber behind it. Therefore, a negative acceleration appears and the piston would decrease and even rebound until the air pressure behind the piston exceeds the pressure before the piston again. This fluctuation also causes t_f and v_e not to increase with the increasing of p_{in}, but has the maps like Fig.10 and Fig.11. Keeping p_{in} constant, t_f displays the general trend to decrease with the increasing of T_0. When the control air temperature is invariant or approximately invariant, the control air pressure should be optimized in order to acquire a fast response and low seating velocity of EPV.

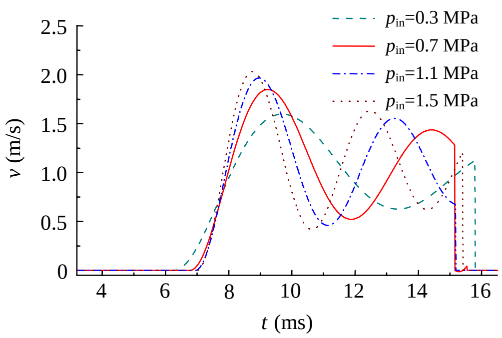

Fig.9 Instantaneous velocity curves of piston for different control air pressures

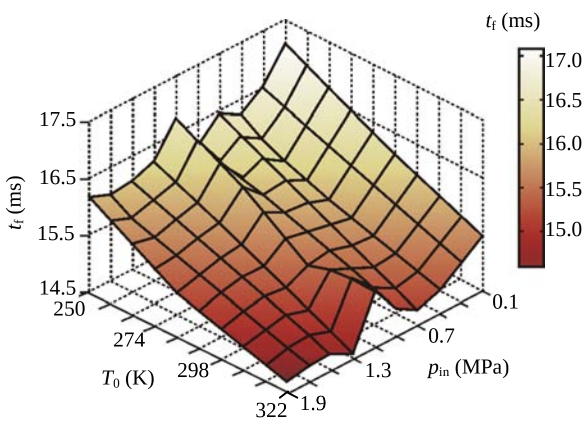

Fig.10 Influence map of temperature and pressure of control air on full opening/closing time

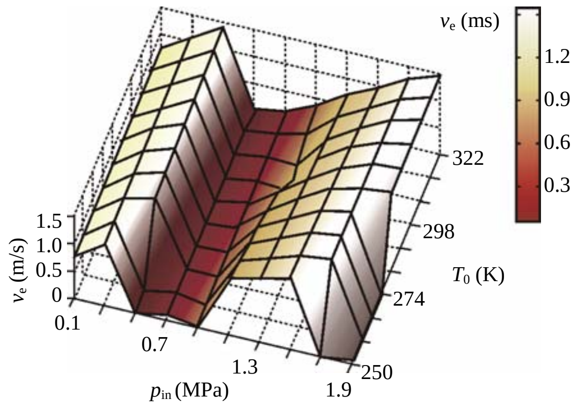

Fig.11 Influence map of temperature and pressure of control air on velocity at end of the stroke

Fig.10 and experimental results also show that the designed EPV could reach the operating frequency of approximate 30 Hz which is equivalent to an engine speed around 2000 r/min. Because APE had the advantages of low compressed-air consumption and large torque at low speed (Zuo *et al.*, 2007), the frequency can satisfy the speed demand of APE.

CONCLUSION

The designed EPV for APE satisfied the frequency requirement and compressed air flux requirement of APE. The mathematical model of EPV developed in this paper had enough accuracy to predict the operation of this type of EPV. Our simulation study showed that increasing pressure of compressed air solely would not make the EPV response faster for the valve piston would fluctuate in high pressure. A further study is under way to develop control systems to achieve soft landing of valve piston by making use of velocity fluctuating of valve piston and to realize the intelligent APE.

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