SENSITIVITY ANALYSIS OF DESIGN PARAMETERS OF ELECTRO-PNEUMATIC VALVE USED IN AIR POWERED ENGINE

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A new electro-pneumatic fully variable valve mechanism used for air powered engine (APE) was proposed, and its working process model was built. The validation tests showed the model had enough accuracy to be used for the dynamic performances analysis. The verified model was combined with in-cylinder thermodynamics model of APE in order to analyze the effects of several design parameters on the dynamic performances of the electro-pneumatic valve in a more practical way. The analysis result shows: the maximum diameter of valve core and the stiffness of the retaining spring should be optimized according to the control air pressure and the flow area of the solenoid valve; reducing the total mass of the moving parts can decrease the response time and the seating velocity of the valve core, however, if the total mass were too small, the intake flow rate of APE would be difficult to control; decreasing the clearance volumes of chambers can also reduce the response time.

Keywords: Electro-pneumatic; Variable valve; Air powered engine; Sensitivity analysis; Design parameter.

1. Introduction

Air powered engine (APE) has gained more and more attentions in recent years for its characteristic of needless fossil oil. However, vehicle driven by APE has very low driving range compared to the traditional vehicle fuelled by fossil oil due to the low energy density of pressurized air and energy utilization efficiency. Up to now, lots of means have been adopted to extend the driving range of the air powered car, which include dual-fuel hybrid propulsion system [1], quasi-isothermal expander engine [2], running in a proper thermal cycle [3], ameliorating the engine structure, and optimized working parameters and so on. Liu et al. [4] conducted a research on the optimal piston trajectory design to improve the high-speed performance of APE, and they expressed the importance of equipping APE with the electronic controlled variable valve system through analyzing the

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impact of valve timing on the performances of APE. Hu [5] made the parametric sensitivity analysis on the performance of APE, and the result shows the parameters of inlet valve are the key parameters to the dynamic performance and fuel economy of APE.

Researchers come into being a new thought of improving the efficiency of APE through reusing the waste heat of traditional internal combustion engine（ICE）considering that the cooling water and exhaust gas take away the energy of fuel up to 60-70% during ICE working period. Browna et al. [6] proposed a low-cost hybrid drivetrain concept, and they think that if waste heat from the engine is used to maintain an elevated tank temperature, efficiencies of nearly 50% may be possible. Huang and Tzeng [7] developed a new type of hybrid pneumatic power-system (HPPS) in order to optimize the running conditions and recycle the exhaust heat of ICE to propel the vehicle. Fang et al. [8] proposed a novel pneumatic-fuel hybrid system which can recover waste heat from cooling water of ICE to optimize the working process of APE, and their experimental results show that the maximum efficiency of APE can be raised from 26.18% to 41.22%. Most of the air hybrid engines should require the use of fully variable valve actuation systems to transfer their working mode among compression braking mode, air motor mode, air-power assist mode and normal firing mode [9]. However, Lee et al. [10] proposed a special cam profile system to achieve air hybrid.

One of the efficient ways to improve the performance of APE and air hybrid is adopting variable valve mechanism. Researches on different types of variable valve actuators for internal combustion engine have been conducted for many years which including electromagnetic [11], electro-hydraulic [12] and electro-pneumatic actuators [13]. Because of the contradiction between fast response and large flow rate, electromagnetic actuator was not fit for its application in APE. Since the APE uses high-pressure air as its energy, the electro-pneumatic valve (EPV) can offer a better alternative to others.

A typical EPV mechanism used in engine includes a pneumatic cylinder, valves, connecting tubes, engine valve and valve spring. To simplify its structure, a newly designed EPV used in APE was proposed in this paper referring to the existence of two-stage on/off valves developed by Jia at el.[14]. The first part of this paper introduced its detailed structure. The second part of the paper was focused itself on modeling EPV opening and closing process. Then one EPV test bench was built up to verify the proposed simulation model. Finally, detailed parameters of sensitivity analysis were carried out on the basis of the simulation model.

2. System Scheme

Fig.1 shows the system scheme of EPV. Two solenoid valves were equipped oppositely on the both ends of valve body directly without tubes in order to shorten the pneumatic pathway and decrease the response time of EPV. Both of solenoid valves have their large nominal diameter of 4 mm and high switching frequency of 120 Hz, and they are almost the same except that one is normally open and the other is normally closed. When both are de-energized, solenoid valve A is open and solenoid valve B is closed, then high-pressure control air would rush into the chamber A and push the valve core to the right end of EPV, which would cut off the pathway between chamber C and port B. When voltage is applied to both solenoid valves, the valve core of EPV would move to left, and high-compressed intake air would pass through port A, chamber C, port B in turn, and then into the cylinder of APE for the inlet EPV, and in the case of outlet EPV, air in the cylinder of APE would rush out from port B, chamber C and port A in turn, and then into the ambient. The shape of two ports was designed to be long and narrow to satisfy the frequency requirement and compressed air flux requirement of APE (Fig. 2).

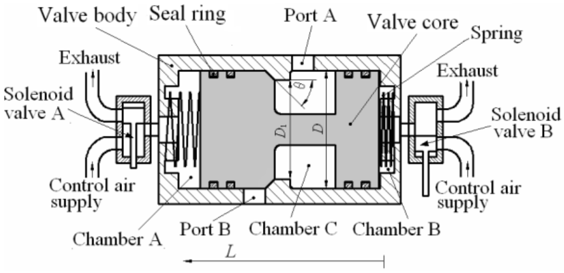


Fig.1 System scheme of EPV

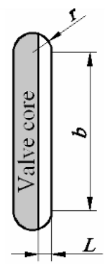


Fig.2 Shape and parameters of the port B

3. Operating Process Model

Because the operating process of EPV could be influenced by the inlet and exhaust air of APE, the whole model should include in-cylinder thermodynamics model of APE and the operating process model of EPV as shown in Fig. 3. The in-cylinder thermodynamics model of APE presented by Chen et al. [15] was used in this article.

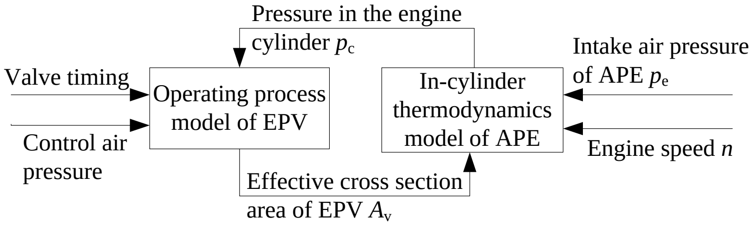


Fig. 3 Structure of simulation model

Dynamic model of valve piston. The kinetic equation of the valve core could be written as Eq.1 based on Newtonian mechanics.

$$(p_B - p_A)\frac{\pi D^2}{4} - F_a + K(L_{max} - 2L) - F_f - \beta\frac{dL}{dt} = M\frac{d^2L}{dt^2}, \tag{1}$$

where p_A, p_B are absolute pressure in chamber A and chamber B respectively, D is the maximum diameter of valve core, K is the stiffness of each spring, L_{max} means the max displacement of piston, L represents the instantaneous displacement of piston, M is the total mass of valve core and seal rings, t is the time, F_f is the Coulomb friction force, β is the viscosity, F_a is the force produced by the air in the chamber C. F_f can be expressed as [16]:

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

where F_s and F_d are the static and dynamic friction force. F_a can be written as:

$$F_a = \begin{cases} \frac{1}{A_v \rho_v}(\frac{dm_v}{dt})^2 \cos\theta & \text{if } L \neq 0 \\ \frac{p_{up}\pi}{4}(D^2 - D_1^2) & \text{if } L = 0 \end{cases}, \tag{3}$$

where m_v and ρ_v are mass and density of inlet and exhaust air of APE, p_{up} is the pressure in chamber C, D_1 is the diameter of inner cylinder near port A as shown in Fig. 1, θ is the cone angle as Fig.1 shows, and A_v is the effective cross section area of EPV. A_v can be expressed as：

$$A_v = \arccos(\frac{r-L}{r})r^2 - (r-L)\sqrt{(2r-L)L} + Lb. \tag{4}$$

Model of gas state within chamber A and chamber B. To simplify the analysis process, it's 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; the operating process is adiabatic. Modeling the chamber consists of three equations regarding the chamber as a thermodynamic system (control volume), and the detailed description can also refer to the in-cylinder thermodynamics model of APE presented by Chen et al[15].

$$pV = mRT, \tag{5}$$

$$\frac{dm}{dt} = \frac{dm_{in}}{dt} + \frac{dm_{out}}{dt}, \tag{6}$$

$$\frac{dT}{dt} = \frac{1}{mC_V}(h_{in}\frac{dm_{in}}{dt} + h\frac{dm_{out}}{dt} - p\frac{dV}{dt} - u\frac{dm}{dt}), \tag{7}$$

where V is the control volume, m is the mass of air enclosed by the chamber, p is pressure, T is gas temperature, R is the gas constant, h and u are the specific enthalpy and the specific internal energy of air within the chamber, and m_{in}, m_{out}, 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, and C_V is the specific heat at constant volume.

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

$$\frac{\mathrm{d}m_{\mathrm{in,out}}}{\mathrm{d}t} = \mu_{\mathrm{in,\,out}} S_{\mathrm{in,\,out}} \psi_{\mathrm{in,\,out}} \sqrt{p_{\mathrm{I}} \rho_{\mathrm{I}}}\ , \tag{8}$$

where variables with subscript "in" and "out" indicate the charging and 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 difference between upstream and downstream the valve.

4. Experimental Validations

The prototypes of inlet and exhaust EPV used for APE were made and their experiments were conducted on the test bench to validate the operating process model of EPV. Fig.4 shows the experimental setup. The outlet port of EPV is always directly connected with atmosphere while the air pressure of its inlet port is set the same as the control air pressure and is kept as constant to simplify the experiment. The parameters of inlet EPV are listed in Table 1. The parameters of exhaust EPV are the same as the inlet one except for M=0.101kg, L_{max}=9.7mm, r=4.5mm and b=27mm. The ambient pressure and temperature are 101325Pa and 287K. The period and duty ratio of pulse width modulation (PWM) to control the solenoid valves are 100ms and 50% respectively.

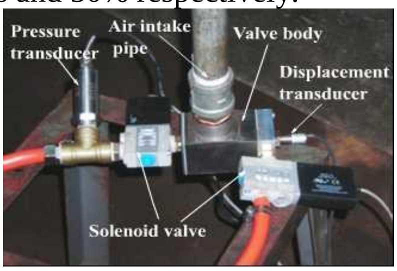


Fig.4 Experimental setup

Table 1

Parameters of inlet EPV

Parameters	Value	Parameters	Value
Clearance volume of chamber A V_{0A} [mm^3]	3525	L_{max} [mm]	7.4
Clearance volume of chamber B V_{0B} [mm^3]	4999	D_1 [mm]	32
D [mm]	38	r [mm]	3.5
M [kg]	0.097	b [mm]	29

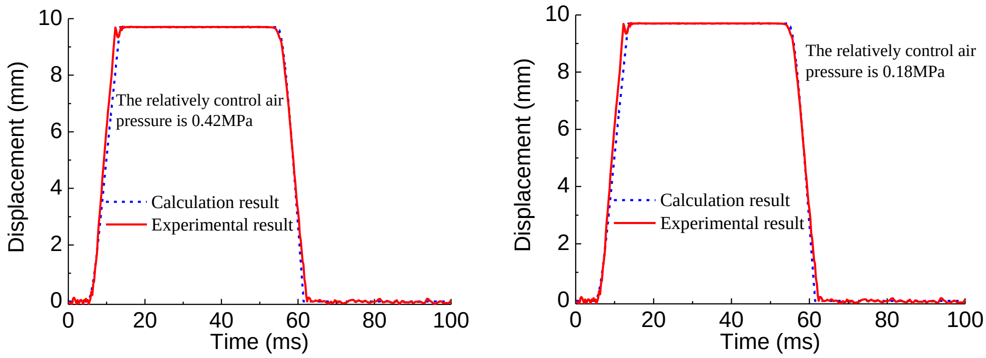
F_s [N]	32.1	θ [°]	45
F_d [N]	30.3	β [N·s/m]	48.9

Table 2

APE parameters used in model

Parameter	Value	Parameter	Value
Diameter of the cylinder [mm]	85	Open angle of inlet EPV[°CA]	-35
Piston stroke [mm]	90	Close angle of inlet EPV [°CA]	25
Ratio of crank radius to connecting rod	0.333	Open angle of outlet EPV[°CA]	150
Clearance volume [mm]	12348.5	Close angle of outlet EPV[°CA]	285
Absolute intake air pressure [MPa]	0.8	Engine speed[r/min]	2000

The operating process model of EPV consists of piston dynamic Eq.1 and gas state Eq.5, mass conservation Eq.6 and energy conservation Eq.7 of chamber A and B. Step-fix 4-order Runge-Kutta method was used to calculate these differential equation sets. The calculation step was 0.01ms. Fig.5 shows both the experimental and calculation results for the instantaneous valve core displacement of inlet and exhaust EPV when the relative pressure of control air to push the valve core of EPV are 0.42MPa and 0.18MPa respectively. There is a close agreement between the calculation and experimental curves, with very good amplitude match and less than 0.4ms and 1.22ms maximum time error for the inlet and exhaust EPV respectively, which confirms that the above model can describe the operating process of EPV and has enough accuracy to be used for its dynamic performances analysis.



(a) Inlet EPV (b) Exhaust EPV

Fig.5 Instantaneous valve core displacements of the EPV prototypes

5. Sensitivity Analysis of Design Parameters

The verified operating process model of EPV was combined with in-cylinder thermodynamics model of APE in order to analyze the effects of several design parameters on the dynamic performances of EPV in a more practical way. The whole model is shown in Fig.3. The parameters used in thermodynamics model of APE were listed in Table 2. The absolute control air pressure of 0.62MPa was not the same as the intake air pressure of APE in the following simulation analysis. Step-fix 4-order Runge-Kutta method was also used to calculate the whole model. The calculation step was 0.05° crank angle (°CA).

The EPV performances were defined by the delay angle (t_{do} and t_{dc}), the full opening/closing angle (t_{fo} and t_{fc}) and the seating velocity (v_{eo} and v_{ec}) as shown in Fig.6. Since the system scheme of inlet and exhaust EPV are the same, the following discussion was focused on the inlet EPV.

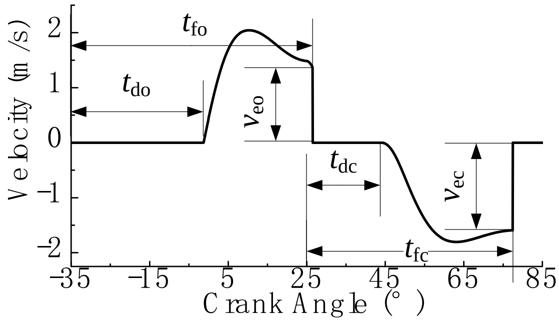
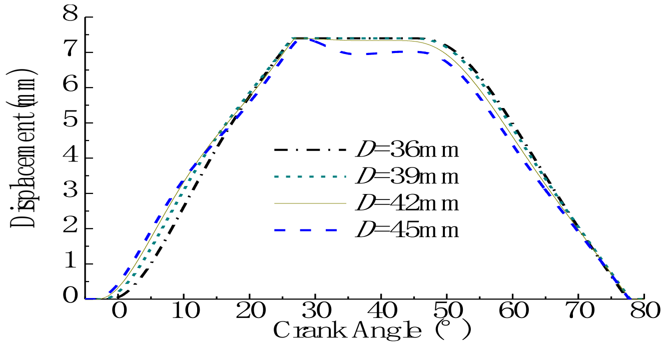
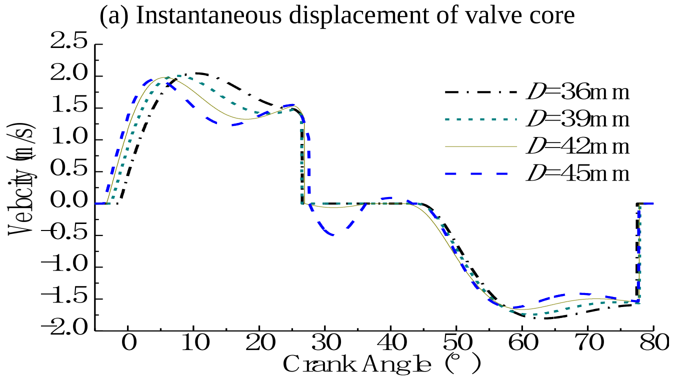


Fig.6 Velocity curves with parameters defining the performances EPV

Maximum diameter of valve core. The maximum diameter of valve core (*D*) is one of the important design parameters of EPV. It was kept increasing from 36mm to 45mm with the step of 3mm while other parameters were defined as constants as Table 1 and Table 2 show to study the influence of *D* independently. The delay angle decreases with the increasing of *D* as shown in Fig.7 (a). Because the changes of instantaneous volumes of chamber A and B increase with the increasing of *D* while the parameters of solenoid valves were kept constant when the valve core is moving, the time for valve core to reach its maximum displacement would increase with the increasing of *D*. Fig. 7 (b) shows that the change of *D* has little effect on the seating velocity.



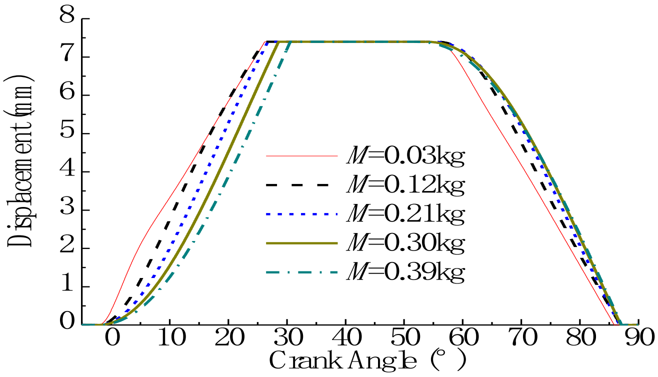
(a) Instantaneous displacement of valve core



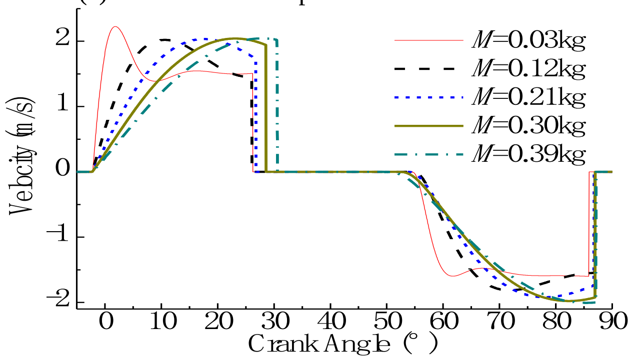
(b) Instantaneous velocity of valve core

Fig.7 Effect of the maximum diameter of valve core on the valve motion

Total mass of valve core and seal rings. In this subsection, the total mass of valve core and seal rings (M) ranged from 0.03kg to 0.39kg with the step of 0.09kg while other parameters were set as constants. The smaller the total mass of valve core and seal rings is, the less energy it needs to push the valve core to move. Hence, decreasing M would be helpful to decrease the response time of EPV as shown in Fig.8 (a). However, the fast-moving valve core would press the air ahead of it and vacuum the chamber behind it, and thus it causes its instantaneous velocity to fluctuate. Although decreasing M could help to reduce the seating velocity of EPV, if M was too small, the intake flow rate of APE would be difficult to control because the velocity fluctuation increases with the decreasing of M (Fig. 8 (b)).



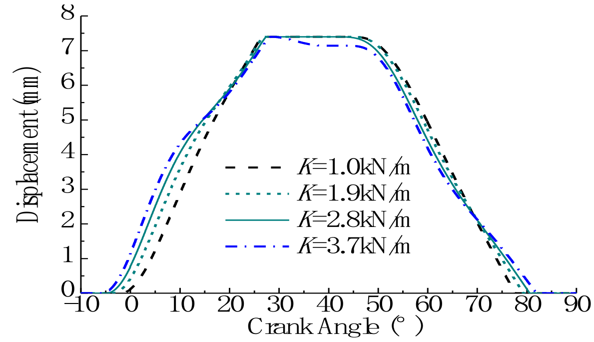
(a) Instantaneous displacement of valve core



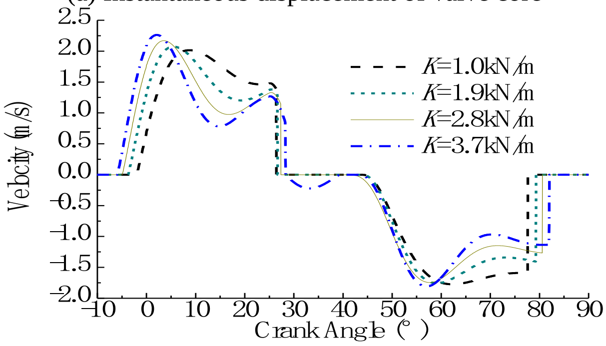
(b) Instantaneous velocity of valve core

Fig. 8 Effect of total mass of valve core and seal rings on valve motion

Stiffness of spring. The main function of spring is to decrease the impact of valve core on the valve body at both ends. Stiffness of spring (*K*) ranged from 1.0kN/m to 3.7kN/m with the step of 0.9kN/m to study the influence of *K* on the performances of EPV. Fig.9 (b) shows that the seating velocity of EPV decreases with the increasing of *K*. Because the direction of spring force is the same as the valve core movement during half of the opening process of EPV while it is opposite during the rest of the opening process, the delay angle of EPV decreases and the full open angle increases with the increasing of *K*. So, does the closing process of EPV, as shown in Fig.9 (b). If the full open/closing angles were used to evaluate the response of EPV, the value of *K* should be optimized since its influence on the response of EPV contradicted the seating velocity of EPV. In addition, it is important to take into account the pressure of control air in determining the value of *K*.



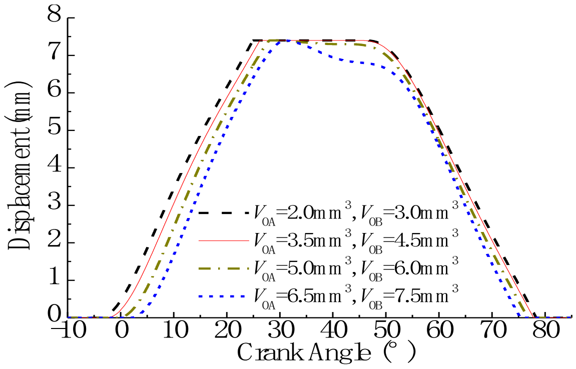
(a) Instantaneous displacement of valve core



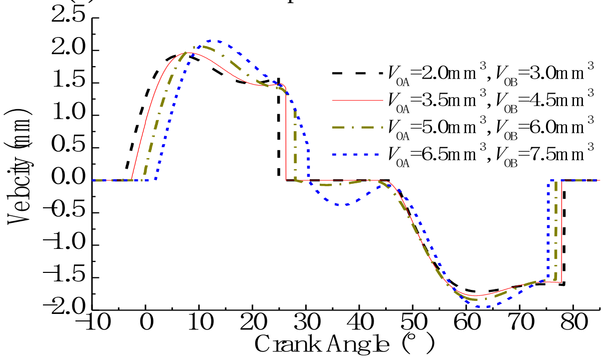
(b) Instantaneous velocity of valve core

Fig.9 Effect of valve spring's stiffness on valve motion

Clearance volumes of chamber A and B. In this subsection, the clearance volumes of chamber A and B (V_{0A} and V_{0B}) ranged from 2.0mm^3 and 3.0mm^3 to 6.5mm^3 and 7.5mm^3 respectively with the step of 0.5mm^3 while other parameters were kept constant. The increasing of clearance volume of chamber B would extend the time for the pressure in chamber B to increase high enough to push valve core to move. That is to say, the increasing of clearance volumes is harmful to increase the response of EPV, but helpful to reduce the seating velocity of EPV as shown in Fig.10. Since the angle gap between the moment when the valve core reaches its maximum displacement and the angle of inlet valve closing (25 °CA) is much smaller, the air pressures in chamber A and B of EPV are still much higher when the crank angle of EPV reaches 25°CA, which causes that the variation of clearance volumes has less influence on the closing process than the opening process of EPV. Generally speaking, reducing the clearance volumes of chamber A and B can decrease the respond time of EPV.



(a) Instantaneous displacement of valve core



(b) Instantaneous velocity of valve core

Fig.10 Effect of clearance volumes of chamber A and B on valve motion

6. Conclusions

The mathematic model of EPV developed in this paper had the precision to predict the operation of this type of EPV. Simulation study using the EPV model combined with the in-cylinder thermodynamics model of APE showed that:

(1) The size of EPV should be reduced to decrease the maximum diameter of valve core, the total mass of valve core and seals and the clearance volumes of chamber A and B with the purpose of improving the response speed of EPV. However, the size of EPV should satisfy the flow capacity for APE to inlet and exhaust.

(2) The stiffness of the retaining spring could be optimized according to the control air pressure and the flow area of the solenoid valve to improve the response speed and to reduce the seating velocity of EPV.

(3) Although decreasing the total mass of valve core could help to reduce the seating velocity of EPV. If the total mass of valve core was too small, it would be difficult to control the intake flow rate of APE because the speed fluctuation increases with the decreasing of the total mass of valve core.

Further study is going along to determine these design parameters to acquire optimal performance of EPV and to realize the intelligent APE.

Acknowledgements

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