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CONTROL PERFORMANCE IMPROVEMENT OF AN EMV SYSTEM USING A PM/EM HYBRID ACTUATOR

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ABSTRACT–In this study, we improved control performance of an EMV (electromechanical valve) system using a PM/EM (permanent magnet/electromagnet) hybrid EMA (electromagnetic actuator) and showed the feasibilities of both soft landing and fast transition of the EMV system using a simple PID control. The conventional EMV systems using only EM show significant nonlinear characteristics. Therefore, it is very difficult to control the valve position and several complex control schemes are used. This paper focused on the control performance improvement using a PM/EM hybrid actuator. In particular, a PM is used as a key design parameter such as a bias current of a magnetic bearing in order to improve the linear characteristic of the actuator, although most PM/EM hybrid actuators use a PM as a power saver during valve-open and -closed states. First, a FE (finite element) analysis was performed to confirm its linear static force characteristics. Then, both a test rig and a valve control system were built in order to prove experimentally the control performance improvement of the actuator. Finally, feasibilities of both soft landing and fast transition of the system were shown experimentally through gain-scheduled PID (proportional derivative integral) control.

KEY WORDS : Electro-mechanical valve (EMV), Electro-magnetic actuator, PM/EM hybrid actuator, Valve motion control

1. INTRODUCTION

At present, the increasing production of vehicles is raising concerns related to environmental pollution and fuel consumption. Therefore, many have come to realize the pressing need for clean and efficient internal combustion engines, and an increasing effort is being made to solve these problems. These efforts have led to the development of the VVT (variable valve timing) and VVL (variable valve lifting) systems. The VVT/VVL system can continuously alter the timing and lifting according to the engine status and the external conditions. Therefore, the VVT/VVL system can significantly improve the performance of the engine by increasing the torque, reducing the pumping loss and emissions, etc. As an alternative to the VVT/VVL system, an EMV system provides flexible controllability of the valve timing and lifting without the need for a mechanical linkage such as a cam system, so it can improve the engine performances and reduce the emissions drastically.

Kreuter *et al.*(1993) proposed an EMV control mechanism at FEV Motoren Technik (Germany). This mechanism was operated by free oscillation with two EMAs, springs and an armature. The mechanism was reported to have a short switching time and high power compared to a conventional valve train. Kluting and Flierl (2000) investigated a similar mechanism at BMW (Germany). In recent years, much research has been focused on EMVs (Henry and Lequesne, 1997; Pischinger *et al.*, 2000; Park *et al.*, 2000).

The existing EMV system has a very similar structure to that of an AMB (active magnetic bearing), although the two EMAs of the EMV system are operated independently. While the AMB system uses a bias current in order to increase the dynamic stiffness and to linearize the current-magnetic force relationship, the EMA of the EMV system does not use a bias current and shows significant nonlinear characteristics. Therefore, it is very difficult to control the valve position, and several complex control schemes, such as adaptive and iterative learning controls, are used to obtain the desired valve trajectory (Peterson *et al.*, 2003; Peterson and Stefanopoulou, 2002; Tai and Tsao, 2003).

In order to save power consumption during open and close states, a bistable PM/EM hybrid actuator was developed at the end of the 1980's and applied to the EMV systems (Theobald *et al.*, 1994). However, upper and lower actuators were controlled independently, as in

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the conventional EMV system, and the permanent magnet was used as a power saver during valve-open and -closed states. Therefore, the actuator is quite nonlinear and, like the conventional EMV actuator, difficult to control during its soft-landing, in particular. In addition, a PM/EM hybrid actuator was applied to an EMV system, where two coils of EMAs were connected serially; however, the PM lowered the effective stiffness and the EMV system only achieved 50 Hz bandwidth (Okada *et al.*, 2004).

This paper presents the control performance improvement using a PM/EM hybrid actuator. In particular, a PM is used as a key design parameter such as a bias current of a magnetic bearing to improve the linear characteristic of the actuator. First, an FE analysis of the proposed EMV system was performed to confirm the actuator static characteristics. Then, both a test rig and a valve control system were built in order to prove experimentally the control performance improvement of the actuator. Finally, the feasibilities of both soft landing and fast transition were proven experimentally through gain-scheduled PID control.

2. EMV SYSTEM USING A PM/EM HYBRID ACTUATOR

2.1. The Existing EMV System

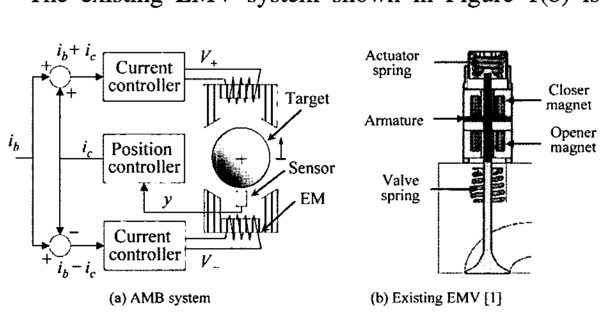
Figure 1 shows an AMB system and an existing EMV system. The AMB system consists of two EMAs, a target, a current sensor, a displacement sensor and a controller, as shown in Figure 1(a). Although the AMB system is unstable in open-loop status, the target is supported without any contact by controlling the currents of the two EMAs using the measured displacement (y) of the target. Therefore, the AMB system usually needs two types of controllers: a position controller produces the control current (i_c) using the measured target displacement, and a current controller produces appropriate voltages (V_+, V_-) for the coils using the measured coil currents. The AMB system adds a bias current (i_b) to the control current in order to linearize the current-force relationship and to increase the dynamic stiffness (Schweitzer *et al.*, 1994).

The existing EMV system shown in Figure 1(b) is similar to the AMB system and consists of an armature, two EMAs (closer and opener magnets), and two springs (actuator and valve springs). The existing EMV system works based on the oscillation of the armature and the springs; the bandwidth of the system is determined by the armature mass and the spring stiffness. The two EMAs take turns producing the attractive force for the valve motion by applying an appropriate voltage to the EMA. The existing EMV system does not use a bias current to reduce power consumption, but has a highly nonlinear magnetic force with the coil current and the armature position. Thus, it is not easy to control the attractive force of the EMA. Moreover, the EMA suffers from magnetic saturation when the armature gets close to one EMA. Therefore, it is very difficult to control the valve motion precisely, and particular challenges are associated with controlling the valve seating velocity for soft landing.

2.2. PM/EM Hybrid EMV System

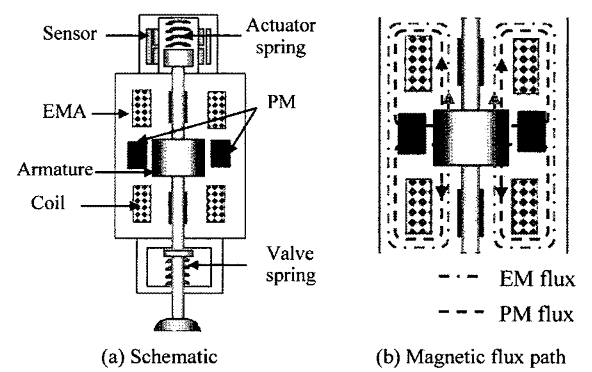
A PM/EM hybrid actuator for the EMV system is proposed in this paper to improve the valve motion control performance of the EMV system while PM/EM hybrid actuator has been used only for power saving. In the case of the AMB system, using a PM, namely, the PM-biased AMB, significantly reduces the power consumption and the heat generation due to the bias current. Moreover, the PM-biased AMB becomes simpler, because there is only one EMA to be controlled. The PM/EM hybrid actuator described in this paper has the same advantages as the PM-biased AMB. The PM/EM hybrid EMV system consists of an armature, two PMs around the armature, an EMA (serially connected upper/lower coils), and upper/lower springs, as shown in Figure 2(a). The flux of the PM flows identically in the upper/lower air gaps and the flux of the EM generates the force required to move the armature, as shown in Figure 2(b). The PM/EM hybrid actuator system has several control benefits over the existing EMV system as follows (Lee *et al.*, 2002; Lee, 2001);

(1) The need for only one EMA to be controlled keeps



(a) AMB system (b) Existing EMV [1]

Figure 1. AMB and existing EMV systems.



(a) Schematic (b) Magnetic flux path

Figure 2. The PM/EM hybrid EMV system.

the system simpler.

(2) Current control is much easier due to the nearly constant inductance by varying the valve position.

(3) The magnetic force characteristic can easily be adjusted by changing the geometry of the PM, which offers greater flexibility in the system design. In particular, the magnetic force characteristic of the actuator can be significantly adjusted by choosing the size of the PM (Chang *et al.*, 2007).

(4) The improved linearity of the proposed EMA enhances the valve motion control performances such as the VVT/VVL and the soft landing control.

3. ANALYSIS OF THE PROPOSED PM/EM HYBRID EMV SYSTEM

FE (finite element) analyses were performed in order to verify the magnetic flux distribution and the static characteristics of the proposed actuator, whose specifications are shown in Table 1. The maximum force was chosen to be the same as that of the existing EMV system, which renders the actuator area. The magnetic coil and the permanent magnet were selected by considering a 105°C temperature rise (insulation class F) and assuming the actuator is cooled down effectively through a water jacket. However, the coil and permanent magnet should be redesigned via a thermal analysis before applying this actuator to a real car system. Only half of the actuator is modeled because the geometric symmetry can be exploited. The FE model consists of the stator core, serially-connected upper/lower coils, armature and permanent magnets. The force applied by the actuator is calculated for various armature positions and coil currents. A flux density distribution result of the FE analysis is shown in Figure 3.

Figure 4 shows the magnetic force of the actuator as determined by using FE analyses for various currents and armature positions. Over the entire range, the magnetic force is much more linear than that of the existing EM system (Lee, 2001) and can be approximated as the linear combination of the current and position terms: a current gain and an open-loop stiffness of the actuator, which is given by

Table 1. Parameter of the PM/EM hybrid EMA used for the FE analysis.

Description	Values
Area of pole	450 mm^2
Area of PM	325 mm^2
Magnetic flux density of PM	1.02 T
Air gap	4.5 mm
Max. coil current	4A
Total coil turns	300
Armature Position	4 mm
Permeability of vacuum	4πe-7 Hm
Relative permeability of PM	1.05
Thickness of PM	25 mm

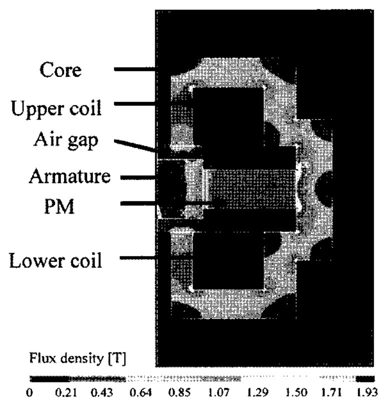


Figure 3. Flux density distribution result of FE analysis.

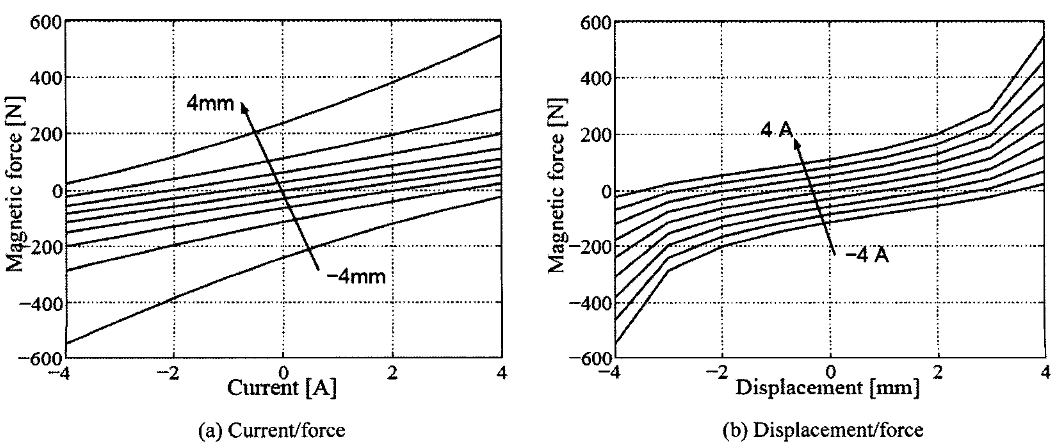


Figure 4. Magnetic force determined by using the FE analyses and dependent on the current and displacement.

$$F_{mag}=K_y y+K_i i \tag{1}$$

where F_{mag} is the magnetic force of the actuator, K_y is the open-loop stiffness, K_i is the current gain, y is the armature displacement, and i is the current of the EMA.

In particular, the open-loop stiffness and the current gain are almost linear within ±3 mm range although the air gap is 4 mm. Therefore, if the air gap is chosen to be 1 mm wider than a desired stroke, the actuator can be operated with linear characteristic and prevent a high magnetic force on the PM. However, the wider air gap will result in a larger actuator size.

4. EXPERIMENTAL SETUP

An experimental setup was built to verify the potential improvement of the valve motion control performance of the proposed EMV system. The experimental setup consists of the PM/EM hybrid actuator, an EMV control system, a control PC, a function generator for position or current reference signal, a 42V DC power supply and a digital oscilloscope, as shown in Figure 5. Nowadays, 42V cars are extensively being investigated worldwide in order both to reduce power loss and to improve the performance of the electric system. Therefore, we assume that the EMV system uses a 42V system.

4.1. Prototype of the PM/EM Hybrid EMV Actuator

A PM/EM hybrid EMV actuator whose schematic was shown in Figure 2(a) was prototyped as shown in Figure 6. Standard E-shaped Si-steel cores were laminated for the stator and solid soft iron, due to the material availability, was used for the armature even though a better candidate is Fe-Cobalt alloy with 2.4T saturation magnetic flux density. Nd-Fe-B rare earth magnets were used and oilless bearings were used to guide the armature. The dimensions of the proposed EMV system are: a total height, including the springs, ofs 174 mm; a width and depth, including the AL housing, of 100 mm and 54 mm, respectively. The size of the actuator is not small enough to install directly a real car. However, if a high magnetic

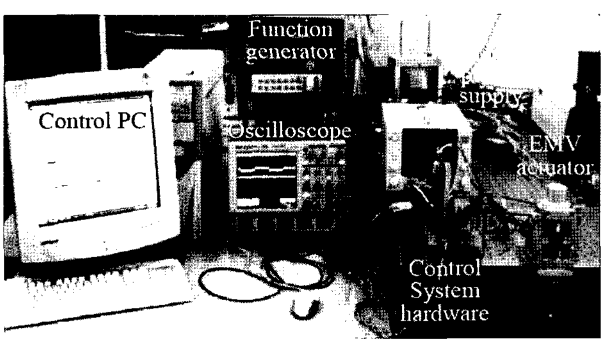


Figure 5. Experimental setup.

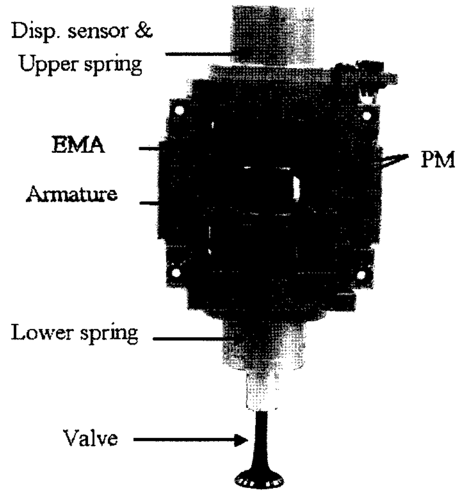


Figure 6. The PM/EM hybrid actuator.

saturation material, e.g., Fe-cobalt alloy, were used and the design further optimized, both the size and the weight could be significantly reduced.

4.2. Linear Capacitive Position Sensor

A linear differential capacitive sensor was developed as the feedback element for the valve motion control. The developed sensor reduces the space required for the installation of the sensor by using the spring supporter as the sensor target and the upper spring housing as the sensor electrode, as shown in Figure 7(a). The sensor consists of two cylindrical sensor electrodes, a guard electrode surrounding the sensor electrodes and a target. The valve position is obtained by measuring the difference in capacitance between the two cylindrical sensor electrodes and the target, as described in equation (2).

$$V_o = G(C_1 - C_2) \tag{2}$$

where, V_o is the output of the sensor, G is the sensor gain, and C_1 and C_2 are the capacitances between the sensor electrodes and the target.

The sensor calibration experiment was performed and the result is shown in Figure 7(b). The developed sensor

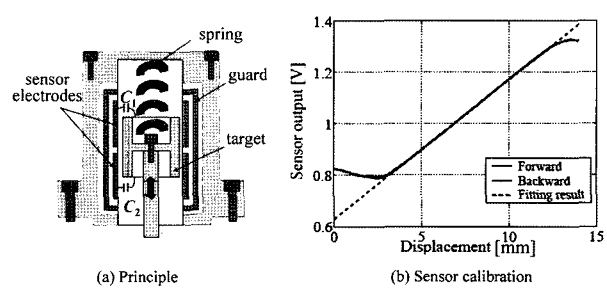


Figure 7. Capacitive displacement sensor.

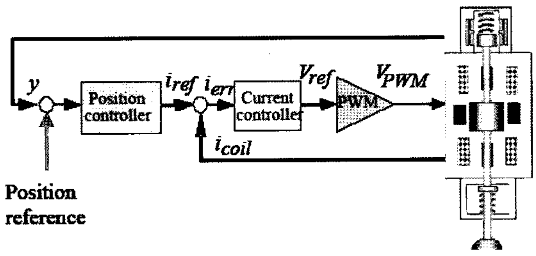


Figure 8. Controller block diagram.

shows excellent linearity in the given operating range. The sensor accuracy, which determines the valve control accuracy, was about 10 μm.

4.3. Control System

The control system block diagram for the developed EMV system is shown in Figure 8. The control system consists of a PID position control, a PI (proportional integral) current control and a PWM generation part. The PID controller produces the current reference (i_{ref}) using the position error between the measured armature position (y) and reference position. Then, the PI controller produces the reference voltage (V_{ref}) using the current error (i_{err}) between the current reference (i_{ref}) and the measured coil current (i_{coil}). Finally, the PWM generation part produces a PWM voltage (V_{PWM}) whose average is equal to the reference voltage (V_{ref}).

The control can efficiently reject disturbance forces such as pressure and friction force variation. In addition, the valve dynamics can be adjusted by the PID position controller: the proportional derivative gains are closely related to the stiffness and damping of the closed-loop system, which means that the closed-loop system can have the desired bandwidth and damping by simply tuning the PID controller. However, the open-loop system shows a narrow bandwidth and low damping like the 1-DOF mass-spring system. The PI current controller usually has a high gain and the current control bandwidth is usually over 500 Hz, which is much higher than the position control bandwidth.

The hardware schematic diagram and the prototype of the control system are shown in Figure 9. The control system hardware consists of a DSP (digital signal processor) system, a power board, a displacement sensor amplifier and an interface circuit. Position, current control and the PWM module are implemented in a TMS320F2812 DSP that has 16 units of 12bit AD and 16 units of 14 bit PWM, which has an IGBT module that consists of six IGBTs, two current sensors and gate drive electronics. One power board can drive two EMAs using the space vector control technique (Yim *et al.*, 2002). Three parts of the control system are mounted on the interface circuit board and interconnected to each other.

5. EXPERIMENTAL RESULTS

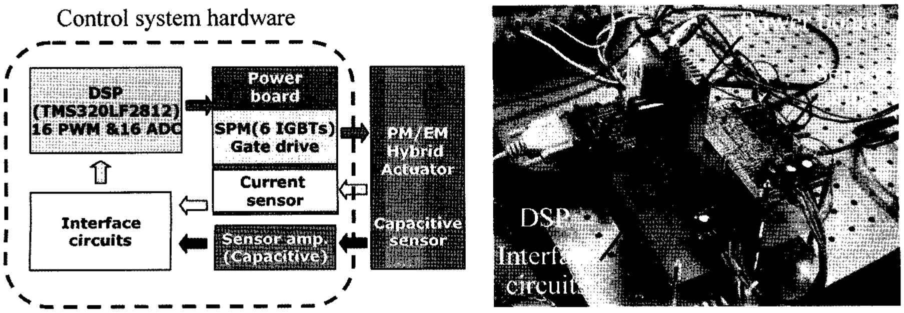
5.1. Identification of the PM/EM Hybrid EMV System

The proposed EMV system can be modeled as a 1-DOF vibration system or as a spring, mass and damper system with a magnetic force of equation (1), as described by equation (3).

$$m\ddot{y}+c\dot{y}+k_s y=F_{mag}=K_y y+K_i i$$
$$\Rightarrow m\ddot{y}+c\dot{y}+(k_s-K_y)y=K_i i \qquad (3)$$

Here, m is the valve mass, c is the damping coefficient, and k_s is the spring stiffness.

In order to verify the simple dynamic model above, the FRF (frequency response function) of the PM/EM hybrid EMV system was measured using the sine chirp signal of the HP dynamic analyzer, 35670A (Ahn *et al.*, 2003). The FRFs both from the current reference to the measured armature position (denotes $Y(s)/I_{ref}(s)$) and from the current reference to the measured coil current (denotes $I_{coil}(s)/I_{ref}(s)$) were measured, and then the EMV system FRF ($Y(s)/I_{coil}(s)$) was indirectly obtained by dividing the two measured FRFs one by the other, as shown in Figure 10. The identification results for various valve positions are shown in Figure 11. The identification result shows



(a) Block diagram

(b) Prototype

Figure 9. EMV control system hardware.

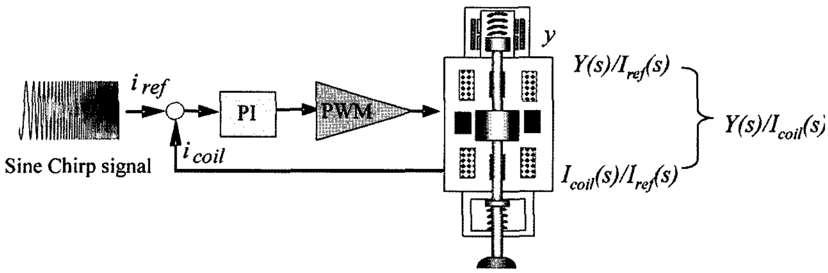


Figure 10. The identification scheme.

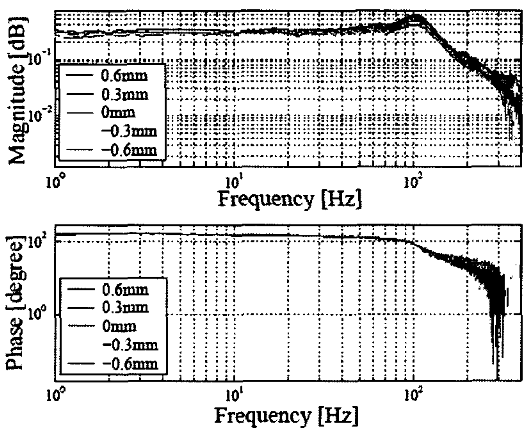


Figure 11. Identification results with various armature positions.

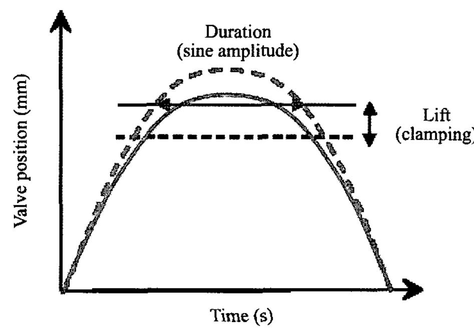


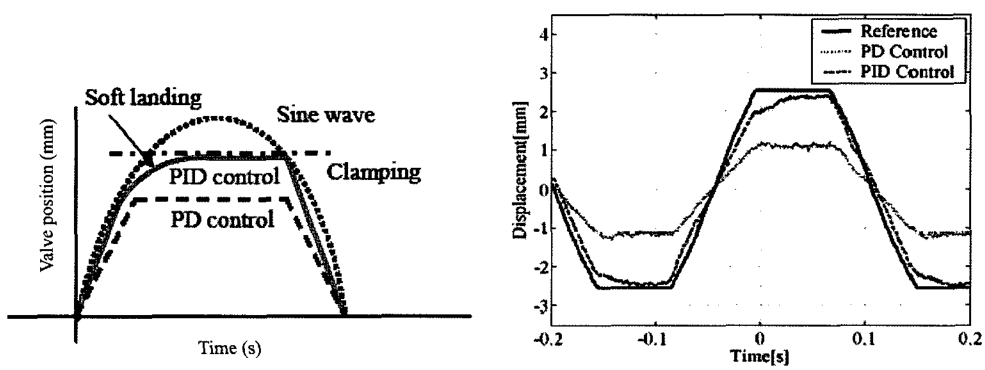
Figure 12. Reference valve motion signal.

the typical FRF of a 1-DOF vibration system, such as that described by equation (3). Although the bandwidth of the open-loop system is less than 100 Hz (about 90 Hz), the bandwidth of the closed-loop system can be increased up to 100 Hz by simply increasing the proportional gain. Furthermore, the FRF variation of the developed EMV actuator with the armature position is much smaller than that of the existing EMV (Tai and Tsao, 2002) since the developed EMV actuator has better linear magnetic force and does not suffer from magnetic saturation.

5.2. Valve Motion Control

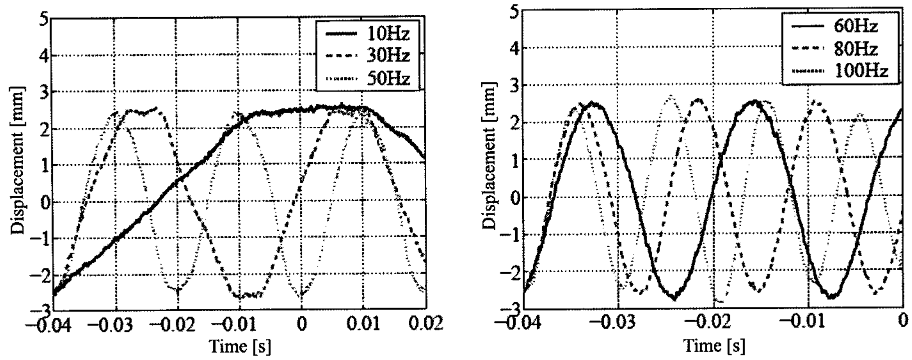
5.2.1. Valve motion reference using a sine wave

The feasibility of the valve motion control was tested using a reference valve motion signal generated from a function generator. A sine wave generated by the function generator, and both the valve duration and the valve lift were varied by simply adjusting the amplitude and clamping magnitude, as shown in Figure 12. Here, the valve lift is usually taken as being less than one-quarter of the valve head diameter (Pulkrabek, 1997). Since the valve diameter is 26 mm, the entire valve lift is chosen to be approximately 5.5 mm, which is still smaller than that of a conventional engine. However, a proper design change such as increasing the nominal air gap would



(a) Principle (b) Experiment

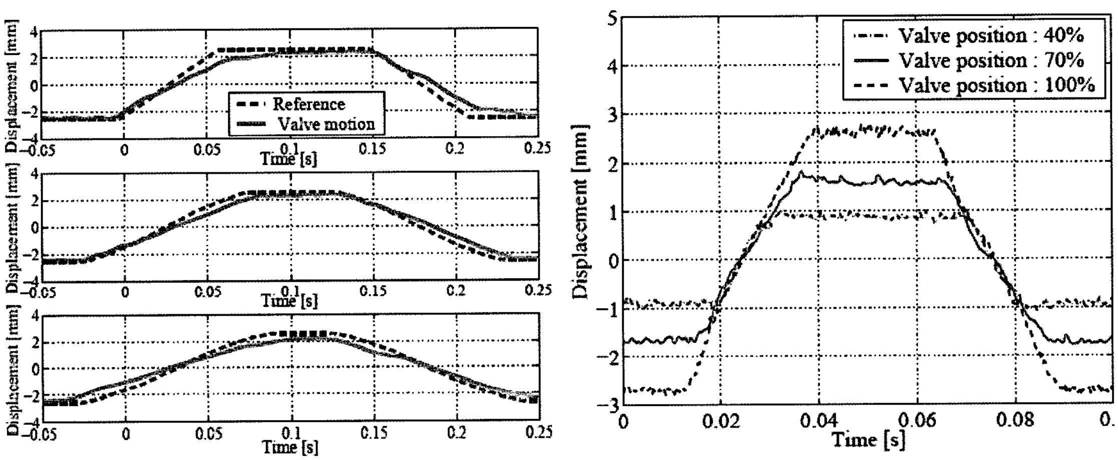
Figure 13. Soft landing control using a PID control.



(a) 10Hz - 50Hz

(b) 60Hz - 100Hz

Figure 14. Valve motion controls at various frequencies.



(a) Valve duration (10Hz)

(b) Valve lift (10Hz)

Figure 15. Adjustment of the valve duration time and the valve lift.

enlarge the valve stroke, which requires an increase in the volumes of the PM and coil and results in a size enlargement of the actuator.

5.2.2. Soft landing and fast transition

The soft landing of the proposed actuator can be achieved by tuning the PID control. The principle of soft landing achieved by the PID control is shown in Figure 13(a). The proportional control increases the fast valve transition speed and the derivative control improves the transient response. Integral control should be added to reduce the tracking error of the PD control. At the same time, soft landing can be achieved by adjusting the integral gain. The results of the soft landing obtained by tuning the PID control gain are shown in Figure 13(b). The PD control always produces steady-state errors and the valve cannot follow the reference valve motion precisely. The integral control allows the valve to follow the reference position as well as achieve the soft landing. Although the valve seating velocity is still higher than a conventional engine and the result in Figure 13(b) is low-speed case, the soft landing with a simple PID control was experimentally shown to be feasible. In addition, valve motion control ranging from 10 to 100 Hz can be implemented by adjusting the PID control gain, as shown in Figure 14. The first proto-type EMV system could operate up to 100 Hz, which is far wider than the existing PM hybrid actuator bandwidth, 50 Hz (Okada *et al.*, 2004). In addition, using the Fe-cobalt alloy and further design optimization could reduce the armature mass and the bandwidth of the system could be further enlarged to cover the entire range of the engine operating speed (usually from 800 rpm to 7000 rpm).

5.2.3. Adjustment of the valve duration time and the valve lift

The valve duration time can also be controlled by adjusting both the reference valve motion signal and the PID control gain. The top graph in Figure 15(a) shows the reference tracking at 100% full lift, and the middle and bottom graphs correspond to lifts of 80% and 60% of the full lift, respectively. The valve lift can also be changed by altering the clamping magnitude of the reference signal. In Figure 15(b), the black dotted line corresponds to the full valve lift and the red and blue dotted lines represent 40% and 70% of the full valve lift, respectively. The result shows that the lift of the proposed EMV system can easily be adjusted by the reference valve position signals.

6. CONCLUSION

This paper presented the control performance improvement using a PM/EM hybrid actuator. In particularly, a PM is used as a key design parameter such as a bias current of a magnetic bearing in order to improve the linear characteristic of the actuator, although most PM/EM hybrid actuators used a PM as a power saver during valve-open and -closed state. First, an FE analysis of the proposed EMV system was performed to confirm the actuator static characteristics. Then, both a test rig and a valve control system were built in order to prove experimentally the control performance improvement of the actuator. Finally, the feasibilities of both soft landing and fast transition were proven experimentally through gain-scheduled PID control.

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