World Academy of Science, Engineering and Technology
International Journal of Biomedical and Biological Engineering
Vol:3, No:8, 2009

One-DOF Precision Position Control using the Combined Piezo-VCM Actuator

Yung-Tien Liu, and Chun-Chao Wang

***Abstract*–**This paper presents the control performance of a high-precision positioning device using the hybrid actuator composed of a piezoelectric (PZT) actuator and a voice-coil motor (VCM). The combined piezo-VCM actuator features two main characteristics: a large operation range due to long stroke of the VCM, and high precision and heavy load positioning ability due to PZT impact force. A one-degree-of-freedom (DOF) experimental setup was configured to examine the fundamental characteristics, and the control performance was effectively demonstrated by using a switching controller. In rough positioning state, an integral variable structure controller (IVSC) was used for the VCM to conduct long range of operation; in precision positioning state, an impact force controller (IFC) for the PZT actuator coupled with presliding states of the sliding table was used to obtain high-precision position control and achieve both forward and backward actuations. The experimental results showed that the sliding table having a mass of 881g and with a preload of 10 N was successfully positioned within the positioning accuracy of 10 nm in both forward and backward position controls.

***Keywords*—**Integral variable structure controller (IVSC), impact force, precision positioning, presliding, PZT actuator, voice-coil motor (VCM).

I. Introduction

PIEZOELECTRIC actuator is a popular electromechanical element that can provide precise motion, generate a large force, and feature a high-frequency response and small size. Benefiting from these advantages, many positioning devices using the PZT actuators are well found in precision industry. Typically, there are three kinds of positioning devices using the PZT actuators classified by the actuating methods [1]. The first one is to use the steady displacement of a PZT actuator to obtain a theoretical unlimited displacement resolution [2]. The next one is to use the dynamic displacement of a PZT actuator to perform stick-slip like motion with a large stroke operation [3, 4]. The third one is to use rapid deformation of a PZT actuator incorporated with a mechanical element, such as spring, pneumatic cylinder, and voice-coil motor, for long range operation [5]-[7]. In this paper, the control performance of a positioning device featuring with one-degree-of-freedom (one-DOF) actuating ability using one set of the combined piezo-VCM is presented.

Y. -T. Liu is with the Department of Mechanical and Automation Engineering, National Kaohsiung First University of Science and Technology, Kaohsiung 811, Taiwan, R.O.C. (e-mail: ytliu@ccms.nkfust.edu.tw).

C. -C. Wang is with the Department of Electrical Engineering, Kao-Yuan University, Kaohsiung 824, Taiwan, R.O.C. (e-mail: jcwang@cc.kyu.edu.tw).

With regard to the combined piezo-VCM actuator [5], it has been reported as a precision adjusting tool featuring tow main characteristics: 1) a heavy load and high-precision actuating ability on the order of 10 nm due to the actuation of PZT impact force, 2) a large operation range of the PZT actuator due to long stroke of the VCM. Therefore, this hybrid actuator can be applied to assembly or alignment works in manufacturing process where a precision adjustment for a component is required. In that study, since the hybrid actuator was not directly connected to the target object and the actuation was through the form of contact force, two hybrid actuators were required to precisely adjust the target object with forward and backward motions [8]. Instead of the function as a precision adjusting tool, the positioning device presented in this paper can obtain one-DOF positioning ability using only on single hybrid actuator. The actuating principle is based on the PZT impact force coupled with a presliding state of the target object.

To achieve a good control performance of a target object being subjected to a heavy load or with dry friction, a suitable controller is required for the actuator to drive the target object smoothly and robustly. The variable structure control (VSC) is one of the major approaches to dealing with the nonlinear system having parameter uncertainties. Phakamach and Akkaraphong designed an integral variable structure controller (IVSC) being applied to the electrohydraulic servomechanism velocity control system [9, 10]. It comprises an integral controller for achieving zero steady-state error and a variable structure controller for enhancing robustness. In this paper, for examining the control performance of the combined piezo-VCM actuator, a switching dual-controller consisting of the IVSC for the VCM and an impact force controller (IFC) for the PZT actuator is configured, and the control performance is verified through experiments.

In the next section, the driving process of the positioning device using the piezo-VCM actuator is described. The theoretical models for the hybrid actuator to perform both rough and precision positioning are presented in Section III. The variable structure controller with integral compensator (IVSC) for the VCM and an impact force controller (IFC) for the PZT actuator are presented in section IV. The experimental setup is given in section V. Results of experiments are provided in section VI. Conclusion is given in Section VII.

II. Driving Process

Fig. 1(a) shows the main components and operation principle of the positioning device. The combined piezo-VCM actuator is directly connected to the sliding table, which is set upon the sliding surface with a frictional constraint. Based on the configuration, the sliding can be actuated to move in ether forward or backward direction by using only the VCM. However, it is very difficult to actuate the sliding table precisely due to stick-slip phenomenon usually existing between the sliding surfaces with dry friction. On the other hand, the actuating method presented in this paper is capable of performing precision positioning by applying a PZT impact force to the sliding table with a presliding state. Detailed operation processes are described as follows:

(a) In the initial stage, the sliding table is standstill and the combined piezo-VCM actuator is electrically neutral.

(b) In the forward rough positioning stage, the VCM is actuated by applying a current to its coil which produces a forward electromagnetic force F_v. If the forward thrust force is larger than the maximum static friction force, the sliding table will start to move with a large travel distance. In this state, however, a high-precision position control is difficult to achieve due to nonlinear characteristic of dry friction.

(c) In the forward precision positioning stage, the actuation for the sliding table is switched to the combination of a small forward thrust force coupled with a PZT impact force F_p. Initially, the small thrust force generated by the VCM is to keep a forward presliding state of the sliding table, i.e., the sliding table does not exactly move. And then, the sliding table is actuated to move precisely by the PZT impact force, which is generated by applying a pulse voltage waveform to the PZT actuator.

(d) In the backward rough positioning stage, the VCM is actuated by applying a reverse current to its coil which produces a backward electromagnetic force F_v. If the backward thrust force is larger than the maximum static friction force, the sliding table will start to move with a large travel distance.

(e) In the backward precision positioning state, the actuation for the sliding table is similar to previous step (c) but with a backward pre-sliding state of the sliding table. Processes (b) ~ (e) are used to control the sliding table undergoing both forward and backward motions with a satisfactory positioning accuracy. Thus, the sliding table may feature one-DOF high-precision positioning ability.

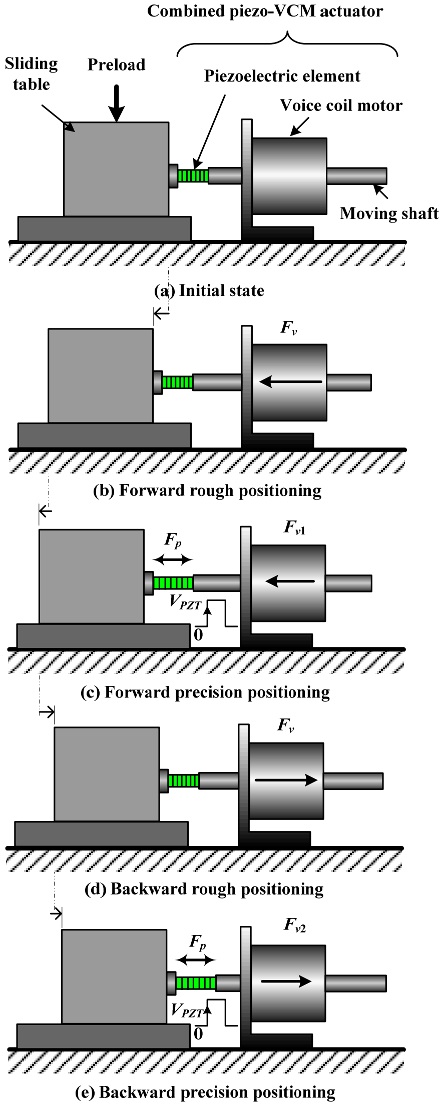


Fig. 1. Main components of the positioning device and driving process.

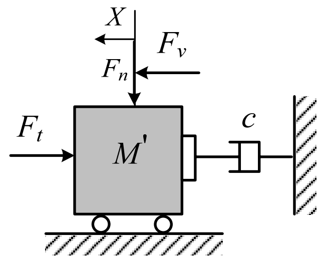


Fig. 2. The mechanical model of the rough positioning.

III. Theoretical Analysis

A. Rough Positioning State

Referring to the main components of the positioning device shown in Fig. 1(a), the analysis model can be represented as shown in Fig. 2. In rough positioning state, since the actuation for the sliding table is only governed by the VCM, the dynamic equation is expressed as follows:

$$M'\ddot{X} = F_v - F_t - c\dot{X}\,. \qquad (1)$$

where M' is the total equivalent mass including moving shaft of VCM, PZT actuator, and sliding table; c is a damping coefficient for expressing the VCM due to the nature of back electromotive force (EMF); F_v is the thrust force caused by the electromagnetic force of the VCM; and F_t is the friction force existing on the sliding surfaces. The thrust force F_v is resulted by applying a current to the coil of the VCM. Referring to the equivalent electrical circuit of the VCM as shown in Fig. 3, the applied voltage V_{VCM} to the VCM can be expressed as follows:

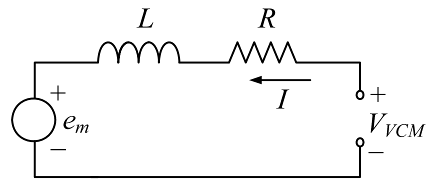


Fig. 3. Theoretical equivalent electrical circuit of the VCM.

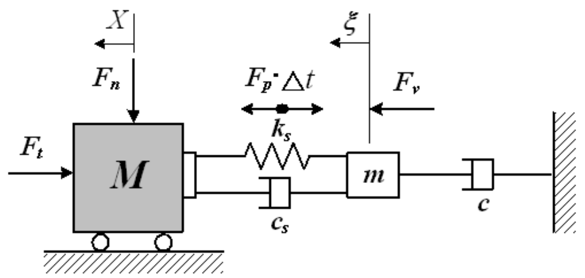


Fig. 4. The mechanical model of the precision positioning.

$$V_{VCM} = RI + L\dot{I} + e_m, \quad (2)$$

where I is the coil current; R is the coil resistance; L is the coil inductance; $e_m (= K_m \dot{x})$ is the back-EMF with a constant K_m. The electromagnetic force F_v is proportional to the current and is expressed as follows,

$$F_v = K_v I, \quad (3)$$

where K_v is a force constant of the VCM.

B. Precision Positioning State

In precision positioning state, the actuation is performed by applying the PZT impact force to the sliding table being in presliding state maintained by the VCM. Referring to the actuating process shown in Fig. 1(c), the analysis model can be described as shown in Fig. 4. Since the positioning device is mainly composed of the actuating part of the VCM and the driven part of the sliding table, it can be simply regarded as a 2-DOF mass-spring-damper mechanical system. Therefore, the physical model is described by the following two-coupled differential equations.

$$\begin{aligned} m\ddot{\xi} &= -c\dot{\xi} + k_s(X - \xi) + c_s(\dot{X} - \dot{\xi}) - F_p + F_v \\ M\ddot{X} &= -k_s(X - \xi) - c_s(\dot{X} - \dot{\xi}) + F_p - F_t, \end{aligned} \quad (4)$$

where M and m respectively represent the masses of sliding table and moving shaft of VCM, and their corresponding displacements are expressed as X and ξ. The PZT actuator is characterized by a spring with stiffness constant k_s and damping coefficient c_s. The impact force F_p is generated by the actuation of the PZT actuator with a short period of elongation of ξ_s, and is expressed as k_s. In precision positioning state, the thrust force of the VCM is used only for producing a presliding state of the sliding table in either forward or backward direction. Therefore, the thrust force is given by a smaller value than the friction force, and satisfies the following condition,

$$F_t \geq F_v > 0. \quad (5)$$

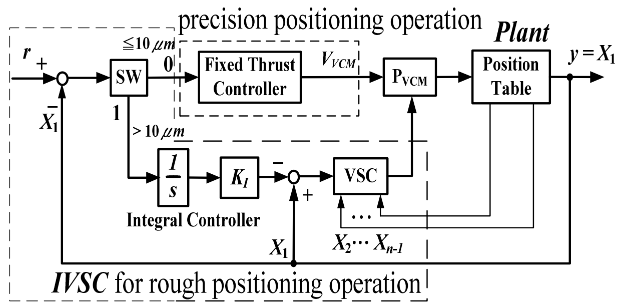


Fig. 5. Block diagram of the switching controller with the IVSC for VCM undergoing rough and precision positioning operation.

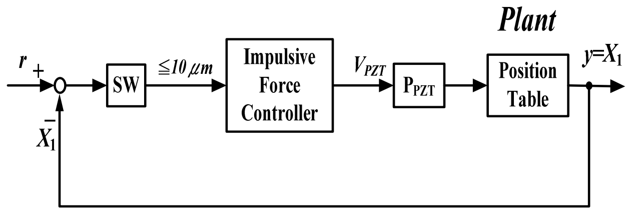


Fig. 6. Block diagram of the switching controller with the IFC for PZT actuator undergoing fine positioning operation.

IV. Control Strategy

A. Design of IVSC for the VCM

As described in Section II of operation principle, the VCM is used to perform rough positioning with long-range operation, and the PZT impact force is used to obtain high-precision positioning ability. Therefore, a dual-controller with switching function shown in Fig. 5 is implemented to attain the performance. The dual-controller consists of a variable structure controller with an integral compensator (IVSC) as well as an impact force controller (IFC). As shown in Fig. 5, the IVSC composes of a VSC with an integral compensator in the feedforward control loop. In rough positioning state, the IVSC is used to control the sliding table with large travel-range and high-speed abilities. Once the steady-state position error ε is within 10 μm, the control practice is switched (SW = 0) to the precision positioning state, which is performed by the IFC for the PZT actuator coupled with a constant thrust force given by the VCM, as shown in the block diagram of Fig. 6. The constant thrust force is to keep the presliding state of the sliding table and thus is smaller than the friction force. Under this condition, the impact actuation based on IFC is carried out by applying a pulse voltage to the PZT actuator. As a result of the combinational actuations, the sliding table can finally move to a specified precision position.

In rough positioning state, the control system is considered as a linear system as shown in Fig. 5, and it can be described as the following forms:

$$\begin{aligned} \dot{X}_i &= X_{i+1}, \qquad i = 1, \cdots, n-1 \\ \dot{X}_n &= -\sum_{i=1}^{n} a_i X_i + bU - d(t) \\ \dot{Z} &= r - X_1 \end{aligned} \quad (6)$$

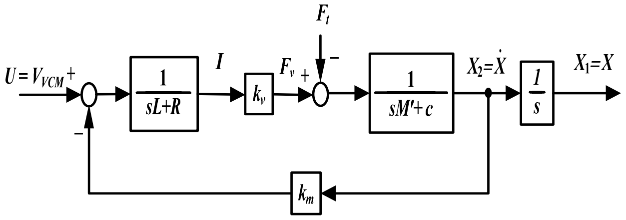


Fig. 7. The block diagram of VCM servo control system.

where r is the input command, X_i is the state variable, X_1 is the output signal, U is the control input, a_i and b are system parameters, d is disturbance, and n is the order of state variable.

Let the control input function U have the switching form of

$$U = s_w U_I + (1 - s_w) U_T \tag{7}$$

where U_I is the control command based on IVSC, U_T is the control command of the constant thrust controller, and

$$s_w = \begin{cases} 1, & \text{if } |r - X_1| \geq \varepsilon \\ 0, & \text{if } |r - X_1| < \varepsilon, \end{cases} \tag{8}$$

where ε (=10 μm) is the steady position error.

The strategy of designing an IVSC controller involves 1) designing an appropriate control function U_{eq} to guarantee the existence of a sliding mode, 2) determining a switching hyperplane and an integral control gain K_I to obtain good control performance, and 3) eliminating chattering phenomena using a supervisor control signal.

The IVSC method was developed by Phakamach and Akkaraphong [9, 10], it combines an integral controller followed by a VSC. In this paper, the IVSC controller is used to examine the control performance of the VCM actuating system. According to (1)-(3), the VCM servo control system can be shown as the block diagram of Fig. 7, and it can be described by the state equations based on (6) as follows:

$$\begin{aligned} \dot{X}_1 &= X_2 \\ \dot{X}_2 &= X_3 \\ \dot{X}_3 &= -a_2 X_2 - a_3 X_3 + bU - d(t) \\ \dot{Z} &= r - X_1 \end{aligned} \tag{9}$$

where

$$[X_1(t) \quad X_2(t) \quad X_3(t)]^T = [X(t) \quad \dot{X}(t) \quad \ddot{X}(t)]^T$$

$$a_2(t) = \frac{K_v K_m + Rc}{M'L}$$

$$a_3(t) = \frac{M'R + cL}{M'L}$$

$$b = \frac{K_v}{M'L}$$

$$d(t) = \frac{R}{M'L} F_t + \frac{1}{M'} \dot{F}_t$$

where $X_1 = X$ is the displacement of the sliding table, and r is the reference input.

From (9), the third order linear controllable canonical form with time-variant system is used to design a variable structure controller with integral compensation. Let the control input U_I satisfy the condition of a sliding mode, it can be defined as

$$U_I = U_{eq} + \Delta U \tag{10}$$

where U_{eq}, called equivalent control, is used to drive the operation point to the sliding surface; ΔU is a supervisor input used to eliminate the influences due to the parametric variations in a_i, b, and disturbances $d(t)$, and to guarantee the existence of sliding mode. Detailed derivations, including U_{eq}, ΔU, and the integral gain K_I, can be referred to previous work [8].

Table 1. Identified parameters of VCM

M' (kg)	c (Ns/m)	L (H)	R (Ω)	K_v (N/A)	K_m (V/(m/s))
63×10^{-3}	1.778	94×10^{-3}	3.657	4.029	4.029

B. Design of IFC for the PZT Actuator

The control model for the PZT impact actuation can be derived based on a heuristic approach, in which it is assumed that the impact actuation is performed between two masses (M and m). According to the conservation law of momentum and further assuming the condition of no energy loss during impact actuation, the predicated displacement ΔX caused by the impact actuation can be derived into the following compact form [6].

$$\Delta X = c \cdot f(V_{\mathrm{PZT}}) \tag{11}$$

where,

$$\begin{aligned} c &= \left.\frac{m}{m+M}\right/ (F_t - F_{\mathrm{v}}) \\ f(V_{PZT}) &= \frac{1}{2} k_s (\xi_s (V_{PZT}))^2, \end{aligned} \tag{12}$$

in which ξ_s is the excited static displacement of the PZT actuator caused by PZT voltage V_{PZT}. Therefore, the control voltage V_{PZT} can be determined by the inverse function of (12) while a predicted displacement ΔX is given. It is a pulse voltage waveform with duty cycle and can be described as

$$\Delta V(n) = f^{-1}\left(\frac{\Delta x}{c}\right) \cdot \delta(n - T) \tag{13}$$

$$\delta(n) = \begin{cases} 1 & , n = T \\ 0 & , n \neq T \end{cases}$$

where $\delta(n)$ is an unit impulse function and T is pulse width.

V. Experimental Setup

Fig. 8 shows the experimental setup of precision positioning device using the combined piezo-VCM actuator. The PZT actuator having the dimension of 5 mm × 5 mm × 20 mm (Tokin) is embedded in a steel protective block with a soft stiffness for preventing breakage from an extension load. The

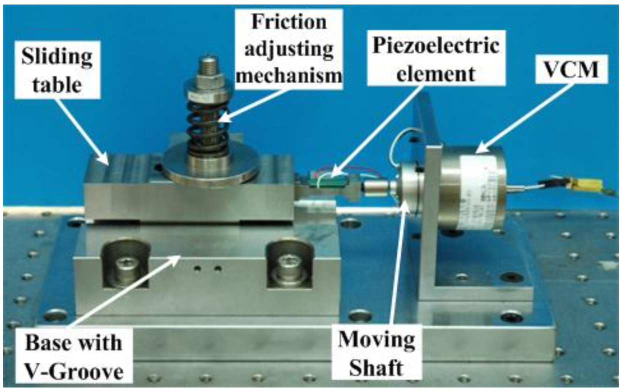


Fig. 8 The Photograph of the experimental setup with hybrid actuator..

steel block is then firmly fixed between the sliding table and the VCM. The sliding table is made of stainless steel with the dimension of 35 × 25 × 130 mm and mass of 881g. The sliding table setting upon the V-grooved base is capable of moving in either forward or backward direction, and thus has one-DOF positioning ability. The sliding surfaces have a roughness of Ra 0.36 m, which was obtained by grinding. A spring-type friction adjusting mechanism with a stiffness of spring being 14.7 N/mm is mounted on the topside of the sliding table for giving a preload to the sliding table.

Fig. 9 shows the schematic experimental setup for performing position control. The driving voltages for PZT actuator and VCM are generated by the MATLAB software and SIMULINK programming language. Two 16-bit D/A converters are used to transform the control voltages into the power amplifier. Since an instantaneous rise in current will occur when a pulse waveform was being applied to the PZT actuator, a heavy-duty power amplifier with the rated current of 10 A and with the frequency bandwidth of 1 MHz was used. The displacement of the positioning table is detected by a capacitance-type gap sensor, with a measuring range of $\pm 250\mu$m and a resolution of 10 nm.

VI. Experimental Results

A. Parametric Identification of the VCM

To design the IVSC, it is necessary to identify the parameters of VCM. Since the VCM control system is considered as a linear system, the VCM was disconnected from the sliding table to form a single actuator in the parametric identification process. As expressed in (2), the VCM is regarded as a mechanical damper and can be represented by an equivalent electrical circuit, as shown previously in Fig. 3. Applying a sinusoidal voltage waveform to the VCM, the displacement of the moving shaft of the VCM and the coil current were recorded, and then parametric estimation was made by the recursive least-square method (RLS) [8]. The VCM parameters identified are summarized in Table I, in which the damping coefficient c is 1.78 Ns/m. Since the coil resistance R = 3.66 Ω and force constant K_v = 4.03 N/A, the thrust force F_v related to the VCM voltage V_{VCM} is therefore derived as 1.10 N/V_{VCM}.

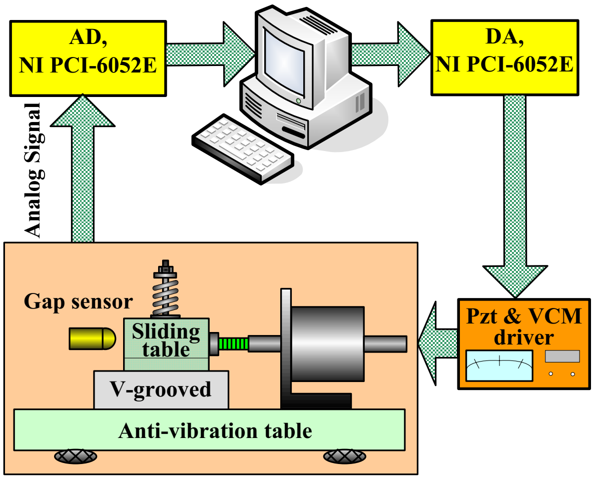


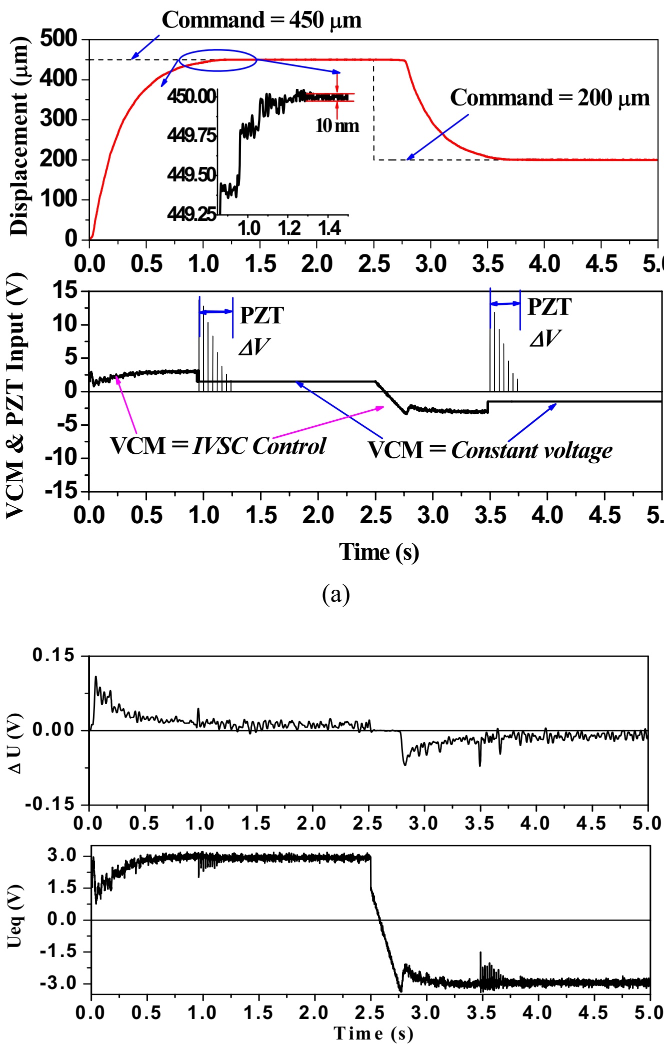
Fig. 9 The schematic drawing of control system.

B. Position control

Although the positioning device might feature high-precision and large operational range of actuating ability, to obtain a satisfied positioning accuracy, the performance of position control has to be examined due to parametric variations of the positioning device. Fig.10 shows the experimental results of position control in accordance with the implemented controller depicted in Fig. 10. In Fig. 10(a), the recorded forward and backward displacements of the sliding table and the control commands for PZT actuator and VCM are shown; in Fig. 10(b), detail commands of IVSC including the equivalent control input U_{eq} and supervisor input ΔU are presented.

Initially, the forward and backward target positions were given as 450 μm at t = 0 s and 200 μm at t = 2.5 s measured from the reference origin, respectively. Referring to the forward position control shown in Fig. 10(a), in rough positioning state using the IVSC algorithm, the VCM carried the sliding table smoothly reaching to the target position of 450 μm within a position error of 10 μm at time axis of 0.95 s. At this moment, the control practice was shifted to the precision positioning state using the IFC for the PZT actuator coupled with a constant thrust force of 2.2 N given by the VCM. Due to the operation of PZT impact force, it took 0.33 s for the sliding table moving from the rough position of 440 μm to the final precise position of 449.99 - 450.01 μm. The enlarged drawing shown in Fig. 10(a) is the recorded displacement obtained by the actuations of PZT actuator. The forward control ended with the positioning accuracy within 10 nm, which is the resolution limitation of the measuring system. The total control time in forward control was 1.28 s.

The backward control started at t = 2.5 s. Similar to the forward control, initially, the VCM control system based on the IVSC algorithm was used to actuate the table moving from the stating position of 450 μm to the final position of 200 μm with a positioning accuracy of 10 μm, and then precision position control was performed based on the PZT impact force coupled with a backward thrust force of the VCM. The total



(b)

Fig. 10 Experimental results of position control (a) displacement and voltage commands of the piezo-VCM actuators, (b) ΔU, and equivalent control Ueq.

control time in backward control was 1.43 s. The average sampling periods for the VCM and PZT control systems were 1 ms and 10 ms, respectively. According to the experimental results, the effectiveness of the configured IVSC and IFC that could be applied to the positioning device using the combined piezo-VCM actuator was successfully demonstrated. Compared with previous study using two sets of the hybrid actuator [8], only one single hybrid actuator was capable of obtaining the same positioning performance in this study. This was due to the PZT impact force coupled with the forward and backward presliding states of the sliding table. In addition, the IVSC used for the VCM control system showed good robustness in different actuating mechanisms provided in previous study and current study.

VII. Conclusion

In this paper, the IVSC and IFC were employed to demonstrate the control performance of a positioning device using the combined piezo-VCM actuator with a frictional constraint. An experimental setup with one-DOF of motion ability was configured to examine the fundamental characteristics and control performance of the positioning device. Main results were obtained as follows:

1) The control algorithm for the positioning device using the combined piezo-VCM actuator was configured with rough and precision positioning states. It was shown as effective in obtaining a large operational range and high-precision positioning ability.
2) In rough positioning state, using the IVSC for the VCM control system, the sliding table could move smoothly to the target position with a positioning accuracy of 10 μm and with a large travel range.
3) In precision positioning state, using the IFC for the PZT actuator coupled with a presliding state of the sliding table, a final positioning accuracy of 10 nm was obtained.
4) The IVSC showed good robustness in the VCM control systems with different actuating mechanisms.

It is shown that proposed positioning device featuring the performance of high-precision, and large operational range has attractive practical applications in the field of precision industry.

Acknowledgment

The authors are grateful to the National Science Council of the Republic of China for the support under Contract No. NSC-96-2628-E-327-003-MY2.

References

[1] Y.-T. Liu, “Nano-Positioning and Sensing Technologies,” *Encyclopedia of SENSORS*, American Scientific Publishers, ISBN: 1-58883-062-4, Vol.6, Me-N, 2006, pp.435-500.
[2] S. -H. Chang, C. -K. Tseng, and H. -C. Chien, “An ultra-precision XYθ_Z piezo-micropositioner part I: design and analysis, “ *IEEE Transaction on Ultrasonic, Ferroelectrics, and Frequency Control*, 46(4), 1999, pp. 897-905.
[3] T. Higuchi, M. Watanabe, and K. Kudoh, “Precise positioner utilizing rapid deformations of a piezo electric element,” *J. Jap. Soc. Prec. Eng.*, vol. 54, no. 11, 1988, pp. 75-80.
[4] C. -L Chu and S. -H. Fan, “A novel long-travel piezoelectric-driven linear nanopositioning stage,” *Prec. Eng.*, 30 (1), 2006, pp. 85–95.
[5] Y. -T. Liu and T. Higuchi, “Precision Positioning Device Utilizing Impact Force of Combined Piezo-VCM Actuator,” (in Japanese), *J. Japan Society Precision Eng.*, vol. 67, no. 1, 2001, pp.70-75.
[6] Y. -T. Liu and T. Higuchi, “Precision positioning device utilizing impact force of combined piezo-pneumatic actuator,” *IEEE/ASME Trans. on Mechatronics*, vol. 4, no. 6, 2001, pp. 467-473.
[7] Y. -T. Liu and C. W. Wang, “A self-moving precision positioning stage utilizing impact force of spring-mounted piezoelectric actuator,” *Sensors and Actuators*, A: Physical;102, Issues 1-2, 2002, pp. 83-92.
[8] Y. -T. Liu, R. -F. Fung, and C. -C. Wang, “Precision Position Control Using Combined Piezo-VCM Actuators,” *Precision Engineering*, Vol. 29, Issue 4, 2005, pp.411-422.
[9] P. Phakamach, S. Tiacharoen and C. Akkaraphong, “Control of an uninterruptible power system (UPS) using a feedforward integral variable structure controller,” in *Proc. IEEE Int. Conf. on Industrial Technology*, vol. 1, Dec. 11-14, 2002, pp. 618–623.
[10] P. Phakamach and C. Akkaraphong, “An optimal feedforward integral variable structure controller for the electrohydraulic position servo control systems,” in *Proc. IEEE Int. Conf. on Tencon*, vol. 4, Nov 21-24, 2004, pp. 459–462.