Design and Experimental Evaluation of An Electromechanical Engine Valve Drive

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***Abstract*— In conventional internal combustion (IC) engines, engine valve timing is fixed with respect to crankshaft angle. Flexibly controlled valve timing offers significant improvements in fuel efficiency, engine performance and emissions. One way to achieve variable valve timing (VVT) is by using an electromechanical valve drive (EMVD). In this paper, we describe the design and experimental evaluation of a new EMVD comprising an electric motor coupled by a nonlinear mechanical transformer to a resonant valve-spring system. Experimental results demonstrate that a 3.3 ms transition time adequate for 6 krpm engine speed, reasonable power consumption, and low valve seating velocity are obtained with the design.**

I. Introduction

In conventional IC engines, engine valves are actuated by cams that are located on a belt- or chain-driven camshaft. Each valve's kinematic profile (i.e., valve position versus time) is of fixed shape and timing relative to the engine crankshaft position. The shape of these cams is determined by considering tradeoffs between engine efficiency, torque requirement and maximum power. Optimized overall performance can only be obtained at certain operating conditions (traditionally at high speed, wide-open throttle operating conditions) [1]. Flexible engine valve timing can improve engine performance, increase fuel economy, and lower emissions at any point of the engine map [2], [3]. IC engines in which the timing of the valve transitions and durations with respect to crankshaft angle can be controlled are said to have variable valve timing (VVT). A new type of electromechanical valve drive (EMVD) realizing variable valve timing was proposed in [4], [5] and it was shown by theoretical analysis and preliminary simulation that the proposed EMVD exhibits a number of advantages over previous EMVD architectures [6], [7], [8], [9], especially in its ability to realize soft valve landing with reasonable power consumption and fast transition times.

In this paper, the design, simulation and experimental evaluation of a prototype EMVD are presented. Both the simulation and experimental results demonstrate the technical feasibility and high performance of the proposed EMVD. VVT is realized while maintaining the essential characteristics of a conventional IC engine valve drive, including a 3.3 ms transition time (the time period during which a valve moves from 5% to 95% of its whole stroke) suitable for 6 krpm engine speed, power consumption of no more than 180 W/valve, and valve seating velocity (velocity at which the valve engages its seat) of less than 30 cm/s.

This paper is organized as follows: in Section II, the general concept of the proposed EMVD is described with its mathematical model. Section III explains the different operational modes, control strategies and simulation results. The experimental set up is presented in Section IV. Section V describes the experimental results and their correlation with the simulations. Section VI concludes the paper.

II. General Concept of the Proposed EMVD

The proposed EMVD comprises an electric motor that is coupled to a resonant valve-spring system via a nonlinear mechanical transformer (NMT), as illustrated in Fig. 1. Rotation of the motor rotor in the θ domain is transformed into translation of the engine valve in the z domain.

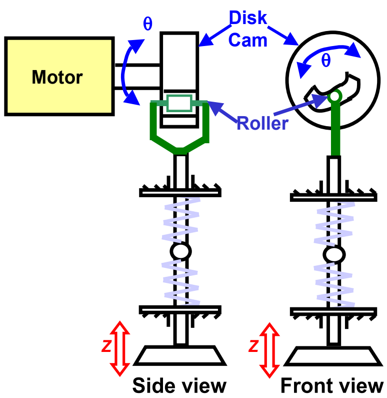

Fig. 1. A schematic of the proposed EMVD in which the NMT is implemented as a disk-cam with a nonlinear relation prescribed in its slot.

In this EMVD, the motor offers control over the whole stroke in both directions, providing the desired VVT behavior. The springs store and regenerate the inertial energy (from potential energy in the springs to kinetic energy in the valve) as the valve moves between the two ends of the stroke, sharply reducing the power requirement of the motor for producing a fast transition time. Most importantly, the characteristic of the NMT can be designed such that low seating velocity, zero holding power and low drive current can be achieved.

We now analyze the relationship created by the NMT. Assume the relation between displacements in the θ domain and the z domain is,

$$z = f(\theta) \tag{1}$$

then it is easy to show that the following relations of velocity and acceleration hold between θ and z,

$$\frac{dz}{dt} = \frac{dz}{d\theta} \cdot \frac{d\theta}{dt} \tag{2}$$

$$\frac{d^2z}{dt^2} = \frac{d^2z}{d\theta^2} \cdot (\frac{d\theta}{dt})^2 + \frac{dz}{d\theta} \cdot \frac{d^2\theta}{dt^2} \tag{3}$$

where $\frac{d\theta}{dt}$ is the rotational speed in the θ domain, $\frac{dz}{dt}$ is the velocity in the z domain, $\frac{d^2\theta}{dt^2}$ is the rotational acceleration, $(\frac{d^2z}{dt^2})$ is the acceleration in the z domain and $\frac{dz}{d\theta}$ is the slope of the NMT characteristic.

By assuming 100% efficiency of the NMT, we can equate the energy in both domains, yielding the relation between the torque in the θ domain and the force in the z domain as:

$$\tau_\theta = \frac{dz}{d\theta} \cdot f_z \tag{4}$$

where τ_θ is the input torque in the θ domain and f_z is the output force in the z domain.

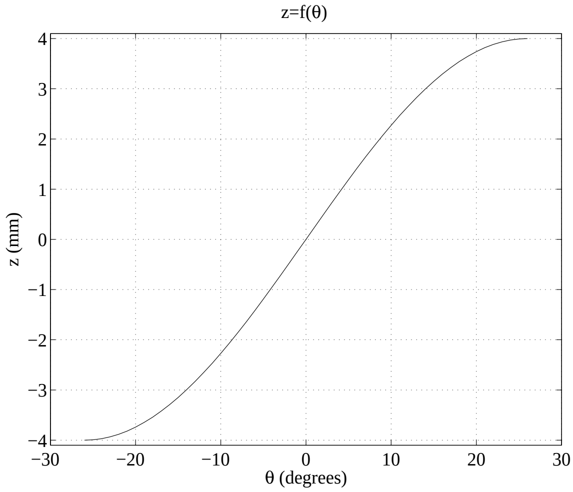

Fig. 2. A desirable transfer characteristic for the NMT. The slope of this NMT profile is the gear ratio of the proposed EMVD when viewed from the θ domain.

A key point of NTF profile design is that, at either end of the stroke, the slope of the NTF characteristic, $\frac{dz}{d\theta}$, will be very close to zero, as shown in Fig. 2. There are a number of advantages to system behavior, owing to this flatness. First, very low seating velocity in the z domain will be obtained even if the rotation speed is high in the θ domain, which is obvious from (2). Secondly, the large spring force at the ends of the stroke is reflected as a small torque in the θ domain for the same reason, as shown in (4), allowing the valve to be held at the ends of the stroke by a low current. As will be shown in the experimental section, with static friction at the end of the stroke, the valve can be held at full open or closed position without current input. Similarly, because the gas force on the exhaust valve is highest at the beginning of the valve closed-to-open transition, the gas force is also reflected as a small torque in the θ domain, reducing the peak torque requirement on the motor. Therefore, the NMT design promises low seating velocity and small holding current while reducing the motor torque requirement.

The equations of motion in both domains for the EMVD including friction and gas forces in both domains are shown below:

$$f_z = m_z \cdot \frac{d^2z}{dt^2} + b_z \cdot \frac{dz}{dt} + K_z \cdot z + F_{gas}(z) \tag{5}$$

$$J_\theta \cdot \frac{d^2\theta}{dt^2} + b_\theta \cdot \frac{d\theta}{dt} + \tau_\theta = K_\theta \cdot i \tag{6}$$

where $F_{gas}(z)$ is the gas force, J_θ is the inertia in the θ domain, m_z is the mass in the z domain, b_θ is the friction coefficient in the θ domain, b_z is the friction coefficient in the z domain, K_θ is the motor torque constant, K_z is the spring constant, θ is the displacement in the rotational domain, and z is the displacement in the vertical domain.

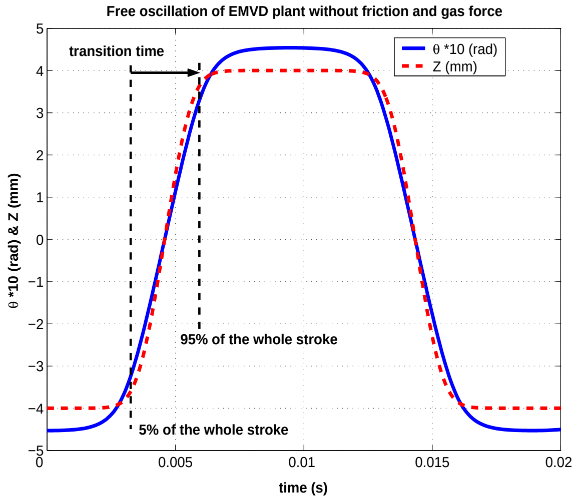

Fig. 3. Trajectory of free oscillation of the EMVD without friction or gas forces.

Fig. 3 shows the free oscillation trajectory of the EMVD, obtained by simulation, ignoring friction and gas forces. The system is a nonlinear mass-spring assembly because of the nonlinear transformation of the θ domain inertia into the z domain mass. But it is still a resonant mechanical mass-spring system having an equilibrium position in the middle of the stroke. The springs provide the inertial power needed to accelerate and decelerate the valve mass and motor inertia during each transition between closed and open positions. Hence, the main control strategy behind each transition is that a position feedback closed loop drives the motor to overcome the friction and gas forces such that the valve trajectory follows the trajectory of the ideal case shown in Fig. 3. This resonant operation greatly reduces the peak power requirement of the motor.

III. Three Operation Modes and Two Control Strategies

The operation of the EMVD can be broken into three modes: initial, transition and holding. In the initial mode, the valve is moved from its resting position (the middle of the stroke) to one end of the stroke (closed or open) upon system startup. The valve is moved from one end of the stroke to the other end during the transition mode, and is held at the arrival end during the holding mode. The three modes are illustrated in Fig. 4 [11].

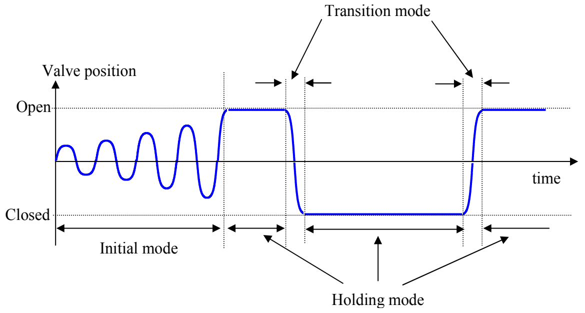

Fig. 4. The three modes of engine valve motion: initial mode, holding mode, transition mode.

Two controllers were designed for these three modes: one was designed to realize both transition and holding mode by shaping the reference input appropriately, while the other was designed to carry out initial mode control.

A. Transition Mode Control

As mentioned in Section II, the transition mode controller is a position feedback closed loop, shown in block diagram form in Fig. 5. The reference input is the desired motor position, and the system feedback signal is the actual motor position. The difference between the actual motor position and the reference position is passed into a controller which provides a current control input to the motor drive. The motor drive then supplies the desired current to the motor.

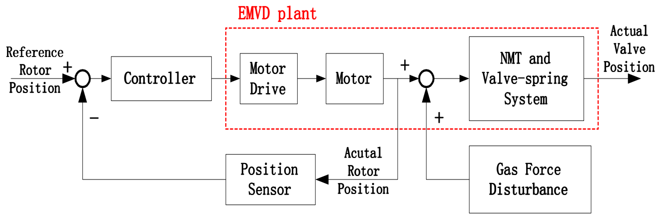

Fig. 5. Block diagram of the feedback-controlled EMVD apparatus. The EMVD plant is comprised of the motor drive, motor, disk cam, and the engine-valve spring system.

A lead compensator was designed in the prototype. The transfer function of the lead compensator is given by:

$$G(s) = K\frac{1+\frac{1}{Z}s}{1+\frac{1}{P}s} \tag{7}$$

where K is the controller gain, Z is the lead compensator zero location, and P is the lead compensator pole location. These three constants are initially chosen by approximate linear model analysis, which will be discussed below. Their final values are determined by simulation and experiment to obtain the desired transient response. An acceptable set of numbers is: $K = 250$, $Z = 1250$ rad/s, and $P = 5000$ rad/s.

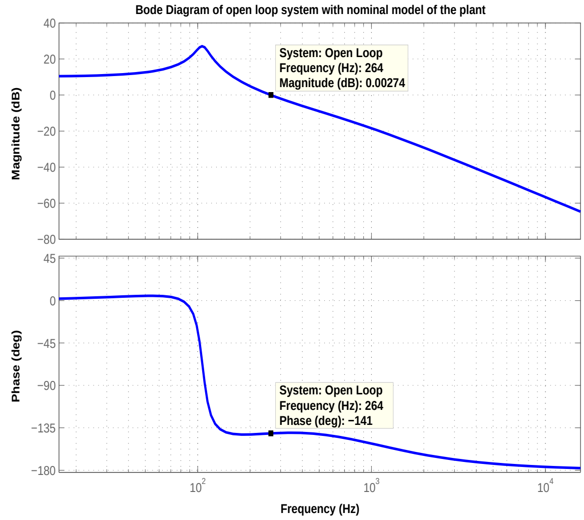

Fig. 6. Bode diagrams for the open loop system.

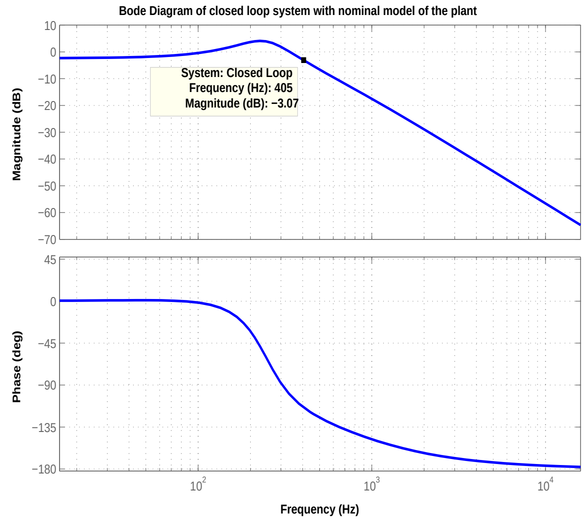

Fig. 7. Bode diagrams for the closed loop system.

To approximately evaluate the closed loop system behavior, we use a linear system analysis. We define the *nominal transformation ratio* as the full valve lift divided by the maximum rotation angle of the motor. If the transformer has this nominal ratio instead of the nonlinear ratio, then we call this system *a nominal model* of the actual system, which is treated as an average model of the nonlinear system in our control analysis. The Bode diagrams for the open loop and closed loop system with this nominal model are shown in Figs. 6 and 7 respectively. The open loop Bode diagram shows that the crossover frequency is about 264 Hz and the phase margin is about $39°$. The closed loop system behaves as a low-pass filter with a bandwidth of about 400Hz.

The objective of the holding mode is to hold the valve at either end of the stroke for a controllable time. This was done by shaping the reference input to stay constant when it reaches the maximum position in either direction, using the same control law as for the transition mode. The simulation and experimental results of transition and holding mode control, with a current limit due to motor thermal constraints is shown in Figs. 12 and 13, which will be discussed in section V.

B. Initial Mode Control

Upon system startup, the valve needs to be moved quickly from the rest position at the middle of the stroke to the open or closed position. We want to avoid requiring the motor to overcome the full static force of the springs directly. An effective strategy for achieving this is to drive the system at its resonant frequency. This is done by applying torque pulses to the EMVD, and changing the sign of these pulses at the moment the motor velocity changes its sign.

A control law implementing this technique is given by:

$$i = \alpha \cdot \omega / |\omega| \tag{8}$$

where ω is the nonzero angular velocity of the motor and α is a pre-determined constant. In a discrete implementation, i keeps its previous value whenever $\omega = 0$.

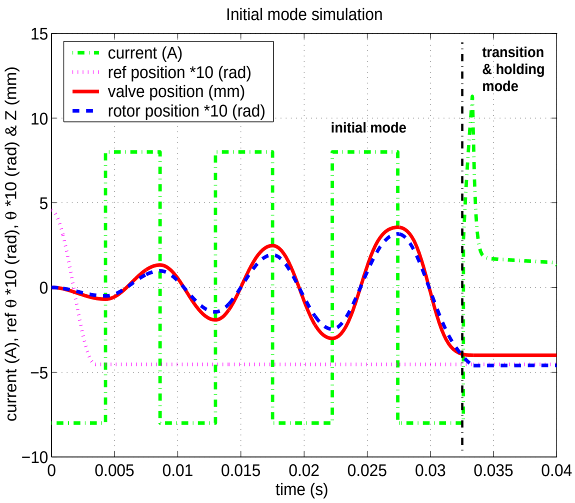

Fig. 8. Simulated initial mode position and current profiles.

Initial mode control will be switched to transition mode control to ensure a soft valve landing when the valve is near its fully open or closed positions. With the controller described above, the initial-mode control phase ends quickly after a few, small-magnitude current pulses. The simulation with $\alpha = 8$A is shown in Fig. 8, in which the valve is moved from its rest position to the fully open position in about 35 ms.

IV. The Experimental EMVD Apparatus

A single-valve experimental apparatus was designed and constructed, as shown in Figs. 9 and 10 [10], to validate that the proposed EMVD is a technically feasible VVT system, i.e., having transition times, power consumption and seating velocity comparable to conventional engine valve drives.

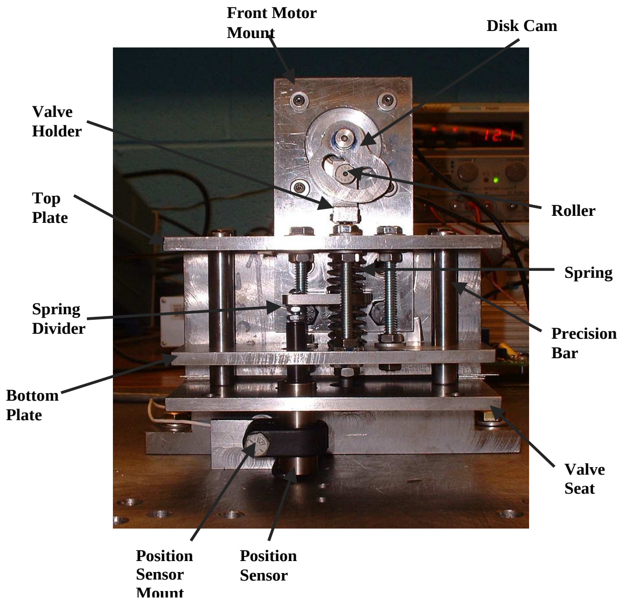

Fig. 9. Front view of the experimental EMVD apparatus, showing the disk cam and engine valve-spring system. The precision bars are used to align the valve-spring system.

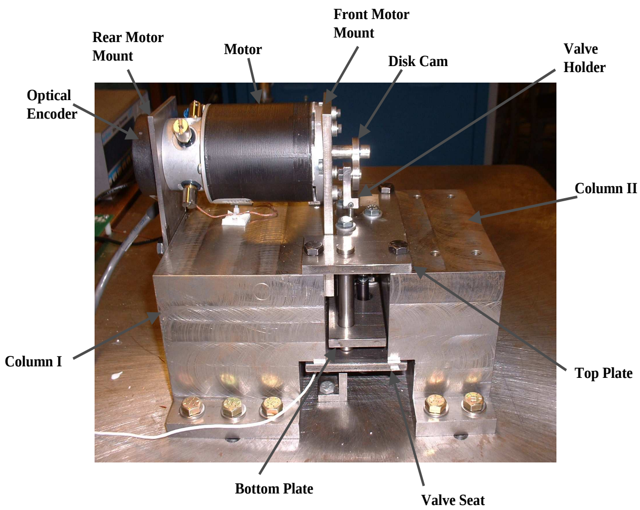

Fig. 10. Side view of the experimental EMVD apparatus.

A standard exhaust valve from a Ford ZETEC 2.0L engine was used in the prototype. Springs producing a combined stiffness of 112 N/mm (56 N/mm each), were selected to meet the required transition time of less than 3.5 ms necessary for 6 krpm engine speed.

The NMT is implemented as a disk cam with a slot in which the nonlinear mechanical characteristic shown in Fig. 2 is realized. As the motor shaft rotates, the disk cam, which is rigidly connected to the motor shaft, also rotates, causing the roller to move within the cam slot. As the roller moves, the engine valve, which is connected to the roller by the valve holder, moves up or down. The valve is free to move vertically but constrained from lateral motions. At either end of the valve stroke, the roller lies on a nearly flat surface at the end of the disk cam slot. At this point static friction holds the valve open or closed without the need for any electrical input to the motor.

A high torque low inertia motor (Pacific Scientific 4N63-100-1) is used in the EMVD apparatus. A 10 kHz bandwidth 1 kW motor drive was developed for the application. It uses a full-bridge inverter topology and a hysteresis current control strategy [5]. We use a 2048-line differential optical encoder in the θ domain to sense motor position for use in the feedback controller. A linear position sensor is installed to track valve position in the z domain. Motor and valve displacements are provided to the ADC channels on a dSPACE DS 1104 digital signal processing board. The position reference, initial mode controller and lead compensator were implemented in MATLAB's Simulink model and run on the dSPACE board, providing a current command to the motor drive circuit via a DAC channel.

V. Experimental Results

System identification, including free oscillation with no current input and open loop system testing, was carried out to obtain a more accurate transfer function and system parameters before the feedback control system was tested. Also, soft springs were used to pretest the system. The details of these experiments are discussed in [10], [11]. The experiments employing the stiff springs, suitable for an engine speed of 6 krpm, are presented in this section, including startup operation (initial mode) and normal operation (transition and holding mode). The experimental data was captured on an oscilloscope, transferred to the computer by LabView software and then processed and plotted using MATLAB.

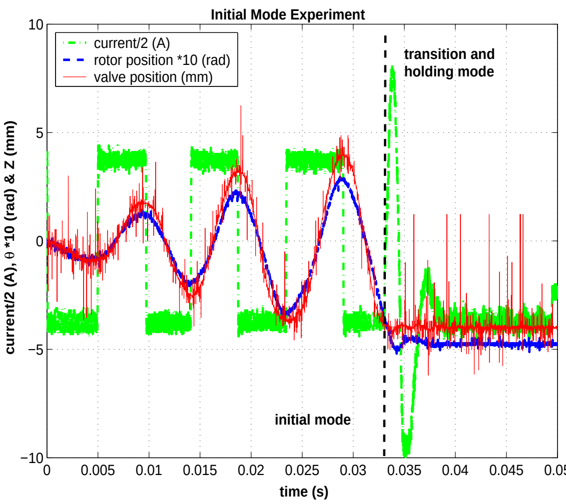

Fig. 11. Experimental results for the initial mode. The initial mode control transient was completed in 35 ms with ± 8 A current pulses.

A. Initial Mode Behavior

Figure 11 shows that the initial mode is completed within 35 ms with ± 8 A current pulses, which is sufficiently fast for practical applications and matches well the simulation results shown in Fig. 8. The average power during the initial mode period is approximately 150 W [11].

When the valve was close to the desired end, the initial mode control was switched to the transition/holding mode control to ensure a soft valve landing. After reaching the stroke end, the valve was retained in the holding mode. A non-zero holding current was required by the controller due to the non-zero static position error, as shown in Fig. 11. However, in the laboratory, we observed that after turning off the motor drive, the valve was still held at the end owing to static friction, demonstrating the zero holding current/power feature of the proposed EMVD, as expected.

B. Transition and Holding Mode Behavior

The key performance parameters of the transition mode are transition time, valve seating velocity, and power consumption. Figure 12 shows the valve and motor position profiles both from the simulation and the experiment, while the corresponding current profiles are shown in Fig. 13. Again, the simulation data and experimental data match quite well. And zero current/power holding was also observed in this experiment after turning off the motor drive.

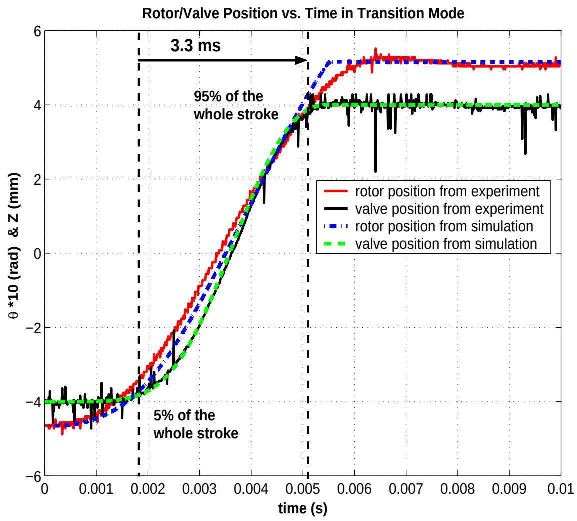

Fig. 12. Rotor and valve position profiles during the transition mode, from both simulation and experimental data.

In the experiment, we limited the motor drive current to ± 18 A due to thermal constraints on the motor, and thus we saw a saturation in motor current during the valve transition. The transition time, defined as the period during which the valve moves from 5% to 95% of its travel, was approximately 3.3 ms, which is adequate for 6 krpm engine speed. The valve seating velocity was measured by a high speed camera repeatedly and ranged from 15.5 to 27.2 cm/s, which is comparable to the seating velocity of 30 cm/s at 6 krpm in a conventional valve drive.

From the experimental data we also estimated the average power required for one transition. The consumed energy is

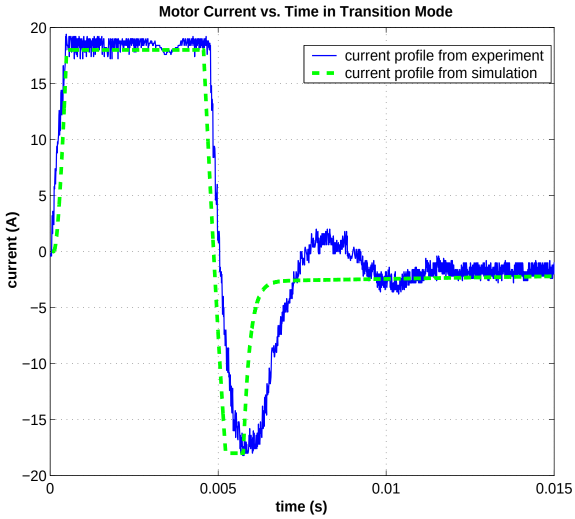

Fig. 13. Current profiles during the transition mode, from both simulation and experimental data.

approximately 1.4 J per transition [11]. At 6 krpm engine speed, the nominal time for one complete cycle of valve operation is 20 ms , including closed-to-open, open-to-closed and holding after each transition. Therefore the average power at 6 krpm is 140 W (2.8 J divided by 20 ms) per valve, since the holding power is negligible as mentioned before. This value applies to the intake valve because gas force was not included in the experiment. The exhaust valve needs an additional 0.8 J per cycle per valve at conditions of 6 krpm and full load to compensate for the gas force in the closed-to-open transition [12]. Hence, the average power is 180 W (3.6 J divided by 20 ms) per exhaust valve at 6 krpm engine speed under wide-open-throttle conditions. Therefore, 2.56 kW is required under these conditions for a 4-cylinder, 8-intake valve and 8-exhaust valve engine, which is comparable to the power requirement of a conventional valve train for the same conditions (about 3 kW). However, for simplicity we used a sinusoidal position reference in our transition mode experiment. We expect lower power consumption if we use the free oscillation trajectory shown in Fig. 3 as the reference, since with a sinusoidal reference, the controller will require more power to make the valve depart from its natural trajectory.

VI. Conclusions

In this paper, we present the design, simulation and experimental evaluation of a new type of EMVD. The results confirm that the proposed EMVD achieves variable valve timing while maintaining the essential characteristics of internal combustion engine valve drives. Several key accomplishments include: (1) a 3.3 ms transition time, fast enough for 6 krpm engine speed; (2) low valve seating velocity, less than 30 cm/s; (3) zero power loss in the holding mode and acceptable power consumption in the transition mode.

Acknowledgment

We would like to thank the Ford Motor Company and the Eaton Corporation for the donation of two engine heads. We also thank Mr. George Yundt and Mr. Bill Fejes of Danaher Motion Corporation, Dr. David Turner of Eaton Corporation, and Dr. Bruno Lesquesne of Delphi Automotive Systems for their advice. This work is funded by a grant from the Sheila and Emanuel Landsman foundation.

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