

Nonlinear System Modeling, Optimal Cam Design, and Advanced System Control for an Electromechanical Engine Valve Drive

Yihui Qiu, *Member, IEEE*, David J. Perreault, *Senior Member, IEEE*, Thomas A. Keim, *Member, IEEE*, and John G. Kassakian, *Life Fellow, IEEE*

Abstract—A cam-based, shear force actuated electromechanical valve drive system offering variable valve timing in internal combustion engines was previously proposed and demonstrated. To transform this concept into a competitive commercial product, several major challenges need to be addressed, including the reduction of power consumption, transition time, and size. As shown in this paper, by using nonlinear system modeling, optimizing cam design, and exploring different control strategies, the power consumption has been reduced from 140 W to 49 W (65%), the transition time has been decreased from 3.3 ms to 2.7 ms (18%), and the actuator torque requirement has been cut from 1.33 N-m to 0.30 N-m (77%).

Index Terms—Open-loop control, optimal cam design, nonlinear friction model, variable valve timing.

I. INTRODUCTION

Despite their simple design and low cost, conventional crankshaft-synchronized cam-driven valve actuation systems can optimize engine performance at only one point on the engine torque-speed operating map. This is due to the fixed valve timing with respect to crank shaft angle independent of different operation conditions. But research has shown that variable valve timing (VVT) can achieve higher fuel efficiency, lower emissions, higher torque output, and throttleless engine control [1]–[6].

Based on the study of previous variable valve actuation (VVA) mechanisms [1]–[12], an electromechanical valve (EMV) system incorporating shear force actuation and nonlinear displacement transformation over a limited angular range was proposed in 2002 [13]. The fundamental contribution of the proposed EMV system is the capability of inherent soft landings at the ends of transitions and zero power consumption between valve transitions owing to a deliberately nonlinear mechanical transformer (NMT). Conceptual feasibility has been demonstrated previously [14][15].

Manuscript received August 06 2010 and revised Dec 01 2010 and Mar 07 2011 respectively. This project was funded by the Sheila and Emanuel Landsman Foundation.

Yihui Qiu, David J. Perreault, Thomas A. Keim, and John G. Kassakian are with the Laboratory for Electromagnetic and Electronic Systems at Massachusetts Institute of Technology, Cambridge, MA, 02139 USA (corresponding author: Yihui Qiu; phone: 617-372-1340; e-mail: qiu@alum.mit.edu, djperrea@mit.edu, tkeim@mit.edu, jgk@mit.edu).

However, to demonstrate the practicality of the proposed system for real engines, several significant technical barriers need to be overcome, including lowering power consumption, reducing actuator size, and providing transition times fast enough for engine speed over 6000 rpm. In this paper, based on nonlinear system modeling, optimizing the NMT design, and a redesigned control algorithm, the power consumption has been reduced from 140 W to 49 W (65%), the transition time has been decreased from 3.3 ms to 2.7 ms (18%), and the torque requirement has fallen from 1.33 N-m to 0.30 N-m (77%). The substantial reduction in power and torque enables the utilization of a much smaller actuator.

We begin this paper with a discussion of the background and motivation of different VVA systems, especially the proposed EMV system in section II. Section III describes the nonlinear friction model, based on which improved system performance is achieved with a more appropriate control strategy. Section IV presents the optimal NMT design, while different control strategies are reported in section V. Section VI concludes the paper.

II. BACKGROUND AND MOTIVATION

In conventional IC engines, the valves are actuated by cams located on a belt- or chain-driven camshaft. As a long developed valve drive, the system has a simple structure, low cost, and offers smooth valve motion. However, the valve timing of the traditional valvetrain is fixed with respect to the crankshaft angle because the position profile of the valve is determined purely by the shape of the cam. If instead, the valve timing can be decoupled from the crankshaft angle and adjusted adaptively for different situations, then the engine performance can be optimized with respect to higher torque/power output, increasing fuel economy, and reducing emissions, at any point on the engine map. According to published research, the main benefits of VVT can be summarized in specific numbers: a fuel economy improvement of approximately 5–20%, a torque improvement of 5–13%, and an emissions reduction of 5–10% in HC, and 40–60% in NO_x [1]–[7].

There are three main categories of VVA: pure mechanical [6], [8], [10], electro-hydraulic [1], and electromechanical [4], [9]–[12]. The various mechanical actuators are mainly

improved designs based on the current valvetrain. These drives are usually simple and widely accepted. But the control flexibility is still very limited and discrete, compared to the ultimate goal of continuously adjusted valve timing for each valve independently.

The electro-hydraulic device, on the other hand, offers much more flexibility in terms of VVT control. But the use of a hydraulic system makes it expensive and cumbersome [1], compromising its practicality for automotive application.

The concept of electromechanical actuation has become more feasible and attractive recently owing to its simple structure, continuous VVT control, and independent action for each valve and each cylinder. A normal force actuated EMV system proposed by Pischinger and its close variants have become a popular research topic and has gotten closest to real engine application [9]-[12]. As shown in Fig. 1, the EMV system proposed by Pischinger et. al. [9], consists of two normal force actuators and a spring-valve system.

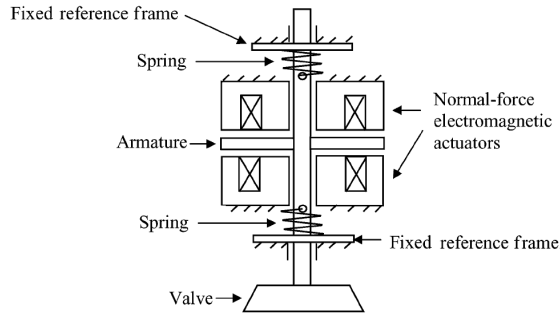


Fig. 1. The normal force actuated EMV system proposed by Pischinger.

The springs are introduced into the system to provide the large force needed for valve acceleration and deceleration during each transition. The force requirements of the actuator are thereby substantially reduced. However, it is difficult to achieve soft landing, i.e., low valve seating velocity at the end of transitions with this system because the normal force actuators are unidirectional with a nonuniform and nonlinear force versus valve position, making control at valve excursion ends very difficult. Soft landing is very critical in terms of acoustic noise and lifespan of the valves. This situation could be more severe in the presence of a high gas force, as will occur with an exhaust valve. In recent literature, soft landing has been first achieved by a complicated nonlinear control scheme without considering gas force [11], and most recently improved algorithms have been reported which show better performance in terms of soft valve landing even in the presence of combustion forces [14][15].

But back to 2002, Dr. Woo Sok Chang and his colleagues at MIT proposed an electromechanical valve drive incorporating a nonlinear mechanical transformer in order to achieve inherent soft landing without complicated control [13]. This EMV system inherits the valve-spring system and its regenerative benefits from the normal force actuated EMV system discussed above, while using a bi-directional shear force actuator with a uniform torque constant. As shown in Fig. 2, the motor shaft is connected to the valve-spring system via a nonlinear mechanical transformer (NMT). The NMT is

implemented by a slotted cam and a roller follower in the slot which are connected to the motor shaft and the valve stem respectively, as shown in Figs. 3 and 4. When the motor swings back and forth within the angle range limited by the cam slot design, the roller follower moves back and forth within the slotted cam, allowing the valve to move up and down between fully open and fully closed positions.

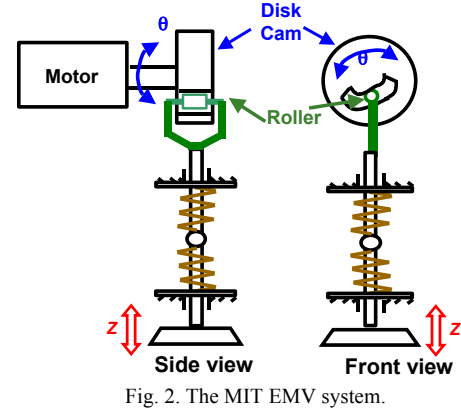


Fig. 2. The MIT EMV system.

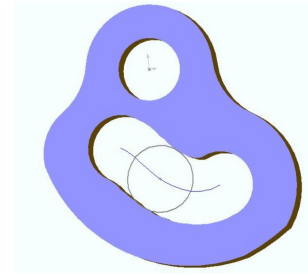


Fig. 3. The nonlinear mechanical transformer.

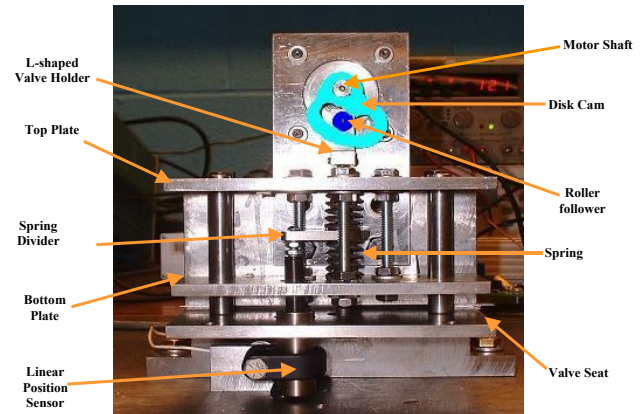


Fig. 4. Front view of the system including motor and spring assembly.

The mechanical transformer with an intentionally nonlinear characteristic leads to attractive features including soft landing at the end of transitions, low torque requirement during transitions, and zero power consumption between transitions [13][16]-[19]. A key feature of the NMT is zero-slope at both ends of the stroke. A sinusoidal function with an effective rotation range of ± 26 degrees was initially chosen by Chang for the desired NMT characteristics, as given by (1) and illustrated in Fig. 5. Note that it will have infinite jerk at the end of the sinusoidal profile. In reality, this would not pose a huge problem, since the cam velocity at the endpoints is low.

$$z = f(\theta) = 4\sin(3.46\theta) \text{ mm}; \theta \leq 26^\circ \quad (1)$$

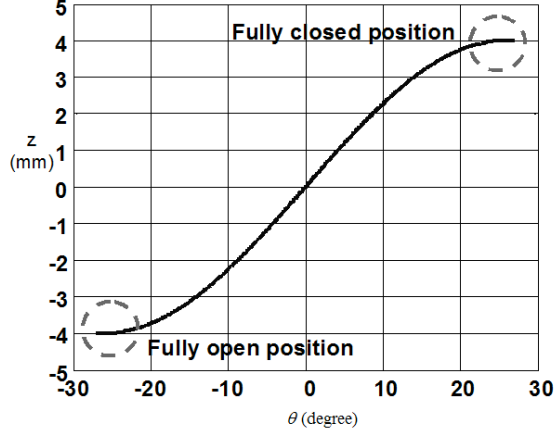


Fig. 5. The initial transfer characteristic for the NMT.

The concept of the proposed EMV system was proved experimentally with an off-the-shelf PM DC brush motor. A position feedback closed-loop control strategy was chosen. The block diagram of the closed-looped EMV system is shown in Fig. 6, where the controller is a lead compensator [16]. The reference position input is a position trajectory, which includes the desired starting and stopping motor positions, corresponding to the valve closed and open positions, respectively. For simplicity a 180° section of a sinusoidal function (from negative peak to positive peak or vice versa) was used as the position reference. The peak-to-peak value is the rotation range, while the frequency is chosen to meet the transition time requirement.

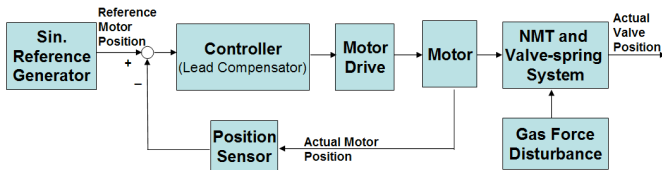


Fig. 6. Block diagram of the closed-looped EMV system.

The position and current profiles from experiment are shown in Figs. 7 and 8 [16]-[19]. In the experiment, the motor drive current is limited to ± 18 A due to thermal constraints on the motor. The most important performance parameters in an engine valve drive are transition time, valve seating velocity, and power consumption, as listed in Table I [16]-[19]. The transition time, defined as the period during which the valve moves from 5% to 95% of the full stroke (8 mm), was approximately 3.3 ms. This is faster than the target 3.5 ms for 6000 rpm engine speed. The valve seating velocity was measured by a high-speed camera with a mean of 21.3 cm/s [16]-[19]. This is less than the seating velocity of 30 cm/s at 6000 rpm in a conventional valve drive, and therefore we have achieved soft landing under such circumstances. Note that seating velocity at this level is acceptable at high engine speeds, but not at idle, compared to the seating velocity of 5cm/s at engine idle achieved by traditional mechanical systems. While this remains a subject of future improvements, Seethaler et. al. have recently addressed this problem for a

similar valve drive system [20]. An average power per valve per half cycle at 6000 rpm was estimated as of 140 W [16]-[19]. Note that all power numbers in this paper are averaged power over half cycle at 6000 rpm engine speed (10ms). The peak torque shown in Table I is not necessarily important for engine performance, but is very crucial as a metric for the size of the motor. For this reason, it is included in the table for future comparison.

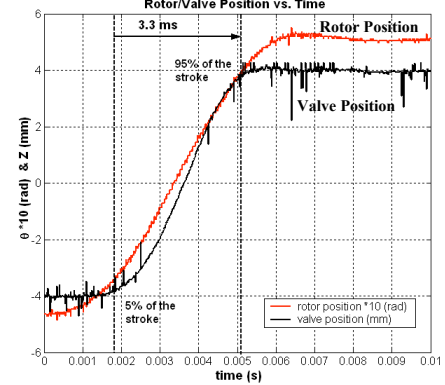


Fig. 7. Rotor and valve position profiles during one transition.

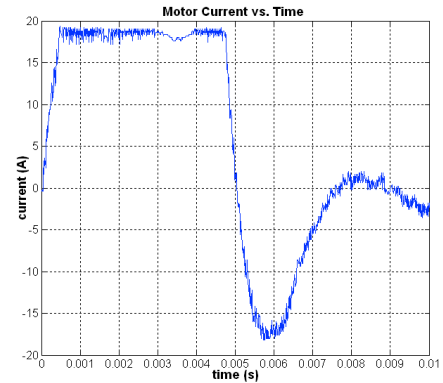


Fig. 8. Current profile during one valve transition.

The experiments confirmed that the system offers consistent VVT with an expected soft landing up to an engine speed of 6000 rpm. However, in order to supply the large power and high torque shown in Table I, a motor of large size and high nominal voltage (42V) was necessary in our prototype, which would be impossible to fit into an engine. We also want to minimize power consumption to be more competitive with other valve actuators and to improve fuel economy. In addition, a faster transition time is very desirable since some modern engines are targeting engine speeds higher than 6000 rpm.

In this paper, based on an improved nonlinear friction model, an optimized cam profile design and different control strategies, power consumption and torque requirement are greatly reduced while faster transitions are achieved. In two papers published separately, we report a novel design and fabrication of a limited-angle actuator for this application in order to tackle the size issue.

TABLE I
PRELIMINARY EXPERIMENTAL RESULTS.

Power Consumption (W)	Peak Torque (N-m)	Transition Time (ms)	Mean Seating Velocity (cm/s)
140	1.33	3.3	21.3

III. AN IMPROVED NONLINEAR FRICTION MODEL

In this section, the nonlinear friction model will be introduced in subsection A. Subsections B and C present a new position reference and the addition of a feed-forward control component. Improved system performance is then predicted and verified.

A. Nonlinear Friction Model

Friction forces play an important role in the EMV system dynamics. In particular, the frictional loss is one of the two main loss sources in the system, electrical loss being the other. Therefore, an accurate model to describe the friction forces in the system is necessary.

In Chang's preliminary analysis, it was assumed that friction could be represented by viscous friction with constant coefficients in both domains [14], as shown in (2).

$$\begin{aligned} f_{\theta} &= B_{\theta} \frac{d\theta}{dt} \\ f_z &= B_z \frac{dz}{dt} \end{aligned} \quad (2)$$

However, based on this friction model, the simulation results could not predict the experimental results well, suggesting the necessity of an improved understanding of friction. We reasoned that this probably is due to the lack of consideration of friction between the roller follower and the cam slot, which is quite complicated because the normal force at each contact point along the transition is different as a result of different slopes, spring forces and inertia forces at different points. Therefore, in our analysis, we keep the assumption that the friction in the z -domain is only viscous friction proportional to the valve velocity, as shown in (2). In the θ -domain, we keep the term proportional to rotation speed to account for windage and contact friction from the shaft bearing but we introduce two additional terms representing viscous friction and coulomb friction between the roller follower and the cam. Therefore, a friction model taking all these effects into consideration is proposed, as shown in (3), with friction due to rotational acceleration and inertia effects of roller and vibration effects of springs assumed to be small enough to neglect:

$$f_{\theta} = B_{\theta} \frac{d\theta}{dt} + F_n \cdot (B_{\theta v} \frac{d\theta}{dt} + B_{\theta c} \operatorname{sgn}(\frac{d\theta}{dt})) \quad (3)$$

In this model, part of the viscous friction, mainly due to windage and contact friction from the shaft bearing, is independent of spring force or normal force, and therefore has a constant coefficient B_{θ} , while the other component of viscous friction, $B_{\theta v} \cdot d\theta/dt$, and coulomb friction, $B_{\theta c} \cdot \operatorname{sgn}(d\theta/dt)$, depend very much on the normal force F_n exerted at the contact surface between the disk cam and roller follower.

The normal force exerted on the rolling surface varies along the valve transition because it is affected by multiple factors, as shown in (4),

$$F_n = (F_s - f_z - f_m) \cdot \cos(\alpha) \quad (4)$$

where $F_s = K_s z$ is the spring force, $f_z = b_z \cdot dz/dt$ is the friction force in the z -domain, $f_m = m_z \cdot d^2z/dt^2$ is the inertial force, and α is the angle between the tangent to the contact surface presented by the line C-C and the horizontal plane which is perpendicular to vertical valve motion presented by the line A-A, as shown in Fig. 9. For a given NMT function, the cam slot surface slope, $\tan(\alpha)$, and its variation, $\cos(\alpha)$, are derived in [21] and [22], which turn out to be nonlinear functions with respect to position.

Obviously, this friction model in the θ -domain is a nonlinear model, not only because the coulomb friction is related to the direction of the valve velocity, but also because a large portion of the friction is related to a varying normal force which is a nonlinear function with respect to θ or z . If we want to summarize the total frictional force in either the θ or z domain, then due to the nonlinear translation of the NMT, an additional layer of nonlinearity is added to the friction model. Therefore, in order to identify system parameters and predict system performance, we have to rely heavily on numerical simulation owing to the inherent nonlinearity of the whole system. The setup of the numerical simulation, experimental apparatus, extraction of friction coefficients, and validation of the proposed nonlinear friction model by comparing simulation to experimental results is discussed in [21] and [22]. Further investigation shows that the nonlinear friction force portion in the θ -domain is a dominant part compared to the linear friction portions both in the θ -domain and in the z -domain.

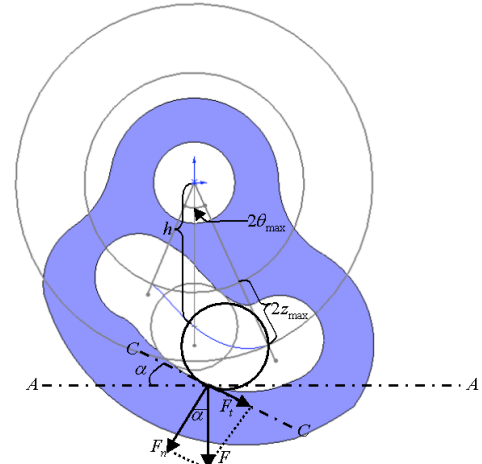


Fig. 9. The NMT used in the prototype.

B. Free-flight Trajectory as Position Reference

As shown in Fig. 8, in the first half of the transition the motor current reaches a positive saturation limit while trying to inject as much energy into the system as possible to accelerate the valve, while in the second half the current reaches the negative limit, trying to pull out as much energy as possible from the system to decelerate the valve. The winding

resistance loss associated with this reactive energy exchange is substantial. The mechanical power loss and the peak torque requirement can also be larger than necessary due to the increased frictional loss and unnecessary fighting with the springs. One reason for this energy waste is the convenient but not optimized sinusoidal position reference, which causes the motor to initially over-accelerate and then over-decelerate.

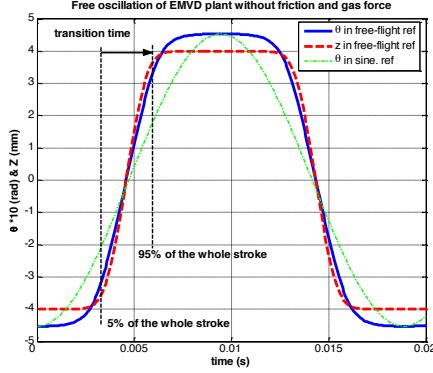


Fig. 10. Free-flight trajectory of the EMV system from simulation.

We believe the free-flight trajectory could be a better reference in order to take full advantage of the springs and reduce the wasted energy discussed above. Figure 10 shows the original sinusoidal reference and the free oscillation trajectory of the EMV system from simulation. Figures 11 and 12 show the simulation and experimental results using the new free-flight trajectory as the position reference. The current profile was improved significantly compared to that with the sine trajectory as the position reference shown in Fig. 8. An identical transition time of 3.3 ms is achieved while the power consumption is reduced from 140 W to 99 W and the peak torque is lowered from 1.26 N-m to 1.05 N-m.

C. A Touch of Feed-forward Control

It is obvious from Figs. 11 and 12 that the simulation predicts the system dynamics very well with the nonlinear friction model. It enable us to calculate total friction torque in the θ -domain, including the friction torque originally generated in the θ -domain and the friction torque reflected into the θ -domain from the z -domain. As shown in Fig. 13, the simulation results indicate that despite all the power/torque reduction with the free-flight trajectory as the position reference, the first over-drive then slow-down problem still exists and so does the unnecessary electrical and frictional losses associated with it. Note that the total friction torque seen by the motor happens to be more or less like a square wave, therefore a touch of feed-forward control --- a current limit matching the square-waved friction torque --- was added to improve this situation. Fig. 14 shows the simulation results with an 8 A current limit. A much lower peak torque and rms torque requirement has been achieved, suggesting much less energy waste and hence a much lower power consumption, which is verified experimentally, as shown in Figs. 15 and 16.

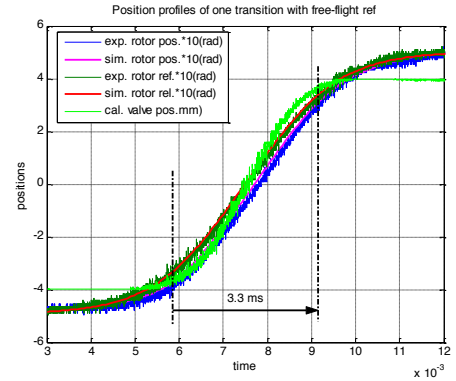


Fig. 11. Simulated and experimental position profiles with free-flight position reference.

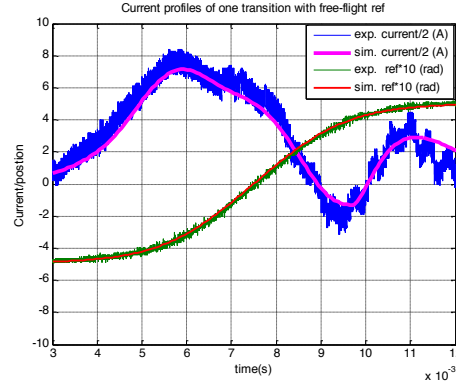


Fig. 12. Simulated and experimental current profiles with free-flight position reference.

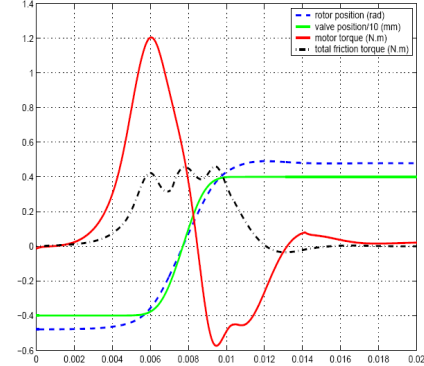


Fig. 13. Simulated results with free-flight reference, no current limit.

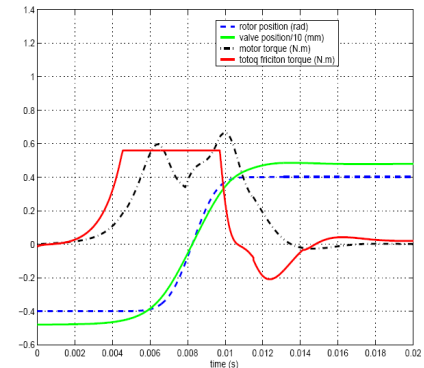


Fig. 14. Simulated results with free-flight reference, 8 A current limit.

The power consumption is reduced further from 99 W to 76 W, peak torque is now reduced from 1.05 N-m to 0.56 N-m while the transition time is 3.4 ms, only a bit slower than

previous experiments. The comparison with different control elements is summarized in Table II below.

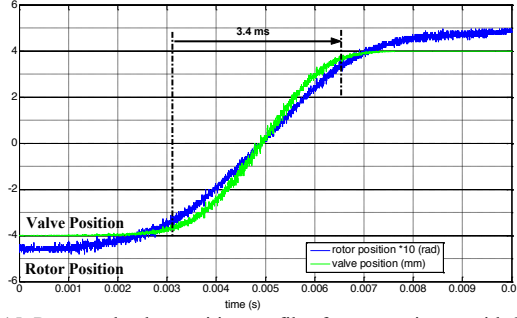


Fig. 15. Rotor and valve position profiles from experiment with 8 A limit.

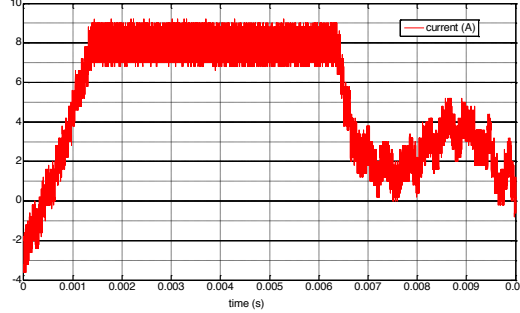


Fig. 16. Current profile from experiment with 8 A limit.

TABLE II
COMPARISON WITH DIFFERENT CONTROL STRATEGIES.

Operation Conditions	Power Consumption (W)	Peak Torque (N-m)	Transition Time (ms)
W/ sin ref and 18A limit	140	1.33	3.3
W/ free-flight ref and no current limit	99	1.05	3.3
W/ free-flight ref and 8A limit	76	0.56	3.4

IV. AN OPTIMAL CAM DESIGN

One of the key advances reported in this paper is the determination of the optimal NMT design. Changing the NMT design will affect the EMV system performance in several ways. First, the friction force will be different in both strength and distribution, resulting in a different motor torque as well as power consumption for valve transitions. Second, the natural frequency of the system may also change, not only because the transformer may have a different moment of inertia but also because the total inertia in the θ -domain will be translated into a different value of mass in the z -domain due to the different transformer ratios of the new NMT design. This change will have an impact on the transition time of the system. It will also affect the friction force related to velocity and hence the torque requirement and power consumption. Therefore, we optimize the design with respect to minimized torque, power and transition time.

A. Two Possible Directions for a Better Design

As discussed in Section II, the current cam presents a sine relation between rotor displacement θ and valve displacement z . As shown in (1), a motor rotation of $\pm 26^\circ$ provides 8 mm

of vertical stroke. Note that in this relation, only the length of the stroke is fixed by the application. Both the function between θ and z as well as the rotating range of the motor in the θ -domain can be designed differently, representing two study directions for designing a better NMT. We refer to these two directions as “function optimization” and “ θ -range optimization” in this paper. Since simulation results showed little performance differences among different cam functions [21][22], our attention is focused on the θ -range optimization.

B. θ -range Optimization

To clarify the intuition behind θ -range optimization in the sense of minimum power consumption and torque requirement, let's begin with a linear transformer which has a constant modulus, rather than a varying one, along the stroke. If $\theta_{\max}$ is the half range of rotor rotation and $z_{\max}$ is the half range of valve lift, then we define the nominal modulus M as in (5),

$$M = z_{\max} / \theta_{\max} \quad (5)$$

Therefore, we can write the transformer equations as below,

$$z = M \cdot \theta \quad (6)$$

$$dz/dt = M \cdot d\theta/dt \quad (7)$$

$$\tau_\theta = M \cdot f_z \quad (8)$$

$$m_\theta = J_\theta / M^2 \quad (9)$$

where θ and z are rotor displacement in the θ -domain and valve displacement in the z -domain respectively, τ_θ is the torque in the θ -domain, F_z is the force in the z -domain, J_θ is the inertia in the θ -domain, and m_θ is the equivalent mass reflected into the z -domain.

From the equations above, we can make two simple arguments. First, if given fixed valve lift and fixed transition time requirements, the average valve velocity during one transition is also fixed. From (7), we will have a smaller average rotating velocity of the rotor with a bigger M and vice versa. This will affect those friction forces that are functions of rotating velocity and hence the torque and power needed to complete a transition. Second, from the perspective of the springs, the total driven mass includes the mass in the z -domain and the reflected mass of the inertia in the θ -domain. From (9), a bigger M results in a smaller reflected inertial mass and hence a smaller total effective mass in the z -domain. Therefore with a given spring constant, the natural frequency of the whole system will be higher, which will very likely result in a faster valve transition. Alternatively, a lower spring constant can be specified, with a possible further reduction of friction.

The impacts on system performance of different ratios M can be more complicated because they will also change the normal force F_n at the rolling contact between the roller surface and cam slot surface. As discussed previously, the normal force F_n is determined by several factors including spring force F_s , z -domain friction force f_z , z -domain inertial force f_m , and modulus M , as shown in (10),

$$\begin{aligned} F_n &= (F_s - f_z - f_m) \cdot \cos(\alpha) \\ \alpha &= \text{atan}(M) \end{aligned} \quad (10)$$

As discussed above, the reflected mass m_θ from the θ -domain inertia J_θ will be smaller if given a bigger ratio M . The decreased m_θ means the inertial force offered by the springs to the cam and rotor, $(F_s - f_z - f_m)$, will decrease, too, if $(F_s - f_z)$ remains unchanged. Furthermore, a bigger cam ratio M will result a smaller slope factor $\cos(\alpha)$. Both changes will reduce the normal force F_n and hence the related friction force.

On the other hand, given the constant travel distance of the valve and the faster transition time, a higher valve velocity and therefore a larger viscous friction forces f_z in the z -domain can be expected. At the same time, from (16), we can tell that this increased valve friction f_z will reduce the normal force F_n and, accordingly, the nonlinear friction force portion in the θ -domain. In other words, with increasing M , the θ -domain nonlinear friction force will decrease while the valve friction force f_z will increase. Additionally, with M increasing, the same amount of friction force in the z -domain will be reflected into a bigger friction torque in the θ -domain, as shown in (8). As a result, when increasing M to a certain point, from the actuator's perspective, the nonlinear friction force portion in the θ -domain will no longer dominate the linear friction force f_z in the z -domain and a minimum friction torque in total can be expected. If we increase M beyond that point, the total friction torque seen by the actuator will begin to increase again.

Therefore, with a bigger M , a higher natural frequency and a shorter transition time is expected; moreover, with a bigger M , the nonlinear friction force in the θ -domain will decrease while the linear friction force in the z -domain will increase. There should be at least one M that will give us the minimum value of total friction force/torque, resulting in the lowest torque requirement and power consumption. Therefore, for the case of a linear transformer, it is possible to find the best ratio M which gives a faster transition with the lowest power consumption and torque requirement. We believe that these conclusions based on a linear transformer apply equally to a nonlinear transformer, except that due to the inherent heavy nonlinearity of the system, we have to rely on a series of numerical simulations to confirm and locate the optimal M , i.e., θ range.

C. An Optimal NMT Design

With the direction pointed out by the analysis above, we conducted a series of numerical simulations for our EMV system using a nonlinear transformer that maintains a sinusoidal relation between a fixed valve lift of 8 mm and an angular range varying from $\pm 26^\circ$ to $\pm 7.5^\circ$ in the θ -domain. The simulation results of peak torque, rms torque and transition time are shown in Fig. 17, from which we can see

that the transition time decreases with decreasing angular range. On the other hand, the peak torque and rms torque both first decrease and then increase with decreasing angular range. The simulation results confirm the previous analysis and the local optimal θ -range is $\pm 15^\circ$. Therefore, the desired z vs. θ characteristics can be described as in (11).

$$z = g(\theta) = 4\sin(6\theta) \text{ mm} \quad |\theta| \leq 15^\circ \quad (11)$$

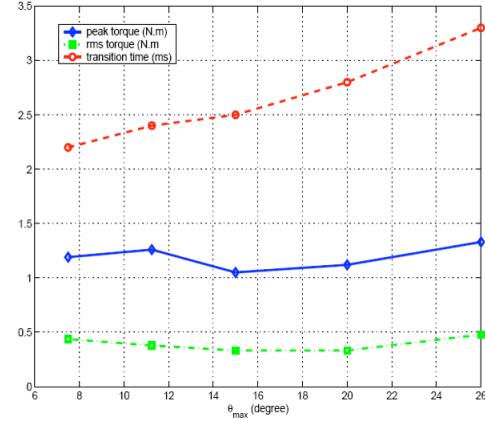


Fig. 17. Peak torque, rms torque, and transition time vs. θ -range.

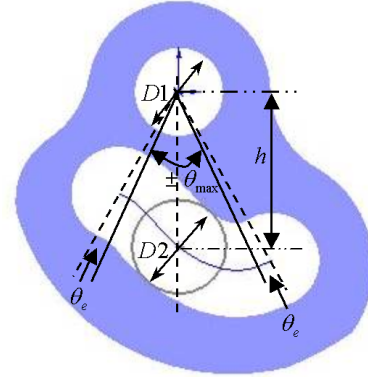


Fig. 18. Important physical parameters in a cam design.

As shown in Fig. 18, there are a number of physical dimensions that we care about in the slot cam design --- the thickness of the cam S (vertical to the picture and not shown in the figure), the diameter of the motor shaft hole $D1$, the diameter of the roller follower $D2$, the effective angular range $\pm\theta_{\max}$, the extended flat length range L_e to allow reasonable overshoot [16]-[19][23] and accordingly the corresponding flat angular range θ_e at each end, and the distance h between the motor shaft and roller follower centers when the cam is located at the equilibrium point $\theta = 0^\circ$. At this point, $D1$ and $D2$ are unchanged and $\theta_{\max} = 15^\circ$ is chosen.

Function-wise the most important element of the cam follower is the cam slot. Our basic approach to slot design is as follows: first, we put a cross section of the roller follower inside the desired slot at the equilibrium point $\theta = 0^\circ$, as shown in Fig. 18; then we let the circle roll to the two end

positions $\theta = \pm\theta_{\max}$ respectively with its center trajectory following the expected modulus function [21]. By keeping all the circumferences of the circles at each position during the full transition, a slot shaped for the desired motion is obtained, as shown in Figs. 19 and 20.

However, if we keep decreasing $\theta_{\max}$ to certain point with the distance h unchanged, the roller follower will begin to roll backwards when reaching both ends in order to maintain the desired relation between z vs. θ . This phenomenon will result in a non-smooth slot surface with abrupt turnings at both ends, as shown in Fig. 19. In order to obtain a smooth slot design between the two ends, we have to increase h to compensate for the problem caused by a decrease of $\theta_{\max}$. But we also want to keep h as small as possible in order to maintain a low cam inertia. After careful calculation, we chose $h = 28.75$ mm for the new cam, as shown in Fig. 20 [19][20]. Note that at this point, no evidence shows that the optimal $\theta_{\max}$ is independent of h , which remains a topic for future investigation.

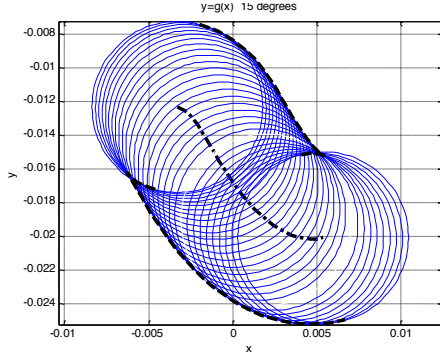


Fig. 19. Roller trajectory with $h = 16.75$ mm and $\theta_{\max} = 15^\circ$.

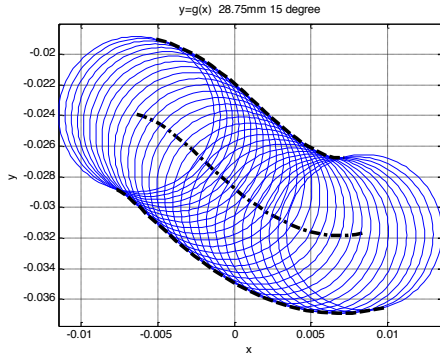


Fig. 20. Roller trajectory with $h = 28.75$ mm and $\theta_{\max} = 15^\circ$.

We designed the extend area at both ends with $L_e = 2$ mm to accommodate a reasonable overshoot of the rotor, which results in an extended angular range $\theta_e = 4^\circ$ at the valve closing end and $\theta_e = 5^\circ$ at the valve opening end.

In order to maintain the same inertia as the old disk cam, we had to reduce the thickness of the cam from 6 mm to 4.75 mm and add a nonfunctional hole. Finite element analysis in SolidWorks® has been done to make sure the new cam still has enough mechanical strength for this application. However,

complete mechanical constraints, especially contact angle and contact stress between the roller and cam, need to be addressed in the future for a more mechanically reliable design [24].

The final design of the new cam with $\theta_{\max} = 15^\circ$ is shown in Fig. 21. The physical parameters of both the old cam and the new cam are summarized in Table III.

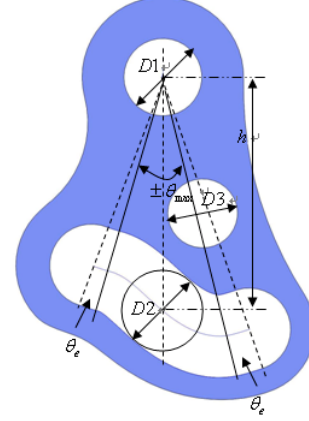


Fig. 21. Final design of the new $\pm 15^\circ$ cam.

TABLE III
COMPARISON OF THE OLD CAM AND THE NEW CAM DESIGNS.

	Old Cam	New Cam
Shaft Hole Diameter $D1$ (mm)	9.5	9.5
Roller Hole Diameter $D2$ (mm)	10	10
Effective Range $\pm\theta_{\max}$ (degree)	$\pm 26^\circ$	$\pm 15^\circ$
Extended Range L_e (mm)	2	2
Distance h (mm)	16.75	28.75
Thickness s (mm)	6	4.75
Extra Hole Diameter $D3$ (mm)	N/A	8.5
Cam Inertia J ($\text{Kg} \cdot \text{m}^2/\text{s}$)	$1.23 \cdot 10^{-5}$	$1.23 \cdot 10^{-5}$

D. Experimental Verification

After replacing the old $\pm 26^\circ$ cam with the new $\pm 15^\circ$ cam in our EMV system, the power consumption is greatly reduced from 76 W to 60 W, the transition time is substantially shortened from 3.4 ms to 2.6 ms, with the torque requirement remaining the same. Figures 22 and 23 show the position and current profiles during one transition.

V. DIFFERENT CONTROL STRATEGIES

The results reported above were achieved simply by substituting an optimized cam with the same control strategy described at the end of section III. We used closed-loop rotor position control with the free flight trajectory as the position reference. Additionally, we imposed a current limit of ± 8 A, as feed-forward control component. The closed-loop position controller forces the actual position to follow the reference position at each point along the transition. However, for this application we do not care about the position in the initial and middle part of the valve trajectory as long as the valve arrives

at the desired final position in a fast-enough time with an acceptably low velocity. Therefore, open-loop control may be attractive during part of the transition if power consumption or transition time can be further reduced. Exploration of alternative control strategies is reported in this Section

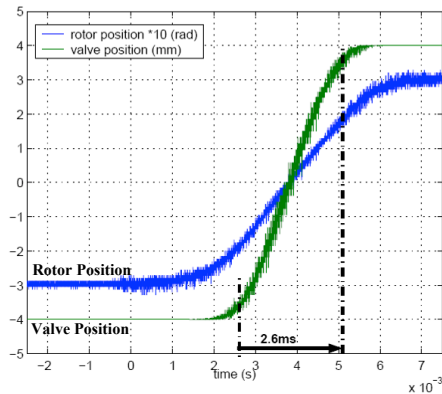


Fig. 22. Position profiles with new cam and closed-loop control.

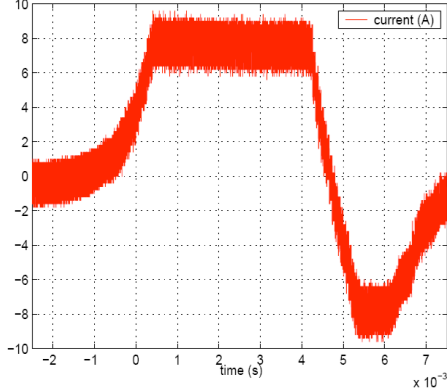


Fig. 23. Current profile with new cam and pure closed-loop control.

A. Combination of Closed-loop and Open-loop Control

With this combined control strategy, we use open-loop control at the initial and middle part of the transition and switch back to closed-loop control during the later part of the transition to ensure a smooth and accurate seating process. This combination control, as shown in Fig. 24, sometime will be referred to in this paper as the “kick-off and capture” strategy, because the initial open-loop current pulse will kick off the transition, the later closed-loop control will capture the valve at its designated position.

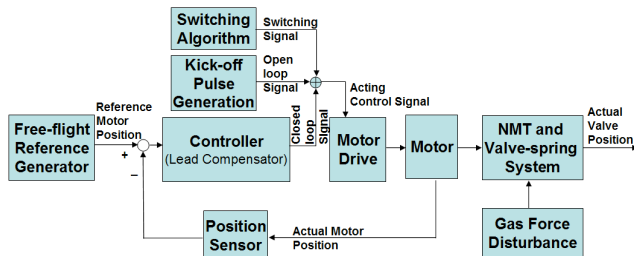


Fig. 24. Block diagram of the combination control.

The main tradeoff in this strategy is power/torque vs. transition time. In other words, it is possible to obtain a faster transition with a relatively high but short kick-off current pulse at the cost of power/torque requirement or to achieve lower power consumption with a relatively low but long kick-

off current pulse and the resulting slower transition. Both possibilities were explored experimentally. First, in order to speed up the transition, we start the transition by with an 8 A kick off current pulse followed by closed-loop control of capture, which results in a transition of 2.5 ms and a power consumption of 82 W. Figs. 25 and 26 show position and current profiles for this case. Next, in order to further reduce power consumption and the torque requirement, we start with a 5 A current pulse and then switch to closed-loop control with a ± 5 A current limit, which give us a much lower power consumption of 49 W and a bit slower transition time of 2.7 ms. These experimental results are shown in Figs. 27 and 28.

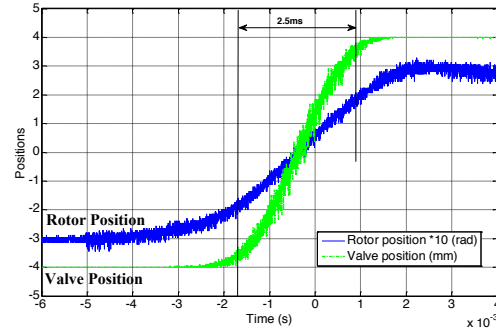


Fig. 25. Position profiles with 8 A kick off current pulse.

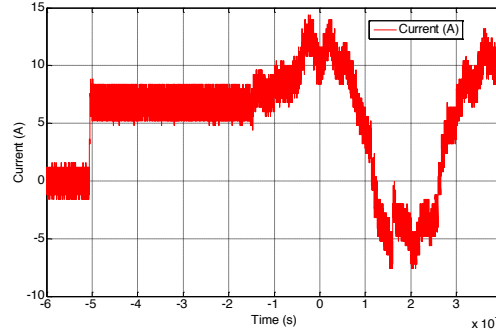


Fig. 26. Current profile with 8 A kick off current pulse.

We have been emphasizing that low power consumption is highly desirable and the ability to achieve it over most engine operating conditions is very valuable. Nevertheless, at the very highest engine speed, a fast transition is required, and an acceptable penalty in power consumption may be a reasonable price to pay for it. Therefore the capability to obtain a quicker transition with an acceptable power increase for higher engine speeds will be another attractive benefit owing to the involvement of open-loop control. Further discussion of this opportunity will be presented in subsection B.

B. Pure Open-loop Control

The study of combinations of initial open-loop control and later closed-loop capture control not only gives us another option to achieve a successful transition, but also points to another way to think about this project from an energy point of view. We will apply the energy view to the EMV system control by investigating the possibility of using pure open-loop control based on a single square-shaped current pulse, as shown in Fig. 29. By adjusting the duration and amplitude of the open-loop current pulse, we achieve successful transitions with different power consumptions and transition times

suitable for different engine conditions. Besides the control flexibility it offers, open-loop control is also a simpler and less expensive option than closed-loop controls and may be preferred by the automotive industry for this reason.

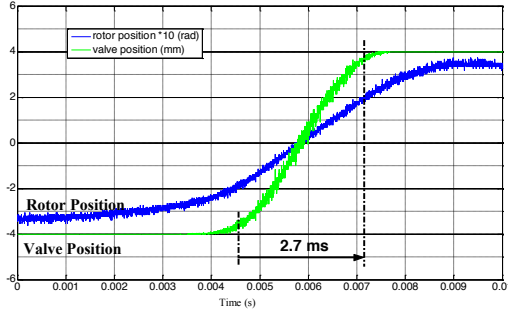


Fig. 27. Position profiles with 5 A kick off current pulse.

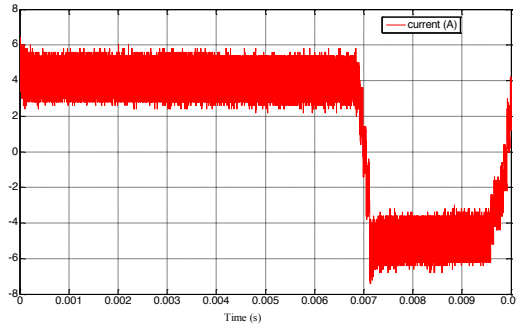


Fig. 28. Current profile with 5 A kick off current pulse.

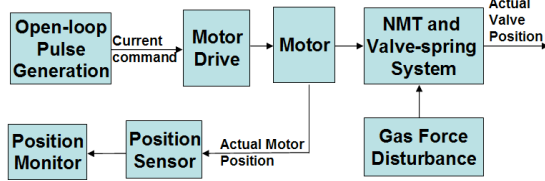


Fig. 29. Block diagram of open-loop control.

Satisfactory transitions have been achieved repeatedly with our EMV system with pure open-loop control. The lowest observed power and torque required to guarantee a successful transition were 49 W and 0.3 N-m respectively, achieved by a current pulse of 4.2 A and 9 ms duration. The experimental profiles are shown in Figs. 30 and 31.

Compared to the pure closed-loop control and the combined control strategy, this experiment with pure open-loop control gives us the best results in both power and torque so far, as indicated in Table IV. Although the transition is a bit slower than the fastest we have achieved, it is still fast enough for 6000 rpm engine speed and much faster than what we achieved with the old cam. These low power and torque requirements establish a reasonable starting point for us to design a much smaller actuator for independent valve actuation, as reported in separate papers [25][26].

At this point, parameters of the open-loop pulse in both strategies, including amplitude, duration, and switching timing if applicable, are adjusted manually via a trial and error approach. Self-learning and automatic tuning will be an interesting direction future investigations, especially when we

want to study how to achieve this performance reliably under random working conditions, such as different friction forces, temperature, vibration, engine speed, gas forces, and so on, since the repeated transitions with a kick off current pulse has been achieved only under static conditions.

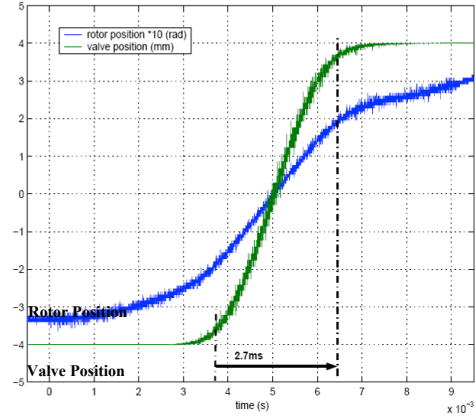


Fig. 30. Position profiles with pure open-loop control.

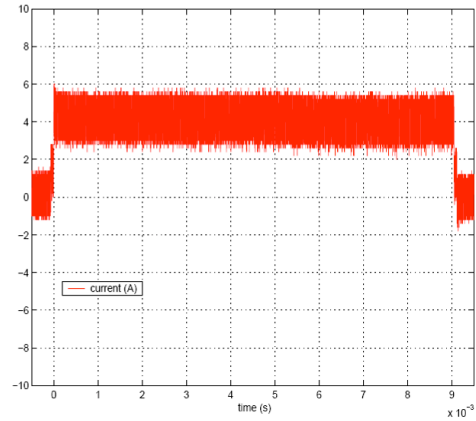


Fig. 31. Current profile with pure open-loop control.

VI. CONCLUSIONS

From the work discussed above we can conclude that based on a more accurate nonlinear system model, an optimal cam design, and the exploration of different control options, power consumption and torque requirements can be reduced substantially, and transition times reduced. Furthermore, an even faster transition can be achieved if an energy requirement penalty can be accepted.

In two published papers [25][26], we reported the design, fabrication, and experimental verification of a novel limited-angle actuator with nominal voltage of 12V for this application in order to tackle the motor size issue. The issue of lash adjustment remains a topic for future work.

TABLE IV
COMPARISON OF WITH DIFFERENT CAMS AND CONTROL STRATEGIES.

Exp. Setup	Power Consumption (W)	Peak Torque (N-m)	Transition Time (ms)
Starting Point: Old cam, closed-loop control w/ sine ref	140	1.33	3.3
Old cam, closed-	76	0.56	3.4

loop control w/ free-flight ref, 8A current limit			
New cam, closed- loop control w/ free-flight ref, 8A current limit	60	0.56	2.6
New cam, kickoff- capture control w/8A current pulse	82	0.88	2.5
New cam, kickoff- capture control w/ 5 A current pulse	49	0.35	2.7
New cam, open- loop control w/ 4.2 A current pulse	49	0.30	2.7

REFERENCES

- [1] M. B. Levin and M. M. Schlechter, "Camless Engine", *SAE Technical Paper Series*, Paper 960581, 1996.
- [2] P. Barkan and T. Dresner, "A Review of Variable Valve Timing Benefits and Modes of Operation," *SAE Technical Paper Series*, Paper 891676, 1989.
- [3] T. Ahmad and M. A. Theobald, "A Survey of Variable-Valve-Actuation Technology," *SAE Technical Paper Series*, Paper 891674, 1989.
- [4] M. Pischinger, et al., "Benefits of the Electromechanical Valve Train in Vehicle Operation," *SAE Technical Paper Series*, Paper 2001-01-1223, 2001.
- [5] C. Tai, et al., "Increasing Torque Output from a Turodiesel with Camless Valve Train," *SAE Technical Paper Series*, Paper 2002-01-1108, 2002.
- [6] M. Sellnau, and E. Rask, "Two-Step Variable Valve Actuation for Fuel Economy, Emissions, and Performance," *SAE Technical Paper Series*, Paper 2003-01-0029, 2003.
- [7] G. B. Parvate-Patil, et al., "An Assessment of Intake and Exhaust Philosophies for Variable Valve Timing," *SAE Technical Paper Series*, Paper 2003-32-0078, 2003.
- [8] R. Steinberg, et al., "A Fully Continuous Variable Cam Timing Concept for Intake and Exhaust Phasing", *SAE Technical Paper Series*, Paper 980767, 1998.
- [9] F. Pischinger, et al., "Electromechanical Variable Valve Timing," *Automotive Engineering International*, 1999.
- [10] R. Flierl, and M. Klütting, "The Third Generation of Valvetrains --- New Fully Variable Valvetrains for Throttle-Free Load Control," *SAE Technical Paper Series*, Paper 2000-01-1227, 2000.
- [11] K. Peterson, et al., "Nonlinear Self-tuning Control of Soft Landing of an Electromechanical Actuator," *Proc. 2nd IFAC Conference on Mechatronics Systems*, Berkeley, CA, 2002.
- [12] D. O. Wright and J.A. Nitkiewicz, "Engine Valve Actuation Control System," *US Patent Number: 5,873,335*, 1999.
- [13] Woo Sok Chang, et al., "A new electromagnetic valve actuator," *IEEE Workshop on Power Electronics in Transportation*, pp. 109–118, Oct.2002.
- [14] Ryan R. Chladny, and Charles R. Koch, "Flatness-Based Tracking of an Electromechanical Variable Valve Timing Actuator With Disturbance Observer Feedforward Compensation," *IEEE Transactions on Control Systems Technology*, VOL. 16, NO. 4, JULY 2008
- [15] Junfeng Zhao and Rudolf J. Seethaler, "A Fully Flexible Valve Actuation System for Internal Combustion Engines," *Accepted by IEEE/ASME Transactions on Mechatronics*, 2010
- [16] Woo Sok Chang, "An Electromechanical Valve Drive Incorporating a Nonlinear Mechanical Transformer," PhD Dissertation, Massachusetts Institute of Technology, 2003.
- [17] Woo Sok Chang, Thomas A. Keim, John, G. Kassakian, "Electromechanical valve drive incorporating a nonlinear mechanical transformer", *US Patent No: 6,755,166*, 2004
- [18] Yihui Qiu, et al., "Design and Experimental Evaluation of an Electromechanical Engine Valve Drive," *Proc. 35th IEEE Power Electronics Specialists Conference*, pp. 4838-4843, Aachen, Germany, June, 2004.
- [19] Tushar Parlikar, et al., "Design and Experimental Implementation of an Electromagnetic Engine Valve Drive," *IEEE/ASME Transactions on Mechatronics*, VOL. 10, NO. 5, October, 2005
- [20] Rudolf Seethaler and Konrad Duerr, "Electromagnetic Valve Actuation System with Two Configurations," *Proceedings of the ASME 2009 International Design Engineering Technical Conferences & Computers and Information in Engineering Conference (IDETC/CIE)*, DETC2009-87141, California, USA, 2009
- [21] Yihui Qiu, "Advanced Modeling, Control, and Design of an Electromechanical Engine Valve Drive System with a Limited-angle Actuator," *Ph.D. thesis, Massachusetts Institute of Technology*, 2009.
- [22] Yihui Qiu, et al., "Optimal Cam Design and System Control for an Electromechanical Engine Valve Drive," *Proceedings of the International Conference on Industrial Technology (IEEE-ICIT)*, p556-572, Chile, March, 2010
- [23] Tushar Parlikar, "Experimental Implementation of an Electromagnetic Engine Valve", *Master Thesis*, Massachusetts Institute of Technology, 2003.
- [24] Konrad Duerr and Rudolf Seethaler, "Time Optimal Cam Synthesis for Spring Driven Cams," *Proceedings of the ASME 2008 International Design Engineering Technical Conferences & Computers and Information in Engineering Conference (IDETC/CIE)*, DETC2008-49440, New York, USA, 2008
- [25] Yihui Qiu, et al., "A Custom-designed Limited-angle Actuator for An Electromechanical Engine Valve Drive, Part I: Conceptual Design," *Proceeding of the 5th IET International Conference on Power Electronics, Machines and Drives (PEMD)*, UK, April, 2010
- [26] Yihui Qiu, et al., "A Custom-designed Limited-angle Actuator for An Electromechanical Engine Valve Drive, Part II: Fabrication and Evaluation," *Proceeding of the 5th IET International Conference on Power Electronics, Machines and Drives (PEMD)*, UK, April, 2010