Optimal Cam Design and System Control for an Electromechanical Engine Valve Drive

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Abstract-An electromechanical valve drive system using a cambased mechanical transformer has been proposed to achieve variable valve timing in internal combustion engines. This technique promises substantial improvements in fuel economy and emissions. However, there are several challenges to transform this concept into an attractive commercial product, especially achieving acceptable power consumption and actuator size. In this paper, significant reduction in power consumption, torque requirement, and transition time are achieved. These improvements are based on an effective nonlinear system model, optimized design of the cam --- a key system component --- and exploration of different control strategies to maximize performance.

I. INTRODUCTION

As early as a few decades ago, people recognized that despite the simple design and low cost of conventional crankshaftsynchronized cam driven valve actuation, it can offer optimized engine performance at only one point on the engine torquespeed operating map mainly due to the fixed valve timing with respect to the crank shaft angle under different operating conditions. On the other hand, research has shown that variable valve timing (VVT) can achieve higher fuel efficiency, lower emissions, larger torque output, and other possible benefits. Based on the study of previous proposals for variable valve actuation (VVA), an electromechanical valve (EMV) system incorporating shear force actuation over a limited angular range was proposed several years ago [13]. As will become evident shortly, the fundamental contribution of the proposed EMV system is the capability of VVT with inherent soft landing at the end of transitions and zero power consumption between valve transitions owing to a deliberately nonlinear element called a nonlinear mechanical transformer (NMT). It had been proved that it is feasible for the proposed system to offer successful valve transitions in less than 3.5 ms, fast enough for up to 6000 rpm engine speed [13]-[15]. However, there still are several challenges that need to be addressed in order to put the proposed system into a real engine, such as high power consumption, high torque requirement, big actuator size and not-fast-enough transition for engine speeds over 6000 rpm. For this purpose, an effective nonlinear system model had been established and initial exploration on control had been discussed in [15], which reduced system power consumption and torque requirement substantially. This paper will follow on and again lower both power and torque numbers substantially

while offering an even faster valve transition by optimizing NMT design and manipulating control strategies.

We begin this paper with background on different variable valve actuation systems, especially the proposed EMV system, as presented in Section II. Section III reviews the nonlinear system model due to the nonlinear mechanical transformer and the nonlinear component of friction force. The optimal NMT design is reported in Section IV, while different control strategies are discussed in Section V. Finally, Section VI concludes this paper.

II. BACKGROUND AND MOTIVATION

In conventional IC engines, the valves are actuated by cams that are 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 can be adjusted adaptively for different situations, then the engine performance can be optimized with respect to higher torque/power output, increased gas mileage, and reduced emissions, at any point of the engine map. This flexibly controlled valve timing is called variable valve timing (VVT) and the corresponding valve drive system is called variable valve actuation (VVA). From the research of engine scientists, the main benefits from VVT can be summarized in specific numbers: a fuel economy improvement of approximately $5 \sim 20\%$, a torque improvement of $5 \sim 13\%$, an emission 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 valve drives [4], [9]-[12]. The various mechanical actuations 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, compromising its practicality for automobile manufacture.

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. The Pischinger EMV system and close variants on that concept have become a popular research topic and has gotten closest to real engine application [9]-[12]. As shown in Fig. 1, the Pischinger EMV system, proposed by Pischinger et al. [9], consists of two normal force actuators and a spring-valve system.



The springs are introduced into the system in order 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 actuators with a nonuniform and nonlinear force constant versus valve position. 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 achieved by a complicated nonlinear control scheme under certain circumstances [11].



Fig. 2. The NMT EMV system.

To solve the landing problem, A new type of electromechanical valve drive has been proposed [13]. This EMV system inherits the valve-spring system and its regenerative benefits from the Pischinger EMV system, 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. 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. Inherent soft landing is guaranteed by a special design of the NMT, as will be discussed shortly. This EMV system is referred to hereafter as the NMT EMV system.

III. NONLINEAR SYSTEM MODELING

In order to achieve soft landing at the end of transitions, low torque requirement during transitions, and zero power consumption between transitions, the mechanical transformer was designed with an intentionally nonlinear characteristics, as will be discussed in subsection A. A cam profile with the desired nonlinearity is shown in Fig. 3. This nonlinear design not only leads to nonlinear relationship between rotor and valve dynamics, but also results in a substantial nonlinear component of friction force, as will be presented in subsection B.



Fig. 3. A desirable transfer characteristic for the NMT.

A. Nonlinear Mechanical Transformer (NMT)

Before we start to analyze the nonlinear design for the transformer, let us define the displacement of the rotor as θ and that of the valve as z. Obviously, θ is a function of z and vice-versa, as defined by the transfer characteristic. Assume that the NMT implies the following relation between θ and z,

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

$$\frac{dz}{dt} = \frac{dg}{d\theta} \cdot \frac{d\theta}{dt}$$
(2)

$$\frac{d^2 z}{dt^2} = \frac{d^2 g}{d\theta^2} \left(\frac{d\theta}{dt}\right)^2 + \frac{dg}{d\theta} \frac{d^2 \theta}{dt^2}$$
(3)

Assume the NMT provides an ideal coupling between the zdomain and the θ -domain, i.e., there is no power loss or energy storage inside the coupling. Therefore we can equate the power in the z and θ domains and obtain the following relations as shown in (4) by using the NMT characteristic as shown in (2), where τ_{θ} is the torque in the θ -domain and F_z is the force in the z-domain.

$$\tau_{\theta} \cdot \frac{d\theta}{dt} = F_z \cdot \frac{dz}{dt} \implies \tau_{\theta} = \frac{dg}{d\theta} F_z \tag{4}$$

There are essential benefits obtained by using this nonlinear mechanical transformer. At either end of the stroke, the slope of the cam characteristic $dg/d\theta$ is designed to be very close to zero, as shown in Fig. 3. Thus, the reflected motor inertia in the z-domain is very large, creating inherently smooth valve kinematic profiles since the valve is slowed down by the large effective inertia near the ends of the stroke. This characteristic not only assures soft landing but also makes it easier to control the motor velocity near the ends of the stroke in the sense that possible high rotation speed and hence overshoot in the θ domain will not prevent small seating velocity of the valve at the end of each transition. Also, overshoot of the rotor is allowed by extending the flat slope area in the θ -domain. Moreover, the large spring forces at the ends of the stroke in the z-domain are transformed into small torques in the θ -domain, also due to the flat end of the transformer. Therefore static friction is enough to hold the valve at open or closed positions without any power or torque input from the motor. This is a big energy saver for the engine, especially at lower speed conditions where the valve spends much more time at closed or open positions than in a transition. In addition, because the gas force on the exhaust valve is largest at the beginning of the opening transition, the reflected gas force obtained in the θ domain is also small, making it easier to open the exhaust valve against a large gas force. Compared to the linear case, the peak torque is both reduced in magnitude and delayed until later in the transition.

Therefore, the design of the nonlinear characteristic of the transformer to have flat ends will ensure an inherent soft landing, allow some overshoot of the motor, realize zero torque/power valve holding, and reduce peak torque and hence the rms torque requirements of the motor.

B. 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 of the system, electrical loss being the other. Therefore, an accurate model to describe the friction forces of the system is very necessary.

In Chang's preliminary analysis, it was assumed that friction could be represented by viscous friction with constant coefficients in both domains [13], as shown in (5).

$$f_{\theta} = B_{\theta} \frac{d\theta}{dt}$$

$$f_{z} = B_{z} \frac{dz}{dt}$$
(5)

However, based on that assumption, the simulation results could not predict the experimental results very well, suggesting an improved understanding of friction is necessary, especially as regards the θ -domain. Therefore, in our analysis, we keep the assumption that the friction in the *z*-domain is only viscous friction proportional the valve velocity exclusively, as shown in (5). In the θ -domain, the situation is somewhat complicated for three reasons. First, there are at least three friction sources --- windage, contact friction from the shaft bearing, and contact friction from the roller follower. Secondly, contact friction

includes not only viscous friction but also coulomb friction. Thirdly, due to the spring force acting between the disk cam and roller follower, contact friction between them is affected by normal force, which is not constant during the transition. Therefore, a friction model taking all these effects into consideration is proposed, as shown in (6):

$$f_{\theta} = B_{\theta b} \frac{d\theta}{dt} + F_n \cdot (B_{\theta v} \frac{d\theta}{dt} + B_{\theta c} \operatorname{sgn}(\frac{d\theta}{dt}))$$
(6)

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 $B_{\theta b}$, while the other portion of viscous friction, $B_{\theta v} \cdot d\theta/dt$, and coulomb friction, $B_{\theta c} \cdot \text{sgn}(d\theta/dt)$, depend very much on the normal force F_n exerted at the contact surface owing to contact friction 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 (7),

$$F_n = (F_s - f_z - f_m) \cdot \cos(\alpha) \tag{7}$$

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, $f_m = m_z \cdot d^2 z/dt^2$ 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. 4. For a given NMT function, the cam slot surface slope, $\tan(\alpha)$, and its variation, $\cos(\alpha)$, is derived in [15], which turns out to be a nonlinear function by itself with respect to position.



Fig. 4. The NMT used in the previous prototype.

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 [15]. Further investigation shows that the nonlinear portion of friction force in the θ -domain dominates the linear portion as well as the linear viscous friction force of valve in the *z*-domain.

IV. OPTIMAL CAM DESIGN

One of the key advances reported in this paper is the optimization of the NMT characteristics. Changing the design of the NMT will affect the performance of the whole EMV system in several ways. First of all, the friction force, including amplitude and distribution, will be different, which in turn will require a different motor torque output to achieve the transition. This change will also affect the power consumption of the motor due to the differences in both frictional and winding losses. Secondly, the natural frequency of the system may also be changed, not only because the transformer may have a different value 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 ratio. The change in the natural frequency will have an impact on transition time of the system. It will also affect the friction force related to velocity and hence the torque requirement and power consumption.

As a result of the previous discussion, we will summarize our design objective as follows. We will pick a design which will not only have a smaller friction force requiring a smaller torque and power requirement and but also offer a smaller effective mass in the z-domain for a faster transition.

A. Two Possible Directions for a Better Design

In our first experimental apparatus, the relation between θ and z over the range of valve stroke is a sinusoidal function, as shown in (8),

$$z = g(\theta) = 4 \frac{\sin(3.46\theta)}{\sin(0.9996\pi/2)} \text{ mm } |\theta| \le 0.4538 \, rad \, (26^\circ) \quad (8)$$

With a motor rotation within $\pm 26^{\circ}$, the valve will move up and down ± 4 mm to provide a stroke of 8 mm. In order to assure an exact valve displacement, a coefficient 0.9996 is inserted in the denominator to cancel out numerical error due to the round-off coefficient 3.46 in the numerator.

Note that in the relation represented by the NMT, only the valve stroke is fixed by the application as 8 mm while the function between θ and z as well as the rotating range of the motor are both flexible and can be designed differently to meet different purposes. This represents two directions in NMT optimization. We call these two directions as "function optimization" and " θ -range optimization". Simulation results showed that there were no big performance differences among different cam function with the same θ -range. This result leads us to focus our attention on the θ -range optimization.

B. θ -range Optimization

To clarify θ -range optimization in the sense of minimum power consumption and torque requirement, let's begin with a mechanical transformer which has a constant modulus, rather than a varying one, along the stroke. If θ_{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 (9),

$$M = z_{\max} / \theta_{\max}$$
(9)

Therefore, we can rewrite the transformer equations (1), (2), and (4) as follows,

$$z = M \cdot \theta \tag{10}$$

$$\frac{dz}{dt} = M \cdot \frac{d\theta}{dt} \tag{11}$$

$$\tau_{\theta} = M \cdot f_z \tag{12}$$

And the inertia in the θ -domain, J_{θ} , can be reflected into the z-domain as m_{θ} , via the relation shown in (13),

$$n_{\theta} = J_{\theta} / M^2 \tag{13}$$

From the equations above, we can make two simple arguments. First, if given fixed valve lift of 8 mm and fixed transition time requirement of 3.5 ms, the average valve velocity during one transition is also fixed. From (11), 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 function of rotating velocity and hence the torque and power needed to complete a transition. Secondly, 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 (13), a bigger M results in a smaller reflected inertial mass and hence a smaller total effective mass in the zdomain. 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 from having different ratio M can be more complicated because it 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 (14),

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

$$\alpha = \operatorname{atan}(M)$$
(14)

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 (14), we can tell that this increased value friction f_z will reduce the normal force F_n and, accordingly, the nonlinear friction force f_{cam} in the θ -domain. In other words, with M increasing, the nonlinear friction force f_{cam} will be decreasing while the valve friction force f_z will be increasing. 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 (12). As a result, when increasing M to certain point, from the actuator's perspective, the nonlinear friction force f_{cam} 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.

From the discussion above, we can make two main points regarding a linear transformer with a differing nominal cam ratios M. First, with a bigger M, a higher natural frequency and a shorter transition time is expected; Second, with a bigger M, the nonlinear friction force related to the normal force in the θ domain will be reduced while the linear friction force in the zdomain will be increased. There should be at least one M that will give us the minimum value of total friction force/torque from the motor's perspective, which will result in the lowest torque requirement and power consumption. Therefore, for the case of a linear transformer, it is very promising to find the best ratio M which gives a faster transition with the lowest power consumption and torque requirement. We believe that the principles discussed above for a linear transformer apply equally to a nonlinear transformer, although we have to rely on numerical simulations to confirm those possible improvements.

C. An Optimized Cam Design

With the direction pointed out by the analysis above, we conducted a series of numerical simulations of our EMV system using a nonlinear transformer which maintains a sinusoidal relation between a fixed valve lift of 8 mm and a angular range varying from $\pm 7.5^{\circ}$ to $\pm 26^{\circ}$ in the θ -domain. The simulation results of peak torque, rms torque and transition time are shown in Fig. 5, 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 in (15).

$$z = g(\theta) = 4\sin(6\theta) \, mm \quad \left|\theta\right| \le 0.2618 rad \ (15^{\circ}) \tag{15}$$



Fig. 5. Peak torque, rms torque, and transition time with different θ -range.

D. Design of the New Cam

As shown in Fig. 6, there are at least six physical dimensions that we care about in a cam design --- the thickness of the cam s (vertical to the picture and not shown in the figure), the diameter of motor shaft hole D1, the diameter of roller follower D2, the effective angular range $\pm \theta_{max}$, the extended flat length range L_e to allow reasonable overshoot 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 $\theta = 0^{\circ}$.



Fig. 6. Important physical parameters in a cam design.

At this point, D1 and D2 are unchanged and $\theta_{\text{max}} = 15^{\circ}$ is assumed as explained in the subsection C.

Function-wise the most important element of the cam follower is the cam slot. Our basic approach of slot design is as follows: first, we will put a cross section of the roller follower inside the desired slot at the equilibrium point $\theta = 0^{\circ}$, as shown in Fig. 6; then we will let the circle roll to the two end positions $\theta = \pm \theta_{\text{max}}$ respectively with its center trajectory following the expected modulus function. By keeping all the circumferences of the circle at each position during the full transition, a slot shaped for the desired motion is obtained, as shown in Figs. 7 and 8.



Fig. 7. Roller trajectory with h = 16.75 mm and $\theta_{\text{max}} = 15^{\circ}$.



Fig. 8. Roller trajectory with h = 28.75 mm and $\theta_{\text{max}} = 15^{\circ}$.

However, if we keep the distance h constant while decreasing θ_{\max} , there will be a point where the roller follower begins to roll backwards at both end areas to meet the z vs. θ relation, which will result a non-smooth slot surface with abrupt turnings at both ends, as shown in Fig. 7. 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 θ_{\max} . Meantime, we want to keep it as low as possible in order to maintain low enough cam inertia. After careful calculation, we chose h = 28.75 mm for the new cam, as shown in Fig. 8 [15].

We designed the extend area at both ends with $L_e = 2mm$, which results in an extended angular range $\theta_e = 4^o$ at the closing end and $\theta_e = 5^o$ at the opening end.



Fig. 9. Final design of the new $\pm 15^{\circ}$ cam.

In order to maintain the same inertia compared to the old disk cam, we have to reduce the thickness of the cam from 6 mm to 4.75 mm and add the nonfunctional hole shown in Fig. 9. Finite element analysis in SolidWorks[®] has been done to make sure the new cam still has enough mechanical strength for this application. The final design of the new cam with $\theta_{max} = 15^{\circ}$ is shown in Fig. 9. The physical parameters of both the old cam and the new cam are summarized in Table I.

	Old Cam	New Cam	
Shaft Hole Diameter D1 (mm)	9.5	9.5	
Roller Hole Diameter D2 (mm)	10	10	
Effective Range $\pm \theta_{max}$ (degee)	± 26°	±15°	
Extended Range L_e (mm)	2	2	
Distance h (mm)	16.75	.75 28.75	
Thickness s (mm)	6 4.75		
Extra Hole Diameter D3 (mm)	N/A 8.5		
Cam Inertia J (Kg·m ² /s)	1.23.10-5	1.23.10-5	

Table I Comparison of the old cam and the new cam designs.

E. 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 76W to 60W (time average for engine running at 6000 rpm), the transition time is substantially shortened from 3.4ms to 2.6ms, with the torque requirement remaining the same. Figs. 10 and 11 show the position and current profiles during one transition. Refer to [15] for details of the experimental set-up.



Fig. 11. Current profile with new cam and pure closed-loop control.

V. DIFFERENT CONTROL STRATEGIES

RotoFhesitionesults reported above were achieved solely by substituting an optimized cam. The control strategy for the new came was the same as that used with the previous cam. We used closed-loop rotor position control with the free flight trajectory (i.e. that occurring in the absence of friction) as the position reference. Additionally, we imposed a current limit of ± 8 A, as thoroughly discussed in [15]. 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 valve trajectory as long as it arrives at the desired final position in the required time with the required velocity. Therefore, open-loop control may be attractive if power consumption or transition time can be further reduced. Exploration of alternative control strategies is the second important result reported in this paper.

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 kind of combination of closed-loop and open-loop control sometime will be referred to in this paper as the "kick 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.

The main tradeoff in this control strategy is power/torque vs. transition time. In other words, it is the possible to obtain a faster transition with a relatively high but short kick-off current pulse or a lower power consumption with a relatively low but long kick-off current pulse. Two different directions of this control strategy were explored in experiments. First, in order to speed up the transition, we start the transition by an 8 A kick off current pulse followed by closed-loop control, which offers a transition as fast as 2.5 ms while the power consumption is 82 W. Figs. 12 and 13 show position and current profiles for this case. Then, in order to further reduce power consumption and torque requirement, we start with a 5 A current pulse and then switch back 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. The experimental results are shown in Figs. 14 and 15.

We have been emphasizing that low power consumption is a highly desirable attribute 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 very desirable, and an acceptable penalty in power consumption may be a reasonable price to pay for faster transition. Therefore while keeping low power consumption at common engine speeds, the ability to achieve a quicker transition with an acceptable power increase at fairly high engine speeds will be another attractive benefit owing to the involvement of open-loop control. More discussion in this aspect will be presented in subsection B.

B. Pure Open-loop Control

The study of combinations of initial open-loop control and later closed-loop control not only gives us another option to achieve a successful transition, but also indicates to us another way to think about this project from an energy point of view.



Fig. 15. Current profile with 5 A kick off current pulse.

We will apply the energy view to the EMV system control by investigating the possibility of using pure open-loop control based on single square-shaped current pulse. 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 closedloop control and may be preferred by the automotive industry for this reason.

Satisfactory transitions have been achieved repeatedly with ourverMy 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 with a 4.2 A amplitude and 9 ms duration. The experimental profiles are shown in Figs. 16 and 17.



Fig. 17. Current profile with pure open-loop control.

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 II. Although the transition is a bit slower that 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 will be reported in future papers.

Table II Performance comparison with different cams and control strategies.

	Power	Peak	Transition
Exp. Setup	Consumption	Torque	Time
	(W)	(N-m)	(ms)
Old cam, closed-loop control	76	0.56	3.4
New cam, closed-loop control	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

VI. CONCLUSIONS

Via an optimized cam design, the power consumption, torque requirement, and transition time have been substantially improved in the NMT EMV system. Additionally, by application of different control strategies, including partial and full open-loop control, the power consumption and torque requirement have been reduced further. Also, an even faster transition can be achieved if a certain penalty of power can be accepted.

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