# A CUSTOM-DESIGNED LIMITED-ANGLE ACTUATOR FOR AN ELECTROMECHANICAL ENGINE VALVE DRIVE PART I: CONCEPTUAL DESIGN

# Yihui Qiu, David J. Perreault, John G. Kassakian, Thomas A. Keim

Laboratory for Electromagnetic and Electronic Systems, Massachusetts Institute of Technology, MA, USA Email: qiu@alum.mit.edu

**Keywords:** Variable valve timing, electromechanical valve drive, limited-angle actuator, low inertia armature, aluminium conductor.

# Abstract

Research has shown that variable valve timing (VVT) can improve significantly the performance of internal combustion (IC) engines, including higher fuel efficiencies, lower emissions, and larger torque outputs at each point of the engine map. To achieve independent and continuous VVT for each valve, an electromechanical valve drive (EMV) system was proposed previously, whose feasibility with low seating velocity had been demonstrated. In order to further improve the practicality of the EMV system, especially for a much smaller system package, a novel rotary actuator with limited angular range is proposed in this paper. Four main design challenges are discussed before a limited-angle actuator with a special yoke structure and a hollow basket-shaped aluminium-based armature is presented. Based on Maxwell simulation with optimized physical dimensions, the actuator is capable of meeting the time, power, torque, and size constraints.

# 1 Introduction

Despite the simple design and low cost of the conventional crankshaft-synchronized cam driven valve actuator, it can offer optimized engine performance at only one point on the engine torque-speed operating map, commonly at the high load and high speed condition. On the other hand, VVT can offer optimal engine performance with significantly improved fuel economy, emissions and torque under any operating condition [1]-[7].

Based on the study of previous work of variable valve actuation (VVA), such as pure mechanical systems [6], [8], [12], an electro-hydraulic system [1], and electromechanical systems [4], [9]-[16], a novel EMV system using a shear force actuator was proposed several years ago [17]. Feasibility of the concept has been validated and performance enhancements have been achieved by previous work [18]-[20]. However, a much smaller actuator is needed to fit into the limited space over the engine head. A custom-designed limited-angle actuator turns out to be a promising solution to this challenge.

Section 2 reviews the background and motivation for the proposed EMV system as well as the custom-designed limited-angle actuator. Section 3 presents the conceptual design of the limited-angle actuator after discussing four main design challenges and comparing several possible topologies. Simulation results of the EMV system employing the new actuator are described in Section 4 and show promising performance. Section 5 concludes the paper.

# 2 Background and motivation

## 2.1 VVT and VVA

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 at any point of the engine map, achieving a fuel economy improvement of 5~20%, a torque improvement of 5~13%, an emission reduction of 5 ~10% in HC, and 40~60% in NO<sub>x</sub> [1]-[7]. This flexibly controlled valve timing is called variable valve timing (VVT) and the corresponding valve drive system is called variable valve actuation (VVA).

The concept of electromechanical actuation has become 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 come closest to achieving commercial application [9]-[15]. As shown in Fig. 1, the Pischinger EMV system 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, and combined with the valve mass form an oscillatory system. The power 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 for its affect on 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. Soft landing has been achieved for the Pischinger system by a complicated nonlinear control scheme under certain circumstances [14][15].



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. In order to achieve soft landing at the end of transitions, a low torque requirement during transitions, and zero power consumption between transitions, the mechanical transformer was designed with an intentionally nonlinear characteristic, as discussed previously [17]-[22]. This EMV system is referred to hereafter as the NMT EMV system.



#### 2.2 The NMT EMV system with commercial motors

In order to validate the feasibility of the proposed EMV system, an experimental apparatus was built and integrated into a computer-controlled experimental test stand. As shown in Fig. 3, the test stand consists of a computer-controlled digital signal processor (DSP) board (the DS1104 from dSPACE, Novi, MI), a motor drive, a shear-force motor with an optical shaft angle encoder, a valve-spring assembly, and a disk cam and roller follower to implement the NMT. More details on the experimental design can be found in [17]-[22].



Fig. 3 The whole EMV system [20].

Two commercial motors, shown in Fig. 4, have been tested in the NMT EMV system. A permanent magnet dc motor (the 4N63-100 from Pacific Scientific, Rockford, IL), was picked for our first prototype because it has a very high torque-torotor inertia ratio while meeting other requirements such as power consumption, torque capacity, and appropriate speed rating. However, this motor is too large to fit into the limited space over the engine head, especially if we want to pursue independent control for each valve and cylinder to maximize the benefits.



Fig. 4. The valve, the brush dc motor, and the brushless dc motor.

Therefore a brushless dc motor (Portescap B1118-050A), which has an even lower inertia (about one third that of the old brush dc motor) and a much smaller size (about one seventh that of the old dc brush motor), was picked for our second experiment. This brushless motor is satisfactory in term of size, but its low efficiency is a fatal flaw. As shown in Table I, the power consumption with the brushless motor jumps to 138 W, while with the brush motor only 49 W is required. Within the power consumption of 138 W, 118 W is the winding loss, the cause of the extremely low efficiency. Since we are already using a carefully selected control strategy to minimize the torque requirement and winding loss, there is little room for improvement using this motor. This

suggested that a conventional motor design is probably not the best choice for this application and motivated the custom design of an actuator, taking advantage of the application's special requirement of a limited angular rotation. The goal of the customized actuator is to meet all the requirements of engine valve actuation, including small physical dimensions, low power consumption, and fast valve actuation.

Exp. Setup	Power Consumption (W)		Peak Torque	Transition Time
Exp. Solup	Electrical loss	Frictional loss	(N-m)	(ms)
Excellent performance	49		0.30	2.7
w/ brush dc motor	31	18		
High loss w/ brushless dc motor:	138		0.30	2.4
	118	20		

Table I: Performance comparison with brush and brushless dc motor.

### **3** The limited-angle actuator

In this application, the rotor only needs to swing back and forth within a limited angular range, which allows us to design a yoke structure and hence a flux path that could not be used in a conventional motor. In this section, we explore design possibilities for an actuator with such a yoke structure as well as other design techniques. We refer to such actuators as *limited-angle actuators*. We first discuss the challenges and objectives of this custom actuator design. Several possible topologies of a limited-angle actuator for valve actuation will then be proposed before the nominal actuator design is chosen. Considerations of armature design and optimization of design parameters will then be presented.

#### 3.1 Design Challenges

There are at least four big challenges that we need to face in order to design a satisfactory actuator.

First of all, the inertia of the rotor needs to be extremely low. As discussed in [17]-[22], the transition time is mainly determined by the stiffness of the springs, the mass of the valve and the inertia of the rotor and the disk cam. The springs used in our system have a fairly high stiffness. A large increase would require many changes to our proposed valve actuation system. There is not much that can be done to reduce the mass of the valve, either. We have already tried very hard to reduce cam inertia as discussed in [21]-[22]. Therefore, the only controllable variable left is the inertia of the rotor. We chose the low inertia of the brush dc motor ----  $3.5 \cdot 10^{-6}$  kg·m<sup>2</sup>/s as our design's upper limit.

Secondly, the requisite torque output must be high enough to overcome the friction and gas forces during transitions. Preferably, an even higher torque is desired, for faster transition and more robust response. From previous experience, we set 0.3 N-m as the lower end of the rated torque output of the actuator.

Third is high efficiency, implying low ohmic winding and low frictional losses, is another very important requirement of this design. If the ohmic loss or frictional loss is too high, not only will the temperature of the armature reach an undesirable level, requiring higher temperature insulation material and a higher performance cooling system, but also will the resulting high power consumption detract from the benefits of our EMV system. From previous work, the frictional loss in the system seems to be well controlled with the optimized cam discussed in [21]-[22]. Therefore, our focus in this study was how to minimize the ohmic loss of the winding. Our total power goal in this respect is no more than 100 W/valve/transition, i.e., 1.6 kW for a 4-cylinder 16-valve engine, at 6000 rpm engine speed.

Fourthly, to fit the EMV system into the limited space over the engine head and to realize independent valve actuation, i.e., one actuator for each valve, we need to make the actuator small. The dc brushless motor has a feasible size that fulfils this requirement, so its size will be used as our benchmark. Special attention will be paid to meet the dimensional constraints coming from valve pitch, which is about 37.5 mm in a common 4-cylinder 16-valve IC engine. This means the width of the actuator is constrained to no more than 37.5 mm, which will allow two actuators side by side to drive two valves independently.

Needless to say, there are some other inevitable requirements for the actuator design such as low cost, robustness, being easy to maintain or maintenance free, and so on. But this paper will focus on how to meet the four challenges described above and summarized in Table II below.

Table II: Design objectives of the actuator

	Inertia	Power	Torque	Size
	(Kg·m <sup>2</sup> /s)	(W)	(N-m)	(mm*mm*mm)
Objective	<=3.5x10 <sup>-6</sup>	<=100	>=0.3	<=70560 (28*28*90) Side dimension <=37.5 mm

#### **3.2 Nominal Topology**

From our study, there are 5 possible topologies that we have taken into consideration, as shown in Figs. 5-9 [21]. In all topologies, the armature is the only moving element (shown in green), which helps to minimize rotor inertia.

Fig. 5 shows a cross section of the first topology. There are two permanent magnets placed 180° apart with opposite poles facing each other. The dotted contours shows the flux paths. Fig. 6 presents a cross section of the second topology, which looks just like half of a regular motor. The two permanent magnets are placed such that they are separated by 90°. Note that the angle between the permanent magnets, defined as  $\sigma$ , can be different from 90°, which will affect the feasible width of the permanent magnets and the winding and hence the rotational angle and torque constant of the actuator. Fig. 7 shows a cross section of the third topology, in which the two permanent magnets are placed parallel to each other with opposite magnetic poles facing each other. Unlike the first two topologies, the two permanent magnets and two active portions of the winding used here are different in size in order to maintain the same angular dimension at different radii.

design is that its slim width will help meet the valve pitch requirement when two actuators are placed side by side for two valves. Therefore we chose topology I as our nominal topology for further study.



Fig. 5. Topology I of a limited-angle actuator.



Fig. 6. Topology II of a limited-angle actuator.



Fig. 7. Topology III of a limited-angle actuator.

Figs. 5-7 are actuators which have radial flux fields. Fig. 8 presents a fourth topology with an axial flux field, i.e., a variation of the conventional disk motor.

The fifth topology, a combination of axial flux field and radial flux field is presented in Fig. 9, which is actually a combination of designs shown in Fig. 5 and Fig. 8.

All five topologies discussed above have been modelled and simulated in Maxwell® for torque constant estimation. The initial designs are based on size constraints and a single-turn winding design. The differences in size and torque generation of each topology are summarized in Table III, from which it is quite obvious that Topology I has the highest torque output while meeting the size constraints. Another advantage of this





(a) whole isometric view (b) isometric view without back from (c) isometric view with



Fig. 9. Topology V of a limited-angle actuator.

Topology	Actuator Size	Torque Constant
1 05	(mm*mm*mm)	(N-m/100 A)
Ι	60*36*28	0.143
П	60*35*18	0.130
III	60*34*30	0.118
IV	54*38*50	0.098
V	60*46*24	0.113

Table III: Comparison of different topologies.

#### 3.2 Armature Design Considerations

As discussed before, the only moving element in this actuator is the one-phase hollow winding, which is centered on, and rotates relative to, the axis of the actuator. This design minimizes the rotor inertia.

Fig. 10 shows an isometric view of the conceptual armature design. The red cylinder is the actuator shaft. It is attached to the conductor/polymer structure of the armature in green while passing through the end turns --- one end will be connected to the optical encoder and the other will be connected to the disk cam. The detail of the attachment will be defined when discussing how to build the actuator in an companion paper.



Fig. 10. Conceptual design of the moving armature.

Fig. 11 shows a close-up isometric view of a portion of the armature. The portion shown is presented as if cut from the effective portion of the armature which passes axially through the arc-shaped gap in the iron/permanent magnet structure. The armature is constructed of an array of conductors with rectangular cross section. These conductors are bonded together by a polymer, which forms the structural element of the armature. The green portion of the figure represents the polymer, which provides mechanical support for the winding. The rectangular cross section of the wire allows a higher packing factor than round wire, while the polymer allows a low mass density structure.



Fig. 11. A closer look of the active portion of the armature.

As for the material choice of the winding conductor, aluminium is preferable to copper due to the special requirements of extra low inertia for fast transitions and low resistance for acceptable winding loss. As developed in detail in [21], for the aluminium conductor we choose ASM grade 1350 with a conductivity 0.61 times and a mass density 0.31 times those of copper, respectively. When the total weight of a conductor is constrained to a given value to limit the inertia of the moving winding, the electrical resistance achieved with an aluminium conductor is about half of that achievable with copper.

#### **3.3 Optimal Geometric Dimensions**

After choosing the nominal topology of our limited-angle actuator and the armature structure, we need to optimize the physical dimensions in terms of torque output, rotor inertia, and physical size. These dimensional parameters are shown in Fig. 12 and the final design choices are listed in Table IV.

A search program in Matlab<sup>®</sup> based on the first-order principles of a magnetic circuit was used to find the best combination of the first ten dimensions shown in Table IV. Maxwell<sup>®</sup> is used to determine a proper value for the

thickness of the vertical air gap,  $\delta_{\nu}$ , which offers a low enough

winding inductance without affecting torque output. We chose  $\delta_v = 1$  mm for our final design.



Fig. 12. Nominal topology with geometric parameters labeled.

Dimension Description	Math Symbol
Thickness of the circumferential air gap	$\delta = 0.2 \text{ mm}$
Thickness of the permanent magnets	$h_m = 4.5 \text{ mm}$
Angular range of the permanent magnets	$\theta_m = 90^{\circ}$
Thickness of the armature	$h_a = 1.5 \text{ mm}$
Angular range of the armature	$\theta_a = 45^{\circ}$
Thickness of the iron yoke	$h_v = 4 \text{ mm}$

w = 28 mm

 $\theta_v = 101^\circ$ 

l = 60 mm

d = 8.4 mm

 $\delta_v = 1 \text{ mm}$ 

Table IV: Final design parameters of the limited-angle actuator.

## **4** Expected Performance

Side width of the iron yoke

Angular range of the iron yoke

Length of the permanent magnets

Diameter of the shaft

Thickness of the vertical air gap

The EMV system simulation with the custom designed actuator predicts a transition of 2.8 ms, a power consumption of 56 W (ohmic loss 34 W and frictional loss 22 W, much lower than our upper bound ---- 100 W). This is huge improvement over the 138 W power consumption of the second brushless motor (ohmic loss 118 W and frictional loss 20 W). Fig. 13 shows a SolidWorks<sup>®</sup> design, demonstrating the possibility of packaging this actuator in a 4-cylinder 16-valve IC engine.

A summarized comparison between the two commercial motors and the limited-angle actuator is shown in Table V. All the normalized numbers are based on the brushless motor. Note that the inertia of the limited-angle actuator shown in Table V is the inertia of only the winding at this point,

calculated by SolidWorks<sup>®</sup>. After we figure out the detailed design of the shaft, the connectors between the shaft and the winding, and the optical encoder we are going to use, a larger total rotor inertia is expected [21], as will be presented in a companion paper. Also note that the entries in the bottom three rows are not functions of the actuator alone, but of the actuator, the balance of the EMV, and the control strategy. These entries present each option as favourably as possible and the limited-angle actuator stands out with the low enough power consumption, fast enough transition, and small enough package.

	Brush motor (experiments)	Brushless motor (experiments)	LA actuator (simulation)
Size (mm*mm*mm)	69*69*119	28*28*90	28*36*60
Volume (mm <sup>3</sup> /normalized)	566559/8	70560/1	64480/0.86
Rotor Inertia (10 <sup>-6</sup> Kg·m <sup>2</sup> /s /normalized)	3.5/2.9	1.2/1	1.3/1.08
Winding Loss (W/normalized)	31/0.26	118/1	34/0.29
Power Consumption (W/normalized)	49/0.36	138/1	56/0.405
Transition Time (ms)	2.7	2.4	2.8

Table V. Comparison of two commercial motors and the LA actuator.



Fig. 13. Independent valve actuation system mounted on an engine head (courtesy to ITRI).

# 5 Conclusions

This paper addresses a custom-designed limited-angle actuator, which fits in an electromechanical valve drive to provide VVT in IC engines within a package small enough to fit in the limited space over the engine head, with a transition time fast enough to accommodate faster engine speed, and a low power consumption. The customized limited-angle actuator enabled the projection of independent valve actuation for a 4-cylinder 16-valve IC engine with reasonable power consumption and high engine speed.

#### Acknowledgements

This project is financially sponsored by the Sheila and Emanuel Landsman Foundation, the Herbert R Stewart Memorial Fund, and the Industrial Technology Research Institute (ITRI) of Taiwan. We thank Profs. Mark Zahn and James Kirtlery of the MIT Laboratory for Electromagnetic and Electronics Systems for their help and support.

## References

- M. B. Levin, and M. M. Schlecter, "Camless Engine", *SAE Technical Paper Series*, Paper 960581, 1996.
  P. Barkan, et. al., "A Review of Variable Valve Timing Benefits and
- [2] P. Barkan, et. al., "A Review of Variable Valve Timing Benefits and Modes of Operation," *SAE Technical Paper Series*, Paper 891676, 1989.
- [3] T. Ahmad, et, al., "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, et. al., "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., "Electromechanically Operating Actuators," U.S. Patent 4,455,543, 1984.
- [10] F. Pischinger, et. al., "Electromechanical Variable Valve Timing," Automotive Engineering International, 1999.
- [11] M. Gottschalk, "Electromagnetic Valve Actuator Drives Variable Valvetrain," Design News, November 1993.
- [12] R. Flierl, and M. Klüting, "The Third Generation of Valvetrains ----New Fully Variable Valvetrains for Throttle-Free Load Control," SAE Technical Paper Series, Paper 2000-01-1227, 2000.
- [13] S. Butzmann, et. al., "Sensorless Control of Electromagnetic Actuators for Variable Valve Train," SAE Technical Paper Series, Paper 2000-01-1225, 2000.
- [14] 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.
- [15] K. Peterson, et. al., "Rendering the electromechanical valve actuator globally asymptotically stable," Proc. 42nd IEEE Conference Decision and Control, Maui, HI, Dec. 2003, pp. 1753–1758.
- [16] D. Durrieu, et. al., "Electro-Magnetic Valve Actuation System: First Steps Toward Mass Production," SAE Technical Paper Series, Paper 2008-01-1360, 2008.
- [17] W. S. Chang, "An Electromechanical Valve Drive Incorporating a Nonlinear Mechanical Transformer," Ph.D. thesis, Massachusetts Institute of Technology, 2003.
- [18] T. A. Parlikar, "Experimental Implementation of an Electromagnetic Engine Valve", Master Thesis, Massachusetts Institute of Technology, 2003.
- [19] Y. H. Qiu, et. al., "Design and Experimental Evaluation of An Electromechanical Engine Valve Drive," in Proceedings of the 35th IEEE Power Electronics Specialists Conference, pp. 4838-4843, Aachen, Germany, June, 2004.
- [20] T. A. Parlikar, et. al., "Design and Experimental Implementation of an Electromagnetic Engine Valve Drive", in IEEE/ASME Transaction on Mechatronics, Vol.10, NO.5, October 2005.
- [21] Y. H. 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] Y. H. Qiu, et. al., "Optimal Cam Design and System Control for an Electromechanical Engine Valve Drive", in proceedings of the.IEEE International Conference on Industrial Technology, 2010, accepted.