ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

Development and control of Camless engine with an electromechanical valve actuator

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Abstract

In an electro-mechanical valve actuated engine, the valves are driven by solenoid-type actuators and cam-shaft is eliminated. Individual control of each valve provides flexibility in valve timings over all engine conditions and fully exploits the benefits of variable valve timing. In this paper, a control system architecture is presented, capable of dealing with all these issues. It is shown that a position tracking controller is needed: a key point is the design of the reference trajectory to be tracked. Actuator physical limitations strongly influence the feasible trajectory when low valve seating velocity is required, thus affecting valve transition time. Owing to the same limitations, valve electromagnets have to be energized for a significant part of the trajectory, thus allowing valve position reconstruction starting from electrical measurements only. A method for position reconstruction is presented, which makes use of auxiliary coils to reconstruct electromagnets fluxes; it is shown via sensitivity analysis that the functional characteristics of position reconstruction and its accuracy are compatible with the required applications. The trajectory design is then addressed as an optimization problem that explicitly considers the tradeoff between fast dynamic performance and system robustness. The solution of this optimization problem enlightens the limitations on achievable dynamic performance, which are presented and discussed.

Keywords—Automotive, camless engine, electromechanical actuator, optimization, tracking control.

1. Introduction

INTERNAL combustion engines traditionally use mechanically driven camshaft to actuate intake and exhaust valves. Their lift's profiles are a direct function of the engine crank angle and cannot be adjusted to optimize engine performance in different operating conditions. This results in a compromise design that impacts on achievable engine efficiency, maximum torque and power, and pollutant emission [1].

Growing needs to improve fuel economy and reduce exhaust emissions led to the development of alternative valve operating methods, which aim to alleviate or completely avoid the limitations imposed by a fixed valve timing [2-3].

Variable valve timing (VVT) solutions use a wide spectrum of different technologies. Camshaft-based variable-valve mechanisms, in which add-on devices shift the camshaft phase [4] or switch to different cam profiles [5], [6], offer significant improvements on the engine performances (see [7] and the references therein).

Cite this article as: Dr. Mangi Naveen Kumar, Kethavath Srinivas, Sitaram Reddy & R Rajunayak,"Development and control of Camless engine with an electromechanical valve actuator", International Journal & Magazine of Engineering, Technology, Management and Research, Volume 6 Issue 4, 2019, Page 166-174.

To fully exploit the possibilities offered by a complete VVT system, camless engine valve trains, in which the valve motion is completely independent of the piston motion, are currently the topic of an in tensive search activity, both from academia and several manufacturers [8], [9]. Major advantages of camless engines reported in the literature are:

- fuel savings of 20% or higher, mainly coming from the possibility of controlling the intake process at part load, eliminating intake air throttling and associated pumping losses;
- increasing of 10% in maximum torque and up to 50% at low speed torque, resulting in a nearly flat torque characteristic that leads to better drive ability of the vehicle; this is the result of optimizing valve timing at all operating conditions, avoiding the compromise of mechanically driven camshafts;
- pollutant emission reduction, especially in NO$_x$production; the possibility of achieving high internal exhaust gas recirculation (EGR) rate, through both retarding or advancing exhaust valve closing, significantly reduces the cylinder peak temperature, resulting in NO$_x$reduction up to 90%;
- other improvements, such as increased burn rate, variable compression ratio, improved combustion stability at low speed, and reduced energy consumption.

Electrohydraulic [10] or electromechanical [11], [12] valve actuators have been proposed for camless engines. In this paper electromechanical actuators are considered. They still present open control problems, [13], and references therein) and significant system cost.

The biggest control difficulty comes from the valve seating velocity, i.e., the valve velocity when approaching the closing position. It should be very low to avoid acoustical noise and wear and tear of mechanical components, with typical values between 0.05–0.1 m/s at idle speed. Solving this issue is complicated by the fact that the system itself is unstable near valve's terminal positions, hence, closed-loop control of this highly nonlinear system is mandatory [14].

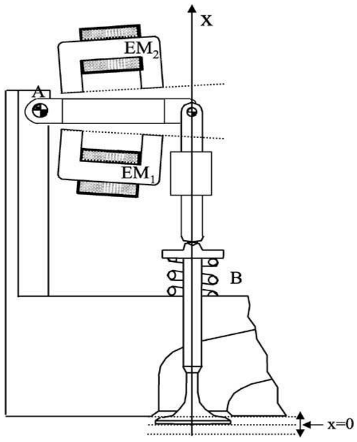

Fig. 1. Valve sketch.

Dynamical response of the system, especially the travel time required for a transition between the two terminal positions of the valve, is another key point. At high engine speed, the time available for the intake and exhaust process becomes low, hence, opening and closing of the valve should be very fast to keep engine performances. Typical values for transition time range between 3–5 ms, with an average valve velocity between 1.5–2.5 m/s.

In order to become a real commercial alternative, system costs for electromechanical valve control should be kept at reasonable levels. The most critical element from this point of view has been identified in the valve position sensor. Consequently, great effort has been spent to design a control system without position sensors [15], [16].

Limited seating velocity, short transition time, and unavailability of position sensor, make the design of the closed-loop control system a very complex task. In particular, removing the position sensor has a big impact on the valve controller, which needs to be carefully designed so as not to penalize the achievable dynamic performance.

In this paper, the control system architecture for an electromechanical valve actuator is presented. The considered actuator is depicted. It is composed of a mobile mechanical part (valve, levers), a spring B counteracted by a torsion bar A, two mechanical stops and two electromagnets, each developing a unidirectional force needed to move the valve along the vertical axis x. Electromagnet EM $_2$is used to close the valve and electromagnet EM$_1$to open it. Torsion bar A delivers a force in the x negative direction, and spring B gives force in the opposite direction [16]. The spring/torsion bar system is preloaded to an amount such that when no other forces are present, the valve is in the center of its stroke ($x = 0$). Owing to cost and space limitations, no position sensor is present in the system.

To reduce the transition time, ideally the electromagnet that keeps the valve against the starting mechanical stop should be switched off instantaneously. In a perfectly symmetrical system without damping the valve would reach the opposite mechanical stop with zero speed. Switching on the corresponding electromagnet at the proper current (flux) level would then keep the valve in the reached position [17]. This ideal behavior cannot be pursued since it neglects the electrical dynamics, assuming that currents can be freely imposed. For the considered system the supply voltage comes from the vehicle battery, limited to relatively low values, and electrical dynamics cannot be neglected.

Supply voltage saturation destroys the system symmetry. Due to voltage drop on electrical resistances, if the same voltage level would be applied to both magnets, the switching-off time of the releasing magnet would be faster than switching-on time of the catching one. In Section V, it is shown that, due to this asymmetry, if the same voltage levels are applied during the transients, the mechanical energy supplied to the system by the catching magnet is higher than the one removed from the system by the releasing magnet. Since the mass/springs system is practically conservative due to the very low damping coefficient, under these operating conditions soft landing cannot be achieved. The potential energy stored in the springs is the same at the two terminal points, hence, the excess of supplied mechanical energy must correspond to a kinetic energy greater than zero at the final mechanical stop.

Consequently, soft landing can only be achieved by slowing down the natural response of the system during the release phase, using a reduced voltage to switch off the magnet. This nonintuitive new result affects both the transient performances (when soft landing is required) and the control system design. On the other side, the need to energize the magnets during a large part of the transient opens the possibility of reconstructing the position of the valve from the electromagnetical status of the magnets, hence avoiding the use of a dedicated position sensor.

Control system architecture, position reconstruction, and achievable dynamical performance all contribute to the solution and cannot be analyzed separately. The joint analysis of these aspects is the main goal of the paper.

Even if no *a priori* specifications are given on the trajectories followed by the valve during transition between opened and closed positions, a position tracking controller is adopted. Reference trajectory design for this tracking controller is one of the most critical point, due to the need of braking the valve during the release phase, the reconstruction of the valve position and the large number of system constraints. A detailed analysis of the electromechanical energy stored in the system is the key for the solution.

In Section II, the system mathematical model is presented. A good model is needed to analyze system performance and to design the closed-loop controller. Following, the proposed model differs from previously published for VVT in the adoption of the coils' fluxes as explicit variables.

In Section III, the control system requirements are presented. A mixed open-loop/closed-loop control is adopted. Although the controller design is not addressed in the paper, a controller structure is presented to show the integration of the reference trajectory inside the controller itself as well as constraints imposed by the controller to the trajectory design.

In Section IV, a method is presented for reconstructing the valve's position from electrical measurements. Currents in the main valve coils and voltages on purposely added auxiliary coils are used. Constraints on auxiliary coils design are discussed, sensitivity analysis is performed to show the feasibility of the method, and experimental results are presented to confirm validity of the solution.

In Section V, the trajectory design is presented. The existing tradeoff between achievable transient performance and robustness against actuator saturation is discussed. Section VI concludes the paper.

2. SOLENOIDS AS A VALVES IN 2 STROKE ENGINE

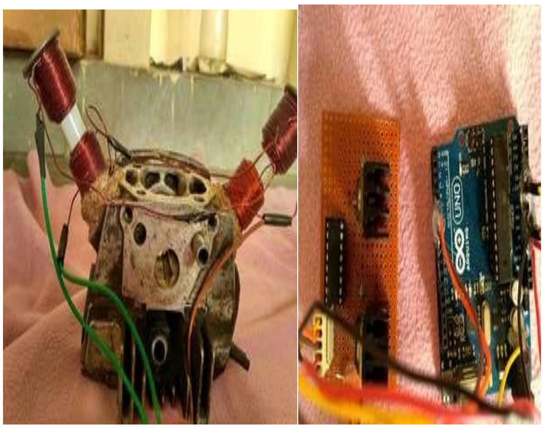

Fig.2. solenoid and Arduino circuit

A **solenoid** is simply a specially designed electromagnet. A solenoid usually consists of a coil and a movable iron core called the *armature*. Here's how it works. When current flows through a wire, a magnetic field is set up around the wire. If we make a coil of many turns of wire, this magnetic field becomes many times stronger, flowing around the coil and through its center in a doughnut shape. When the coil of the solenoid is energized with current, the core moves to increase the flux linkage by closing the air gap between the cores. The movable core is usually spring-loaded to allow the core to retract when the current is switched off. The force generated is approximately proportional to the square of the current and inversely proportional to the square of the length of the air gap.

When an electrical current is passed through the coils windings, it behaves like an electromagnet and the plunger, which is located inside the coil, is attracted towards the centre of the coil by the magnetic flux setup within the coils body, which in turn compresses a small spring attached to one end of the plunger. The force and speed of the plungers movement is determined by the strength of the magnetic flux generated within the coil.

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When the supply current is turned “OFF” (de-energised) the electromagnetic field generated previously by the coil collapses and the energy stored in the compressed spring forces the plunger back out to its original rest position. This back and forth movement of the plunger is known as the solenoids “Stroke”, in other words the maximum distance the plunger can travel in either an “IN” or an “OUT” direction, for example, 0 to 30 mm.

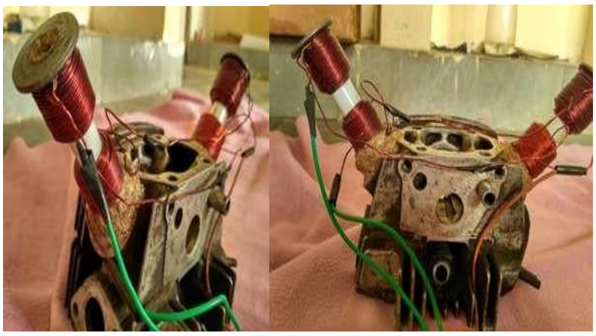

Fig.3.Side View and Front view of Solenoid Operated Camless Engine

3. CONTROL SYSTEM REQUIREMENTS

The main goal of the control system is to move the valve from the fully closed to the fully open position (and vice-versa) avoiding noisy and wearing hits against the hard mechanical stops. Since the positions of the mechanical stops are uncertain, this “soft-landing” behavior can only be achieved by controlling the speed of the valve when it is approaching the mechanical stops. This calls for a position tracking control along a reference trajectory, which has to be suitably designed.

Unfortunately, trajectories characterized by low velocity in that operating regionare unstable. This directly followsfrom the instability of the system equilibrium points at narrow air gaps. Detailed stability analysis of equilibrium points. Since the air gap is positive and $\partial\Re_2/\partial x$ has small values, two equilibrium points can exist for a constant flux value, as shown in Fig. 3. This figure is obtained with $\varphi_1 = 0$ and considering three different constant values for the flux φ_2. For low constant flux values (i.e., $\varphi_2 = 1.00$ mWb) only one equilibrium state exists. This equilibrium state is stable, as can be intuitively understood considering that any positive perturbation of the position is compensated by a negative net force ($F_{\mathrm{magnet}} - F_{\mathrm{spring}}$) and any negative perturbation is compensated by a positive net force. For greater constant flux values (e.g.,), two equilibrium points P_1 P_2, exist.The state is stable for the same reasons of the previous case, whereas the state is unstable because any positive perturbation of the position is amplifiedby a positive force($F_{\mathrm{magnet}} - F_{\mathrm{spring}}$) and any negative perturbation is amplified by a negative force. The resulting stability properties of the equilibriums are reported in Fig. 3. The unstable equilibriums lead to unstable trajectory at low air-gap and low speed, i.e., during the “soft landing.”

Closed-loop position control is, therefore, needed when approaching the landing position, but when the valve is in the central part of the transition (high air gap with respect to both magnets) closed-loop control is not necessary. Forces supplied by the magnets in these control positions are not significant due to small values of $\partial\Re_j/\partial x$. Very high currents should be imposed to increase these forces, leading to magnetic saturation. Closed-loop control with such limited control action would be useless.

The valve transition can be divided into several steps. As an example consider a transition from open to closed valve position. The control first enters RELEASE state. When in this state, closed loop position control is needed to prevent unpredictable valve movement due to instability. EM $_1$is the active actuator [switches at position “b]. When the valve distance from the releasing magnet is greater than a fixed threshold x_{off} (central region of the transition), control enters FLY state where open-loop position control is applied [switches at position “a”]. The releasing magnet $EM_1$is switched off, while the catching magnet EM $_2$will be switched on at time defined by the reference trajectory design.

From time up to the exit from the FLY state ($x \geq x_{\mathrm{on}}$), closed-loop flux control on the catching magnet is used, in order to guarantee that the system enters LANDING state with a proper flux level. Here closed loop position control is activated on the magnet EM_2until the mechanical stop is reached.

The flux and position measurements during the RELEASE and the LANDING states are needed in the controller. Cost, space, and reliability issues prevent the use of a standard position sensor in the actuator. Direct flux measurement by means of Hall sensor is unfeasible owing to hostile working conditions, mechanical constraints, and tight cost limits.

4. EXPERIMENTATION

A Regulated 5V DC power supply is feed to Arduino board and IC 7805 Voltage regulator. All microcontrollers operate at low voltages and require a small amount of current to operate while solenoids require higher voltages and current. Hence current cannot be supplied to the solenoid from the microcontroller .This is the primary need for IC L293D.A diode(**IN4007**) and a voltage regulator (**7805**) IC are connected in the path ,the diode is used as a one-way check valve. Since these diodes only allow electrical current to flow in one direction.IC 7805 is a 5V Voltage Regulator that restricts the voltage output to **5V** and draws 5V regulated power supply. A digital signal generated by Arduino based on the input program is feed to the L293D IC .L293D Is a voltage amplifier that amplifies the 5V into 12V.The L293D IC receives signals from the micro controller and transmits the relative signal to the solenoids .A L293D IC consists of 16 pins in total. 4 ground pins,4-input pins,4-output pins,2 voltage and enable pins. The digital signal output from 7 pin of arduino is feed to 10th pin of L293D(input),output from 7th pin is feed to 14th pin of L293D(output).The 4th,5th and the 13th,12th pins of L293D are grounded. L293D has an enable facility which helps you enable the IC output pins. If an enable pin is set to logic high, then state of the inputs match the state of the outputs. If you pull this low, then the outputs will be turned off regardless of the input statesDepending upon our power requirements we can use Transistors/MOSFETs as switches.

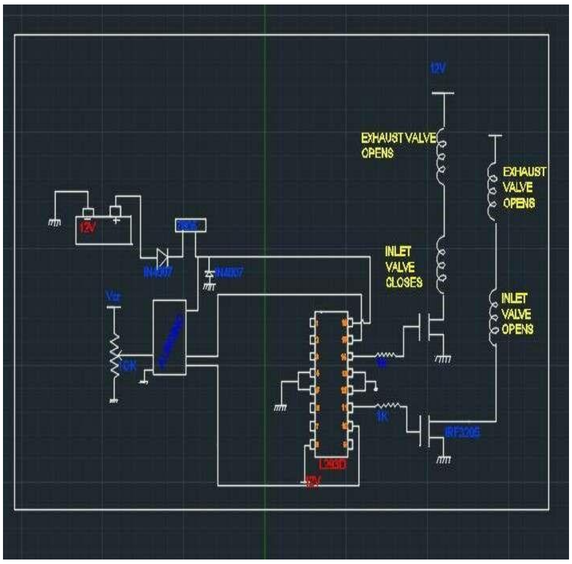

Fig.4. LINE DIAGRAM FOR VALVE ACTUATION OF CAMLESS ENGINE

The MOSFETs used are (IRF3205) which act as current amplifiers and amplify the current from 1 amp to 3 amps. Two solenoids are placed on the inlet and exhaust valves the piston of the solenoid is directly connected to the valve using a rubber tubing for motion transfer. Each solenoid consists

IJMETMR
ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
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of two set of copper windings with 12 mm dia,20 turns and 8 layered both the solenoid are oppositely connected and when actuated two sets of opposite windings get magnetized ,the piston inside solenoid moves up closing the valve the alternate valve is opened. The solenoids are rigidly placed over the cylinder head with the help of wood powder and glue which turns into concrete strong upon drying up. A solenoid is simply a specially designed electromagnet. A solenoid usually consists of a coil and a movable iron core called the *armature*. Here's how it works.

When current flows through a wire, a magnetic field is set up around the wire. If we make a coil of many turns of wire, this magnetic field becomes many times stronger, flowing around the coil and through its center in a doughnut shape. When the coil of the solenoid is energized with current, the core moves to increase the flux linkage by closing the air gap between the cores. The movable core is usually spring-loaded to allow the core to retract when the current is switched off. The force generated is approximately proportional to the square of the current and inversely proportional to the square of the length of the air gap.

5. Observations from The Tests Conducted Solenoid Force

The actual force required in the application is need to move the engine valve along with spring that must be considered.

The force can be calculated by:

$F = (N*I)^2 \mu_0 A / (2 g^2)$, Where:

- $\mu0 = 4\pi10-7$
- F is the force in Newtons
- N is the number of turns
- I is the current in Amps
- A is the area in length units squared
- g is the length of the gap between the solenoid and a piece of metal.

For different N values we get different solenoid force for valve operating

I=5amp, g=0.5, A=πdl. (d =2mm, l=5cm,)

Table 1: Valve Frequency

SL.NO	NUMBER OF TURNS	SOLENOID FORCE	VALVE FREQUENCY PER SECOND
1	100	0.512 N	14
2	120	0.738 N	18
3	140	1.001 N	22
4	160	1.310 N	25
5	180	1.661 N	28
6	200	2.050 N	33

Valve Frequency

At average speed i.e the valve opening or closing time is 40 ms, for 1 sec 25 openings and closings is possible

For 1 min for one valve 25*60=1500. With a force of 1.31N the inlet valve opens for 1500 times and exhaust valve opens fo 1500 times.

Table 2: Average speed

SL.NO	TIME TAKEN FOR ONE OPENING OR CLOSING IN MILLI SECONDS	NO OF OPENINGS OR CLOSINGS IN ONE SECOND	NO OF OPENINGS OR CLOSINGS IN MINUTE
1	71.4	14	840
2	55.5	18	1080
3	45.45	22	1320
4	40	25	1500
5	35.7	28	1680
6	30.30	33	1980

6. CONCLUSION

Looking back on this project, the overall outcome of results to be observed. This can be evaluated by looking at how well our objectives were met. Our first objective is to control the engine valve of an engine, select a linear actuator that meets specifications, and construct an electronic control system, deal with the design aspect of our project and were all almost achieved. More specifically, next objective, the electronic control system we constructed is able to read engine speeds from 0 to 3600 rpm and vary the valve timing depending on engine speed and operator inputs. However, our final objective, to obtain gains in horsepower, torque, and efficiency of 2% was not met because of not setting up in an engine but theoretically it should be done. We are confident though that this objective of installing in an engine can be met if more time for testing and facilities is given. There is a lot we could say about the need for variable valve timing. This design is very realistic for the future of the automotive industry as well as our education.

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