

Experimental Investigation of a Double Magnet EMVA at Key-On Engine: A Mechanical Resonance Based Control Strategy

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ABSTRACT

Nowadays, developing of effective camless engine systems, allowing Variable Valve Actuation (VVA), is one of the fundamental automotive challenge to increase engine power, reduce fuel consumption and pollutant emissions, as well as improve the engine efficiency significantly. Electromechanical devices based on double electromagnets have shown to be a promising solution to actuate engine valves during normal engine cycle due to their efficient working principle. Conversely, this solution requires special care at the key-on engine for the first valve lift, when the valve must be shifted from the middle equilibrium position to the closing one with limited coil currents and power requirements as well. Despite the central role of the first catching problem, few attempts have been done into the existing literature to tackle it systematically.

This paper presents an experimental validation of a control strategy for the first lift of a double magnet Electro Mechanical Valve Actuator (EMVA) system. The key idea is to induce valve oscillations exploiting the resonant behavior of the mechanical part. Since the actuator produces a magnetic force as a nonlinear function of the coil currents and valve position, a proper force-to-current inversion algorithm, previously presented by the authors, has been here adopted to transform control magnetic force into reference coil currents actuated via a multi-hysteresis control logic. Results of a wide experimental investigation show that performance strongly depend on the control parameters, i.e. the amplitude and the frequency of the exciting periodic force, with respect to different performance indexes. The performance of the

proposed control strategy have been compared experimentally with those provided by an empirical solution involving pulsing coil currents with constant amplitude.

INTRODUCTION

Currently better and better performance are demanded to vehicles equipped with gasoline engine, both in terms of engine efficiency (fuel economy, torque production and pumping losses reduction) and emission reduction in order to meet the more and more restrictive emission legislation [13]. A promising solution to improve engines of the next generation is provided by VVA [23, 28, 43, 44]. Indeed, the current engine technology is still based on mechanical cams to actuate the intake and exhaust valves and on the throttle body to regulate the amount of air fed to the cylinders. Despite of the simplicity of this approach, throttling strongly increases pumping work that turns into a worsening of the combustion process due to the lower levels of turbulence inside the combustion chamber and consequently engine efficiency decreases. To overcome such a limitation novel camless engine systems have been studied for the last two decades as a viable solution to achieve VVA allowing throttle free operations [12, 13, 42].

In automotive literature different kind of solutions, ranging from mechanical [5, 50], electro-hydraulic [2, 3, 8, 42], pneumatic [30, 33, 41, 53], motor-driven [24, 34, 40] to electromechanical [6, 7, 38, 39, 46], have been proposed to achieve Variable Valve Timing (VVT) and Variable Valve Lift (VVL). Among them, EMVA systems have shown to provide valuable improvements in terms of full load torque curve, fuel economy (particularly at low loads), emission

reduction (especially during engine cranking) and engine warm up (see [16, 20, 21] and references therein) by allowing timing variations, valve event, late closure of the intake valve and cylinder or port deactivation. By exploiting EMVA systems the low-end torque improves by 10% - 20% and the fuel economy increases by 16% - 19% with a substantial reduction of pollutant emissions [13].

An EMVA system is mainly formed by two opposed magnets and two balanced springs working in parallel (see also Fig. 1). Each magnet consists of a coil and a core where the magnetic field flows. A ferromagnetic armature, connected to the engine valve by a non-ferromagnetic stem, is placed between the two magnets. The armature initially is in the middle equilibrium position between the two magnets. At engine startup the armature is pulled up near to the upper/lower magnet and hold there, storing in the springs the mechanical energy spent for their deformations. When the upper magnet is deactivated, the elastic energy is released and the EMVA system approximately behaves as an “harmonic oscillator” in free evolution. The moving part, at a distance of less than about one millimeter, is captured and held by the lower electromagnet. A similar series of events occur when the moving part is released and the upper magnet catches it. Since electromagnets can produce high forces at low distances, power requirements of the system are quite low.

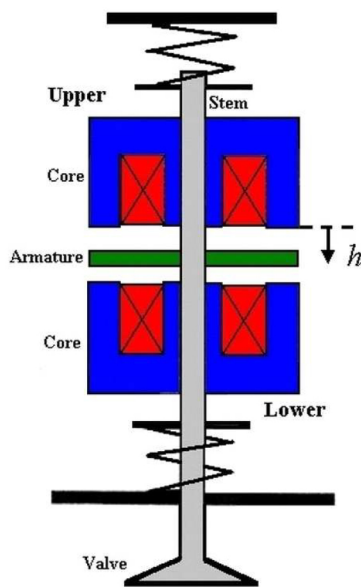


Figure 1. A schematic view of the EMVA.

To achieve the potential benefits promised by EMVA systems in replacing the traditional camshaft, two crucial control problems have to be tackled and solved efficiently by car manufactures, i.e. the control of the valve seating velocity (Soft Landing Control problem, SLC) and the first lift manoeuvre (First Catching Control problem, FCC). (Notice that, the high nonlinear coupling between the mechanical and

magnetic parts embedded into the valve actuator and the coupling with the engine make these control tasks cumbersome obstacles to overcome for a low-cost mass production of this camless technology.) More in detail, the control objective in the case of the SLC problem is the reduction of the impact velocity during valve opening and closing in order to increase the lifespan of the valve actuator and at the same time to reduce acoustical noise. In fact, an acceptable landing velocity should be less than 0.03 [m/s] at 600 [rpm] engine speed and less than 0.3 [m/s] at 6000 [rpm] engine speed [1], although in camshaft design the seating velocity is limited below 0.05 [m/s] [45]. This specification becomes more arduous to accomplish since a valve stroke has to occur in a very short time (about 3 [ms]) to guarantee a proper load variation ability of the valvetrain [21].

To solve this control problem, different control strategies can be found in the technical literature. For example in [20, 21, 45, 52] EMVA system is controlled in open loop via series of properly tuned current patterns. Since open loop control is sensitive to disturbances and suffers poor repeatability, closed loop strategies have been also extensively used on the basis of different EMVA modeling. For instance in [35] the soft landing is addressed using a nonlinear feedback strategy, where control parameters were tuned by means of an extremum seeking method. In [46] a Linear-Quadratic Regulator feedback controller was developed for soft seating while Flatness-Based tracking method is adopted in [10]. In [15], based on nonlinear force model reproduced via proper polynomials, a sensorless sliding mode control strategy was used to track a reference trajectory designed for the soft-landing. In [49] the quiet seating control is designed based on a linearized model of EMVA and in [27] a simplified nonlinear model was used to design an exact linearization based control law. A physics-based model for a single magnet developed in [52] was used to set up valve position controllers [26, 51], observer based [36] and adaptive strategies [25, 37]. Moreover, a linear grey-box modeling including a valve lash was used to design a third order linear controller coupled to cycle-to-cycle repetitive learning control action [45, 47,48,49] and in [22] a sensorless approach is proposed.

The second control problem, here referred to as FCC, consists of moving the valve from its resting position (middle equilibrium position) close to an end-stroke position (closed/opened valve position) at the engine start-up. Taking into account that the magnetic force decreases rapidly and nonlinearly with the air-gap (between the armature and cores) and no potential elastic energy is initially stored in the system, the control authority on the armature is low. Hence, the first lift manoeuvre turns to be a complex and arduous task to be accomplished with feasible coil currents provided by automotive electrical power circuits and then the challenge of the FCC problem arises [55]. Despite the central role, few attention has been made by the existing automotive literature

to solve the problem of the first lift as far as we are aware. In [54, 56] special designs of EMVA based on Permanent Magnetics (PM) have been proposed to attain the first lift manoeuvre with admissible coil currents. We highlight that, even though this approach simplifies the FCC task, the demagnetization of the PM have to be efficiently addressed to avoid system faults.

Some solutions to prevent the PMs from demagnetization can be found in [29, 57]. When conventional EMVAs are adopted a simple and viable solution to catch the valve at engine startup is instead presented in [4]. Here, in the case the temperature is within a predetermined normal range, armature oscillations are induced by alternatively supplying current to electromagnets at the mechanical resonant frequency of the system. The currents levels are regulated as function of the approach of the armature toward a pole face. Instead, if the temperature is below that range, a high current pulse is supplied to an electromagnet. We point out that to implement effective managements of the entire system of valves, power requirements to accomplish the first catching play a crucial role. In [11], in order to provide effective approaches for the initialization of the engine valves, the maximum number of valve that can leave the equilibrium position is computed on the basis of the actual power supply capability, which depends on the battery charge state, battery life, as well as ambient and engine temperatures. It is then apparent that such a number increases if the energy to complete the first lift per valve decreases. Again, in order to reduce the instantaneous power requirements, inventors propose in [11] to initialize engine valves sequentially. As a consequence the time needed to complete the first lift of each valve becomes crucial to meet customer satisfaction in terms of waiting time at the key-on.

In [18] authors have solved both the control tasks via model-based decoupling control strategies. The key point of the entire control approach is a detailed and predictive FEM-based dynamic model of the EMVA system presented in [19]. This model describes accurately self and mutual magnetic effects and has been used in [18] to design a Force-To-Current (FTC) inverse algorithm allowing to determine the equivalent coil currents generating a given magnetic force when the armature position is known. The resulting decentralized control architecture consists of two nested controllers. The inner feedback controller regulates the upper and lower coil currents to demanded values. Instead, the outer controller is chosen among a given set of subcontrollers by a scheduler on the basis of the control task to be performed, i.e. the first lift manoeuvre or the tracking of a soft landing trajectory. The magnetic force is then used as control variable by each subcontroller and converted by means of the FTC inverse algorithm into feasible reference coil currents fed to the inner control loop. In so doing, the outer controllers can be designed neglecting the magnetic and electrical dynamics of the actuator which results decoupled from the mechanical

part. Relating to the FCC problem, in [18] authors have shown the infeasibility of solving it as a tracking control problem due to current constraints on the power actuator and they proposed a mechanical resonance frequency strategy as viable energy solution for the first lift manoeuvre.

In this paper we propose for the first time experimental results of an EMVA system controlled via the decoupled approach in [18]. In particular the first catching problem is addressed by exploiting the resonant frequency of the mechanical part. Hence, the control input generated by the outer controller for the first lift manoeuvre takes a sinusoidal shape. Since the resonant frequency as well as the resonant peak are sensitive to system parameter variations, a sensitivity analysis is also carried out. Experimental results are then analyzed with respect to different performance indexes based on energetic consideration, maximum coil currents as well as first catching time with the aim to characterize experimentally the system behaviors during first lift using its frequency response. Finally, performance of the proposed control strategy have been compared experimentally with those provided by an empirical solution involving pulsing coil currents with constant amplitude.

EXPERIMENTAL SETUP

This work is based on the electromechanical valve prototype designed in [17, 19,20,21] and developed by the Istituto Motori of National Research Council of Italy in cooperation with “Dell’Orto S.p.A.” company. EMVA prototype is shown in Fig. 2. The setup consists of four main components: i) the EMVA system; ii) the interface with a Rapid Control Prototyping (RCP) module; iii) a sensor position; iv) a power actuator.

The valve is one of intake of a four valves spark ignition 2 liter engine (see Fig. 2c). It is worth noting that the valve actuator has been designed with particular attention to the choice of technologically advanced materials (sintered materials), such as those based on iron powders with high electrical resistivity and good magnetic performances. In so doing, undesirable eddy currents are strongly limited and the magnetic response of the valve actuator improves consequently.

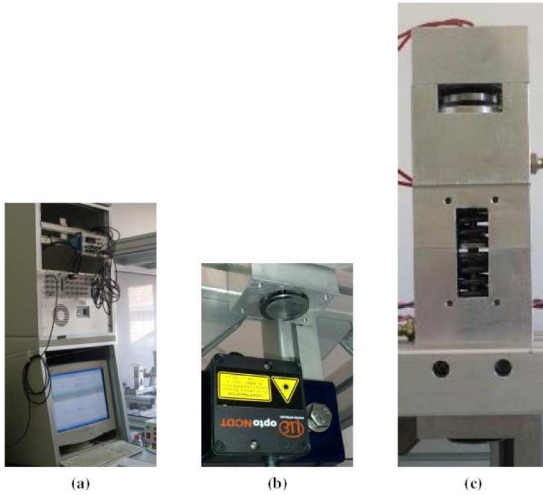


Figure 2. (a)-(b) Experimental setup for double magnet EMVA system. (c) Prototype of EMVA system.

Power circuit has the important task to track (by means of a properly multi-hysteresis control logic [18]) the currents in the coils at the reference values yielded by the outer controllers. It follows that the performance of these last strongly depend on those of power circuit. Control of the system is done via a RCP dSPACE based station shown in Fig. 2a, composed by a processor board (DS1005), including a PowerPC RISC μ P running at 1GHz, equipped with a digital (DS1004) and an analog I/O (DS2201) boards. The control station is programmed in Matlab/Simulink (MathWorks) environment and the experiments are managed and instrumented by ControlDesk application (dSPACE). A laser position sensor LD1627 (Micro-Epsilon) [32] depicted in Fig. 2b, that works in the range 0-10 [mm] with cut-off frequency of 37 [kHz], is used for the on-line measurement of the valve position. Sampling time is set to 70 [μ s], given enough interval time to make digital I/O signal processing and computed control actions, as well as all others auxiliary services and monitoring tasks of the EMVA plant.

MATHEMATICAL MODEL

Among many models proposed in technical literature [9, 14, 21, 45, 52] in this work we exploit the lumped parameter dynamic model derived from a hybrid analytical-FEM approach in [19] for its high predictive capabilities shown in reproducing the strongly nonlinear dynamics characterizing such a magnetic valve actuator. The model reproduces precisely all electrical and magnetic nonlinearities, as self and mutual inductances, back-ElectroMotive Forces (back-EMFs) and magnetomotive force for a wide range of magnetization (linear and saturation) and for all armature positions.

According to [19] and as schematized in Fig. 3b, EMVA dynamics are the result of an high nonlinear interaction between the electrical, the magnetic and the mechanical

subsystems. More in detail the electrical subsystem describes the coil currents dynamics, while the mechanical one takes into account armature dynamics. The interaction between the electrical and mechanical parts is due to the magnetic force generated by the magnetic field (magnetic block in Fig. 3b) which is applied on the armature.

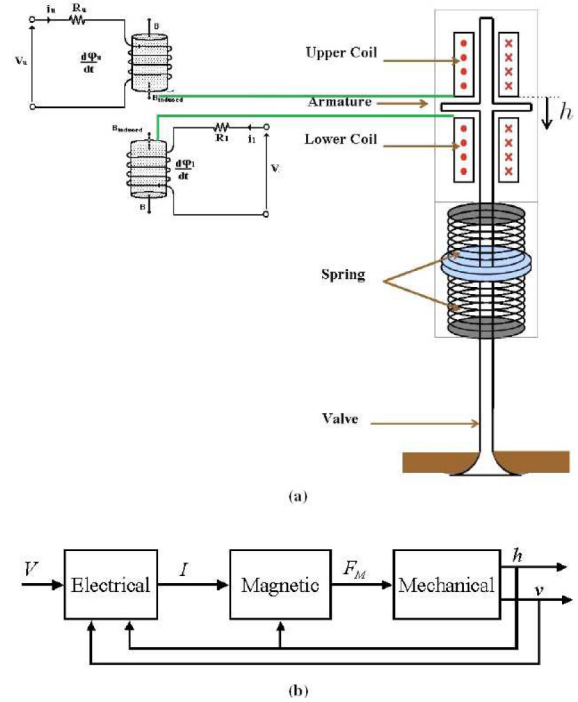


Figure 3. (a) EMVA system full scheme. (b) Block scheme of the EMVA model.

From the previous discussion and assuming the pair armature-valve as a rigid body, plant dynamics are described as:

$$\frac{dI}{dt} = \Lambda^{-1}(I, h)(V(t) - RI - Z(I, h)v) \quad (1)$$

$$\frac{dv}{dt} = \frac{1}{m}(-k(h - h_{eq}) - \sigma v + F_M(I, h)) \quad (2)$$

$$\frac{dh}{dt} = v, \quad (3)$$

where: $I = (i_u \ i_l)^T$ is the vector of the upper, i_u [A], and lower, i_l [A], coil currents, respectively; $V = (V_u \ V_l)^T$ [V] is the vector of the voltages applied to upper and lower coils in Volt; $R = \text{diag}(r_u \ r_l)$ [Ω] is the diagonal matrix of the coil electrical resistances; h [m] and v [m/s] are the position and

the velocity of the armature, respectively; h_{eq} [m] is the position of the valve at equilibrium in the absence of external forces; $k(h - h_{eq})$ [N] is the sum of the upper and lower spring forces, with k [N/m] being the equivalent spring stiffness; σv [N] is a linear viscous friction with σ [Ns/m] being the viscous coefficient; m [kg] is the equivalent mass of all moving parts, included the effect of non negligible spring masses; the matrix $\Lambda(I, h)$ [H] models the self and mutual coil inductances, whereas the vector $Z(I, h)$ [V·s/m] collects self and mutual back-EMFs caused by the motion of the armature in a magnetic field.

Exploiting the symmetry of the system, the magnetomotive force, say F_M [N] in model (1), (2), (3), can be written as:

$$\begin{aligned} F_M(i_u, i_l, h) &= F_{M_u}(i_u, h) + F_{M_l}(i_l, h) + F_{M_c}(i_u, i_l, h) \\ F_{M_u}(i_u, h) &= -F(i_u, h), \\ F_{M_l}(i_l, h) &= F(i_l, g_{max} - h), \end{aligned} \quad (4)$$

where F_{M_u} , F_{M_l} and F_{M_c} are the magnetic force due to the upper coil, the lower coil and the interaction between them, respectively, $F(i, g)$ is the force given by a single magnet for a given current i [A] and air-gap g [m], and g_{max} [m] is the maximum air-gap value. The mathematical expressions of the terms $\Lambda(I, g)$, $Z(I, h)$, $F(i, h)$ and $F_{M_c}(i_u, i_l, h)$ are complex, hence, for the sake of brevity we report here only that of magnetic force $F(i, g)$:

$$\begin{aligned} F(i, g) &= m_F(g)i + \frac{2}{\pi} n_F(g) \cdot \\ &\cdot \arctan\left(\frac{\pi}{2}(s_{0_F}(g) + s_{1_F}(g)i)i\right) \end{aligned} \quad (5)$$

The reader is referred to [19] for further details. We point out that eddy currents have not been taken into account in the derivation of the magnetic force model (4). The eddy current effects can be neglected since sintered materials were adopted for most of the volume (two cores) of the actuator. Nevertheless in [19] has been shown that the lumped parameter model (1), (2), (3) reproduces accurately dynamics obtained via a FEM-based magnetic analysis including also eddy currents effects. In the next section we show that the proposed magnetic force model allows to reproduce valve dynamics with a great accuracy. Hence it is here validated experimentally for the first time.

We note that, since some residual air-gaps could be considered between the armature and cores at closing and opening valve positions (i.e. to ensure an adequate airtight of the valve on its seat), we suppose that valve position h can

vary in the interval $[h_{min}, h_{max}]$ with $0 \leq h_{min} < h_{max} \leq g_{max}$, with h_{min} and h_{max} being the end-stroke positions. This implies, under the assumption that the two springs are equally preloaded, the nominal equilibrium position is $h_{eq} = (h_{max} + h_{min}) / 2$ and actually valve stroke is $h_{str} = h_{max} - h_{min}$. According to this notation, the residual air-gaps between the armature and upper magnet/lower magnet (at closed/opened valve) can be expressed as $g_{r_u} = h_{min}$ and $g_{r_l} = g_{max} - h_{max}$, respectively. Notice that these parameters strongly affect the magnetic force since it rapidly decreases with air-gap for a constant current [19]. One can deduce that small residual air-gaps become a prerequisite to limit current requirements in closed/opened position, whereas the highest spring force ($k \cdot h_{str}/2$) could easily trigger instability conditions.

MODEL VALIDATION

In this section we validate on the basis of experimental data the model introduced previously. In particular, to control the valve motion according to the decoupling control approach presented in [18] and briefly summarized in the next section, it is enough to validate the mechanical part of the model and the magnetomotive force, i.e. equations (2), (3) and (4). The identification and model validation activities are organized in three steps: 1) estimation of the mechanical parameters during free response; 2) tuning of the residual air-gaps; 3) validation of the whole model in forced response.

1stStep - To tune the mechanical parameters in (2), i.e. k , m and σ , we have considered different time series of the valve position acquired during free oscillations, i.e. when the coil currents are set to zero (absence of external forces), with initial valve position, $h_0 := h(t_0)$, equal to the first peak oscillation, occurring at the instant t_0 after switch off of the lower magnet where valve was initially hold, and where initial velocity, $v_0 := v(t_0)$, is zero. An optimization technique based on a least-square algorithm [31] has been then used to estimate the mechanical parameter. A priori information on parameters value set were exploited to initialize the algorithm for finding the minimum of cost index (sum of squared errors between model data and experimental ones). The identification routine returns as optimal values $m = 0.153$ [kg], $k = 96993$ [N/m] and $\sigma = 11.5$ [N·s/m]. Fig. 4 shows the high matching between experimental data and model predictions and confirms the effectiveness of the mechanical valve modeling as well as the identification method.

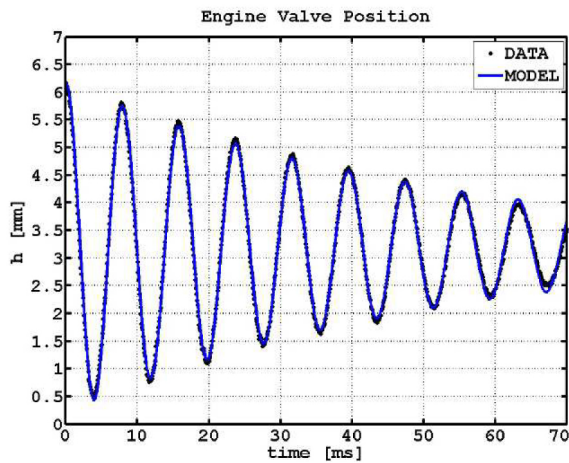


Figure 4. Model validation during free oscillation.
Engine valve position: model predictions (blue dashed line) and experimental data (black solid line).

2nd Step - The residual air-gap parameters g_{r_u} and g_{r_l} are tuned as those values such that the magnetic force (5) equals that elastic in closed and opened position, respectively. Speaking in math terms, the subsequent nonlinear equations have to be solved:

$$F(i_u^*, g_{r_u}) = k(\underbrace{h_{eq} - h_{min}}_{h_{str_u}}), \quad (6)$$

$$F(i_l^*, g_{r_l}) = k(\underbrace{h_{max} - h_{eq}}_{h_{str_l}}), \quad (7)$$

where i_u^* and i_l^* are the minimum current values (experimentally observed) able to hold magnetically the armature against the upper and lower cores respectively, and the h_{str_u} and h_{str_l} are the effective valve displacements (experimentally measured) from equilibrium position. Equations (6),(7) give as solutions $g_{r_u} = 171.6$ [μm] and $g_{r_l} = 170.2$ [μm], here expressed in micrometers.

3rd Step - To validate the model when it is excited by a magnetic force different from zero, we have considered different experimental time histories of the coil currents fed to magnetomotive force model (4) when the valve position h is computed by means of (2)-(3) and the mechanical parameters are those identified previously. Fig. 5a shows the precise matching between experimental valve position data and model predictions when the measured coil currents take the shape in Figs. 6a and 6b, and the magnetomotive force predicted by the model (4) is traced in Fig. 5b.

We remark that the magnetic force model (4) here adopted is the one derived using FEM data [19] and no further experimental parameter tuning has been required. Hence the effectiveness of the FEM based modeling approach proposed in [19] is also proven indirectly.

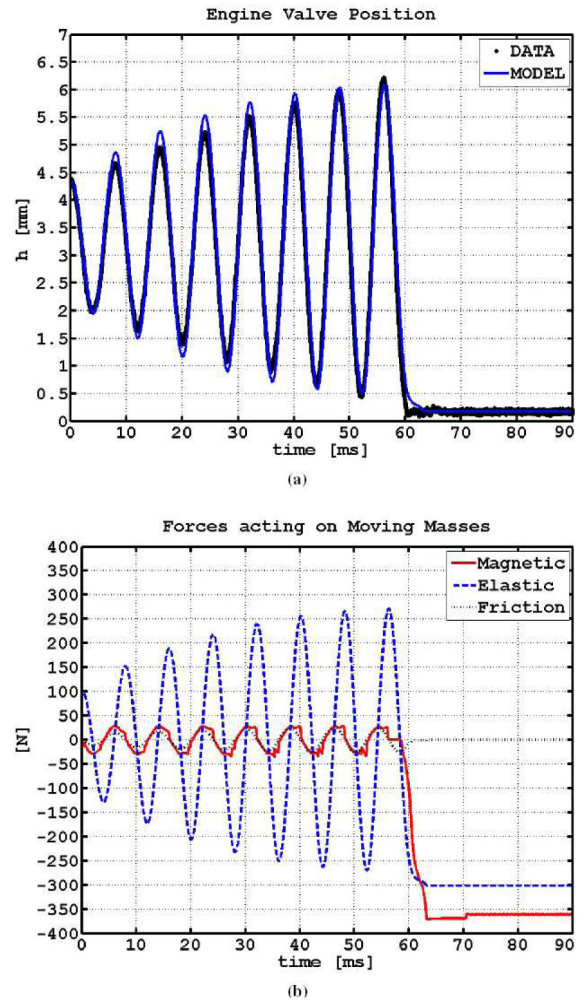
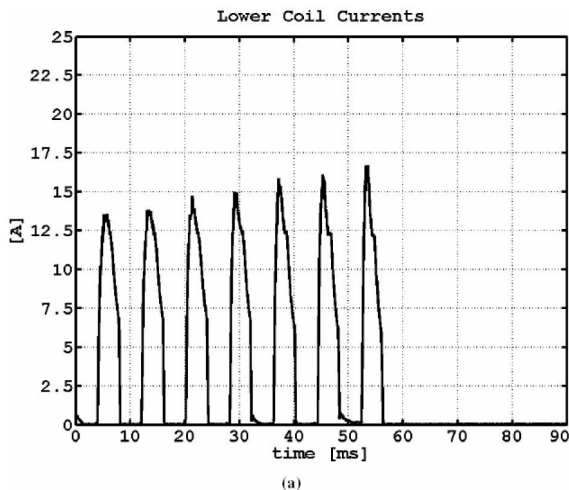
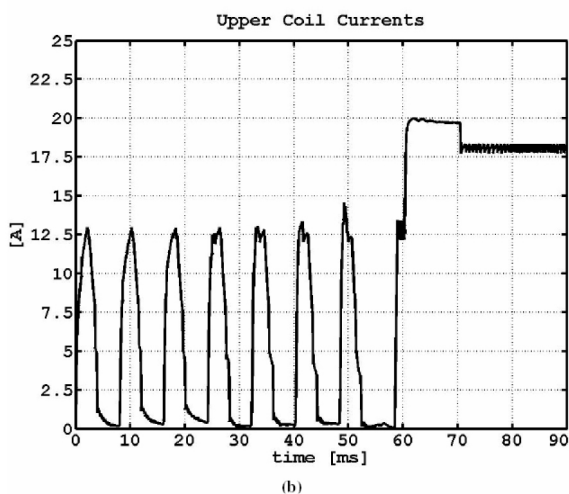


Figure 5. Model validation for a nonzero magnetic force.
(a) Simulated valve position (blue dashed line) and experimental data (black points). (b) Forces: magnetic force (red solid line), elastic force (blue dashed line) and friction force (black dotted line).



(a)



(b)

Figure 6. Model validation for a nonzero magnetic force.
Measured currents: (a) lower coil current and (b) upper coil current.

THE DECOUPLING ARCHITECTURE TO CONTROL EMVA SYSTEMS

The decoupled control architecture for EMVA systems presented by authors in [18] mainly consists of two nested control loops and can be schematized as in Fig. 7.

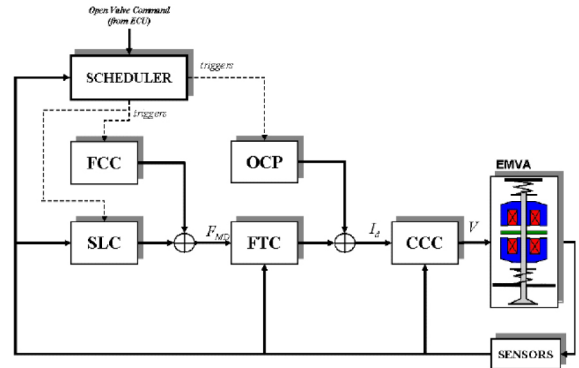


Figure 7. Block scheme of the EMVA control unit.

The inner feedback controller (CCC, Coil Current Controller) regulates to demanded values, I_d , the upper and lower coil currents, I , by varying the coil voltages, V . Instead, the outer controller is chosen by the supervisor (SCHEDULER) on the basis of the control task to be performed and EMVA operating conditions. In particular the first catching controller (FCC) is scheduled at the engine startup to transfer the engine valve from the initial middle position to that closing/opening. Instead, the soft landing controller (SLC) is activated to control the valve motion during the last part of the valve stroke to perform soft catching operations. We note that, the output of both these subcontrollers is the demanded magnetic force to apply on the armature. To link the outer and the inner control loop a proper force to current inverse algorithm is implemented into the FTC block in order to transform the demanded magnetic force into a pair of feasible reference coil currents to be tracked by the CCC. Finally, the Open-loop Current Pulses (OCP) block provides directly reference setpoint coil currents to the inner current control loop to perform other valve operation modes.

In what follows some details about the CCC strategy and the FTC inverse algorithm will be given. The reader can be found further details on the supervisor policy, OCP block and on SLC block in [18].

COIL CURRENT CONTROL BASED ON A MULTI-HYSTERESIS STRATEGY

The *Coil Current Controller* block in Fig. 7 has the important task to regulate the currents for both coils to a desired current

vector $I_d(t) = [i_{u_d}(t) \ i_{l_d}(t)]^T$. To achieve this control task with an implementable low cost circuitry, the multi-hysteresis control strategy proposed in [18] has been implemented in hardware (see also Fig. 8a).