

Model-based Decoupling Control of a Magnet Engine Valve Actuator

Alessandro di Gaeta, Veniero Giglio and Giuseppe Police

Istituto Motori, National Research Council of Italy

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ABSTRACT

The use of Variable Valve Actuation (VVA) systems offers many advantages in terms of increased engine power, reduced fuel consumption and pollutant emissions, accomplishing a significant improvement of the global efficiency of the engine. In the last decade different devices have been proposed to implement advanced and innovative VVA managements on four-stroke engines. ElectroMechanical Valve Actuator (EMVA) formed by two opposite magnets and two balanced springs seem to be a very promising solution among several camless actuation systems. This type of valve actuator is characterized by highly nonlinear and strongly coupled dynamics which makes very difficult to govern engine valve motion during the last part of the closing and opening strokes, where an unstable behavior is exhibited.

In this regard the control problem of the EMVA is tackled in this paper. A predictive electromagnetic model based on a hybrid analytical-FEM approach is used to decouple and linearize the nonlinear cross-coupled electric, magnetic and mechanical dynamics of the actuator. The model takes into account self and mutual magnetic effects (inductances, back-EMFs, magnetomotive forces) over linear and saturation regions of the materials and for

all engine valve positions. The proposed EMVA control scheme results in two nested state feedback controllers: the inner one regulates the currents of two coils, and the outer one controls the engine valve position through the magnetomotive force. An inversion algorithm transforms and splits the control force produced by the outer controller into the desired coil currents to be tracked by the inner controllers. Particular attention is paid in the design of a desired valve motion trajectory addressing a soft landing task. The control problem related to the first lift manoeuvre of the engine valve is also tackled exploiting the mechanical resonance of the actuator. Simulation results are presented to show the performances of the proposed decoupling control strategy. Eventually, the power required for EMVA operations is evaluated for different coil current control strategies.

INTRODUCTION

One of the major drawbacks of gasoline engines is the necessity to partialize load reducing the amount of the mixture introduced in the cylinder. By means of this, throttling causes a strong increase of pumping work and a worsening of combustion process because of the lower levels of turbulence inside the combustion chamber. A quite ef-

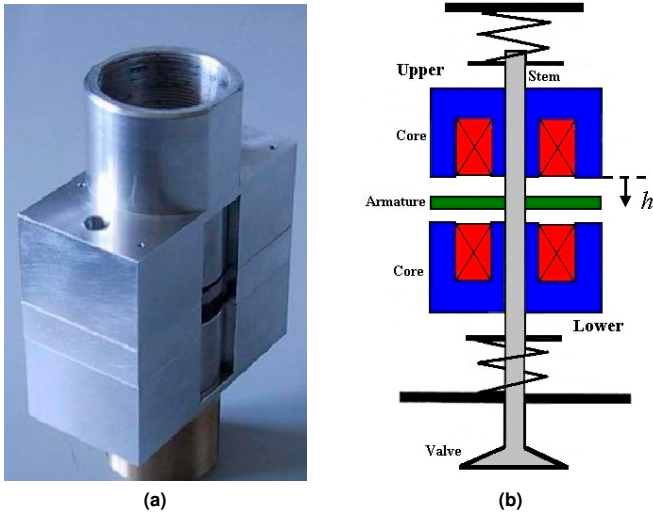


Figure 1 – (a) A prototype and (b) a schematic view of the EMVA.

fective way to reduce pumping work at partial loads is the adoption of a variable valve actuator to realize throttle free operations. The mechanical cams, that drive the motion of the engine valves, would be replaced by an actuator for each engine valve.

Different devices (mechanical [5, 15, 17, 24, 30, 42], electrohydraulic [2, 3, 7, 8, 37], pneumatic [21, 22, 25, 35, 45], motor-driven [26, 34] and electromechanical [6, 31, 32]) have been proposed in the past years to control the inlet and the outlet phases of four strokes internal combustion engines. The actuator systems are often classified depending on how they allow to realize load variation. For example, a variable lift actuator system allows a load control varying the maximum valve lift, whilst a fixed lift actuator system allows a load variation acting on the valve opening duration. The flexibility of the various systems may include: valve timing variation, valve lift and event variation, single port deactivation, cylinder and valve deactivation.

In this paper the attention is focused on the control of an electromechanical actuator depicted in Fig. 1. It is formed by two opposed magnets and two balanced springs working in parallel. 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 upper magnet pulls up and holds the armature, and the springs store 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 “armonic 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 exert high forces at low distances, power requirements of the system are quite low. Energy is spent for Joule effect and to restore the losses of potential energy due to friction. The traveling time of the EMVA depends by first approximation on the mechanical characteristic of the system, while the total opening/closing time will depend on the controller performances.

From the engine management viewpoint, the EMVA allow the variation of timing, valve event, late closure of intake valve and cylinder or port deactivation strategies. These systems are highly flexible, even if their working principle can give some problems in load variation [33]. The advantages in the adoption of electromechanical valve actuators have been largely described in technical literature [15, 16]. Among other, we remember the improvement of full load torque curve, the reduction of fuel consumption, particularly at low loads, the reduction of pollutant emissions, the possibility to reduce the emissions during cranking and to faster warm up.

One of the problems related to EMVA operations is the impact between moving parts during the opening and closing phases, producing acoustical noise and reducing the lifetime of the actuator. 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] [38]. This specification becomes more arduous to accomplish given that a valve stroke has to occur in a very short time (about 3 [ms]) to guarantee an adequate load variation ability of the valvetrain [16]. The control of the valve seating velocity has been identified to be crucial for the application of such an EMVA. In this regard different works have been published in the last years, some of them are briefly cited in the following.

Open-loop controllers. A first and simple way to approach the valve seating problem would consist of controlling the system with a series of properly tuned open-loop current pulses so that valve pushes near its seat as close as possible with a limited velocity [16, 33, 38, 44]. Unfortunately, this simple solution fails since the plant exhibits an unstable behavior [38], the valve response is sensitive to perturbation and suffers poor repeatability. Moreover, under real operating conditions, some disturbances make difficult to obtain a soft landing. This control issue is increased by the fact that this type of actuator is a strongly nonlinear system [12, 14, 44]. It is therefore mandatory to set-up feedback control strategies to attain a soft landing manoeuvre for the engine valve.

Sensorless controllers. For cost reasons it would be desirable to conceive position sensorless control solutions that are based only on low cost coil current measurements. In [6] a simplified modeling of the plant was used to derive an intuitive sensorless control based on the regulation of the ratio between the coil current and the derivative during the end-stroke of the valve. On the contrary, a

more sophisticated nonlinear magnetic model [14], incorporating hysteresis and mutual forces, was experimentally regressed and used to design a sliding mode control strategy based on a nonlinear observer [13]. A flatness-based tracking control was firstly presented in [11] and later coupled to a cycle-by-cycle feedforward control for the rejection of the gas force disturbance [10], where the flux measurement in an auxiliary coil was used to reconstruct position [23] and a nonlinear observer was furthermore developed to estimate velocity and gas force disturbances [10].

Closed-loop controllers. A closed-loop control of the valve position is mandatory to ensure stability, repeatability and robustness properties during valve operations. A physics-based model [44] for a single magnet, where the magnetic flux and force are described by piecewise functions in linear and saturation regions, was used to set up different valve position controllers [19,28,43], observer based [27] and coupled to a cycle-to-cycle adaptive strategy [18,29]. On the contrary, 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 [38–41]. A simplified nonlinear model was used to design an exact linearization based control law [20]. Being the controller based on lie-derivatives of the model equations, its robustness can be rather weak. The use of a highly predictive dynamic model for such a nonlinear plant becomes mandatory.

Valve seating control strategies proposed in previous works have been all conceived according to a centralized control approach. The regulation is performed by a controller designed for the plant between the coil voltage (inputs) and valve position (output). Therefore, all strongly coupled and nonlinear dynamics (electrical, magnetic and mechanical) characterizing such an EMVA should be taken into account in the controller synthesis even if this leading to a rather complicated design.

In this paper we use a detailed and predictive FEM-based dynamic model [12] of the EMVA to set up a model-based decoupling control strategy of the EMVA that is implicitly organized according to a decentralized framework. The model, describing self and mutual magnetic effects (inductances, back-EMFs, magnetomotive forces) over linear and saturation regions of the materials and for all engine valve positions, is actually used as a tool to decouple and linearize the nonlinear cross-coupled electric, magnetic and mechanical dynamics of the actuator.

The whole EMVA control schema results in two nested feedback controllers. Two inner controllers (CCC, Coil Current Controller) regulate the upper and lower coil currents, and an outer one has the task to track the valve position along a soft landing trajectory (SLC, Soft Landing Controller). The SLC outputs a demanded magnetic force to exert on the valve, force that is then transformed and splitted into desired coil currents to be tracked by the internal CCC. A proper Force-To-Currents (FTC) inversion algorithm has been developed to this aim.

In this framework the design of the SLC and CCC can be performed separately, although the performances of the inner controllers can alter those of outer one. In this regard, different CCC have been designed and tested in this work. A model based control has been firstly designed and tested actuating the voltage as continuous and pulse width modulated. Subsequently, a multi-hysteresis function based controllers has been designed and tested.

Furthermore, the control task related to the first lift manoeuvre of the engine valve has been also tackled (FCC, First Catching Control). An open-loop valve position control exploiting the mechanical resonance has been designed with the aim to limit the current peaks in the coils and energy losses. The performances have been compared with the ones of a quasi steady first valve lift manoeuvre.

The model-based decoupling control strategies are extensively tested in simulations and the results in terms of valve seating velocities, rising time and energy losses for different coil current controllers, are summarized in the last section.

A FEM-BASED NONLINEAR MODEL OF THE EMVA

Among many models proposed in technical literature [9,14,16,38,44] in this work we adopted a lumped parameter dynamic model derived from a hybrid analytical-FEM approach [12] for his high predictive capabilities shown in reproducing the strongly nonlinear dynamics characterizing such a magnetic valve actuator. The model reproduces quite well 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. In the following we briefly present the EMVA model equations (see [12] for all details).

The model can be decomposed into three strongly interconnected subsystems as shown in the block scheme of Fig. 2. The dynamics of the upper and lower coil currents (i_u and i_l) are described in the electrical subsystem, while the valve position (h) and velocity (v) dynamics are described in the mechanical block. The central magnetic block determines the total magnetomotive force exerted on the armature as combined action produced by the two magnets and depending on both currents and position.

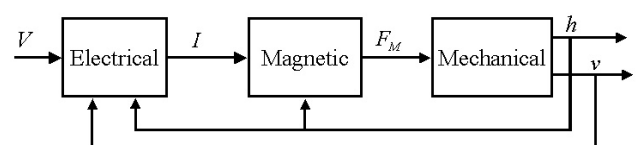


Figure 2 – Block scheme of the EMVA model.

The model equations written in a compact form are:

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

$$\frac{dv}{dt} = \frac{1}{m}(-K(h - h_{eq}) - \sigma(v)v + F_M(I, h) + F_{ext}(t)) \quad (1b)$$

$$\frac{dh}{dt} = v, \quad (1c)$$

where: $I = (i_u \ i_l)^T$ groups the coil currents; $V = (V_u \ V_l)^T$ is the vector of the voltages applied on the coils; $R = \text{diag}(r_u \ r_l)$ is the matrix of the coil electrical resistances; $K_s(h - h_{eq})$ is the sum of the spring forces; σv is a linear viscous friction; $F_{ext}(t)$ represents the sum of possible external forces (disturbance), basically it refers to gas pressure forces acting on the engine valve. All the variable meanings and the related symbols are detailed in Tab. 1.

Symbol	Unit	Value
g_{max}	mm	7
h_{min}	mm	0.05
h_{max}	mm	7
h_{eq}	mm	3.525
r_u	Ω	0.3647
r_l	Ω	0.3647
m	g	80.9
K	N/mm	100
σ	Ns/m	3.5

Table 1 – Values of the EMVA model parameters.

Since some residual air-gap could be considered between the armature and cores at opening/closing 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 $0 \leq h_{min} < h_{max} \leq g_{max}$, and where $h_{min, max}$ are the two end-stroke positions. This implies that the equilibrium position is $h_{eq} = (h_{max} + h_{min})/2$ having assumed that the two springs are equally preloaded.

The strong nonlinearities in the model (1) are: the matrix function $\Lambda(I, h)$, composed by the self and mutual coil inductances functions; the vectorial function $Z(I, h)$ grouping the self and mutual back-EMFs caused by the motion of the armature in a magnetic field; the total magnetomotive force $F_M(I, h)$ sum of the forces yielded by the single magnets and their mutual effects. The analytical expressions of such nonlinearities are:

• the **inductance matrix**,

$$\Lambda(I, h) = \begin{pmatrix} L(i_u, h) & M(i_l, g_{max} - h) \\ M(i_u, h) & L(i_l, g_{max} - h) \end{pmatrix}, \quad (2)$$

where

$$L(i, h) = m_s(h) + \frac{n_s(h)s_s(h)}{1 + \left(\frac{\pi}{2}s_s(h)i\right)^2} \quad (3)$$

and

$$M(i, h) = m_m(h) + \frac{n_m(h)s_m(h)}{1 + \left(\frac{\pi}{2}s_m(h)i\right)^2}, \quad (4)$$

are respectively the self and mutual inductance functions and g_{max} is the maximum air-gap between the armature and upper core;

• the **back-EMFs vector**,

$$Z(I, h) = \begin{pmatrix} z(i_u, i_l, h) & -z(i_l, i_u, g_{max} - h) \end{pmatrix}^T, \quad (5)$$

where

$$z(i_u, i_l, h) = z_s(i_u, h) - z_m(i_l, g_{max} - h) \quad (6)$$

is given by

$$z_s(i, h) = m'_s(h)i + \frac{n_s(h)s'_s(h)i}{1 + \left(\frac{\pi}{2}s_s(h)i\right)^2} + \frac{2}{\pi}n'_s(h) \arctan\left(\frac{\pi}{2}s_s(h)i\right) \quad (7)$$

and

$$z_m(i, h) = m'_m(h)i + \frac{n_m(h)s'_m(h)i}{1 + \left(\frac{\pi}{2}s_m(h)i\right)^2} + \frac{2}{\pi}n'_m(h) \arctan\left(\frac{\pi}{2}s_m(h)i\right), \quad (8)$$

respectively, the self and mutual back-EMF terms for each magnet;

• the **magnetomotive force**,

$$F_M(i_u, i_l, h) = F_{M_u}(i_u, h) + F_{M_l}(i_l, h) + F_{M_c}(i_u, i_l, h), \quad (9)$$

where

$$F_{M_u}(i, h) = m_F(h)i + \frac{2}{\pi}n_F(h) \times \arctan\left(\frac{\pi}{2}(s_{0F}(h) + s_{1F}(h)i)i\right) \quad (10)$$

and

$$F_{M_l}(i_l, h) = -F_{M_u}(i_l, g_{max} - h) \quad (11)$$

are the main forces exerted by the upper and lower magnets respectively, and

$$F_{M_c}(i_u, i_l, h) = i_u z_m(i_l, g_{max} - h) \quad (12)$$

is a second order mutual term.

The functions of the valve position present in (3), (4), (7), (8) and (10) are called *shaping factors*. Their behavior strongly depends on the magnetic circuit path given by each position of armature and the ferromagnetic materials as well. To this aim FEM simulations data were used to estimate these factors for each armature position and then regressed with polynomial functions. Nevertheless, we preferred to implement them by means of look-up tables better addressed for simulation purposes. Further details on the identification and modeling of the shaping factors can be found in [12].

We point out that FEM analysis shown that the percentage ratio of the mutual inductance with respect to the self one ($M/L \cdot 100$) varies between 1% and 11% for primary current in the range [1; 10] [A] and air-gap in the range [0; g_{max}], reaching value near 16% for current of about 20 [A].

OUTLINE OF THE EMVA CONTROL UNIT

The outline of the control unit of the EMVA (see Fig. 3) reflects the decoupling control approach here proposed.

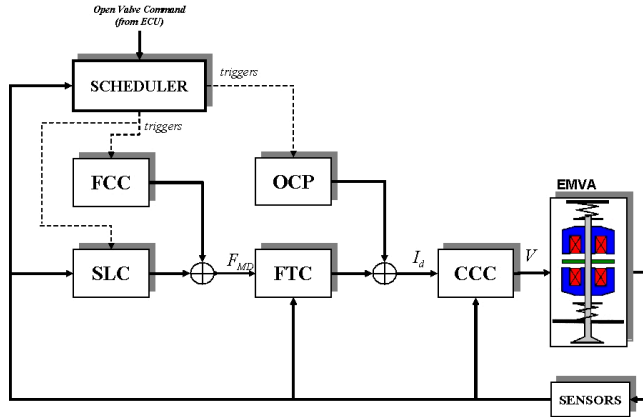


Figure 3 – Block scheme of the EMVA control unit.

The *Scheduler* enables/disables the control tasks in order to accomplish all engine valve operations. The *First Catching Controller* (FCC) and the *Soft Landing Controller* (SLC) govern the mechanical dynamics of the EMVA. In particular, FCC brings the engine valve from the initial half-stroke position to the closing one, and the SLC controls the valve motion during the last part of the valve stroke. The output variable of these controllers is the desired magnetic force to exert on the armature. The *Force - To - Currents* (FTC) block transforms and splits the desired magnetic force produced by the previous controllers into the references for the downstream *Coil Current Controllers* (CCC) which has the task to drive the current in the coils. The *Open-loop Current Pulses* (OCP) block yields those desired current values required to perform all other valve operations as for example its holding in full closing or opening positions. The description of each control block is given in what follows.

SCHEDULER

The *Scheduler* ensures the synchronization of all control functions for the right execution of all valve operations. The scheduler has been designed and coded as a finite state machine whose evolution is driven by the valve position and external open/close valve commands sent by the Engine Control Unit (ECU). A brief description of the scheduler states/actions is now given.

First Catching State. The scheduler is in this state at key-on where the engine valve is in the initial half stroke position. The scheduler enables the FCC until the armature does not reach the upper magnet. The FCC is then disabled and the scheduler transits in the *Close State*.

Close (Open) State. The scheduler is in this state when the valve is fully closed (opened). Here, the OCP is enabled to yield a proper current value i_C (i_O) for the upper

(lower) CCC enough to hold the valve closed (opened). The opposite lower (upper) magnet is not energized in this state. The scheduler remains in this state waiting for an open (close) valve signal from the ECU. When the command arrives the scheduler transits in the *Release State*.

Release State. The scheduler is in this state during the first part of the valve opening (closing). Here, the coil currents are controlled at zero and the valve proceeds in “free” evolution towards the lower (upper) magnet. The scheduler transits in the *Catch State* when the distance between the armature and the end of stroke becomes less than a given threshold Δ_{catch_O} (Δ_{catch_C}).

Catch State. In this state the OCP is enabled to yield a proper current value i_{catch_O} (i_{catch_C}) enough to create the better magnetic conditions for the next soft landing activities. The scheduler exits from this state when the distance between the armature and the end-stroke becomes less than a given threshold Δ_{sl_O} (Δ_{sl_C}).

Soft Landing State. In this state the scheduler enables the SLC that hooks the armature and controls the valve motion along a soft landing trajectory $h_{dO}(t)$ ($h_{dC}(t)$). When the valve is close to the end-stroke h_{max} (h_{min}) the scheduler transits in the open (close) state.

OPEN-LOOP CURRENT PULSES

The *Open-loop Current Pulses* (OCP) block yields proper coil current references when the upstream position controllers (FCC and SLC) are disabled. It is triggered by the scheduler and its output $I_d = [i_{u_d} \ i_{l_d}]^T$ depends on the EMVA working conditions coded by the scheduler.

The values $I_{dO} = [0 \ i_O]^T$ and $I_{dC} = [i_C \ 0]^T$ are outputted when the valve is fully opened and closed, respectively. The values of these desired currents have to be tuned so that the produced magnetic force is enough at least to balance the spring force in that positions. We point out that the values for i_O and i_C are in general different, given that the residual air-gaps between the armature and upper/lower magnet (i.e. h_{min} and $g_{max} - h_{max}$) could be different and so the magnetic force that strongly depends on it.

The values $I_{dO_{catch}} = [0 \ i_{O_{catch}}]^T$ and $I_{dC_{catch}} = [i_{C_{catch}} \ 0]^T$ are instead yielded when the catch states occur respectively during the valve opening and closing. These currents create the better magnetization conditions for the subsequent soft landing task. Roughly speaking, from the control viewpoint, this action increase the controllability of the system. The $i_{O_{catch}}$ and $i_{C_{catch}}$ values have to be tuned so that the armature approaches to the end of stroke avoiding to impact on the magnet.

From the position control viewpoint, the injection of these catching current pulses can be likened to a feed-forward action whose intensity increases with the rest of the valve

stroke (i.e. the (9) evaluated on I_d and $h(t)$) allowing a quickly compensation of the spring forces and the friction as well. Nevertheless, the valve seating velocities would appear too high ($0.2 \div 0.5$ [m/s], see [16, 33, 38, 44]) for engine applications. Therefore, a closed-loop control on the valve position becomes mandatory to reduce further the landing velocities as well as the sensitivity with respect to parameter uncertainties and perturbations.

SOFT LANDING CONTROLLER

The scheduler enables the *Soft Landing Controller* when the engine valve is in the last part of its opening/closing stroke (i.e. the soft landing state). The controller has the arduous task to hook the armature and to stop the valve motion smoothly in a very short time. The design of a high dynamic tracking controller becomes therefore mandatory to cope this task.

Fig. 4 shows a block scheme of the proposed soft landing controller. It is formed by a model-based feedback action, compensating the spring and friction forces, coupled to a PD action on the tracking error compensating all other model uncertainties.

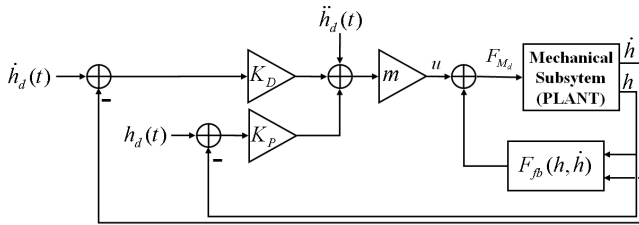


Figure 4 – Block scheme of the soft landing control.

From the equations (1b) and (1c) we get the feedback control law:

$$F_{Md}(h, v, u) = F_{fb}(h, v) + mu(t), \quad (13)$$

where $F_{fb}(h, v) = K(h - h_{eq}) + \sigma v$ and the $u(t)$ is given by a PD controller augmented by a feed-forward acceleration term:

$$u(t) = K_P e + K_D \dot{e} + \ddot{h}_d(t), \quad (14)$$

where $e(t) = h_d(t) - h(t)$ is the position tracking error.

Introducing the (13) and (14) in the mechanical model (1b) we get the position tracking error equation

$$\ddot{e} + K_D \dot{e} + K_P e = 0, \quad (15)$$

that is asymptotically stable choosing $K_P = 1/(\tau_1 \tau_2)$ and $K_D = K_P \cdot (\tau_1 + \tau_2)$, where the time constants τ_1 and τ_2 regulate the convergence rate of tracking error.

Numerically, the desired time constants have been tuned to have a settling time of 0.4 [ms] (that is about $1/7 t_R$, where $t_R = \pi \sqrt{M/K} \approx 2.82$ [ms] is the natural rise time). In this case, the values $\tau_1 = 0.1$ [ms] and $\tau_2 = \tau_1/2$ were chosen.

Remarks. The valve velocity term, necessary to compute the control law (13) and (14), can be obtained through a low-pass derivative filtering of valve position $h(t)$ supposed measurable. Since an unavoidable noise is present on the position measurement, the velocity filter will have to be designed accurately, trading-off between bandwidth and signal-to-noise ratio. Nevertheless, to avoid numerical derivatives, model based velocity observers [13, 27] could be adopted in conjunction with a disturbance observer [10].

In this regard, we point out that the control law (13) and (14) has been derived implicitly assuming that no external force disturbances ($F_{ext}(t) \equiv 0$) were present on the valve. Unfortunately, gas pressure forces can affect the valve motion in a real engine. Consequently, if these perturbations are too large the control performances can strongly deteriorate and a compensation of this disturbance would become mandatory. To this aim, a further model based control term $\hat{F}_{ext}(t)$, predicting the external disturbance forces, injected in (13) would improve the robustness of the controller re-establishing acceptable performances. Indeed, the response of the tracking error (15) would differ from the ideal one being now driven by the model prediction error $\delta_{ext}(t) = (\hat{F}_{ext}(t) - F_{ext}(t))/m$. Gas force models proposed in [10, 36] could be used for our control purposes.

LANDING TRAJECTORY. The reference position $h_d(t)$ has to describe a soft trajectory from the actual valve position $h(t_0)$ (where t_0 denotes the instant when SLC is triggered) to the end of stroke one h_{min} (or h_{max}) guaranteeing a smooth clutch of the SLC. A high control effort can in fact occur if the initial tracking errors $e(t_0)$ and $\dot{e}(t_0)$ in (15) are not small and if the desired valve acceleration at t_0 is very different from the valve one.

A signal able to meet all these requirements is given by the response of a third order linear system with unitary gain,

$$H_d(s) = \prod_{j=1}^3 \frac{1}{1 + s\tau_{d_j}}, \quad (16)$$

excited by a step of amplitude h_{min} (or h_{max}), and having as initial conditions h_{d0} , $\dot{h}_{d0}$ and $\ddot{h}_{d0}$, respectively, the position, the velocity and the acceleration of the engine valve at the instant t_0 . The initial condition on the desired acceleration guarantees the continuity of the control force (13) to the actual magnetomotive one (9) acting on the armature just before t_0 . The landing trajectory in terms of slope and duration can be easily shaped tuning properly the desired time constants.

Once the time constants are fixed the landing trajectory becomes a parameterized function of the valve state at the instant t_0 . Unfortunately only the valve position is immediately known since it is measured. An approximation, $\hat{h}(t_0)$, of the valve speed is however provided by the derivative filter used in the controller (13) and (14).