Hybrid Command Governors for Idle Speed Control in Gasoline Direct Injection Engines[1]

Leonardo Albertoni(1), Andrea Balluchi(2), Alessandro Casavola(3),
Claudio Gambelli(1) and Edoardo Mosca(1) and Alberto L. Sangiovanni–Vincentelli(2,4)

(1)Dip. di Sistemi ed Informatica, Università di Firenze, Via S.Marta, 3 - 50139 Firenze, Italy
Tel. +39-055-4796258 - Fax. +39-055-4796363, {albertoni,gambelli,mosca}@dsi.unifi.it
(2)Parades, Via San Panteleo, 66 - 00186 Roma, Italy
Tel. +39-06-68807923 - Fax. +39-06-68807926, {balluchi,alberto}@parades.rm.cnr.it
(3)Dip. di Elettronica, Informatica e Sistemistica Università della Calabria, Arcavacata di Rende - (CS), 87037 Italy
Tel. +39-984-494724 - Fax. +39-984-4947, casavola@si.deis.unical.it
(4)EECS Dept., University of California at Berkeley, CA 94720, USA, alberto@eecs.berkeley.edu

Abstract

The idle speed control problem for automotive GDI engines is formalized as a constrained optimal control problem for a hybrid model of the GDI engine, where fuel consumption is the cost function to be minimized. A sub-optimal but effective and easily implementable solution is obtained by resorting to the *Command Governor* methodology for a discrete-time abstraction of the hybrid model. Simulation results of the hybrid closed-loop system are presented.

1 Introduction

The main targets of the design of 4-stroke gasoline engines for passenger cars are: improvement of safety, driveability and comfort, minimization of fuel consumption and compliance with the emissions standards. Besides the direct economic benefit for customers, reduction of fuel consumption results in reduction of the combustion product CO_2, which is a critical issue due to the well known effects of CO_2 on global environmental warming.
High fuel economy, as well as high driving performances, can be achieved by modern Gasoline Direct Injection (GDI) engines (see [4] for an extensive description). GDI engines operate either in homogeneous charge (with stoichiometric air/fuel ratio) or stratified charge (with lean mixtures at high air/fuel ratio). Direct injection is characterized by: 1) low pumping and heat losses, which increase thermal efficiency, 2) low temperature of charge air, producing high volumetric efficiency and anti-knock characteristics and 3) high response and superior transient driveability (due to direct fuel injection into the cylinder). Using the stratified mode at low loads and engine speed (i.e. in idle speed), fuel consumption can be reduced by 20-25%.
In this paper, the idle speed control problem is formalized as a fuel consumption minimization problem, subject to constraints on engine speed and air-to-fuel ratio, for a hybrid model of the GDI engine. An effective sub-optimal solution is developed by resorting to the *Command Governor* (CG) approach. A CG is a nonlinear device which is added to a pre-compensated control system. The latter, in the absence of the CG, is designed so as to perform satisfactorily in the absence of constraint violations. Whenever necessary, the CG modifies the reference to the closed-loop system so as to avoid violation of constraints. Specific merits of the CG approach in dealing with constraints are that it can handle absolute and incremental constraints on input and state-related variables of the plant and that the numerical burdens of the on-line computation can be modulated according to the available computing power. Studies along these lines have been illustrated in [5]-[3]. An idle speed controller for diesel engines, based on the CG methodology, has been presented [6].
The paper is organized as follows: in Section 2, a hybrid model of a GDI engine is presented and the idle speed control problem is formalized. In Section 3, the proposed approach based on a hybrid CG is illustrated. In Section 4, a sub-optimal implementable solution is synthesized and simulation results of the closed-loop hybrid system model are reported.

2 Hybrid model of GDI engines

In this section a nonlinear hybrid model of a 4–stroke 4–cylinder inline GDI engine is presented (see [2] for an extensive discussion on hybrid modeling in powertrain control). The proposed model has been developed and identified in collaboration with Magneti Marelli Powertrain (Italy). To achieve fuel consumption minimiza-

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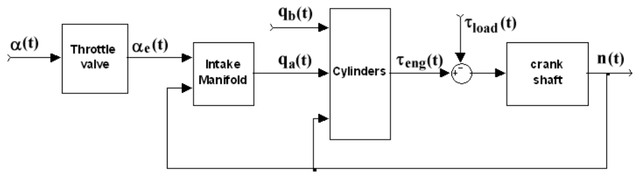

Figure 1: GDI engine hybrid model.

tion, we assume that the GDI engine operates in stratified charge only.
As depicted in Figure 1, the GDI engine hybrid model is composed of four interacting subsystems: the *throttle valve*, the *intake manifold*, the *cylinders* and the *crankshaft*. The control inputs are: the command to the throttle valve[1], referred to as α, and the mass of fuel injected in each engine cycle, referred to as q_b. In stratified charge, spark ignition has to be synchronized with fuel injection and cannot be used as a control input.

The dynamic of the *throttle valve* is modeled as a first-order lag with input delay

$$\dot{\alpha_e}(t) = \frac{1}{\tau_\alpha}\alpha_e(t) + \frac{1}{\tau_\alpha}\alpha(t - d_\alpha) \ , \qquad (1)$$

where α and α_e denote, respectively, the throttle valve command and position, and d_α models the actuator delay.
The *intake manifold* dynamics, described in terms of the intake manifold pressure p, is

$$\dot{p}(t) = K_{gas}[F_{th}(\alpha_e(t)) - F_{cyl}(p(t), n(t))] \ , \qquad (2)$$

where F_{th} and F_{cyl} are nonlinear functions that model the input air–flow rate into the manifold and the output air–flow rate, resp., and n denotes the crankshaft revolution speed.
The *cylinders* hybrid model describes the generation of the engine torque $\mathcal{T}_{eng}(t)$, which is modeled as a piecewise constant signal synchronized with the dead center events. In 4-cylinder inline engines, pistons reach either bottom or top dead-centers when the crankshaft angle θ is multiple of 180^0. For the k–th expansion stroke, the amount of the engine torque $\mathcal{T}_{eng}(k)$ depends in a nonlinear fashion on: the mass of injected fuel $q_b(k)$, the normalized air–to–fuel ratio $\lambda(k)$ of the loaded mixture, and the value of the engine speed at the beginning of the stroke $n(t_k)$:

$$\mathcal{T}_{eng}(t) = \mathcal{T}_{eng}(k) = \mathcal{T}_{eng}(q_b(k), \lambda(k), n(t_k)) \qquad (3)$$

for $t \in [t_k, t_{k+1})$. The normalized air–to–fuel ratio $\lambda(k)$ of the mixture during the k-th expansion stroke is

$$\lambda(k) = \frac{q_a(k-1)}{q_b(k)} \left(\frac{q_{a0}}{q_{b0}}\right)^{-1} \qquad (4)$$

[1]Used to control the amount of air loaded by the cylinders during intake strokes.

where $\frac{q_{a0}}{q_{b0}}$ stands for the stoichiometric air–to–fuel ratio[2].
Finally, the *crankshaft* block describes the evolution of the crankshaft revolution speed n (in rpm) and the crankshaft angular position θ (in degrees),

$$\begin{aligned} \dot{n}(t) &= K_J(\mathcal{T}_{eng}(t) - \mathcal{T}_{load}(t)) \qquad (5)\\ \dot{\theta}(t) &= 6\ n(t) \qquad (6) \end{aligned}$$

where $\mathcal{T}_{load}$ is the load torque, which consists of the sum three distinct amounts: the pumping torque $\mathcal{T}_p$ (a constant), the friction torque $\mathcal{T}_{fr}$ (linearly depending on n), and the torque $\mathcal{T}_d$ due to the auxiliary subsystems powered by the engine (e.g. air conditioning compressor, steering pump, electric generator, etc.). Furthermore, for reasons that will be clear in the following, it is convenient to split $\mathcal{T}_d$ as follows

$$\mathcal{T}_d(t) = \mathcal{T}_{unp}(t) + \mathcal{T}_{pr}(t)$$

where $\mathcal{T}_{unp}$ collects all *unpredictable* but bounded disturbance torques, whereas $\mathcal{T}_{pr}$ represents *predictable* amounts, usually larger than $\mathcal{T}_{unp}$, and

$$\mathcal{T}_{unp}(t) \in \mathcal{D}_1, \qquad \mathcal{T}_{pr}(t) \in \mathcal{D}_2, \qquad \forall t \ .$$

For instance, the air conditioning subsystem generates a load which can be considered predictable. In fact, we can assume to know both the time of the air conditioning switching and the corresponding value of load. This information can be exploited in order to achieve less conservative results. The interesting interactions in the hybrid models are due to the fact that the evolution of the event–based dynamics (3–4), which are triggered by dead-center events, depends on the continuous evolution of the crankshaft dynamics (5–6) that in turn is subject to the event–based engine torque $\mathcal{T}_{eng}$.

2.1 Problem formulation

The goal of this paper is the design of an idle speed control for GDI engines, which minimizes fuel consumption, maintains system variables within prescribed operative constraints and prevents engine stalls. Fuel minimization should be achieved both in steady-state conditions and during transients caused by disturbance torques acting on the crankshaft. Usually, the specification for the idle speed control is to maintain the engine speed n around a reference value n_0. In steady-state fuel consumption is strictly related to the engine speed reference value n_0, in that the lower n_0 the lower the fuel consumption. The reference value n_0 is determined by trading-off between fuel consumption minimization and the need of avoiding the engine to stall

[2]Notice that in (4), the normalized air–to–fuel ratio $\lambda(k)$ depends on the amount of air $q_a(k-1)$ loaded in the cylinder during the previous intake stroke: the one step delay models the compression stroke.

during transients due to load disturbances. In our approach, to achieve minimization of fuel consumption, we allow the engine speed n to vary in an interval around the nominal value n_0.
The idle speed control specification is as follows:

$$\min_{\alpha(t), q_b(k)} \sum_{0}^{\infty} q_b(k)$$
$$n(t) \in [n_0 - 40, n_0 + 40]\ (rpm), \quad n_0 = 750\ (rpm)$$
$$\alpha(t) \geq 0\ (degree), \qquad \dot{\alpha}(t) \in [-5, 5]\ (degree/s)$$
$$\lambda(k) \in [0.8, 3.5], \qquad q_b(k) \geq 1\ (mg)$$
$$\mathcal{T}_{unp}(t) \in \mathcal{D}_1 = [3, 8]\ (Nm)$$
$$\mathcal{T}_{pr}(t) \in \mathcal{D}_2 = [0, 12]\ (Nm)$$

The lower-bound of 710 rpm on $n(t)$ is imposed to prevent the engine from stalling, whereas the upper bound is dictated by fuel economy.

3 The proposed approach

The throttle valve is controlled with a discrete–time feedback with sampling period $T_c = 10\ ms$. The engine hybrid model in Section 2 is linearized about the operating point corresponding to the nominal idle speed n_0 and the disturbance torque $\mathcal{T}_d = \mathcal{T}_{d0} = 8.5\ Nm$. Then, the linearized model is discretized at the throttle valve control period T_c. The time between two subsequent dead centers is approximated with its value at the engine speed n_0, i.e. $40\ ms$, and expressed as 4 times T_c (assuming a synchronization of the engine cycle with the throttle valve control). The time delay d_α of the throttle valve actuation is assumed to be $2T_c$.
With this elaboration, the engine model becomes

$$\begin{cases} x_p(t+1) = Ax_p(t) + Bu(t) + B_d d(t) \\ y(t) = Cx_p(t) \end{cases} \tag{7}$$

where

$$x_p(t) = \begin{bmatrix} n(t) - n_0 \\ \cdots \\ n(t-4) - n_0 \\ p(t) - p_0 \\ \cdots \\ p(t-4) - p_0 \\ \alpha_e(t) - \alpha_{e0} \\ q_b(t-1) - q_{b0} \\ \cdots \\ q_b(t-2) - q_{b0} \\ \alpha(t-1) - \alpha_0 \\ \cdots \\ \alpha(t - d_\alpha) - \alpha_0 \end{bmatrix} \qquad u(t) = \begin{bmatrix} q_b(t) - q_{b0} \\ \alpha(t) - \alpha_0 \end{bmatrix}$$

$$d(k) = \mathcal{T}_d(t) - \mathcal{T}_{d0} \qquad y(t) = \begin{bmatrix} n(t) - n_0 \\ \lambda(t) - \lambda_0 \end{bmatrix}$$

and A, B, B_d and C are obtained from the linearization and discretization of the engine hybrid model.
To achieve minimization of the fuel consumption, we design a tracking controller for the engine speed n and the normalized air–to–fuel ratio λ, with reference signals produced on–line by a command governor (CG). The CG ensures that the prescribed constraints are never violated, irrespective of all possible load disturbance occurrences, and fuel consumption is optimized. Using varying reference signals we avoid to force the engine speed to be remarkable higher than strictly necessary to keep the engine running. This situation often occurs in the standard approach where a constant reference engine speed is used.
For the engine at hand, it results that fuel consumption is minimized when $n = n_r = 710\ rpm$ and $\lambda = \lambda_r = 2$. Then, the basic strategy underlying the use of a CG will be that to apply the optimal reference values n_r and λ_r and let the CG to modify them on-line whenever their application would lead to constraint violation. Details on CG theory are reported in the next session.

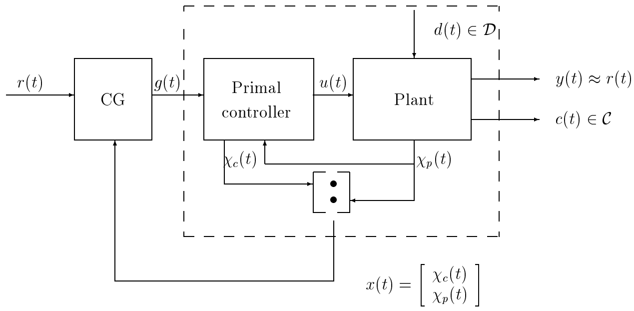

Figure 2: Command Governor structure.

3.1 The Command Governor (CG) approach

A CG control scheme, with plant, primal controller and CG device, is depicted in Fig. 2. A state-space description of the plant regulated by the primal controller is

$$\begin{cases} x(t+1) &= \Phi x(t) + Gg(t) + G_d d(t) \\ y(t) &= H_y x(t) \\ c(t) &= H_c x(t) + Lg(t) + L_d d(t) \end{cases} \tag{8}$$

In particular, $x(t) \in \mathbb{R}^n$ is the state which includes plant and compensator states; $g(t) \in \mathbb{R}^m$, which would be typically $g(t) = r(t)$ if no constraints were present (no CG present), is the CG output, viz. a suitably modified version of the reference signal $r(t) \in \mathbb{R}^m$; $d(t) \in \mathbb{R}^{n_d}$ an exogenous disturbance satisfying $d(t) \in \mathcal{D}$, $\forall t \in \mathbb{Z}_+$, with $\mathcal{D}$ a specified convex and compact set such that $0_{n_d} \in \mathcal{D}$; $y(t) \in \mathbb{R}^m$ is the output, viz. a performance related signal which is required to track $r(t)$; $c(t) \in \mathbb{R}^{n_c}$ the vector to be constrained, viz. $c(t) \in \mathcal{C}$, $\forall t \in \mathbb{Z}_+$, with $\mathcal{C}$ a specified convex and compact set. It is assumed that

1. System (8) is asymptotically stable;
2. System (8) is offset free, viz. $H_y(I - \Phi)^{-1}G = I_m$

CG problem consists of finding, at each time t, a command

$$g(t) := \underline{g}(x(t), r(t)) \tag{9}$$

as a function of the current state $x(t)$ and reference $r(t)$, in such a way that $g(t)$ is the best approximation of $r(t)$ at time t, under the constraint $c(t) \in \mathcal{C}$, $\forall t$, and all possible disturbance sequences $d(t) \in \mathcal{D}$. Moreover, it is required that: 1) $g(t) \to \hat{r}$ whenever $r(t) \to r$, with $\hat{r}$ the best feasible approximation of r; and 2) the CG have a finite settling time, viz. $g(t) = \hat{r}$ for a possibly large but finite t whenever the reference stays constant after a finite time.

By linearity, one is allowed to separate the effects of initial conditions and input from those of disturbances, e.g. $x(t) = \bar{x}(t) + \tilde{x}(t)$, where $\bar{x}$ is the disturbance-free component (depending on initial state and input only) and $\tilde{x}$ depends on the disturbances only. Then, denote the disturbance-free steady-state solutions of (8), for a constant command $g(t) \equiv w$, as follows

$$\begin{array}{rcl} \bar{x}_w & := & (I_n - \Phi)^{-1} G w \\ \bar{y}_w & := & H_y (I_n - \Phi)^{-1} G w \\ \bar{c}_w & := & H_c (I_n - \Phi)^{-1} G w + L w \ . \end{array} \tag{10}$$

Consider next the following set recursion

$$\begin{array}{rcl} \mathcal{C}_0 & := & \mathcal{C} \sim L_d \mathcal{D} \\ \mathcal{C}_k & := & \mathcal{C}_{k-1} \sim H_c \Phi^{k-1} G_d \mathcal{D} \\ \mathcal{C}_\infty & := & \bigcap_{k=0}^{\infty} \mathcal{C}_k \end{array} \tag{11}$$

where $\mathcal{A} \sim \mathcal{E}$ is defined as $\{a \in \mathcal{A} : \ a + e \in \mathcal{A}, \forall e \in \mathcal{E}\}$. It can be shown that the sets $\mathcal{C}_k$ are non-conservative restrictions of $\mathcal{C}$ such that $\bar{c}(t) \in \mathcal{C}_\infty$, $\forall t \in \mathbb{Z}_+$, implies $c(t) \in \mathcal{C}$, $\forall t \in \mathbb{Z}_+$. Thus, one can consider only disturbance-free evolutions of the system and adopt a "worst case" approach. Next consider, for a small enough $\delta > 0$, the sets

$$\begin{array}{rcl} \mathcal{C}^\delta & := & \mathcal{C}_\infty \sim \mathcal{B}_\delta \\ \mathcal{W}^\delta & := & \{w \in \mathbb{R}^m : \ \bar{c}_w \in \mathcal{C}^\delta\} \end{array} \tag{12}$$

where $\mathcal{B}_\delta$ is the ball of radius δ centered at the origin. In particular, $\mathcal{W}^\delta$, which we assume non-empty, is the set of all commands whose corresponding steady-state solution satisfies the constraints with margin δ.

The main idea is to choose at each time step a constant *virtual*, command $v(\cdot) \equiv w$, with $w \in \mathcal{W}^\delta$, such that the corresponding virtual evolution fulfills the constraints over a semi-infinite horizon and its distance from the constant reference of value $r(t)$ is minimal. Such a command is applied, a new state is measured and the procedure is repeated. In this respect we define the set

$$\mathcal{V}(x) = \{w \in \mathcal{W}^\delta : \ \bar{c}(k, x, w) \in \mathcal{C}_k, \ \forall k \in \mathbb{Z}_+\} \ , \tag{13}$$

where

$$\bar{c}(k, x, w) := H_c \left(\Phi^k x(t) + \sum_{i=0}^{k-1} \Phi^{k-i-1} G w \right) + L w \tag{14}$$

is to be understood as the disturbance-free virtual evolution at virtual time k of c from the initial condition x at time 0 under the constant command $v(\cdot) \equiv w$. As a consequence, $\mathcal{V}(x) \subset \mathcal{W}_\delta$. Moreover, if non-empty, it represents the set of all constant virtual sequences in $\mathcal{W}_\delta$ whose evolutions starting from x satisfies the constraints also during transients. Thus, taking as a selection index a quadratic cost, the CG output is chosen according to the solution to the following constrained optimization problem

$$g(t) = \arg \min_{w \in \mathcal{V}(x(t))} \|w - r(t)\|_\Psi^2 \tag{15}$$

where $\Psi = \Psi' > 0_p$ and $\|w\|_\Psi^2 := x' \Psi x$. The reader is referred to [1] for full details.

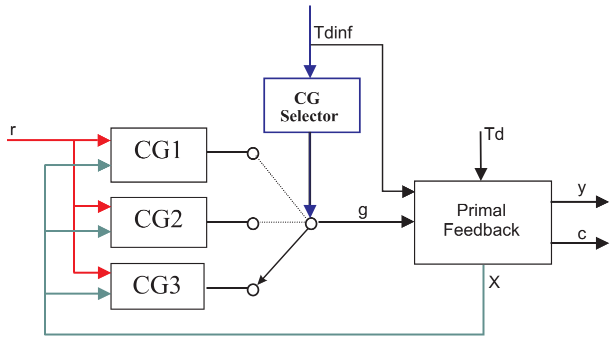

Figure 3: Hybrid CG structure.

4 Controller synthesis

In this section the design of a CG idle speed controller is presented. The controller consists of two nested loops:

- a switching LQ controller in the inner loop, whose objective is the minimization of fuel consumption during transients;
- a CG in the outer loop, whose objective is the minimization of fuel consumption during steady states and the verification of the constraints.

The inner loop and the outer loop are, respectively, described in Section 4.1 and in Section 4 below. Simulation results of the closed–loop hybrid system are reported in Section 4.3.

4.1 Primal Control

The CG approach requires preliminarily the design of a primal stabilizing controller which, because is supposedly to be used along with a CG, is designed without tacking into account the prescribed constraint. The one used here is depicted in Fig. 4. In order to have zero tracking error in steady-state we require the use of

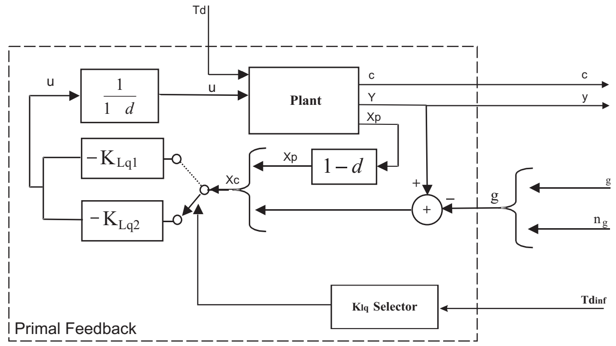

Figure 4: Primal hybrid feedback control structure.

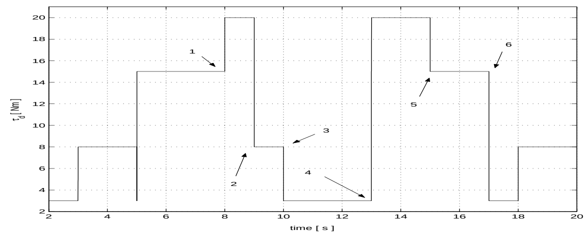

Figure 5: Profile of the load disturbance.

an integral action. This is done be resorting to the incremental model approach which consists of rewriting the model (7) in terms of the extended state

$$x_c(t) := \begin{bmatrix} \delta x_p(t) \\ \varepsilon(t-1) \end{bmatrix} \qquad \begin{array}{l} \delta x_p(t) := x_p(t+1) - x_p(t) \\ \varepsilon(t-1) = y(t-1) - g(t-1) \end{array}$$

being $g(t)$ the reference signal, and the incremental input $\delta u = u(t+1) - u(t)$

$$\begin{array}{rcl} x_c(t+1) & = & \Phi x_c(t) + G\delta u(t) \\ \varepsilon(t) & = & H x_c(t) \ . \end{array} \qquad (16)$$

Then, optimal LQ state feedbacks of the form

$$\delta u(t) = -K_{Lq} x_c(t) \ , \qquad (17)$$

which minimizes the following quadratic cost

$$J = \sum_{t=0}^{\infty} \|\varepsilon(t)\|^2_{\Psi_\varepsilon} + \|\delta u(t)\|^2_{\Psi_u} \qquad (18)$$

with $\Psi_\varepsilon = \Psi_\varepsilon' \geq 0$, $\Psi_u = \Psi_u' > 0$ can be easily determined. In particular, we have found convenient to determine two different LQ state feedback control laws:
Lq1: for predictable disturbance in steady–state;
Lq2: to handle on/off or off/on transitions of the predictable disturbance.
A supervisor (K_{lq} Selector) is in charge to identify when each specific controller has to be put in the loop on the basis of the input T_{dinf} that indicates in advance the state of the (ON-OFF) predictable disturbance. The main reason for using two state feedback control laws instead of a single one is that of having different gains during large transient occurrences and steady-state operations. This is convenient for trading-off between fuel consumption minimization and fast transients achievement. In fact, for fuel consumption minimization the weight Ψ_u in the cost has to be chosen remarkably larger than Ψ_ε. Under small disturbances this choice ensures low fuel consumptions. The embedded integral action ensures zero tracking error in steady-state.

4.2 CG application (External Loop)

Accordingly to the above primal control structure, we have designed a bank of three command governors, referred to as CG1, CG2 and CG3, each one in charge to deal with a different situation, as described below:

CG1: predictable disturbance off with Lq1 in the loop;

CG2: transitions of the predictable disturbance with Lq2 in the loop;

CG3: predictable disturbance on with Lq1 in the loop.

The selection of the CG to be applied is handled by the block "CG selector" (see Figure 3) on the basis of the state of the input T_{dinf}. The CGs have been designed on the incremental model (16) with the LQ primal controller specified as above. The only difference has regarded the admissible load disturbance ranges. Specifically,

CG1: $\mathcal{D}_1 \in [3, 8]$ (Nm) and $\mathcal{D}_2 = \{0\}$ (Nm);

CG2: $\mathcal{D}_1 \in [3, 8]$ (Nm) and $\mathcal{D}_2 = [0, 12]$ (Nm);

CG3: $\mathcal{D}_1 \in [3, 8]$ (Nm) and $\mathcal{D}_2 = \{12\}$ (Nm).

4.3 Simulations

In this section we report some simulation results obtained applying the proposed LQ-CG hybrid control strategy, illustrated in Sections 4.1 and 4, to the nonliner hybrid model of the plant described in Section 2. Simulations show that the discrete-time approximation of the plant described in Section 3 is good enough since the performances of the hybrid closed loop system are satisfactory, both in terms of fuel consumption and constraints fulfilment. In the results here reported, the reference set-point $r = [710, 2]$ was applied. This choice leaves the CG free of choosing the lower possible values of crankshaft speed to minimize fuel consumption and guarantee the verification of the constraints in any circumstance. Under the action of the load disturbance T_d depicted in Figure 5, the evolution reported in Figure 6 was obtained. The idle speed n is always at the lowest level compatible with loads and constraints (compare upper and lower plots in Figure 6 at time instants $t = 4, 14, 20$) and the constraints are always satisfied.

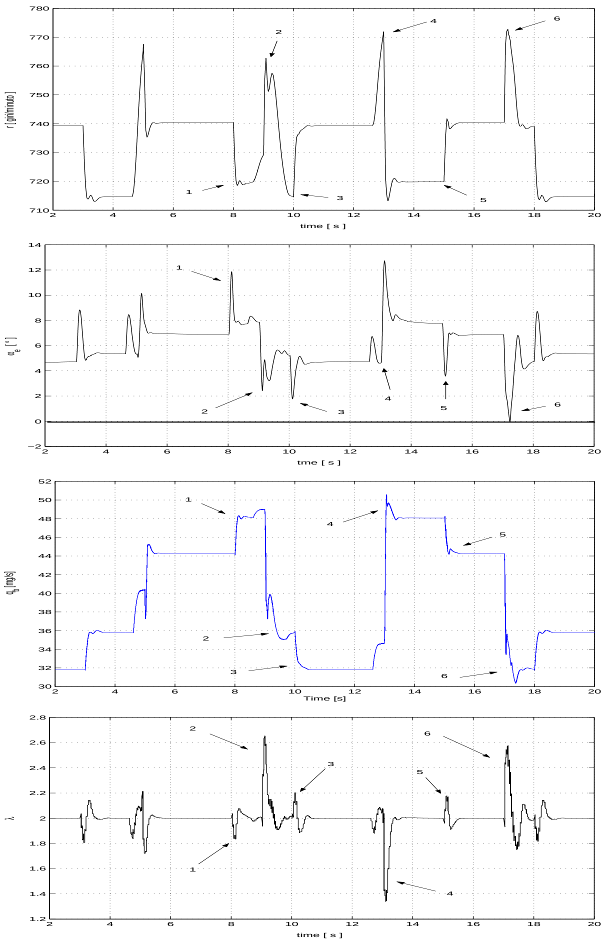

Figure 6: From the top: engine speed n, throttle valve angle α, injected fuel q_b and normalized air–to–fuel ratio λ.

Conclusions

In this paper the design of an idle speed control for automotive GDI engines has been considered. The main control objective was fuel consumption minimization. Load variations and constraints fulfilment on relevant system variables have been explicitly taken into account in the control design as well as the requirement for computationally inexpensive and easily implementable solutions.
A highly accurate nonlinear hybrid model for the stratified mode of operation of a GDI engine has been used to derive a low-dimensional linear discrete-time system used for control design purposes. The hybrid nature of the problem has reappeared in the linear discrete-time representation, essentially due to the presence of predictable load disturbances, and has led to the design of a hybrid CG unit for constraints fulfilment which uses two switching LQ optimal controllers as a primal control structure for ensuring nominal closed-loop stability and performance under linear regimes.
The CG approach has been instrumental not only for achieving lower fuel consumptions but also for improving the designer ability of explicitly taking care of prescribed constraints in the design phase, avoiding optimality degradation and extensively recurring to simulations for the assessment of the solution. It results that the fuel consumption during transients was reduced about of 50 %, essentially for the freedom in tuning the primal LQ switching control structure without the need of taking into account the constraints. The overall consumption reduction was about 2 %. This allows us to conclude that the proposed technique can achieve, over linear control methods, improvements on fuel consumption in the presence of constraints up to an extent which justifies the increase of computing burdens required by the hybrid CG algorithm.

Acknowledgments

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