IDLE SPEED CONTROL – A BENCHMARK FOR HYBRID SYSTEM RESEARCH [1]

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Abstract: The design of engine control systems has been traditionally carried out using a mix of heuristic techniques validated by simulation and prototyping with approximate mean–value models. However, the ever increasing demands on passengers' comfort, safety, emissions and fuel consumption imposed by car manufacturers and regulations call for more robust techniques and the use of cycle–accurate models. The use of hybrid methodologies is then natural because of the rich combination of time and event-based behaviors exhibited by a controlled engine. While there is no doubt that hybrid modeling is relevant for this application, its efficiency in providing industrial strength solutions is still debated. For this reason, it is important to corral the hybrid system research community to provide evidence of the quality of the proposed control solutions. In this perspective, we present a hybrid benchmark problem on "Idle Speed Control" proposed by the Network of Excellence HYCON. We hope this benchmark problem will also serve as the basis for comparison of different approaches, thus helping industry to identify the best solutions available.

Keywords: Automotive, engine modeling, hybrid systems.

1. INTRODUCTION

In the automotive industry, increased performance, safety and time-to-market pressure require the use of complex control algorithms with guaranteed properties. Best practices in this industry are based on extensive experimentation and tuning. This procedure needs a substantial overhaul to eliminate long re-design cycles and potential safety problems after the car is introduced in the market. Using more accurate models and control algorithms with guaranteed properties reduces greatly the need for extensive experimen-

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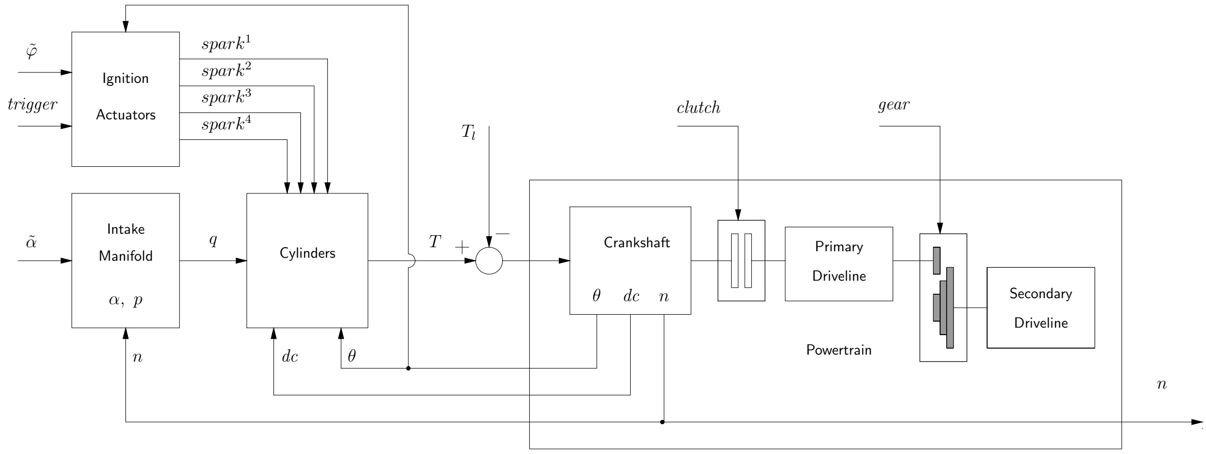

Fig. 1. Engine hybrid model.

tation and points to potential problems early in the design cycle.

In this general scenario, the synthesis of a control strategy for spark–ignition engines at idle speed is one of the most challenging problems. The goal is to maintain the engine speed as close as possible to a reference constant engine speed despite load torque disturbances (due to e.g. the air conditioning system, the steering wheel servo-mechanism) and engagements and disengagements of the transmission occurring when the driver operates on the clutch. In order to achieve the best fuel economy, the reference engine speed is chosen at the minimum value that yields acceptable combustion and emission quality, and noise, vibration and harshness (NVH) characteristics.

A survey on different engine models and control design methodologies for idle control is given in (Hrovat and Sun, 1997). Both time–domain (e.g. (Butts *et al.*, 1999)) and crank–angle domain (e.g. (Yurkovich and Simpson, 1997)) mean–value models have been proposed in the literature. More recently, hybrid system techniques have been applied to the idle speed control problem. The hybrid nature of the problem of engine control comes not only from the digital controllers used to manage an analog plant, but also from the behavior of the plant to be controlled. In fact, an accurate model of a four–stroke gasoline engine has a "natural" hybrid representation because: pistons have four modes of operation corresponding to the stroke they are in, while power–train and air dynamics are continuous–time processes. In addition, these processes interact tightly. In fact, the timing of the transitions between two phases of the pistons is determined by the continuous motion of the power–train, which, in turn, depends on the torque produced by each piston. This problem has been attacked with hybrid approaches in (Balluchi *et al.*, 2000*b*; Balluchi *et al.*, 2002; Balluchi *et al.*, 2004), using hybrid games, formal verification and *control–to–facet* techniques. Other approaches are being developed in several institutions. Given the difficulty and industrial relevance of the engine control problem, together with the availability of models and control algorithms, we believe that offering a complete description and the collateral material will allow researchers to apply their methods to the problem and as a consequence this will push the state of the art considerably.

In this paper, we present a particular benchmark problem in this domain: "Idle Speed Control". This benchmark has been developed under the sponsorship of the Network of Excellence HYCON. The documentation and the simulation files related to this benchmark problem are available at the HYCON web–page www.ist-hycon.org, under "WP2: Performance Evaluation Platform".

2. THE ENGINE HYBRID MODEL

In this section, a hybrid model of a 4–cylinder 4–stroke spark ignition engine equipped with an electronic–throttle is presented. The proposed hybrid model represents accurately the behavior of the engine during idle speed control. The overall system is composed of four blocks, namely the *ignition actuators*, the *intake manifold*, the *cylinders* and the *powertrain* (Figure 1) (to satisfy emission requirements, fuel injection is regulated so that the air and fuel mixture is stoichiometric). The *ignition actuators* deliver the sparks $spark^i$ to the cylinders with a timing defined by the desired spark advance angle $\tilde{\varphi}$. The latter represents the spark ignition control input. When the controller issues a new value $\tilde{\varphi}$, it emits the synchronization event *trigger*. The mass of air q loaded in the cylinders depends on the dynamics of the *intake manifold*. The manifold pressure p is controlled by a throttle valve powered by an electrical motor; α and $\tilde{\alpha}$ denote, respectively, the throttle valve position and the reference to the throttle valve controller. The air charge q is a function of the pressure p and of the crankshaft revolution speed n. The *cylinders* model describes the engine torque generation process. The engine torque T depends on the air charge q and on the timing of spark ignition. The timing sequence of the four strokes of each cylinder is determined by the motion of the piston between the top and the bottom *dead centers* (dc), i.e. the piston uppermost and lowermost positions. The position of the piston is determined by the crankshaft angle θ. Finally, the crankshaft revolution speed n depends on the powertrain dynamics. In idle speed control it is assumed that the gear is idle, while the clutch can be either open or closed. The powertrain is powered by the balance between the engine torque T and the load torque T_l, due to the auxiliary systems driven by the crankshaft (such as e.g. the generator).

2.1 Intake manifold

The intake manifold model gives the cylinder air charge q. A sufficiently accurate model of the air charge can be obtained abstracting away the intake manifold pumping fluctuations due to the periodic motion of pistons and valves. In fact, they are usually filtered out in air charge estimation algorithms for engine torque control. Denoting by α and p the throttle-valve position and the intake manifold pressure p, respectively, we have (see (Hendricks and Sorenson, 1990)):

$$\dot{\alpha} = -\frac{1}{\tau_{thr}}(\alpha - \tilde{\alpha}) \tag{1}$$

$$\dot{p} = \frac{RT_{air}}{V_{pln}}[f_{thr}(\alpha) - f_{cyl}(p,n)] \tag{2}$$

$$q = \frac{30}{n} f_{cyl}(p,n)\ . \tag{3}$$

Equation (1) represents the actuation dynamics of the throttle valve, with $\tilde{\alpha}$ being the reference command. The intake manifold pressure dynamics (2) depends on the balance between the air-mass flow through the throttle valve $f_{thr}(\alpha)$ and through the cylinder valves $f_{cyl}(p,n)$, modeled as follows:

$$f_{thr}(\alpha) = s_0 + s_1\,\alpha + s_2\,\alpha^2 \tag{4}$$

$$f_{cyl}(p,n) = c_0 + c_1\,p + c_2\,n + c_3\,p\,n\ . \tag{5}$$

2.2 Cylinders.

The engine torque T is given by the sum of the contributions T^i by the four cylinders:

$$T = \sum_{i=1}^{4} T^i\ . \tag{6}$$

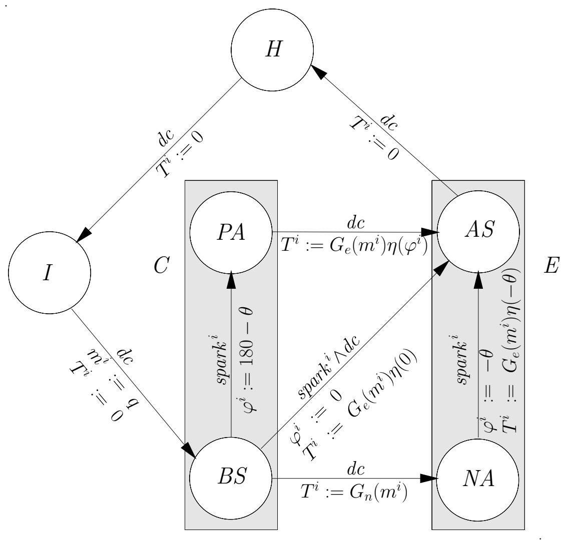

Fig. 2. Hybrid system describing the behavior of the i–th cylinder.

The profile of T^i depends on the current stroke of the cylinder (either *intake*, *compression*, *expansion*, or *exhaust*), the piston position, the mass of air loaded in the cylinder during the intake stroke, and the spark ignition timing. The contributions to the engine torque in the intake, compression and exhaust strokes are quite small and, for small throttle valve angles, only weakly dependent on the air charge. Then, the cylinder torque T^i is modeled as a piecewise constant signal, nonzero in the expansion stroke only, whose value in expansion takes into account the effects of the other three cylinders that evolves in the intake, compression and exhaust strokes.

At every cycle, the mixture is ignited by the spark. Ideally, heat release should occur instantaneously when the piston reaches the compression stroke top–dead–center. However, due to the nonzero combustion time, the maximum engine torque is achieved when spark ignition is given before the piston completes the compression stroke (*positive spark advance*). Delaying spark ignition to the expansion stroke (*negative spark advance*) reduces drastically the engine torque. The spark control input has a very short delay and can be used to reduce the torque much faster than using only the throttle valve. The spark ignition time is commonly defined in terms of the spark advance angle φ^i, which denotes the difference between the angle of the crankshaft at the compression top–dead–center and the one at the ignition time. Hence, the air-fuel mixture is loaded in the cylinder during the intake stroke, while the spark is ignited when the piston is around the compression stroke top dead–center (Hrovat *et al.*, 1998). The delay between mixture intake / ignition and torque generation is represented by the hybrid system depicted in Figure 2, where I, C, E and H stand respectively for the intake, compression,

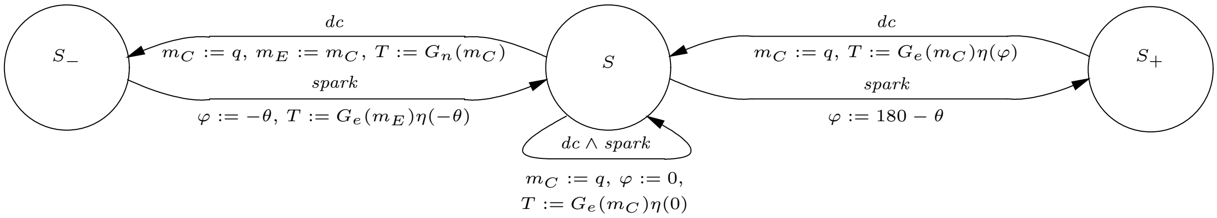

Fig. 3. Hybrid system describing the behavior of a 4-cylinder engine.

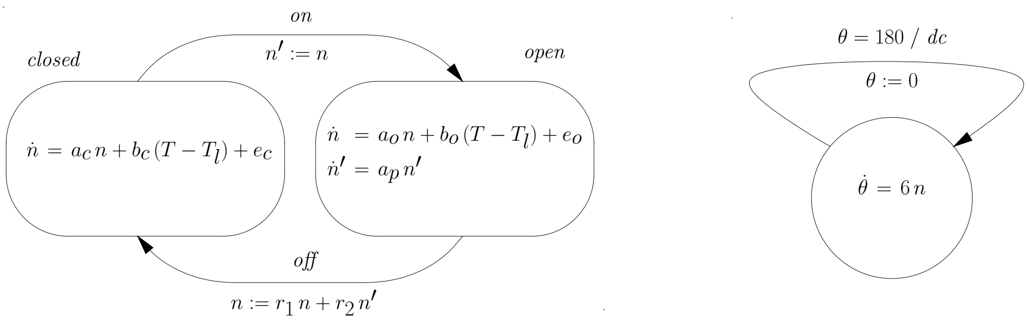

Fig. 4. Powertrain (left) and crankshaft angle (right) hybrid models.

expansion and exhaust strokes. Since spark ignition may occur either during compression or expansion, the macro–states C and E are split as follows:

- *BS* (*Before Spark*). The cylinder is in compression and no spark has been ignited yet.
- *PA* (*Positive Advance*). The cylinder is in compression and the spark has been ignited.
- *NA* (*Negative Advance*). The cylinder is in expansion and the spark has not been ignited yet.
- *AS* (*After Spark*). The cylinder is in expansion and the spark has been ignited.

The hybrid system makes a transition either when the piston reaches a dead–center (dc) or when the spark is ignited ($spark^i$). The spark advance angle φ^i is evaluated when the spark is ignited ($BS \to PA$, $BS \to AS$, $NA \to AS$) and is expressed in terms of the crankshaft angle θ.

At the end of the intake stroke (transition $I \to BS$), the air charge for the current engine cycle, m^i, is set. In case of negative spark advance, the hybrid system enters the state NA, where T^i is positive due to gas expansion and depends on the loaded air mass m^i:

$$T^i = G_n(m^i) = g_0 + g_1\, m^i + g_2\,(m^i)^2 > 0 \;. \quad (7)$$

The hybrid system is in state AS either during the entire expansion stroke, in case of positive spark advance, or just after spark ignition, in case of negative spark advance. In state AS,

$$T^i = G_e(m^i)\,\eta(\varphi^i) \;, \qquad \text{with} \qquad (8)$$

$$G_e(m^i) = h_0 + h_1\, m^i + h_2\,(m^i)^2 \;, \qquad (9)$$

$$\eta(\varphi) = v_0 + v_1\,\varphi + v_2\,\varphi^2 + v_3\,\varphi^3 \;, \quad (10)$$

where $\eta(\varphi) \leq 1$ is the ignition efficiency function and $G_e(m^i)$ is the engine torque produced with loaded air mass m^i and optimal spark advance (i.e. $\eta = 1$). The behavior of a four–cylinder in-line engine can be obtained by composing four cylinder hybrid models as given in Figure 2. However, since at any time each cylinder is in a different stroke of the engine cycle, the model can be significantly simplified and reduced to a three–state hybrid model, with discrete states S, S_+ and S_- as depicted in Figure 3. States S, S_+ and S_- correspond to the following cylinder configurations: $S = (I, BS, AS, H)$, $S_+ = (I, PA, AS, H)$, $S_- = (I, BS, NA, H)$. For details on how the reduced model is obtained see (Balluchi *et al.*, 2000*a*).

2.3 Powertrain

In idle speed control, the gear is fixed in neutral position (idle). Consequently, the secondary driveline is disconnected and does not affect the crankshaft dynamics. Due to the actions of the driver on the clutch pedal, the first part of the driveline is either connected or disconnected from the engine (see Figure 1). The dynamics of the crankshaft speed n is given by the hybrid model depicted in Figure 4, where: the discrete states *open* and *closed* encode the two possible positions of the clutch, the input events *on* and *off* represent the driver action on the clutch pedal, and the continuous dynamics are affine.

When the clutch is *open* the primary driveline speed n' evolves independently from the crankshaft speed n. Instead, when the clutch is *closed*, they evolve at the same speed n. When the clutch pedal is released (*open* $\to$ *closed*), the order of the model is reduced and the common speed state is reset. When the clutch is opened (*closed* $\to$ *open*), the primary driveline speed is appropriately initialized. The continuous dynamics and

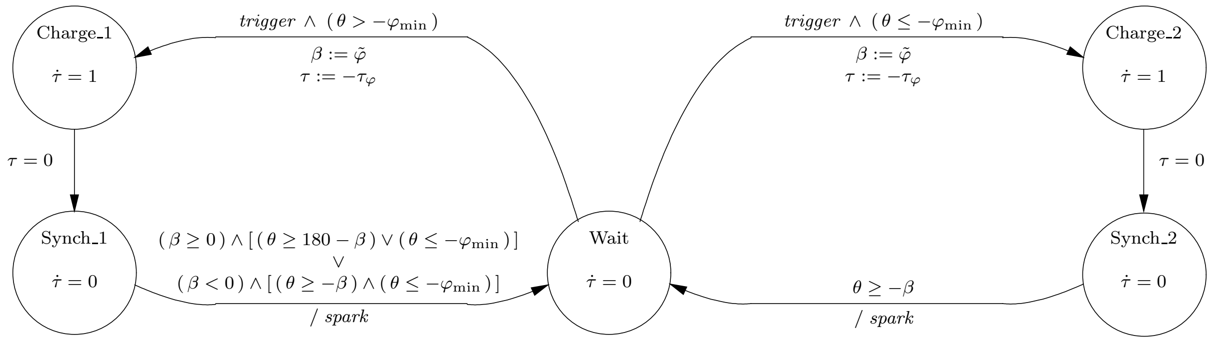

Fig. 5. Hybrid model of the ignition actuator models.

reset parameters depends on inertial momenta and viscous friction coefficients.

The evolution of the crankshaft angle in the interval $[0, 180]$ gives the position of the pistons within each stroke. It is described by the simple hybrid model reported in Figure 4. The dynamics is given by the crankshaft speed n. When the crankshaft angle θ reaches the value 180, it is reset and the dead–center event dc is emitted.

2.4 Ignition actuators

The ignition coil charging time introduces a non–negligible delay in spark ignition control. Due to this delay, the desired spark advance $\tilde{\varphi}$ has to be set with a sufficient advance to allow proper spark actuation. The ignition actuator delay is described by the hybrid model depicted in Figure 5.
The system waits in state *Wait* for the desired value of spark advance angle $\tilde{\varphi}$. It is supposed that the desired spark advance angle $\tilde{\varphi}$ is always issued for each cylinder and that it belongs to the feasible range $[\varphi_{\min}, \varphi_{\max}]$, with $\varphi_{\min} < 0$ and $\varphi_{\max} > 0$. The controller provides the desired $\tilde{\varphi}$ at some time between $180 + \varphi_{\min}$ degree in advance and $-\varphi_{\min}$ degree in delay from the compression dead–center. When the controller issues the new value $\tilde{\varphi}$, it emits the event *trigger* to synchronize with the ignition actuator. If $\tilde{\varphi}$ is issued when $\theta \le -\varphi_{\min}$, then the command is given at the beginning of the expansion stroke (only negative spark advance will be feasible) and the system takes the right cycle. Otherwise, if $\theta > -\varphi_{\min}$, then either positive or negative spark advances could be applied and the system takes the left cycle. After the transition, the system starts charging the ignition coil for a time τ_φ, spent either in state *Charge_1* or state *Charge_2*. When the charging time is elapsed, two cases are in order. (1) the crankshaft angle has not reached the desired spark advance yet ($\theta < 180 - \beta$, for positive spark advance, and $\theta < -\beta$, for negative spark advance): the system remains in state *Synch_1* / *Synch_2* until the desired spark advance is reached; then, it makes the transition to state *Wait* emitting the spark signal. (2) the crankshaft angle has already passed the desired spark advance (the command was issued too late): the system takes the transition to state *Wait* emitting the spark signal.

3. PROBLEM FORMULATION

The purpose of idle speed control is to keep the engine speed n as close as possible to a target value n_0 when the gear is neutral, preventing the engine to stall. For fuel consumption minimization, the target value n_0 is, ideally, the lowest engine speed for which the engine can be robustly controlled avoiding engine stall. More precisely, the goal is to maintain the crankshaft speed n in a specified range, $n_0 \pm \Delta_n$, robustly with respect to two sources of disturbances (a continuous and a discrete one):

- The load torque T_l acting on the crankshaft due to the auxiliary sub-systems;
- The changes on the crankshaft dynamics due to the motion of the clutch.

The control inputs are: the desired spark advance angle $\tilde{\varphi}$ and the throttle valve command $\tilde{\alpha}$. Actuator constraints and dynamics have to be taken into account in the design. Available sensors provide the following feedbacks: throttle valve position α, manifold pressure p, engine speed n.
The design specification can be formalized as a constrained optimal control problem for the hybrid model of the engine described in Section 2. The adoption of a hybrid formalism allows us to represent the cyclic behavior of the engine, thus capturing the effect of each spark command on the generated torque, the interaction between the discrete torque generation and the continuous powertrain and air dynamics, and the discrete changes of the powertrain behavior.
For any action of the torque disturbance $T_l \in [0, T_{l\max}]$ and any switching of the clutch state, the controller has to guarantee that the following constraints are satisfied:

C1 : engine speed

$$|n - n_0| \le \Delta_n \qquad (11)$$

Problem specification								
n_0	800	[RPM]	$T_{l\,\max}$	22.5	[Nm]	$\varphi_{\min}$	-12	$[^o]$
Δ_n	50	[RPM]	$\alpha_{\max}$	15	$[^o]$	$\varphi_{\max}$	15	$[^o]$
Ignition actuator / Intake manifold			**Powertrain**			**i–th and 4–th cylinder**		
τ_φ	$5\,10^{-3}$	[sec]				g_0	-2.93	[Nm]
τ_{thr}	$8.35\,10^{-2}$	[sec]	r_1	0.9	[ ]	g_1	$6.03\,10^4$	$[\frac{\text{Nm}}{\text{Kg}}]$
RT_{air}/V_{pln}	$2.152\,10^5$	$[\frac{\text{mbar}}{\text{Kg}}]$	r_2	0.1	[ ]	g_2	$1.36\,10^9$	$[\frac{\text{Nm}}{\text{Kg}^2}]$
s_0	$7\,10^{-4}$	$[\frac{\text{Kg}}{\text{sec}}]$	a_c	-0.5852	$[\frac{1}{\text{sec}}]$	h_0	-4.168	[Nm]
s_1	$3.9\,10^{-4}$	$[\frac{\text{Kg}}{\text{sec}\,^o}]$	b_c	54.26	$[\frac{\text{RPM}}{\text{sec Nm}}]$	h_1	$1.265\,10^5$	$[\frac{\text{Nm}}{\text{Kg}}]$
s_2	$5.78\,10^{-5}$	$[\frac{\text{Kg}}{\text{sec}\,(^o)^2}]$	e_c	-976	$[\frac{\text{RPM}}{\text{sec}}]$	h_2	$2.145\,10^9$	$[\frac{\text{Nm}}{\text{Kg}^2}]$
c_0	$8.279\,10^{-4}$	$[\frac{\text{Kg}}{\text{sec}}]$	a_o	-0.625	$[\frac{1}{\text{sec}}]$	v_0	$7.74\,10^{-1}$	[ ]
c_1	$3.041\,10^{-6}$	$[\frac{\text{Kg}}{\text{sec mbar}}]$	b_o	59.68	$[\frac{\text{RPM}}{\text{sec Nm}}]$	v_1	$1.729\,10^{-2}$	$[(^o)^{-1}]$
c_2	$8.5\,10^{-8}$	$[\frac{\text{Kg}}{\text{sec RPM}}]$	e_o	-1074	$[\frac{\text{RPM}}{\text{sec}}]$	v_2	$-3.65\,10^{-5}$	$[(^o)^{-2}]$
c_3	$2.245\,10^{-9}$	$[\frac{\text{Kg}}{\text{sec mbar RPM}}]$	a_p	-0.1875	$[\frac{1}{\text{sec}}]$	v_3	$-7.24\,10^{-6}$	$[(^o)^{-3}]$

Table 1. Model parameters

C2 : throttle angle

$$0 \le \alpha \le \alpha_{\max} \tag{12}$$

C3 : spark advance control

$$\varphi_{\min} \le \tilde{\varphi} \le \varphi_{\max} \tag{13}$$

$$\varphi_{\min} \le \varphi \le \varphi_{\max} \tag{14}$$

The cost function to minimize is

$$\min \|n - n_0\|^2_{\mathcal{L}_2} = \int_0^\infty (n - n_0)^2 dt \ , \tag{15}$$

in a transient due to a torque load $T_l = T_{l\,\max}$, starting from the steady state point with $T_l = 0$ and assuming that the clutch is open.

4. CONCLUSION

We presented the benchmark problem on "Idle Speed Control" proposed by the Network of Excellence HYCON. The purpose of the benchmark is to promote the application of hybrid system techniques to automotive control problems and demonstrate the effectiveness of hybrid system methodologies for the automotive industry. The description of the benchmark includes: a hybrid model of the engine, formalized system specification and a model for control algorithm validation.

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