Model-based Control of Automotive Engines and After-treatment Devices

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Abstract: As environmental requirements on automotive vehicle emissions have steadily increased over the last decades, embedded control technology to perform real-time management of the driver's requests has appeared as a key technology and has become ubiquitous. Today, every subsystem that is found under the hood is controlled, most often in closed-loop mode. A key factor in the development of a new engine or a new vehicle is the duration of its design cycle. The design and tuning of all the intricate control loops is very time-consuming. One of the main reasons for this is that tuning procedures are often used to compensate for certain un-modeled effects such as neglected couplings of dynamic variables. In this talk, we explain how models can be used to reduce expensive calibration time significantly. As will be shown in the light of several experiments (involving airpath, fuelpath, and after-treatment systems), simple models bring effective and practical solutions involving mature tools from nonlinear control theory.

1. INTRODUCTION

This paper presents a model-based view on several control problems relating to the field of automotive engines. It exposes several milestones that have been identified and worked on during a research program setup for about a decade between IFP Energies Nouvelles and MINES ParisTech.

As a preliminary, it is necessary to sketch a brief overview of recent advances in automotive engine technologies. A first keyword celebrated by the automotive industry is the term "downsizing". This terminology refers to the important efforts carried out by manufacturers and engine designers to reduce the size of engines. The rationale behind this is as follows. To produce the same level of power (torque) with a smaller engine, one increases the amount of air and fuel entering the cylinders at each stroke. This is accomplished by working with much higher internal pressures, which has the beneficial effect of reducing pumping losses, and as a result, improves the engine efficiency. Downsizing has thus emerged as a promising solution to *reduce consumption*. On the other hand, reduction of pollutant emissions has also received much attention in the past decades. This is particularly true for Nitrogen Oxides (NOx), and Particulate matter (PM). To achieve the desired reduction, one solution is to consider burned-gas recirculation into the combustion chamber. Interestingly, combined with an overabundance of air, exhaust gas recirculation (EGR) gives a cleaner combustion producing less NOx. To further the reduction of pollutant emissions (and treat PM), after-treatment devices are usually considered . These devices, which are located near the outlet in the exhaust line, can be viewed as some small-sized chemical reactors, often of spatially distributed nature. The efforts to decrease the consumption and pollutant emissions have been spurred by the successive normative environmental requirements. Steadily, over three decades, the maximum rates of admissible pollutants emissions, both in NOx and PM, have drastically decreased, for both gasoline and diesel engines. It is often considered that the regulations Euro 6, to appear in 2014, can not be satisfied for diesel engines without advanced after-treatment devices and control strategy.

2. DESCRIPTION OF A MODERN AUTOMOTIVE ENGINE

A modern automotive engine is a relatively compact but complex system. The process of bringing gases and fuel inside the combustion chamber is handled by two subsystems: the airpath and the fuelpath. The airpath controls the amount of inducted air and the EGR. In a typical configuration, it consists of an intake manifold, an EGR valve, a Variable Valve Actuation (VVA) actuator, and a Variable Geometry Turbocharger (VGT). The fuelpath controls the injected fuel. It consists of a high pressure injection system. Then, the after-treatment devices (PM filters, SCR, NOx trap, DOC) are found, usually in series, in the exhaust line. A schematic view of these elements can be found in numerous classic references, e.g. Heywood [1988], Kiencke and Nielsen [2000].

2.1 Objectives of the embedded control system

In the presented context of modern automotive engines, the embedded control system is in charge of the following tasks: provide a smooth real-time management of the driver's requests (which means handling the various control variables to produce the requested torque level), provide stability (this is especially true for engines using very high levels of EGR, such as the HCCI combustion modes), and reduce emissions and consumption. This last task might seem out-of-the-scope of control systems technology, but this is not the case. We now explain why.

In the automatic control perspective discussed in this article, a central trouble is that engines are mostly designed in view of steady-state operation. In other words, the various subsystems discussed above (airpath, fuelpath, after-treatment devices) are designed to fulfill the consumption and pollutants emissions requirements at *steady-state*.Yet, no one drives a car at steady

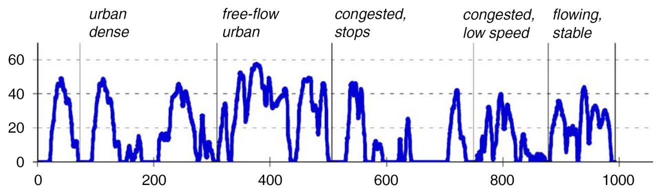


Fig. 1. The Artemis urban cycle (time in s.), reproduced from André et al. [2006]

speed/load. This is, in part, accounted for in the definition of qualification normalized test cycles (e.g. NEDC, Artemis), see Figure 1. Yet, during transients, the internal variables representing the states of the whole engine system are not necessarily synchronized, as they have dynamics with different response times, or lags. These mis-synchronizations of state variables generate temporary (but possibly long lasting) tracking errors.

To illustrate this point, let us discuss a (very) simplified version of the Fuel/Air Ratio (FAR) regulation problem for a Spark Ignited (SI) engine. We model this problem with two states corresponding to the amount of fresh air and fuel available for combustion, respectively. To optimize efficiency, the blend of air and fuel must be close to stoichiometry, which is verified by using a lambda sensor in the exhaust line. In short, it can be considered that the output of interest $y = \frac{x_1}{x_2}$ should remain close to 1 at all times. Now, the inducted mass of air and the injected mass of fuel are assumed to satisfy some first-order linear dynamics with unitary static gain. They only differ through their response times, τ_1 and τ_2, respectively. Consider that a steady-state control design is performed. Then, both dynamics are controlled using the same input signal $u(t)$, which is varied according to the requests of the driver. This symmetric open-loop control strategy of the two ISS systems has the equilibrium output $y = 1$, whatever the values of $\tau_1 > 0$ and $\tau_2 > 0$ are. Yet, when this strategy is used during transients, e.g. with a periodic time-varying signal $u(t)$, oscillations in the output variables appear, when $\tau_1 \neq \tau_2$. These are simply due to the discrepancies of the response times. The FAR is oscillating about 1, which generates unexpected pollutants. Some active control strategy is thus needed.

2.2 *Further requirements*

In view of implementation, some further requirements must be taken into consideration. The controllers (several dozens under the hood) that can be proposed, will, eventually, be tuned by expert technicians and engineers during several months (typically 6-12 months). The tuning procedures, a.k.a. "calibration", consume a lot of time in a typical design cycle of 2-3 years. This work results in experimentally determined lookup tables that are stored aboard an Electronic or Engine Control Unit (ECU). As a result of these facts, it is asked to every control engineer who wishes to propose a new control algortihm/strategy to respect the following constraints: *i)* tuning should remain simple and intuitive, i.e. as much as a typical PID loop can be, and every control-induced complexity should be avoided *ii)* tuning of the various loops should remain decentralized, i.e. cross coupling between the loops should be avoided or should not require any specific tuning operations. A last limitation that must be considered is that the computational power and memory storage capacities of available ECUs (typically Risc 32 bits running at 200 MHz) are very limited.

2.3 *What models can be good for*

In this paper, we advocate the use of models to address the issues discussed above. In more details, we wish to demonstrate that models can be used to *i)* bridge the "dynamical gap" between steady state operation points *ii)* bring insight into puzzling control design cases (e.g. those involving couplings or delays) *iii)* avoid any extension of the required calibration time *iv)* replace/complement existing sensors.

3. SENSORS AND OBSERVERS

A prime example where models can be used in automotive control design lies in the application of observers to replace or complement existing sensors. A main motivation to do so is that getting information from sensors is not as simple as it could seem.

Commercial line sensors embedded in cars produce information which is not easy to exploit. This information can be relatively low-resolution, e.g. the rotation speed of the crankshaft is measured only every 6 deg. Also, information may be only qualitative, e.g the lambda-sensor used in the exhaust line to monitor the air/fuel ratio produces only on/off information, and not a proportional signal. Finally, information can be ambiguous, e.g. the NOx sensors used to measure pollutant emissions are usually cross sensitive to NH_3 which gives a measurement equation $y = x_1 + \alpha x_2$ where x_1 is a signal of interest and x_2 is another state (see Bonfils et al. [2012]).

Further, in engine control systems, the sensors are *usually remote* from the locus of phenomena of interest and "*low-cost*". Sensors are chosen in terms of cost and long-term reliability, not on the sole basis of their bandwidth nor accuracy. To illustrate these two facts, consider the following two examples.

3.1 *Remoteness of the lambda-sensor*

Ideally, an individual sensor should be located as close as possible to the combustion chamber to monitor precisely the air/fuel ratio in each cylinder. This would yield insightful information for balancing the torque production. For cost, maintenance and reliability reasons, a single lambda-sensor is used and located in the exhaust line, downstream the turbine, see Figure 2. It is impacted by a variable delay and the mixing effect of the turbine. Fortunately, it is possible to de-convolute the measurement signal, through the model, and to recover with good accuracy the FAR signals which have a limited amount of overlap over time Chauvin et al. [2006b, 2005].

3.2 *Reconstruction of the combustion torque*

A second example is given in the combustion torque reconstruction problem. For torque balancing, again, it would be highly valuable to have information on the torque produced by each cylinder, individually. Unfortunately, except for a few exceptions, in-cylinder pressure sensors are not available on commercial line engines. On the other hand, the rotation dynamics of the crankshaft are well known and reliable. They write under the form

$$\frac{d}{d\alpha}\left(\frac{1}{2}J(\alpha)\dot{\alpha}^2\right) = T_{comb}(\alpha) - T_{load}$$

where T_{load} is the known load torque, T_{comb} is the unknown combustion torque, α is the crankshaft angle and J is the varying inertia of the crankshaft.

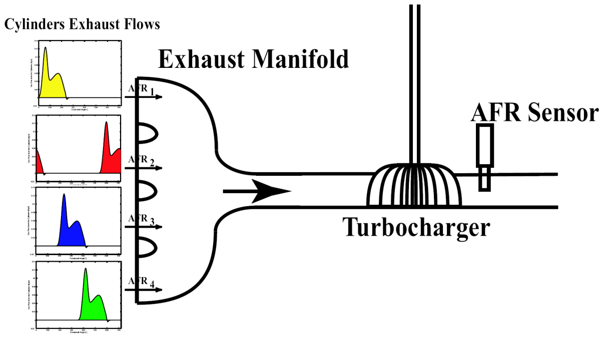


Fig. 2. A lambda-sensor located in the exhaust line, away from the exhaust manifold.

The problem of determining T_{comb} from measurements of α can be recast as a standard input reconstruction for a time-varying linear dynamics Chauvin et al. [2004]

$$\frac{d}{dt}X = A(t)X + A_0(t)w(t), \quad Y = C(t)X$$

$$\begin{cases} \frac{d}{dt}X = A(t)X + A_0(t)\left(\sum_{k\in\mathscr{I}} c_k \exp(ik\omega_0 t)\right), & Y = C(t)X \\ \frac{d}{dt}c_k = 0, \quad c_{-k} = c_k^\dagger \end{cases}$$

By using the periodicity of the dynamics, stemming from the geometric periodicity of the engine cycle, observability properties can be established which, in turn, guarantee convergence of the following general observer, in which the period is known. The observer reconstructs the Fourier decomposition of the sought-after variable T_{comb}

$$\frac{d}{dt}\hat{x} = A(t)\hat{x} + A_0(t)\left(\sum_{k\in\mathscr{I}} \hat{c}_k \exp(ik\omega_0 t)\right) - L(t)\left(C(t)\hat{x} - Y\right)$$

$$\frac{d}{dt}\hat{c}_k = -\exp(-ik\omega_0 t)L_k(t)\left(C(t)\hat{x} - y\right)$$

Depending on the desired level of accuracy, the number of Fourier harmonics can be chosen typically from 5 to 20. Without any guideline, tuning the tens of gains can reveal a tedious and difficult task. This is where the model comes into play. By performing an asymptotic analysis for an infinite number of harmonics, classic tools of functional analysis reveal that the gains should be chosen as follows, to guarantee convergence of the reconstruction in a relevant Sobolev space (through a careful LaSalle invariance principle extension to this infinite dimensional case, see Chauvin et al. [2007b], Chauvin and Petit [2010]).

$$L(t) \triangleq (A(t) - \bar{A})C^{-1}(t)$$
$$L_k(t) \triangleq \beta_k A_0^\dagger(t) P C^{-1}(t)$$

where P is the solution of a Lyapounov equation, $\beta_k = \frac{\gamma}{1+k^2}$ with $\gamma > 0$. The only remaining tuning parameter is the scalar γ which accounts for the sensor noise level. Similar results are obtained with more general assumptions in the reference cited above.

The obtained reconstruction performance stresses that it is possible to accurately estimate in real-time the combustion torque produced by each cylinder using the crankshaft angle measurement and the model as only sources of information.

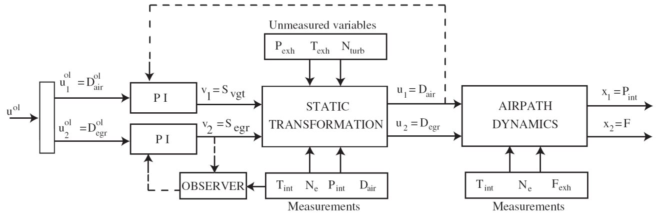


Fig. 3. Control scheme for the EGR.

4. MIMO CONTROL PROBLEMS

In this section, we wish to demonstrate how models can be used to decouple dynamics and cancel non-linearities. A first example under consideration is low-pressure EGR.

4.1 Decoupling control for EGR

Consider the airpath subsystem of a diesel engine, which is governed by the following balance equations, using the pressure x_1 and the burned gas ratio x_2 as state variables and the flowrates u_1 and u_2 as inputs

$$x_1 = P_{int}, \quad u_1 = F_{air}, \quad \alpha_{int} \triangleq \frac{RT_{int}}{V_{int}}$$
$$x_2 = F_{int}, \quad u_2 = F_{egr}, \quad \beta_{int} \triangleq \frac{1}{RT_{int}} V_{cyl} \frac{N_e}{120}$$

$$\begin{cases} \frac{d}{dt}x_1 = \alpha_{int}\left(u_1 + u_2 - \beta_{int}\eta_{vol}(x_1, Ne)x_1\right), \\ \frac{d}{dt}x_2 = \frac{\alpha_{int}}{x_1}\left(F_{exh}u_2 - (u_1+u_2)x_2\right) \end{cases}$$

where the variables T_{int}, V_{int}, Ne, V_{cyl}, F_{exh}, R are either measured or known constants. While being coupled and nonlinear, in particular due to the breathing efficiency function η, these dynamics have some remarkable properties. The system is globally exponentially stable (GES), see Chauvin et al. [2007a], and invertible, see Chauvin et al. [2008]. Invertibility can be easily checked by recovering the two control variables from the values of the states and their derivatives, under the form

$$\begin{cases} u_1 + u_2 = \eta_{vol}(x_1, N_e)\beta_{int}x_1 + \frac{1}{\alpha_{int}}\frac{d}{dt}x_1 \\ -x_2u_1 + (F_{exh} - x_2)u_2 = \frac{1}{\alpha_{int}}\frac{d}{dt}x_2x_1 \end{cases}$$

Consequently, the system is easy to control by a conveniently saturated "motion planning" strategy (see Chauvin et al. [2006a]), and (model inversion) decoupling. The saturation strategy is necessary to account for the physical limitations of the flow rates variables u_1 and u_2 which are governed by physical actuators.

For implementation, it is simply needed to perform some static changes of variables to relate the computed values of u_1 and u_2 in terms of the equivalent actuators positions (EGR choke effective area and position of the VGT). Eventually, an explicit closed-loop is used to cancel model errors in the inversion. The overall control scheme, using commonly available sensors, is reported in Figure 3.

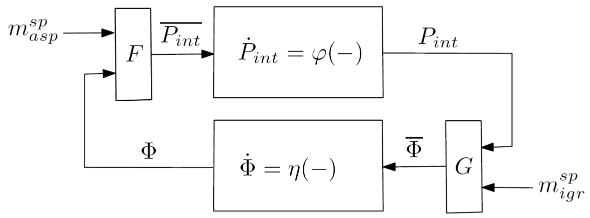


Fig. 4. Interconnecting control strategy for the iEGR problem.

4.2 *Interconnection in internal EGR*

As mentioned in the introduction of this article, it is necessary to coordinate the internal states variables to reduce the detrimental effects of the discrepancies of their response times. To illustrate how this coordination can be effectively implemented under the form of an interconnection of systems, we now consider an example of internal EGR.

Consider an engine with cylinders equipped with VVA actuators Leroy et al. [2009]. Roughly speaking, the cylinder filling during each cycle depends on the intake manifold pressure, and the VVA actuator position (openings). Depending on operation conditions, these two subsystems have relatively different behaviors. Under atmospheric conditions, the VVA are slower than the intake pressure. Conversely, under turbocharged conditions, the VVA is faster than the intake pressure.

Therefore, these two subsystems can not be controlled in a efficient manner by using solely static lookup-tables in a decentralized manner.

This issue can be addressed by interconnecting the two subsystems, according to the following equations

$$\frac{d}{dt}x_1 = \varphi(x_1, y_2, F(y_2, u_1)), \quad y_1 = x_1$$
$$\frac{d}{dt}x_2 = \eta(x_2, G(y_1, u_2)), \quad y_2 = x_2$$

which consists in connecting existing low-level controllers by the scheme of Figure 4. In this scheme, no a-priori information on the relative speeds of the two variables is necessary. By connecting the outputs of each subsystem to the input of the controller of the other one, the two subsystems are accounting for the possible lag of either variable. The scheme still admits the user's request as input, but differs from the conventional approaches by its internal connections. From a theoretical point of view, it is interesting to establish that the constituted interconnection is uniformly globally exponentially stable (UGES). This can be proven, using the "Generalized small-gain theorem" of Jiang et al. [1994]. Depending on the engine configuration, it may be necessary to introduce an additional gain in the loop to attenuate the loop gain.

This interconnection strategy is relatively general. In other scenarios, e.g. Leroy et al. [2008], it is advised to use the general result on λ-UGES of parameterized LTV systems Loria and Panteley [2002] to prove stability of the interconnection.

In the considered case of internal EGR, implementation of the interconnection loop takes the form of a cascaded block onto low-level controllers fed with measurements from the engine system. The experimental results obtained stress that it is possible to shorten the response time of the produced torque

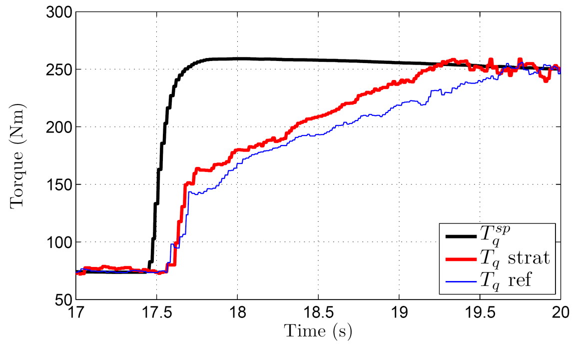


Fig. 5. Interconnection experimental results. The response time of the produced torque is reduced.

by 20%, without any additional tuning burden. The results are reported in Figure 5

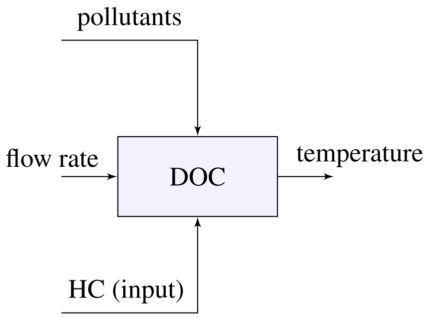
5. DIESEL OXYDATION CATALYST (DOC) TEMPERATURE CONTROL

In this example, ranging in the domain of infinite dimensional systems, we wish to demonstrate that very simple models can be sufficient to explain some surprising experimental observations, and suggest a control design.

The Diesel Oxydation Catalyst DOC is an after-treatment system located upstream of the Particulate Filter (DPF) in Diesel engines. It is in charge of oxidizing CO and hydrocarbons (HC). Its outlet temperature has to be controlled to protect the DPF during its regeneration.

5.1 *Input-output description*

Considering the flowrate of HC as input, the outlet temperature as output, and the pollutants to be treated, and the engine flow rate as disturbances, one can consider the input-output scheme below.



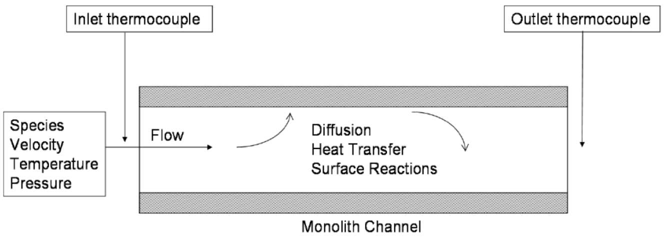
5.2 *Experimentally observed behavior*

On several experiments, the input-output response of the system is reported to have a surprisingly long and varying response time (much higher than the travel time of gas depending on the engine flow rate, as well as a delay. It also has an inverse response. These particularities have long been considered as difficult to explain.

Physically, the DOC is a distributed system having multiple channels. Each channel has a washcoat of catalyst, where the chemical species are adsorbed and react before being released.

The washcoat is distributed along the whole length of the channel.

Considering that all the channels are similar and decoupled, one can represent the DOC as a single distributed parameter system.



5.3 *Model: balance equations*

It has been determined that the chemical reactions take place in the upstream part of each channel. After this reactive part, a simple convection phenomenon comes into play, involving both the gas and the solid phase. Considering heat exchanges between the gas and the monolith (solid part) with distinct transport speed v and 0 respectively, one obtains the following equations

$$\begin{cases} \dfrac{\partial T}{\partial t} + v\dfrac{\partial T}{\partial z} = -k_1\left(T - T_s\right) \\ \dfrac{\partial T_s}{\partial t} = k_2\left(T - T_s\right) \end{cases}$$

where the input and the output are, respectively,

$$u(t) = T(x=0,t), \qquad y(t) = T(x=L,t)$$

Interestingly, this mathematical model has the surprisingly long response time, experimentally observed.

5.4 *Transfer function of the model*

The transfer function of the proposed model is easy to compute. It suffices to solve the linear partial differential equations in the Laplace domain

$$\hat{T}(x,s) = \hat{u}(s)\exp\left(-\frac{x}{v}s - \frac{k_1 x}{v} + \frac{m}{s+k_2}\right)$$

which gives, in the time domain

$$T(x,t) = H(t-\frac{x}{v})\exp(-\frac{k_1 x}{v})\times$$
$$\left[u(t-\frac{x}{v}) + \int_0^{t-\frac{x}{v}} \exp(-k_2\tau)\sqrt{\frac{m}{\tau}}I_1(2\sqrt{m\tau})u(t-\frac{x}{v}-\tau)d\tau\right]$$

where, besides a damping, and a punctual delay, a Bessel kernel over a finite support arises.

This theoretical input-output description of the thermal phenomena can be fitted to experimental data with a high degree of accuracy, as is shown in Figure 6.

So far, only the thermal propagation effects have been accounted for. The chemical reactions take place upstream of the thermal propagation region. To account for the inlet of chemically reactive species, one can simply schedule the length of the thermal propagation region according to their flowrate. In other words, the propagation length appearing in the model above (resp. the upstream reactive length) shall depend on the input of the system. By doing so, one constitutes a system having, on top of the discussed delay, decay rate, and long response time, the experimentally observed inverse response behavior.

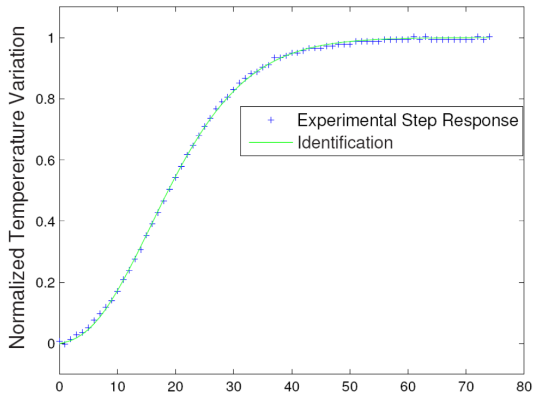


Fig. 6. Fitting experimental thermal response of the DOC.

5.5 *Inversion-based control for transient*

The mathematical input-output model presented above can be readily inverted to provide sharp transients by means of a simple open-loop strategy. The inverse transfer function is

$$\hat{u}(s) = \exp\left(\frac{x}{v}s + \frac{k_1 x}{v} - \frac{m}{s+k_2}\right)\hat{y}(s),$$

where x is the modelled reactive length. A causal motion planning strategy is then easily set-up for any anticipated desired output history f,

$$\hat{y}(s) = \exp\left(-\frac{x}{v}s\right)\hat{f}(s)$$

The open-loop motion planning formula, $t \geq 0$ is $u(t) =$

$$\exp(\frac{k_1 x}{v})\times\left[f(t) - \int_0^t \sqrt{\frac{m}{\tau}}J_1(2\sqrt{m\tau})\exp(-k_2\tau)f(t-\tau)d\tau\right]$$

5.6 *Practical application*

Again, as in the previously presented applications, some preliminary static changes of variables are needed to related the considered input to the actual control variable. Here, the control is not the temperature but the "equivalent" injected mass of fuel. A static nonlinearity, compensated by sensors and lookup tables is implemented. Finally, a feedback is achieved by a "modified" Smith predictor accounting for the modelled variation of the delay. Typically observed results are that the DOC outlet temperature can be controlled within ± 15 deg when the gas flow rate is varied promptly. More details can be found in Lepreux et al. [2008, 2009, 2011].

The presented control strategy has been considered and developed further during a technology transfer phase from IFPEN to PSA Peugeot-Citroën. In this form, it is used onboard the PSA commercial line Diesel Euro 6 passenger cars, starting 09/2014.

6. CONCLUSIONS AND OPEN PERSPECTIVES

As a conclusion of the review that we have presented in this article advocating the use of models for control design of automotive engines, following other approaches Re et al. [2010], van Nieuwstadt and Tennison [2006], Eriksson et al. [2002], Eriksson [2007], van Nieuwstadt et al. [2000], Shaver et al. [2006], Chauvin et al. [2006c], we wish to stress some remarkable facts.

First, it appears, at the light of the numerous applications treated on the subsystems and at more global levels, that first-principles 0D or 1D models (as presented in Heywood [1988] e.g.) are surprisingly realistic. In this context, Bernoulli equation, simple thermodynamics (e.g. ideal gas law), compression equations, simplified transport phenomena Bresch-Pietri et al. [2012a,b], and even equations of combustion (at a macroscopic level Hillion et al. [2009]) appear as valuable tools to describe observed dynamical behaviors.

Second, control theory seems to provide a collection of mature tools to solve problems recurring in the considered area. Static nonlinearities are relatively easy to compensate, e.g. by the geometric tools of feedback linearization (mostly static). Decoupling for MIMO architectures is easily achievable and outperform SISO controllers. Also, the variabilities of response-times are handled by coordination of subsystems using generalized small gain theorems.

On the applications side, one can then conclude that models are good for: replacing or complementing existing sensors, suggesting relevant control designs, and explaining surprisingly complex behaviors.

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