Model-free control of automotive engine and brake for Stop-and-Go scenarios

Sungwoo Choi, Brigitte d'Andréa-Novel, Michel Fliess, Hugues Mounier, Jorge Villagra

Model-free control of automotive engine and brake for Stop-and-Go scenarios

Sungwoo CHOI†, Brigitte d'ANDRÉA-NOVEL, Michel FLIESS, Hugues MOUNIER, Jorge VILLAGRA

***Abstract*— In this paper we propose a complete strategy for the longitudinal control of automotive vehicles in Stop-and-Go situations. Firstly, a upper level grey-box torque control is proposed to compensate for neglected dynamics at chassis level (due for example to road slopes, aerodynamic forces, rolling resistance forces, etc.). Secondly, to obtain the desired torque, we have considered a model-free approach to elaborate the suitable low level engine or braking torque. Convincing simulation results are presented to validate our method.**

***Index Terms*— Stop-and-Go, model-free control, inteligent PID controllers, numerical differentiation, robustness.**

I. INTRODUCTION

Driving assistance systems like Adaptive Cruise Control (ACC) and Stop-and-Go have been extensively studied in recent years [17]. The former is devoted to inter-distance control in highways where the vehicle velocity remains quasi constant, whereas the latter is appropriate for vehicles driving in towns with frequent and sometimes hard stops and accelerations. The constraints of these two situations are quite different in terms of comfort, so they have often been treated as two distinct approaches. The authors of [9] propose a unique non-linear reference model and controller for both ACC and Stop-and-Go scenarios. However, their feedback terms are not sufficiently robust to tackle external disturbances such as road characteristics and aerodynamic forces. A grey-box control strategy was therefore proposed in [18] in order to compensate all the neglected dynamics.

In [9] the authors suppose that the reference acceleration generated by the model can be instantaneously applied to the following vehicle. However, the corresponding braking and accelerating torques are often difficult to obtain because of the uncertainty of the engine/brake models. Different approaches have been proposed to handle the nonlinear dynamics of engine and brake: input/output linearization [16], [15], fuzzy logic ([11], [7]) and sliding mode ([6], [19], [12]) have been proposed for engine and brake control.

S. Choi and B. d'Andréa-Novel are at Mines ParisTech, CAOR-Centre de Robotique, Mathématiques et Systèmes, 60 bd. Saint Michel, 75272 Paris Cedex 06, France. sung-woo.choi@ensmp.fr, dandrea@ensmp.fr

M. Fliess is at INRIA-ALIEN & LIX (CNRS, UMR 7161), École polytechnique, 91128 Palaiseau, France. Michel.Fliess@polytechnique.edu

H. Mounier is at Institut d'Électronique Fondamentale (CNRS, UMR 8622), Université Paris-Sud, 91405 Orsay, France. Hugues.Mounier@ief.u-psud.fr

J. Villagra is at Departamento de Ingenieria de Sistemas y Automatica, Universidad Carlos III, 28911 Leganes (Madrid), Spain. jvillagr@ing.uc3m.es

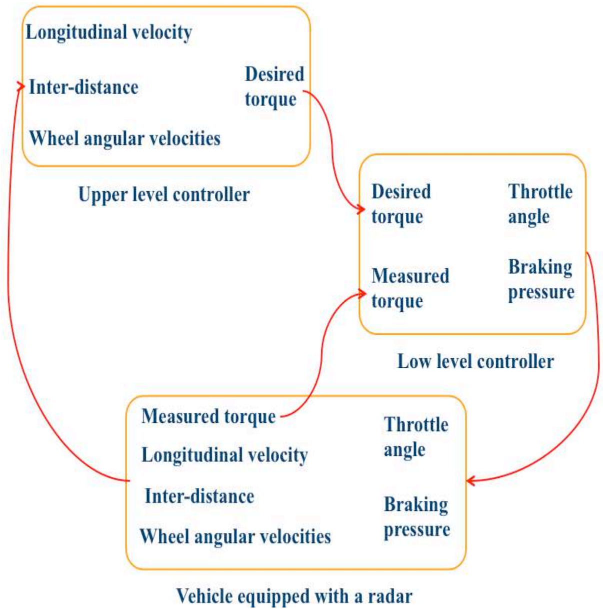

Fig. 1. General Stop-and-Go control scheme

Most of these methods use fixed gains, whereas the model parameters vary during the lifetime of the vehicle. In addition to this, off-line identification for an engine/brake modeling is often complex. We here propose a so called model-free control approach developed in [2], [3], which is inherently robust to the very poorly known engine and brake dynamics.[1]

The remainder of the paper is organized as follows. The general control scheme is presented in Section II. In Section III we explain the model-free control setting. Section IV presents upper level control with the compensation of unmodeled dynamics using the so-called *intelligent* PID controllers [2], [3]. Section V is devoted to lower level engine/brake control and to associated simulations. A robustness study will be presented in Section VI.

II. CONTROL SCHEME

The whole control scheme is graphically summarized in Figure 1. The longitudinal control system architecture for the Stop-and-Go is designed to be hierarchical, with an upper level controller and a lower level controller. The upper level controller determines the desired torque for the vehicle

[1]See [8] for another application to the automotive industry, i.e., to the control of throttles.

using longitudinal velocities, wheel angular velocities and inter-distance measurements by radar. An intelligent PID compensation is developed at this level to deal with the chassis dynamics uncertainties. The lower level controller determines the throttle angle input or the braking pressure input required to track the desired torque generated by the upper level controller. For the model-free control design, measured engine and brake torques are required as an input to this level. For the simulation, a 14 degree-of-freedom vehicle representation with tire models as well as a radar model have been used.

III. MODEL-FREE CONTROL[2]

We only assume that the plant behavior is well approximated in its operational range by a system of ordinary differential equations. The input/output equation looks like for a SISO system:

$$E(t,y,\dot{y},\ldots,y^{(\iota)},u,\dot{u},\ldots,u^{(\kappa)})=0$$

where E is a sufficiently smooth function of its arguments. Assume that for some integer n, $0<n\leq\iota$, $\frac{\partial E}{\partial y^{(n)}}\not\equiv 0$. The implicit function theorem yields then locally

$$y^{(n)}=\mathfrak{E}(t,y,\dot{y},\ldots,y^{(n-1)},y^{(n+1)},\ldots,y^{(\iota)},u,\dot{u},\ldots,u^{(\kappa)})$$

This equation becomes by setting $\mathfrak{E}=F+\alpha u$:

$$y^{(n)}=F+\alpha u \tag{1}$$

where

- $\alpha\in\mathbb{R}$ is a *non-physical* constant parameter, such that F and αu are of the same magnitude;
- the numerical value of F, which contains the whole "structural information", is determined thanks to the knowledge of u, α, and of the estimate of the derivative $y^{(n)}$ (see remark 1 below).

In all the numerous known examples it was possible to set $n=1$ or 2. If $n=2$, we close the loop via the *intelligent PID controller*, or *i-PID* controller,

$$u=-\frac{F}{\alpha}+\frac{\ddot{y}^{*}}{\alpha}+K_{P}e+K_{I}\int e+K_{D}\dot{e} \tag{2}$$

where

- y^* is the output reference trajectory, which is determined via the rules of flatness-based control;
- $e=y-y^*$ is the tracking error;
- K_P, K_I, K_D are the usual tuning gains.

Remark 1: The numerical derivation of noisy signals, which is necessary to implement the feedback loop (2) is borrowed from [4] and already plays an important role in model-based control (see [10] for further important theoretical developments, and, also, [1] for some applications). This approach obviously necessitates derivatives estimation of noisy signals. The estimate of the 1^{st} order derivative of a noisy measured signal y can be expressed as follows:

$$\hat{\dot{y}}=-\frac{3!}{T^{3}}\int_{0}^{T}(T-2t)y(t)dt, \tag{3}$$

where [0,T] is a quite "short" time window. Moreover, a filtered version of the noisy measured signal y yields:

$$\hat{y}=\frac{2!}{T^{2}}\int_{0}^{T}(2T-3t)y(t)dt. \tag{4}$$

The global structure of our scheme is the following:

- the upper level uses an *i-PD* controller of the form (2) to handle external disturbances (road slope, aerodynamic forces, ... see (12)). Equation (3) will also be used to estimate the time derivative of the inter-distance d which is necessary to obtain the leader vehicle velocity.
- The lower level uses the other *i-PD* controllers (20) and (22) to handle the unknown engine and brake dynamics.

IV. UPPER LEVEL CONTROLLER

First of all, a short summary of the control law for the upper level controller is given and then the resulting engine/brake torque is elaborated.

A. Feedforward control

An inter-distance reference model proposed by [9] will act as a feedforward control law for the upper level controller. The inter-distance reference model describes a virtual vehicle dynamics which is positioned at a distance d^r (reference distance) from the leader vehicle. The reference model dynamics is given by

$$\ddot{d}^{r}=\ddot{x}_{l}-\ddot{x}_{f}^{r} \tag{5}$$

where $\ddot{x}_l$ is the leader vehicle acceleration and the follower vehicle acceleration

$$\ddot{x}_{f}^{r}=u^{r}(d^{r},\dot{d}^{r}) \tag{6}$$

is a nonlinear function of the inter-distance reference d^r and of its time derivative $\dot{d}^r$.

Introducing $\tilde{d}\triangleq d_0-d^r$ in (6), where d_0 is the safe nominal inter-distance, the control problem consists in finding a suitable control when $\tilde{d}\geqslant 0$ so that all the solutions of the dynamics (5) fulfill the following comfort and safety constraints:

- $d^r\geqslant d_c$, with d_c the minimal inter-distance, will guarantee collision avoidance.
- $\|\ddot{x}^r\|\leqslant B_{max}$, where B_{max} is the maximum attainable longitudinal acceleration, depending on the driver, the vehicle and the infrastructure, will have an effect on security and comfort.
- $\|\dddot{x}^r\|\leqslant J_{max}$, where J_{max} is a bound on the driver desired jerk, which directly affects the comfort performances.

The work by [9] proposed to use a nonlinear damper model:

$$u^{r}=-c|\tilde{d}|\dot{\tilde{d}},\ \forall\tilde{d}\geqslant 0$$

[2]See [2], [3] for more details.

which leads to the following equation:

$$\ddot{\tilde{d}} = -c|\tilde{d}|\dot{\tilde{d}} - \ddot{x}_l.$$

The previous equation can be analytically integrated and expressed in terms of d^r, assuming that $\dot{x}_l(0) = 0$:

$$\dot{d^r} = \frac{c}{2}(d_0 - d^r)^2 + \dot{x}_l(t) - \beta,\ \beta = \dot{x}_f^r(0) + \frac{c}{2}(d_0 - d^r(0))^2. \quad (7)$$

From (6), the feedforward control law is then obtained:

$$u^r = \ddot{x}^r = c|d_0 - d^r|\dot{d}^r \quad (8)$$

where the reference inter-distance evolution comes from numerical integration of (7). Note that the parameters c and d_0 are algebraic functions of comfort and safety parameters (see [9] for details).

Remark 2: The leader vehicle velocity $\dot{x}_l$ is estimated using $\dot{x}_l = \dot{d} + \dot{x}_f$ where $\dot{d}$ is estimated by (3) from the measured distance d.

B. Upper level closed loop control

The feedforward control should be corrected with feedback terms which can compensate errors induced by measurement noises and external disturbances. We use an algebraic PD compensator[3], so that the corrected acceleration of (8) is given by:

$$u(=\gamma_x) = u^r + K_P e + K_D \dot{e} \quad (9)$$

where e represents the error between the real inter-distance and the reference inter-distance ($e = d - d^r$), and its time derivative will be estimated using (3).

C. Engine/Brake torque generation

The wheel rotation dynamics can be written as follows:

$$I\dot{\omega} = -rF_x + \tau_{e_a} - \tau_{b_a} \quad (10)$$

where I is the rotation inertia moment, ω is the wheel angular velocity, r is the tire radius, F_x is the longitudinal tire force, τ_{e_a} is the applied engine torque, and τ_{b_a} is the applied brake torque, both of them applied at the wheel center.

The sum of the 4 wheels rotation dynamics equations and of the vehicle longitudinal dynamic equation $M\gamma_x = \sum_{i=1}^4 F_{x_i}$ yields

$$\tau_g = I\sum_{i=1}^{4}\dot{\omega}_i + rM\gamma_x \quad (11)$$

where $\tau_g = \tau_e - \tau_b = \sum_{i=1}^4(\tau_{e_a i} - \tau_{b_a i})$ is the generalized total torque, M is the total weight of vehicle and γ_x is the longitudinal acceleration. $\dot{\omega}$ is computed once more with (3) from the measured wheel angular velocities.

Our final reference torque τ_g can be obtained using (11) where $\gamma_x = u$ is given by (9), and an intelligent PID compensation is applied to handle unmodeled external disturbances γ_0 (due to road slope, rolling resistance, wind, etc.):[4]

[3] See [5] for details.
[4] See [18] for details.

TABLE I
ENGINE MODEL VARIABLES AND PARAMETERS

$\dot{m}_{ac_e}$	Mass air flow rate into the manifold (kg/s)
$\dot{m}_{ac_s}$	Mass air flow rate out of the manifold (kg/s)
P_{ad}	Manifold pressure (P_a)
α_p	Throttle angle (°)
w_m	Engine speed (tr/min)
T_m	Internally developed torque (Nm)
T_{ch}	Load torque (Nm)
$T_{ch_{pert}}$	Shaft torque (Nm)
k_p	Manifold dynamic constant
k_n	Rotational dynamics constant

$$\tau_g = I\sum_{i=1}^{4}\dot{\omega}_i + rM(u - \hat{\gamma}_0). \quad (12)$$

V. LOW LEVEL CONTROLLER

In the lower level controller, the throttle angle input and the brake pressure input are calculated in order to track the desired torque determined by the upper level controller. For this study, an engine model and a brake model have been used.

A. Engine model

The engine model we use in this study was derived by Powell *et al.* [14] from steady-state engine maps. The model represents a 1.6 liter, 4-cylinder fuel injected engine. The dynamic equations of the model are:

$$\begin{array}{rcl} \dot{P}_{ad} & = & k_p(\dot{m}_{ac_e} - \dot{m}_{ac_s}) \\ \dot{w}_m & = & k_n(T_m - T_{ch}) \end{array} \quad (13)$$

$$\begin{array}{rcl} \dot{m}_{ac_e} & = & (1 + a_1\alpha_p + a_2{\alpha_p}^2)g(P_{ad}) \\ g(P_{ad}) & = & \left\{ \begin{array}{ll} 1, & P_{ad} \leq 50.66 \\ a_3\sqrt{(a_4P_{ad} - P_{ad}^2)}, & P_{ad} > 50.66 \end{array} \right. \end{array} \quad (14)$$

$$\begin{array}{rcl} \dot{m}_{ac_s} & = & a_5w_m + a_6P_{ad} + a_7w_mP_{ad} + a_8w_mP^2{}_{ad} \\ T_m & = & a_9 + a_{10}\widetilde{m}_{a_s} + a_{11}w_m + a_{12}w_m^2 \quad (15) \\ \widetilde{m}_{a_s} & = & \dfrac{\dot{m}_{ac_s}}{120w_m} \quad (16) \\ T_{ch} & = & (\dfrac{w_m}{263.17})^2 + T_{ch_{pert}}. \quad (17) \end{array}$$

All the quantities in those equations are explained in Table I and we will use all the coefficients (a_i, $i = 1, \ldots, 12$) proposed in [13]. T_m is related to τ_g through the transmission chain which we will not detail here.

B. Brake model

We will use a simple brake model proposed in [19] where the brake hydraulic dynamics have been approximated by a linear second-order system. The dynamic equations for the brake pressure is:

$$\begin{aligned} \dot{z}_1 &= z_2 \\ \dot{z}_2 &= -b_1 z_1 - b_1 z_2 + b_3 P_m \\ P_\omega &= z_1 \end{aligned} \qquad (18)$$

where P_m is the input to the solenoid valve and P_ω is the brake pressure at the wheel.

The brake torque τ_b is considered to be proportional to the brake pressure at the wheel:

$$\tau_b = K_b P_\omega.$$

where K_b is the lumped gain for the entire brake system.

C. Model-free feedback control

Inspired by (1) with $n = 0$, we assume that there is locally a linear relation between the measured engine torque and the throttle angle:

$$\tau_e = k_a \alpha_p + G(t) \qquad (19)$$

where k_a is a constant and $G(t)$ represents neglected dynamics of the engine.

If we use as a throttle input:

$$\alpha_p = \frac{1}{k_a}(a(\dot{\tau}_e - \dot{\tau}_g) + \tau_g - \hat{G}), \qquad (20)$$

where $a \in \mathbb{R}^-$, τ_e is the measured engine torque, and $\hat{G}$ is the estimate of $G(t)$ which can be obtained from (19) in discrete time form:

$$G_k = \tau_{ek} - k_a \alpha_{k-1},$$

this control law leads to the following torque error dynamics:

$$a(\dot{\tau}_e - \dot{\tau}_g) - (\tau_e - \tau_g) = 0.$$

Therefore τ_e will converge exponentially to τ_g.

The brake torque can also be expressed by:

$$\tau_b = k_b P_m + D(t) \qquad (21)$$

where k_b is a constant and $D(t)$ represents neglected dynamics of the brake. And the same technique can be applied for the brake input too:

$$P_m = \frac{1}{k_b}(b(\dot{\tau}_b - \dot{\tau}_g) + \tau_g - \hat{D}), \qquad (22)$$

where $b \in \mathbb{R}^-$, τ_b is the measured brake torque and $\hat{D}$ is the estimate of $D(t)$.

Remark 3: In (20) and (22), we need torque measurements. From a practical point of view, the torque measurement is not an easy task. Instead, it could be estimated using (11). In order to test this, realistic noises have been added on both angular wheel velocities ω and longitudinal accelerations γ_x. To attenuate the perturbation due to these measurement noises, α_p and P_m are finally filtered using (4). Let us also notice that the time derivative terms in (20) and (22) are obtained using (3). Finally we have implemented our resulting control law which does not need any torque measurement but in absence of external dynamic disturbances on a typical Stop-and-Go scenario where several accelerations/decelerations are applied on a flat road. Figure 2 shows that the desired torque is pretty well tracked.

The same scenario is tested once more, with several external disturbances (road slope, rolling resistance and aerodynamic force) which are detailed for example in [18]. Thanks to the grey-box compensation on the upper level controller, external disturbances are efficiently compensated and the desired torque is well tracked too (see Figure 3). Indeed, in that case, we cannot obviously estimate γ_0 from (12) if we suppose that the torque τ_g is not itself measured. These simulation results show that our control strategy exhibits good robustness properties with respect to the external disturbances, if the torque is supposed to be measured.

VI. ROBUSTNESS STUDY OF THE MODEL-FREE CONTROL AGAINST PARAMETER VARIATIONS

In order to show the robustness of our model-free control approach which does not rely on any parametric model, we can compare it with an analytical solution of the engine/brake models presented in Section V.

A. Analytical input solutions of the engine/brake models

Introducing $\widetilde{m}_{a_s}$ of (16) into (15) yields

$$\begin{aligned} &T_m = A\dot{m}_{ac_s} + B \\ &A = \tfrac{a_{10}}{120 w_m}, \ \ B = a_9 + a_{11} w_m + a_{12} w_m^2. \end{aligned} \qquad (23)$$

In a similar way, (23) can be rewritten using $\dot{m}_{ac_s}$ from (13):

$$T_m = A(\dot{m}_{ac_e} - \frac{\dot{P}_{ad}}{k_p}) + B. \qquad (24)$$

If we rearrange (24) in terms of $\dot{m}_{ac_e}$ and we compare it with (14):

$$\dot{m}_{ac_e} = \frac{1}{A}(T_m - B + A\frac{\dot{P}_{ad}}{k_p}) = (1 + a_1\alpha_p + a_2{\alpha_p}^2)g(P_{ad})$$

then, the following second order equation in α_p can be written:

$$\begin{aligned} &a{\alpha_p}^2 + b\alpha_p + c = 0 \\ &a = a_2, \ \ b = a_1, \ \ c = 1 - \frac{T_m - B + A\frac{\dot{P}_{ad}}{k_p}}{Ag(P_{ad})}. \end{aligned} \qquad (25)$$

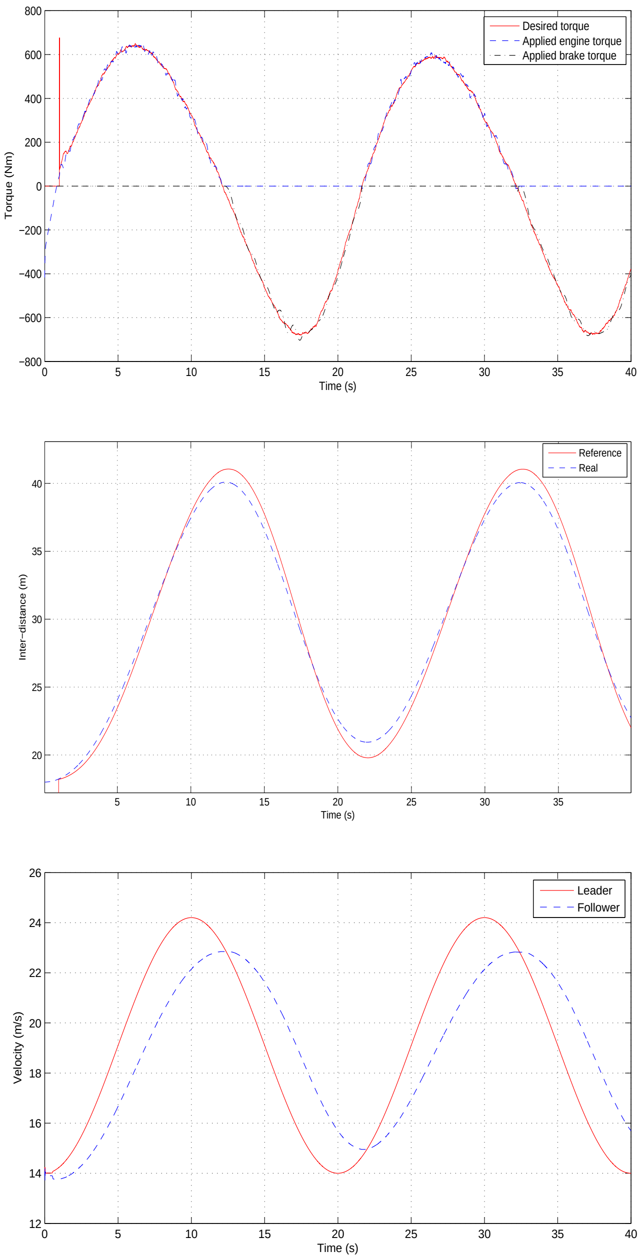

Fig. 2. Torques, inter-distances and velocities in a typical Stop-and-Go scenario

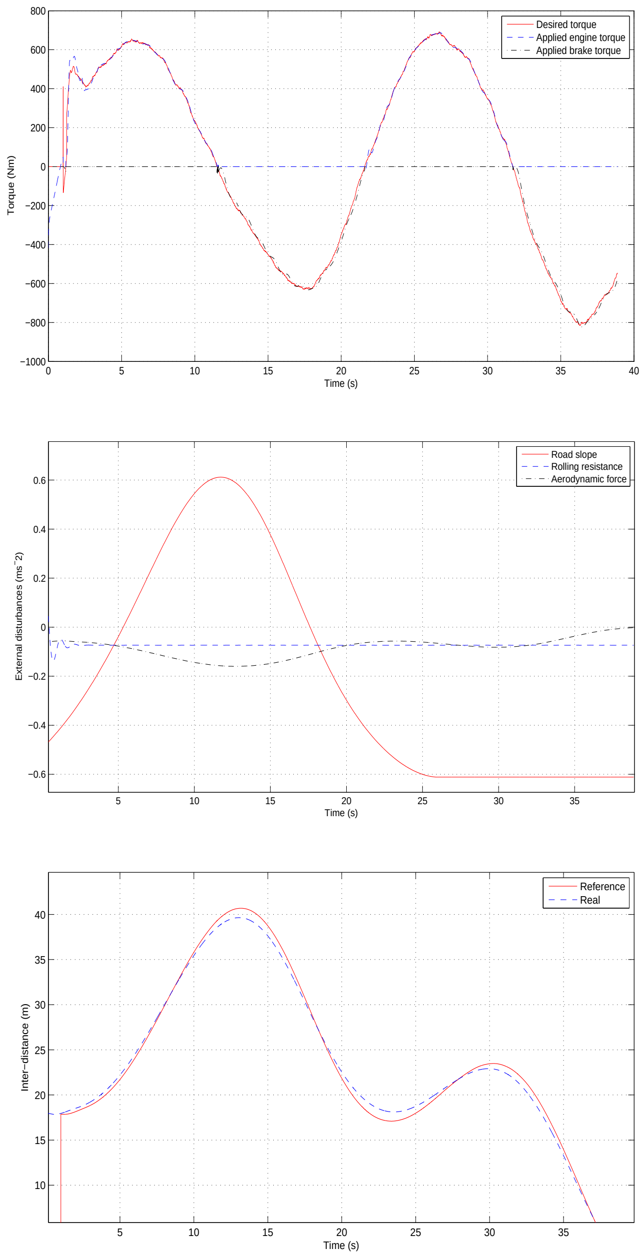

Fig. 3. Torques, external disturbances and inter-distance when the torque is supposed to be measured

Finally, we solve (25) and obtain a throttle angle input which is a function of Tm, P_{ad} and ω_m (the positive root will be taken because physically $\alpha_p \geq 0$):

$$\alpha_p = f(T_m, P_{ad}, w_m) = \frac{-b + \sqrt{b^2 - 4ac}}{2a}. \qquad (26)$$

For the brake input, we can express easily P_m from (18):

$$P_m = \frac{1}{b_3}(\ddot{P}_\omega + b_2 \dot{P}_\omega + b_1 P_\omega) \qquad (27)$$

using algebraic estimates of $\ddot{P}_\omega$ and $\dot{P}_\omega$.

B. Influence of parameter uncertainties

We assume that parameters are not well known. We will increase a_9 and a_{10} up to 20 % in the engine model and b_3 up to 20 % in the brake model. And then, we will compare the model-free control performance using [(20), (22)] with the analytical solution performance using [(26), (27)].

As the analytical solutions are obtained from (26) and (27), they are very sensitive to any parameter variation, and therefore torque tracking quality is quite poor (see Figure 4). On the contrary, the model-free control strategy shows good performances (see Figure 5).

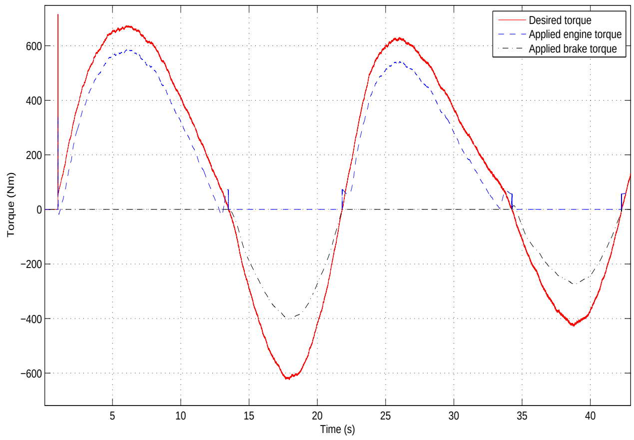

Fig. 4. Influence of the parameter changes on the analytical solutions

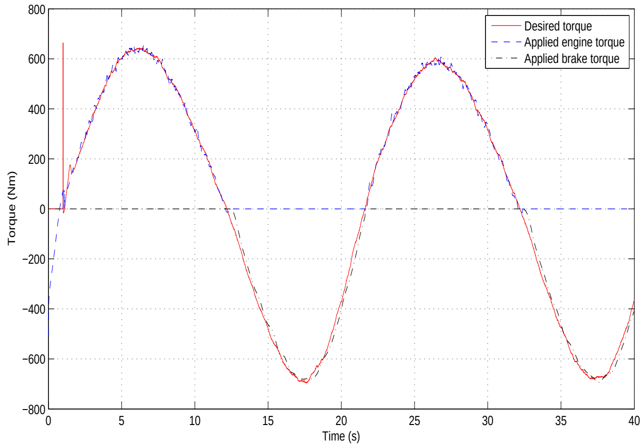

Fig. 5. Robustness of the model-free control against parameter changes

VII. CONCLUSION

The model-free control approach has been applied to develop controllers in Stop-and-Go scenarios. Our control laws are naturally robust not only to unmodeled low level dynamics but also to external disturbances applied to the chassis. It should be pointed out that engine/brake torque measurements are no more needed and can be estimated using algebraic techniques in the disturbance free case.

REFERENCES

[1] S. Choi, B. d'Andréa-novel, J. Villagra, 'Robust Algebraic Approach for Radar Signal Processing: Noise filtering, time-derivative estimation and perturbation estimation' *Proc. EU-Korea Conf. Science and Technology*, pp. 297-305, Heidelberg, 2008.

[2] M. Fliess, C. Join, 'Commande sans modèle et commande à modèle restreint', *e-STA*, vol. 5, No 4, pp. 1-23. 2008. `http://hal.inria.fr/inria-00288107/en/`

[3] M. Fliess, C. Join, 'Model-free control and intelligent PID controllers: towards a possible trivialization of nonlinear control?', *Proc. 15th IFAC Symp. System Identif. (SYSID 2009)*, Saint-Malo, 2009. `http://hal.inria.fr/inria-00372325/en/`

[4] M. Fliess, C. Join, H. Sira-Ramírez, 'Non-linear estimation is easy', *Int. J. Model. Identif. Control*, Vol. 4, pp. 12-27, 2008. `http://hal.inria.fr/inria-00158855/en/`

[5] M. Fliess, R. Marquez, E. Delaleau, H. Sira-Ramírez, 'Correcteurs proportionnels-intégraux généralisés' *ESAIM Control Optim. Calc. Variat.*, Vol. 7, pp. 23-41, 2002.

[6] J.C. Gerdes, J.K. Hedriek, 'Vehicle speed and spacing control via coordinated throttle and brake actuation', *Control Eng. Practice*, Vol. 5, pp. 1607-1614, 1997.

[7] S. Germann, R. Isermann, 'Nonlinear distance and cruise control for passenger cars', *Proc. Amer. Control Conf.*, Seattle, 1995.

[8] C. Join, J. Masse, M. Fliess, 'Étude préliminaire d'une commande sans modèle pour papillon de moteur', *J. europ. syst. automat.*, Vol. 42, pp. 337-354, 2008. `http://hal.inria.fr/inria-00187327/en/`

[9] J. Martinez, C. Canudas-de-Wit, 'A Safe Longitudinal Control for Adaptive Cruise Control and Stop-and-Go Scenarios', *IEEE Trans. Control Syst. Techno.*, Vol. 15, pp. 246-258, 2007

[10] M. Mboup, C. Join, M. Fliess, 'Numerical differentiation with annihiators in noisy environment', *Numer. Algor.*, Vol. 50, pp. 439-467, 2009.

[11] J. E. Naranjo, C. González, R. García, T. de Pedro, 'ACC+Stop&Go Maneuvers With Throttle and Brake Fuzzy Control', *IEEE Trans. Intel. Transport. Syst.*, Vol. 7, pp 213-225, 2006.

[12] L. Nouvelière, S. Mammar, 'Experimental vehicle longitudinal control using a second order sliding mode technique', *Control Eng. Practice*, Vol. 15, pp. 943-953, 2007.

[13] L. Nouvelière, 'Commandes robustes appliquées au contrôle assisté d'un véhicule à basse vitesse', *Thèse*, Univ. Versailles - Saint-Quentin, 2002.

[14] B.K.Powel, D. Hrovat, 'Optimal idle speed control of an automotive engine', *Amer. Control Conf.*, Chicago, 2000.

[15] H. Raza, Z. Xu, B. Yang, P. Ioannou, 'Modeling and control design for a computer-controlled brake system', *IEEE Trans. Control Syst. Technol.*, Vol. 5, pp. 279-296, 1997.

[16] D. Swaroop, K. Hedrick, C. Chien, P. Ioannou, 'Comparison of spacing and headway control laws for automatically controlled vehicles', *Vehicle Syst. Dyn.*, Vol. 23, pp. 597-625, 1994.

[17] A. Vahidi, A. Eskandarian, 'Research advances in intelligent collision avoidance and adaptive cruise control', *IEEE Trans. Intel. Transport. Systems*, Vol. 4, pp 143-153, 2003.

[18] J. Villagra, B. d'Andréa-Novel, M. Fliess, H. Mounier, 'Robust grey-box closed-loop stop-and-go control', *Proc. IEEE Conf. Decision Control*, Cancun, 2008. `http://hal.inria.fr/inria-00319591/en/`

[19] K. Yi, J. Chung, 'Nonlinear Brake Control for Vehicle CW/CA Systems', *IEEE/ASME Trans. Mechatronics*, Vol. 6, pp. 17-25, 2001.