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Longitudinal Control for Automated Vehicle Guidance

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Abstract: this paper presents an automated vehicle guidance strategy with a special focus on the longitudinal control. Indeed, the longitudinal control plays an important role in the automated guidance strategy to ensure safety and comfort of automotive passengers. Sophisticated longitudinal control allows also to enhance fuel consumption reduction. Thus, a nonlinear longitudinal control design based on a Lyapunov approach is proposed taking into account the throttle, the brake and the gear ratio. The proposed longitudinal controller is coupled to a previously developed lateral guidance and the whole control architecture (coupled longitudinal and lateral control) performance is highlighted through simulations.

Keywords: Automotive vehicle control, Longitudinal control, Automated guidance.

1. INTRODUCTION

During last years, the field of automotive vehicle is experiencing an important evolution due to an increasing use of individual vehicles in everyday life. This increasing use poses new challenges such as security and comfort of car passengers, traffic management, reduction of fuel consumption and pollutant emissions. As discussed in [Guzzella, 2009], the automatic control plays a particular role in solving these problems. Indeed, the automatic control can help to develop Advanced Driver Assistance Systems (*ADAS*), Global Chassis Control (*GCC*), engine control, automated driving and guidance (longitudinal and lateral control), etc. The automated driving seems to be an interesting research way to cope with the mentioned problematics. In fact, the concepts developed in this framework can be also transposed to develop new *ADAS* as well as generic control architectures.

The automated guidance of a vehicle is a non trivial problem due to the strong couplings in the vehicle dynamics. Indeed, in the vehicle motion several longitudinal and lateral couplings arise:

- Kinematic and dynamic coupling of the longitudinal and lateral motions due to the yaw motion caused by the wheels steering.
- The load transfer phenomenon due to the longitudinal and lateral accelerations. Theses accelerations affect the normal forces on the tires and so the longitudinal and lateral tire forces.
- Tire-road coupling forces constrained by the so-called friction circle/ellipse . In fact, the maximal available tire-road friction is distributed between lateral and longitudinal forces.

The use of strongly nonlinear coupled models is necessary to capture these couplings in a wide operational range [Lu and Hedrick, 2004]. In this work the dynamic coupling of the longitudinal and lateral motions as well as the tire-road forces are handled by the nonlinear models used for controllers synthesis.

Hence, the design of a coupled longitudinal/lateral control strategy becomes particularly arduous. In order to cope with this problem, some authors attempt to consider simultanousely the couplings to propose coupled longitudinal and lateral control strategies. The main idea is to obtain an only one controller ensuring both longitudinal and lateral control. A sliding mode technique for coupled longitudinal and lateral control is proposed in [Lim, 1998] and recently [Menhour et al., 2011, Nehaoua and Nouvelière, 2012] propose a solution based on the flatness control theory. Notice however that in these works the engine map and the gear ratio management are not taken into account.

This problem can be also tackled considering the lateral and the longitudinal control of the automotive vehicle in a decoupled way. The lateral control can be investigated using different techniques. Among them, model-matching synthesis [Kwon et al., 2005], techniques issued from artificial intelligence [Onieva et al., 2010] and the predictive control technique [Falcone et al., 2007, Besselmann and Morari, 2009]. The longitudinal control meanly deals with the improvement of the classic Cruise Control (*CC*) which ensures only the regulation of the cruise speed. In [Nouvelière and Mammar, 2007], an advanced longitudinal control design based on a sliding mode technique is proposed and experimentally validated. An Active Cruise Control (*ACC*) based on a gain-scheduling control technique is recently proposed in [Shakouri et al., 2010]. The *ACC* ensures the regulation of the vehicular inter-distance and the vehicle speed. Those systems are intensively studied for automated highway driving phases where the main goal is the regulation of both speed and the inter-vehicular distance [Delprat et al., 2008]. In the

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quoted works, the authors attempt to consider the engine model and the shift gear but the tire saturation is not considered. To enhance the longitudinal performance, [ElMajdoub et al., 2012] proposes an improvement of the longitudinal modeling taking into account tire model and saturation. In fact, sophisticated applications such as platooning and collaborative driving [Rajamani and Shladover, 2001] require a high longitudinal vehicle capacities.

Our main purpose is to propose global control strategies and generic architectures for automated vehicle guidance. A first issue of a coupled longitudinal and lateral guidance strategy taking into account security aspects has been recently proposed by the authors in [Attia et al., 2012]. The lateral guidance, based on a Nonlinear Model Predictive Controller (*NMPC*), is the main contribution of that work. Nevertheless, the longitudinal control was not well addressed. Hence, the present paper aims to go further in the longitudinal control design to improve the performance of the proposed guidance strategy. Indeed, driving safety as well as passengers' comfort of automotive largely depend on the longitudinal motion of the vehicle. It can be remarked that the longitudinal control is synthesized independently of the lateral one to reduce synthesis complexity. However, the lateral control strategy takes into account the both dynamics via the chosen prediction model.

This paper addresses the longitudinal control of an automotive vehicle in a global guidance strategy. For that purpose, a nonlinear longitudinal model of the vehicle is considered and a nonlinear controller is synthesized. The paper is organized as follows. Section 2 presents a brief overview of the proposed vehicle guidance strategy. The whole coupled longitudinal/lateral control strategy is recalled in Section 3. Section 4 details the proposed longitudinal control design based on a Lyapunov approach. Simulation results of the proposed nonlinear longitudinal control are presented in Section 5 and a discussion on the role of longitudinal control in reducing fuel consumption is finally done in Section 6.

2. VEHICLE GUIDANCE STRATEGY

The architecture of the proposed automated guidance strategy is given by Fig. 1 and can be synthesized into three main modules namely *Perception*, *Reference Generation* and *Control*.

The *Perception* of the vehicle environment is of the utmost importance in the control architecture as it gives the frame in which the vehicle evolves. Its role is to provide relevant information related to the vehicle/environment to the *Reference Generation*.

The *Reference Generation* provides two kinds of profiles necessary to *Control* considering the informations provided by the *Perception*. The first profile is related to the geometric path to be followed by the vehicle and the second one is the longitudinal speed reference. Several considerations such as legal speed limit, safety and comfort could be taken into account for the reference speed generation [Daniel et al., 2009]. In addition, eco-aspects based, for example, on road topology can also be considered to minimize the fuel consumption.

The *Control* ensures the automated vehicle guidance along the references provided by the *Reference Generation*. This task is accomplished by providing the appropriate control signals i.e. the steering angle for the lateral control and the traction torque for the longitudinal control. Consequently, a coupled longitudinal and lateral control becomes necessary to ensure an efficient vehicle guidance. The lateral control strategy is based on a *NMPC* design previously presented in [Attia et al., 2012]. It has been shown that this controller ensures well tracking performance in normal driving situations as well as in some critical maneuvers such as double lane change maneuver (a brief recall of the lateral controller is presented in Section 3). The lateral controller is coupled with a longitudinal controller allowing an adaptation of the longitudinal speed to improve rolling performance. In a previous work [Attia et al., 2012] the reference speed tracking is ensured with a controller providing moderately good tracking speed performance. Here, the longitudinal control performance is enhanced by considering a more complete longitudinal model taking into account the real physical inputs of the vehicle i.e. the throttle opening angle and the brake. Thus, the engine, the gearbox as well as the driveline are taken into account in the model and controller synthesis as shown in Section 4. It is expected that thanks to these enhancements the vehicle guidance strategy can be improved.

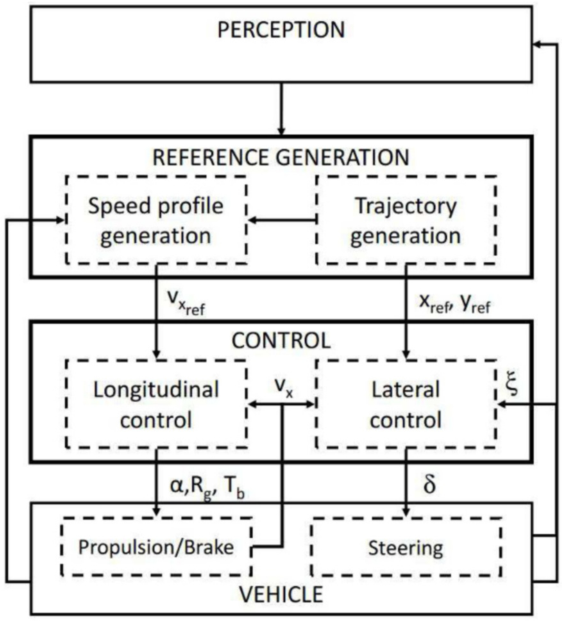

Fig. 1. Architecture of the control strategy

3. LATERAL CONTROL

This section recalls the vehicle model and the controller synthesis for lateral guidance based on a *NMPC* design.

3.1 Vehicle Dynamic Model for Guidance

For guidance purpose, the following bicycle nonlinear model is considered [Kiencke and Nielsen, 2000]:

$$m\ddot{x} = m\dot{y}\dot{\psi} + F_{x_f} + F_{x_r} \tag{1a}$$

$$m\ddot{y} = -m\dot{x}\dot{\psi} + F_{y_f} + F_{y_r} \tag{1b}$$

$$I\ddot{\psi} = l_f F_{y_f} - l_r F_{y_r} \tag{1c}$$

where the longitudinal and lateral directions in the vehicle frame are x and y respectively, X and Y the longitudinal and lateral directions in the inertial frame, ψ the yaw angle in the $\{x, y\}$ frame and Ψ the heading angle in the $\{X, Y\}$ frame. The vehicle mass and the moment of inertia are given by m and I, the front and rear CoG-distance by l_f and l_r and the forces in the x and y directions by $F_{x_{f/r}}$ and $F_{y_{f/r}}$ [1]. These forces depend on the longitudinal and lateral tire-road friction forces $F_{l_{f/r}}$ and $F_{c_{f/r}}$ as follows:

$$F_{x_f} = F_{l_f}\cos(\delta) - F_{c_f}\sin(\delta) \tag{2a}$$
$$F_{y_f} = F_{l_f}\sin(\delta) + F_{c_f}\cos(\delta) \tag{2b}$$
$$F_{x_r} = F_{l_r} \tag{2c}$$
$$F_{y_r} = F_{c_r} \tag{2d}$$

where δ is the front wheel steering angle. Here, the tire-road behavior model characterizing the longitudinal and lateral forces $F_{l_{f/r}}$ and $F_{c_{f/r}}$ is the well-known Burckhardt's model given by [Kiencke and Nielsen, 2000]. The dynamics of the wheels is described by:

$$I_{w_{f/r}}\dot{\omega}_{f/r} = -F_{l_{f/r}}R_{f/r} + T_{t_{f/r}} \tag{3}$$

where $T_{t_{f/r}}$ is the total torque applied on the wheel (engine and brake torques), $I_{w_{f/r}}$ the moment of inertia of the wheel, $R_{f/r}$ the wheel radius. Finally, the chassis kinematic model is given by the following equations:

$$\dot{X} = \dot{x}\cos\Psi - \dot{y}\sin\Psi \tag{4a}$$
$$\dot{Y} = \dot{x}\sin\Psi + \dot{y}\cos\Psi \tag{4b}$$
$$\dot{\Psi} = \dot{\psi} \tag{4c}$$

The state representation of the model (1)-(4) is used as a prediction model for lateral control purpose. Notice that this model captures the longitudinal and lateral couplings. Therefore, this controller ensures good tracking performance at variable longitudinal speeds.

3.2 Nonlinear Model Predictive Control

The nonlinear model predictive controller allows reference tracking for nonlinear systems considering constraints on both control inputs and state variables [Re et al., 2010]. The controlled inputs are calculated at each sampling time thanks to an on-line optimization minimizing the tracking errors between the predicted and reference outputs on a prediction time-horizon N_p. The predicted outputs are obtained from a prediction model obtained from the discretization of the continuous-time vehicle model (1)-(4). The sampling period used here is $20\,ms$.

A usual cost function employed in the optimization problem is given by:

$$J = \sum_{n=1}^{Np} \left\|h(k+n) - h_{ref}(k+n)\right\|_Q + \sum_{n=0}^{Nc-1} \|u(k+n)\|_R \tag{5}$$

where $h = [X\,Y\,\psi]^T$ and h_{ref} are respectively the measured and the reference outputs, $u = \delta$ the control input and N_c the *control time-horizon* fixing the optimization problem dimension to be solved. Matrices Q and R respectively represent the weights regarding the tracking errors and the control input energy.

The *NMPC* problem to cope with the lateral control is formulated as follows [Attia et al., 2012]:

$$\arg\min_{\Delta U} J(\xi(k), \Delta U) \tag{6a}$$
$$\text{s.t } \xi(k+1) = \xi(k) + T_s f(\xi(k), u(k)) \tag{6b}$$
$$h(k) = [X(k)\ Y(k)\ \Psi(k)]^T \tag{6c}$$
$$u_{min} \le u(k) \le u_{max} \tag{6d}$$
$$\Delta u_{min} \le \Delta u(k) \le \Delta u_{max} \tag{6e}$$
$$u(k) = u(k-1) + \Delta u(k) \tag{6f}$$
$$\Delta u(k) = 0 \quad \text{for } k = N_c, \ldots, N_p$$

where ξ is the state-space vector of the prediction model, $\Delta U = [\Delta u(k), \ldots, \Delta u(k+N_c-1)]$ the optimization vector, u_{min} and u_{max} the lower and upper limits of the control input u, Δu_{min} and Δu_{max} the lower and upper limits of the control input variation. Finally, the closed-loop control law is calculated from the solution ΔU^* of the optimization problem (6), i.e.:

$$u(k) = u(k-1) + \Delta U^*(1) \tag{7}$$

at time $k+1$ the same operation is repeated on the time horizon $[k+1, k+N_p+1]$.

The lateral controller thereby obtained takes into account the longitudinal dynamics since the yaw cross coupling is taken into account in the prediction model. Therefore, the proposed lateral controller allows path-following at variable longitudinal speeds.

4. NONLINEAR LONGITUDINAL CONTROL

This section details the design of the longitudinal control. The nonlinear longitudinal model used for control law synthesis is firstly presented. Secondly, the longitudinal controller issued from a Lyapunov-based synthesis and the strategy for the management of the different longitudinal actuators are developed.

4.1 Vehicle Longitudinal Motion Modeling

The resultant of the forces acting on the vehicle in straight motion allows to write the longitudinal motion dynamics equation as follows:

$$m\dot{v} = F_p - F_a - F_g - F_{rr} \tag{8}$$

where m is the total vehicle mass, v the longitudinal speed, and the forces on the right side of (8) are defined hereafter:

- F_p is the propelling force which is the controlled force produced by engine or brake,
- F_a is the aerodynamic force given by:
$$F_a = \frac{1}{2}\rho C_d v^2 \tag{9}$$
with ρ the air density and C_d the drag coefficient,
- F_g is the gravitational force due to road slope:
$$F_g = mg\sin(\theta) \tag{10}$$
with θ the road slope and g the gravitational acceleration,
- F_{rr} is the rolling resistance force given by:
$$F_{rr} = C_r mg\cos(\theta) \tag{11}$$
with C_r the rolling resistance coefficient,

The equation (3) related to the wheels dynamics is slightly modified to distinguish the brake torque T_b and the propelling torque T_p as follows:

$$I_w\dot{\omega} = -F_l R + T_p - T_b \tag{12}$$

where R is the wheel radius and F_l the tire-road friction force due to sliding.

[1] The indices $\{f, r\}$ refer to the $\{front, rear\}$ wheels.

Assumption 1. A non-slip rolling is considered then the following relationships hold:

$$v = R\omega \tag{13a}$$
$$F_p = F_l \tag{13b}$$

Assumption 2. No losses are considered between the engine and the final driveshaft, then:

$$\omega = R_g \omega_e \tag{14a}$$
$$T_e = R_g T_p \tag{14b}$$

where R_g is the gearbox ratio, ω_e the engine speed and T_e the engine torque.

Now, substituting (13b) into (12) yields:

$$F_p = \frac{1}{R}(T_p - T_b - I_w \dot{\omega}) \tag{15}$$

The longitudinal dynamics becomes:

$$(m + \frac{I_w}{R^2})\dot{v} = \frac{T_p - T_b}{R} - F_a - F_{rr} - F_g \tag{16}$$

using (13a) and (15) into (8). Finally, by taking into account Assumption 2 the longitudinal dynamics is:

$$\frac{(mR^2 + I_w)R_g}{R}\dot{v} = T_e - R_g(T_b + R(F_a + F_{rr} + F_g)) \tag{17}$$

It is important to remark that the controlled input T_e depends on the throttle opening ratio α and the engine speed ω_e. This relationship is often characterized by a static map-model of the engine, an example of a gasoline engine map is shown on Fig. 2 [].

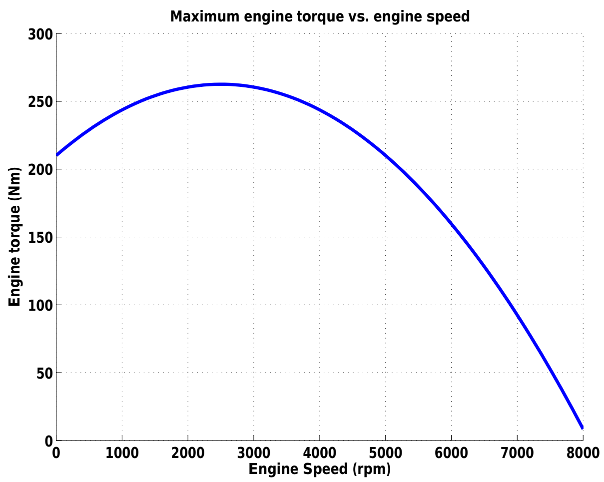

Fig. 2. Engine map model

The nonlinear relation between the engine torque and the speed is given in this figure.

The nonlinear static relationship between T_e and ω_e can be captured by the following polynomial expression [Guzzella and Onder, 2004]:

$$P = a_1 + a_2\omega_e + a_3\omega_e^2 \tag{18}$$

where $P = T_e \times \omega_e$ is the engine power and a_1, a_2 and a_3 are empirical model parameters. For a given engine speed, the maximum available torque is:

$$T_{emax} = \frac{P_{max}}{\omega_e} \tag{19}$$

Assumption 3. It is assumed for control purpose that the engine power is proportional to the throttle opening ratio α.

4.2 Longitudinal Control Strategy

This section is devoted to the longitudinal control design. The reference speed generation, the management of the gear shift and the switching between throttle and brake are discussed. Then, a Lyapunov-based approach is adopted to synthesize the nonlinear longitudinal controller.

Reference Speed Generation: the reference speed to be tracked by the vehicle is issued from the reference generation module (cf. Section 2). Here, the reference speed provided by the reference generation module is assumed to be a continuously differentiable signal available in real-time.

Longitudinal Control Actuators Management: as above mentioned, the available control inputs are the throttle opening ratio α, the brake torque T_b and the gear ratio R_g. A management policy should be defined to handle the working exclusivity between throttle and brake. A gear shift policy also has to be fixed.

The switching between throttle and brake is defined using a policy taking into account the throttle opening value given by the nonlinear controller and the speed tracking error. The brake is activated if the throttle is inactive ($\alpha = 0$) and the vehicle speed is greater than the reference speed.

The automatic gear shift management, i.e. determining the adequate gearbox ratio at each instant time, is a complex optimization problem [Guzzella and Sciarretta, 2005] and will not be handled in this work. However, several studies show that the optimal engine operating point for small road gradients is reached around $2750\,rpm$ [Luu et al., 2010]. Based on this observation, the solution adopted here results of an automatic gearbox-like system modeled by the statechart shown in Fig. 3.

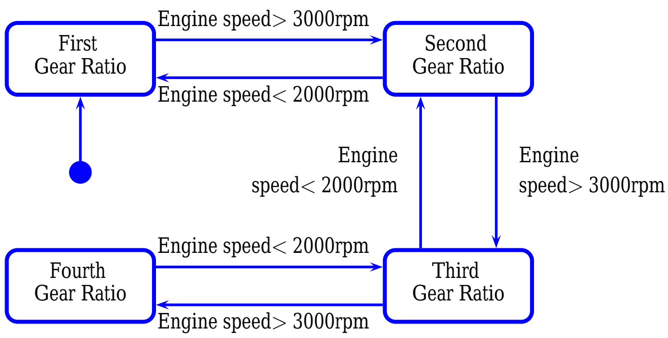

Fig. 3. Gear shift strategy

Nonlinear Longitudinal Controller: the proposed controller is synthesized using a Lyapunov-based approach. Consider the speed tracking error given by:

$$e = v_{ref} - v \tag{20}$$

where v_{ref} and v are the reference and the actual speeds. The dynamics of this error is:

$$\dot{e} = \dot{v}_{ref} - \dot{v} \tag{21}$$

which can be rewritten as:

$$\dot{e} = \dot{v}_{ref} - \frac{1}{M_t}(T_e - R_g(T_b + R(F_a + F_{rr} + F_g))) \quad (22a)$$

$$M_t = \frac{(mR^2 + I_w)R_g}{R} \quad (22b)$$

using the expression of $\dot{v}$ given by the the nonlinear longitudinal model (17).

In order to ensure the convergence towards zero of the tracking error, a Lyapunov approach is employed. For that purpose, consider the following definite positive Lyapunov candidate function:

$$V = \frac{1}{2}e^2 \quad (23)$$

and its time-derivative:

$$\dot{V} = e\dot{e} \quad (24)$$

The tracking error convergence towards zero is ensured if the following condition is verified:

$$\dot{V} \leq -kV \quad (25)$$

where k can be considered as a *decay rate* ensuring exponential convergence.

Replacing (22a) in (24) gives:

$$\dot{V} = e(\dot{v}_{ref} - \frac{1}{M_t}(T_e - R_g(T_b + R(F_a + F_{rr} + F_g)))) \quad (26)$$

Considering (26) and $T_b = 0$ (when throttle is active the brake is inactive), the stability condition (25) suggests the following control law:

$$T_e^* = M_t(ke + \dot{v}_{ref}) + R_g R(F_a + F_{rr} + F_g) \quad (27)$$

Notice that the controlled input applied to the vehicle is the throttle opening α. This last is obtained from the required engine torque T_e^* provided by the control law (27) using:

$$\alpha = \frac{T_e^* \omega_e}{P_{max}} \quad (28)$$

where ω_e is the actual engine speed and P_{max} the maximum engine power.

Robust Stability Analysis: the stability condition $\dot{V} = -kV$ is met for the control torque T_e^* when the model matches exactly with the physical system. To overcome this strong assumption, the control law (27) is robustified by considering in the controller synthesis uncertainties on the model parameters.

Consider the robustified control law

$$\hat{T}_e = \hat{M}_t(ke + \dot{v}_{ref}) + R_g\hat{R}(\hat{F}_a + \hat{F}_{rr} + \hat{F}_g) + \rho \quad (29)$$

where the hat designates the considered values for the parameters (nominal values) which may be different from the effective ones. The robustification term ρ is determined to ensure robust convergence of the tracking error. Considering the control law (29), the tracking error dynamics (21) becomes:

$$\dot{e} = -\frac{\hat{M}_t}{M_t}ke + \frac{1}{M_t}\left((M_t - \hat{M}_t)\dot{v}_{ref} + (Rg(M_r - \hat{M}_r) - \rho)\right) \quad (30)$$

where:

$$M_r = R(F_a + F_{rr} + F_g) \text{ and } \hat{M}_r = \hat{R}(\hat{F}_a + \hat{F}_{rr} + \hat{F}_g) \quad (31)$$

The definite positive Lyapunov candidate function is considered:

$$V = \frac{1}{2}e^2 \quad (32)$$

using the tracking error dynamics (30), the time derivative of the Lyapunov candidate is:

$$\dot{V} = -\frac{\hat{M}_t}{M_t}ke^2 + \frac{1}{M_t}\left(\tilde{M}_t\dot{v}_{ref} + R_g\tilde{M}_r - \alpha\right)e \quad (33)$$

where $\tilde{M}_t = M_t - \hat{M}_t$ and $\tilde{M}_r = M_r - \hat{M}_r$.

To ensure the negativeness of the time-derivative of the Lyapunov candidate function (32), the following robustification term is considered:

$$\rho = \Delta sign(e) \quad (34)$$

where $\Delta > \tilde{M}_t\dot{v}_{ref} + R_g\tilde{M}_r$

The robustified control law is then given by:

$$\hat{T}_e = \hat{M}_t(ke + \dot{v}_{ref}) + R_g\hat{M}_r - (\tilde{M}_t^{max}\dot{v}_{ref} + R_g\tilde{M}_r^{max})sign(e) \quad (35)$$

where $\tilde{M}_t^{max}$ is the maximum uncertainty on the parameter M_t and $\tilde{M}_r^{max}$ the maximum uncertainty on the total resisting moment M_r. From a practical point of view, the *sign* function is replaced by a saturation function $sat(e/\delta)$ to avoid the chattering phenomenon.

5. TEST AND SIMULATION

This section presents the simulation results of the proposed longitudinal control design. The validation of the longitudinal control is achieved using the full car drivetrain benchmark available in $Matlab/Simulink$. In this way, the synthesis model previously presented is not the same as the validation model. Indeed, this validation model considers an engine map as well as a mechanical modeling of the driveline and the gearbox as it can be seen in Fig. 4. Besides, a gasoline engine and a four ratios gearbox have been considered.

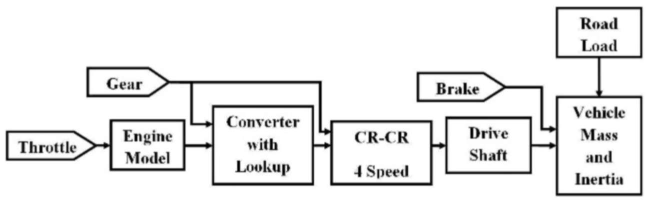

Fig. 4. Full vehicle longitudinal model -simulation model-

The control design is tested using two simulation tests. In the first test, the performance of the longitudinal control is evaluated through a speed tracking test on a flat track. In the second one, the proposed longitudinal controller is coupled to the lateral controller already proposed. The whole guidance strategy is evaluated in this manner.

5.1 Longitudinal Control Test

This test consists in tracking different reference speeds. Fig. 5 shows the evolution of the vehicle speed to track the reference in acceleration and deceleration phases. Notice the gear shifts at the instants $20s, 30s, 45s, 61.5s, 71.4s, 81s$. In fact, in acceleration phase the increasing reference speed needs more propelling torque, then the engine speed increases and reaches the maximum value where a gear shift becomes necessary. Notice that the time response of the controlled system depends on the design parameter k. In fact, this parameter determines the convergence rate of the Lyapunov function.

Fig. 6 shows the evolution of the engine speed, the gear shift and the applied torque. It can be noticed that the engine speed remains in the neighborhood of the considered *optimal* operating range $[2000\,rpm, 3000\,rpm]$ thanks to the proposed gear shift policy.

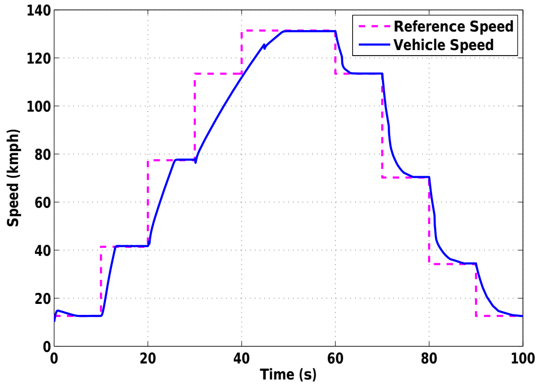

Fig. 5. Reference speed tracking

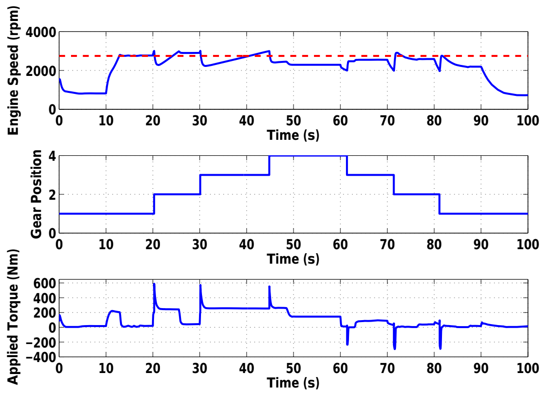

Fig. 6. Engine speed, Gear shift and Applied Torque

Fig. 7 shows the throttle and brake inputs in acceleration and deceleration phases. The switching between throttle and brake is effective in deceleration phases. Note that at the origin time ($t = 0$) a slight brake torque is applied to track the reference speed. In fact the initial vehicle speed ($v(0) = 16\,kmph$) is greater than the reference ($v_{ref}(0) = 12\,kmph$).

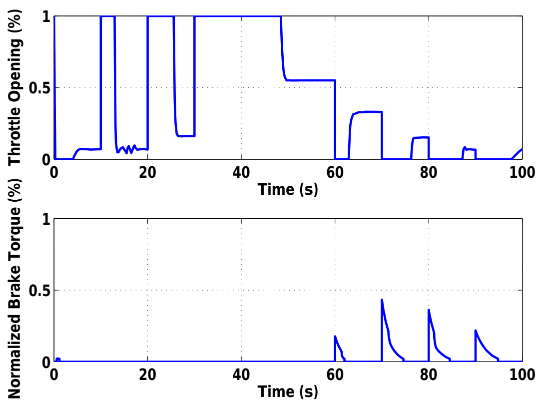

Fig. 7. Throttle and brake control signals

5.2 *Coupled Longitudinal and Lateral Test*

In this test, the longitudinal controller is considered in the whole guidance strategy. The reference trajectory corresponds to a highway exit, it is obtained from the Reference Generation module. The longitudinal speed reference is calculated considering safety and comfort criteria as discussed in [Attia et al., 2012]. Fig. 8 shows the reference trajectory and the tracking performed by the vehicle. Fig. 9 shows the tracking errors, it can be noticed that the lateral position and the heading angle errors are very acceptable. It can be also seen that thanks to the proposed longitudinal control the reference speed is well tracked.

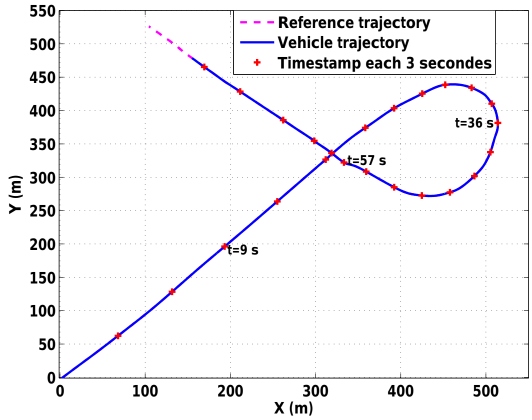

Fig. 8. Test trajectory

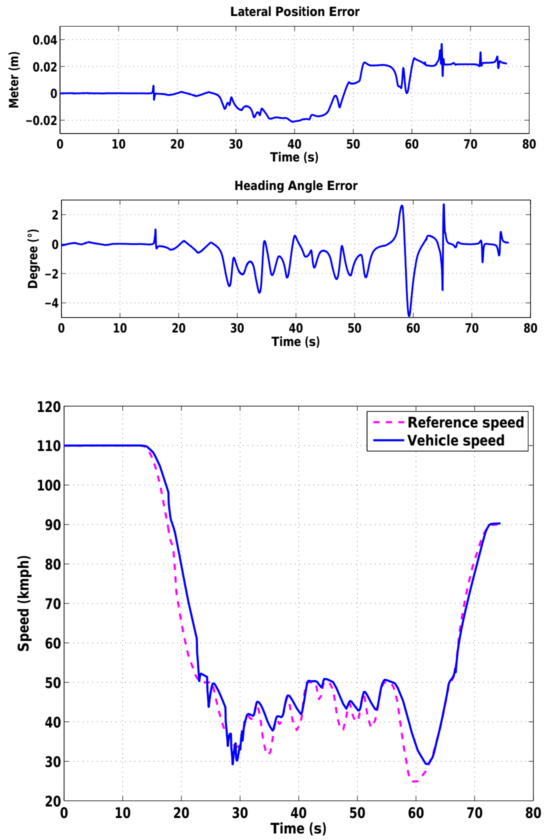

Fig. 9. Coupled longitudinal and lateral guidance

6. FUEL CONSUMPTION REDUCTION AND LONGITUDINAL CONTROL

This section discusses the role of longitudinal control in reducing the fuel consumption. The developed control strategy (in its longitudinal and lateral dimensions) aims to provide generic and evolutionary vehicle guidance strategies suitable for consideration of fuel consumption aspects . The energy management could be considered at two distinct levels:

- The first one is the reference generation level. In fact, the generation of smooth reference speeds allows to guarantee an eco-friendly driving and contributes to the minimization of fuel consumption. Till now the reference speed generation is done without considering the road slope. The authors aim to generalize the approach proposed in [Daniel et al., 2011] by considering the third dimension z which represents the variation of the road slope. The proposed longitudinal control is able to handle this additional information.
- The second one is the gear shift strategy. The assumption done in this work allows to keep the engine speed in an acceptable range not far from the optimal operating point. This problem kept independent form the one of reference generation in order to keep the genericity of the control architecture.

Several works in the literature propose solutions to cover this aspect [Guzzella, 2009, Luu et al., 2010]. In [Luu et al., 2010] the problem of minimizing fuel consumption on a known road part using a dynamic programming approach is proposed. [Guzzella, 2009] propose to solve the problem by using the same technique i.e. dynamic programming by introducing a look-ahead strategy. Indeed, the problem of choosing the appropriate gear ratio to meet some energetic cost is a mixed integer optimization problem for which real-time solution risks to be hard. The authors aim to propose a solution to reduce fuel consumption by adapting a model predictive like approach.

7. CONCLUSION

Following a Lyapunov-based nonlinear design, a nonlinear longitudinal controller is proposed. This controller ensures tracking of time-varying reference speeds. The performance of the obtained control is shown in simulation. In a sake to complete a previously presented work, the proposed longitudinal controller is tested in a coupled longitudinal and lateral guidance strategy. The whole guidance strategy is tested in simulation and it shows promising results.

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