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Combining Active Steering and Independent Wheels Braking for CIVIC Lateral Assistance

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Abstract— **This paper addresses the problem of integrated vehicle-infrastructure-driver control (CIVIC). Both vehicle handling improvement and lane keeping support are considered. The control synthesis procedure uses a linear driver-vehicle-model model which includes the yaw motion and disturbance input with speed and road adhesion variations. The synthesis procedure allows the separate processing of reference signal tracking, robust stabilization and disturbance rejection. The control action is performed as a combination of additional steering angle and a yaw moment generated by differential wheel braking. It uses a combination of the driver input, feedback of the yaw rate and vehicle positioning. The synthesized controller is tested for different speeds and road conditions on a nonlinear model in both disturbance rejection, driver imposed yaw reference tracking maneuvers and lane keeping. Preliminary validation using data from an experimental test track is included.**

I. Introduction

Single vehicle accidents which represent 30% of fatalities in France occur generally on rural roads and are due to inadequacy between vehicle dynamics and road geometry which is hardly constrained by ground relief. Addressing vehicle yaw dynamics improvement is of primary importance regarding driver inadequate actions or over-reaction. Vehicle dynamics variations due to own vehicle parameter or road interaction variations have also to be compensated in order to make the driver to feel with almost invariant vehicle. However, accidents due to excess yaw dynamics or loss of control are only part vehicle alone accidents. Lane departures represent also a major ratio of such type of accident and generally occur with less than 5deg of relative yaw angle.

In this paper, we address the problem of yaw dynamics handling improvement and driver lateral support by combination of differential braking and active front steering. This work is done within the framework of the French ARCOS project and is part of the CIVIC[1] concept developments.

The method presented uses both feedback and feedforward controllers. Using this configuration, we address with the same framework, both handling improvement and assistance according to the vehicle sensed environment: lane, other vehicles, which define the admissible trajectories. The feedback components use available information from proprio and/or exteroceptive sensors while the feedforward parts process the drivers input and are used for model reference tracking and compensation of the estimated road curvature. Using H_∞ performance index criteria, it is shown that this configuration allows achievement of robust model matching against parameters variations and rejection of lateral forces and torque disturbances which may rise from wind forces. Lane keeping in enhanced.

The remainder of the paper is organized as follows: section 2 introduces the model used for control synthesis. This linear model includes simple dynamics of both the vehicle and the driver, it is completed by the positioning equations against the lane centerline. Control synthesis and simulation results are provided respectively in sections 3 and 4.

[1]Conduite Intégrée Véhicule Infrastructure Conducteur

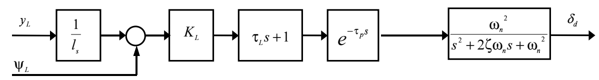

Fig. 1. Simple Driver model

II. VIC model

The driver assistance approach which uses individual wheel braking and active steering is developed on the basis of a low complexity VIC (Vehicle-Infrastructure-Conducteur) model version and then tested on the high level complexity [3]. This low complexity model is presented here.

A. Simple Human driver model

Modelling of human driver is a difficult task. However, several components can be identified [9]. The first one is called structural model. This component represents the high frequency driver compensation component, modelled by a dead time $\tau_p = 0.15$ sec representing inherent human processing time and neuromotor dynamics, and a second order low pass filter with damping factor $\xi_n = 0.707$ and natural frequency $\omega_n = 10$ rad.s^{-1}. The second component corresponds to the driver lead and predictive actions. It is modelled by a first order lead filter, where the time constant τ_L is representative of the driver mental load. The third component is a simple gain representing the proportional action of the driver face to the perceived vehicle positioning relative to the driving environment. This positioning is expressed in terms of lateral displacement at some look-ahead distance and the relative yaw angle (Figure 1).

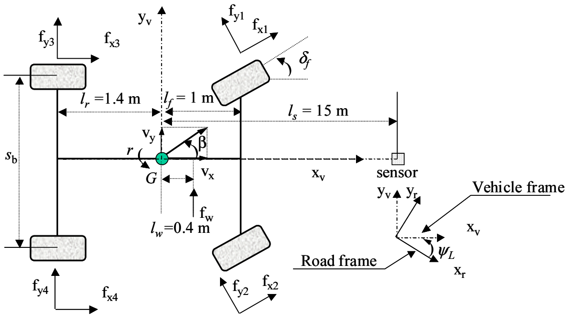

Fig. 2. vehicle dynamics model

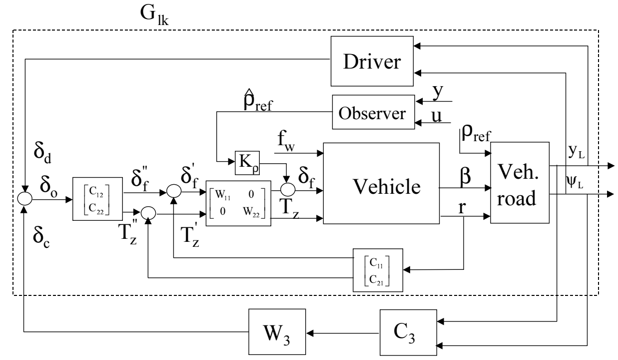

Fig. 3. Control architecture.

B. Vehicle dynamics for handling

The model used for control synthesis is derived from the high complexity model in which the longitudinal velocity v is assumed to be constant and not influenced by the yaw torque control input T_z used in differential braking. All the angles are also considered sufficiently small in order to allow linear approximations. The sideslip angle β is used as the first state variable, the second is the yaw rate r. It is also assumed equal cornering stiffness for the two front wheels $\frac{c_f}{2} = 25.2$ K.N/rad and the rear ones $\frac{c_r}{2} = 25.2$ K.N/rad. When the track width is neglected, the left and right tires slip angles are equal at front and rear wheels. The model takes the form

$$\dot{\bar{x}} = \bar{A}\bar{x} + \bar{B}_w\bar{w} + \bar{B}_u u \quad (1)$$

where $\bar{x} = [\beta, r]^T, \bar{w} = f_w$ is the disturbance wind force, $u = [\delta_f, T_z]^T$ is the control input where δ_f is the front tire steering angle.

$$\bar{A} = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix}, \bar{B}_w = \begin{bmatrix} b_{w_1} \\ b_{w_2} \end{bmatrix}, \bar{B}_u = \begin{bmatrix} b_{u_1} & 0 \\ b_{u_2} & \frac{1}{J} \end{bmatrix} \quad (2)$$

with

$$\begin{array}{llll} a_{11} = -\frac{c_r + c_f}{mv} & a_{12} = -1 + \frac{l_r c_r - l_f c_f}{mv^2} & b_{w_1} = \frac{1}{mv} & b_{u_1} = \frac{c_f}{mv} \\ a_{21} = \frac{l_r c_r - l_f c_f}{J} & a_{22} = -\frac{l_r^2 c_r + l_f^2 c_f}{Jv} & b_{w_2} = \frac{l_w}{J} & b_{u_2} = \frac{c_f l_f}{J} \end{array} \quad (3)$$

Vehicle parameter variations, mainly the cornering stiffness and the speed and represented in an linear fractional transformation (LFT) form by defining extra input and output on the system connected by a diagonal perturbation [7]. The mass $m = 1400$ Kg and the moment of inertia $J = 2750$ Kg.m^2 are constant. In the following, it is assumed that the additional steering angle is achieved by steer-by-wire and the yaw moment by differential wheel braking. The necessary control logic for obtaining the desired actions are outside the scope of this paper and is not addressed in the remainder.

C. Additional dynamics for lane keeping assistance

The previous model has to be expanded with two supplementary equations for lane tracking. Let $\psi_L = \psi - \psi_d$ be the yaw angle error which is the angle between the vehicle heading and the tangent to the road (figure 2). The differential equation for ψ_L is

$$\dot{\psi}_L = \dot{\psi} - \dot{\psi}_d = r - \dot{\psi}_d \quad (4)$$

The road reference curvature ρ_{ref} is defined by $\dot{\psi}_d = v\rho_{ref}$. Denoting l_s the look-ahead distance, the equation giving the evolution of the measurement of the lateral offset y_L from the centerline at sensor location is obtained by

$$\dot{y}_L = v(\beta + \psi_L) + l_s r \quad (5)$$

The new state vector is $x = [\beta, r, \psi_L, y_s]^T$, $w = \left[f_w, \rho_{ref}\right]^T$ is the disturbance input, the control input remain the same. In addition to the yaw rate, it is assumed that y_L and ψ_L are measured using a video sensor and are then available for feedback. The measurement vector is thus $y = [r, y_L, \psi_L]^T$.

III. Control synthesis

The control philosophy processes an internal loop for handling improvement and an external loop for lane keeping assistance. A combination of active steering and differential braking is used. In vehicle active steering, the front tires steering angle is set in part by the driver steering angle δ_d through the vehicle classical steering mechanism while an additional steering angle is set by the controller. The yaw torque generated by differential braking is directly set by the controller (Figure 3).

The internal loop uses dynamic feedback controller C_1 of the yaw rate and a dynamic feedforward controller $C_2$of the steering angle $\delta_o = \delta_d + \delta_c$ to respectively the tire steering angle and the yaw moment. The inner loop control takes thus the form

$$\begin{bmatrix} \delta_f' \\ T_z' \end{bmatrix} = \begin{bmatrix} C_1 & C_2 \end{bmatrix} \begin{bmatrix} r \\ \delta_o \end{bmatrix} = \begin{bmatrix} C_{11} & C_{12} \\ C_{21} & C_{22} \end{bmatrix} \begin{bmatrix} r \\ \delta_o \end{bmatrix} \quad (6)$$

where $\left[\delta_f', T_z'\right]^T$ is such that $\left[\delta_f, T_z\right]^T = W_1 \left[\delta_f', T_z'\right]^T$, and W_1 is a shaping filter of control inputs which will be defined below. The steering angle δ_o is commanded by the outer loop.

As lane keeping is a disturbance rejection problem, the outer loop uses a feedback controller C_3 of the lateral displacement from the lane centerline at a look-ahead distance

l_s and the relative yaw angle. This controller produces a steering angle $\delta_c = W_3 C_3 \left[y_L, \psi_L\right]^T$ which is added to that of the driver. The action of each control component is as follows

- The feedback controller C_1 ensures robust stability of the feedback loop with guaranteed damping enhancement on the yaw rate. It has also in charge fast disturbance rejection within driver reaction time. Controller C_2 acts as a prefilter of the reference signal by adding the feedforward action. This controller is synthesized to make the vehicle yaw rate response robustly follows as close as possible the response of a reference model. This constitutes robust model matching [5].
- From the vehicle point of view, a lane keeping maneuver requires the controller to reject lateral acceleration and yaw rate disturbances caused by changes in the radius of curvature. In fact, in this configuration, the reference curvature is an external input for the system. This is achieved by controller C_3.
- Finally, a constant gain K_ρ is added in order to compensate the road curvature effect by feedforward action from the estimated curvature $\hat{\rho}_{ref}$.

In the following a two stages approach is adopted for the synthesis of feedback and feedforward components. At the first stage the feedback part C_1 of the controller is computed using the H$_\infty$ coprime factors based loop shaping method of [8]. Afterwards, the new vehicle model which incorporates the feedback controller is computed, thus the feedforward part C_2 is synthesized from a second H$_\infty$ optimization. The procedure used for the synthesis of C_1 is also used for C_3. The inner loop is processed first and then the outer one. This ensures that handling improvement is still optimal even when the lane keeping assistance is not connected because of insufficient accurate video detection for example or driver choice. All the controllers are synthesized on a nominal linear system at speed of 20 m/s and full road adhesion.

A. Synthesis of $[C_{11}, C_{21}]^T$

We consider first the sub-system $G_{r[\delta_f, T_z]^T} = \left[G_{r\delta_f}, G_{rT_z}\right]$ which maps the front wheels steering angle and the yaw moment T_z to the yaw rate r. In order to reject a constant step input perturbation on the yaw rate, a diagonal weighting compensator is added on the inputs of the system $(W_1(s) = diag\{W_{11}, W_{22}\})$. The compensators are of the form of a PI filter for δ_f and a combination of two lead filter for T_z. This choice makes the yaw moment negligible after driver reaction time.

$$W_1(s) = diag\left\{0.3\frac{s+5}{s}, 375\frac{0.2s+1}{0.1s+1}\frac{100s+1}{10s+1}\right\} \quad (7)$$

Let now G_s be the shaped plant $(G_s = G_{r[\delta_f, T_z]^T} W_1)$. It has been verified that according to the gap-metric, the stability of the vehicle is guaranteed for admissible parameters variations. The stabilizing H_∞ feedback controller $C_1 = [C_{11}, C_{21}]^T$ is computed using the non-iterative procedure in [8] with a relaxed value of the maximal stability margin ($\gamma_1 = 1.5$) such that

$$\left\| \frac{1}{(1 - G_s C_1)} \begin{bmatrix} 1 & G_s \\ C_1 & C_1 G_s \end{bmatrix} \right\|_\infty = \gamma_1$$

This controller C_1 provides the needed phase lead for close loop stabilization and ensures robust stability for all systems variation. The controller is implemented as shown in figure 3.

B. Synthesis of $[C_{12}, C_{22}]^T$

The weighting compensators are left at the system input, the shaped system described by $r = G_{r\delta_f} W_{11} \delta'_f + G_{rT_z} W_{22} T'_z$ is closed with the feedback controller C_1 such that the control input are respectively $\delta'_f = C_{11} r + \delta''_f$ and $T'_z = C_{21} r + T''_z$. The mapping from the two inputs $\left[\delta''_f, T''_z\right]^T$ to r is $r = G_{ff_1}\delta''_f + G_{ff_2} T''_z$. We seek now a single input, two outputs feedforward controller $[C_{12}, C_{22}]^T$ such that $\left[\delta''_f, T_z\right]^T = [C_{12}, C_{22}]^T \delta_o$. The controller C_{12} is set as a static speed depend controller such that the DC-gain from steering angle δ_0 to yaw rate is equal to that of the conventional vehicle without feedback control and when yaw moment input is zero. As the compensator W_{11} contains an integral action, one has to choose $C_{12}(v) = G_{r\delta_f}(0, v) C_{11}(0)$. The dynamic feedforward controller C_{22} will be designed with robust model matching purposes. Let T_0 be the desired transfer function between δ_o and r. In order to ensure at nominal speed, the same steady state value for the controlled and the conventional car, the reference model is chosen as a first order transfer function with the same steady state gain as the conventional car. It is of the form $T_0 = \frac{G_{r\delta_f}(0,v)}{0.15s+1}$. The settling time is about 0.5 sec. A first order model also avoids overshot on vehicle responses. The error signal z is computed from $(z = r - T_0 \delta_o)$. The feedforward controller C_{22} has to keep the error signal z small in H_∞ sense for the class of perturbed systems according to vehicle parameter variations [2], [7]. When including the controller C_{22}, the error signal is thus $z = \left(G_{ff_1} C_{12} + G_{ff_2} C_{22} - T_0\right) \delta_o$. It can be rewritten in an LFT form which is suitable for H_∞ optimization

$$z = lft\left(\begin{bmatrix} G_{ff_1} C_{12} - T_0 & G_{ff_2} \\ I & 0 \end{bmatrix}, C_{22}\right) \quad (8)$$

C. Synthesis of C_3

After designing the controller C_{22}, the transfer function from δ_o to r is $G_h = G_{ff_1} C_{12} + G_{ff_2} C_{22}$. The control input δ_o is set in part by the driver and by the controller which performs lane keeping. The model is first completed with the two state equations for vehicle positioning relative to the lane The new measurement variables are the lateral displacement y_L at the look-ahead distance l_s and the relative yaw angle ψ_L. As shown on figure 3, the nominal system G_{lk} used for controller synthesis is obtained by feeding back the model with the driver model. This model has the steering angle δ_c as the control input and two output $[y_L, \psi_L]^T$. As lane keeping is a disturbance rejection problem, this system is shaped at the input by the pre-compensator $W_3 = \frac{0.1}{0.1s+1}$. A stabilizing controller is finally synthesized for the shaped plant with ($\gamma_3 = 2.2$), from

$$\left\| \begin{bmatrix} C_3 \\ I \end{bmatrix} (I - G_{lk} W_3 C_3)^{-1} \begin{bmatrix} G_{lk} W_3 & K_3 \end{bmatrix} \right\|_\infty = \gamma_3$$

The controller C_3 is finally implemented as shown in figure 3.

D. Estimation and Feedforward of road curvature

The available measurements are the lateral displacement, the relative yaw angle and the yaw rate. In the following, we seek a full order observer on the basis of the linear system driven from equations (1), (4) and (5), assuming that f_w and T_z are zero. The model is thus of the form

$$\begin{cases} \dot{x} = Ax + B\delta_f + E\rho_{ref} \\ y = Cx \end{cases}$$

with the objectives of state and unknown input estimation represented by ρ_{ref}

Assuming that the road curvature is almost constant, we choose a Proportional Integral (PI) observer which is able to estimate the curvature if we can approximate the unknown input ρ_{ref} as a constant disturbance [6]. The approximation error can be reduced by increasing the observer bandwidth. The PI observer has the following form

$$\begin{cases} \dot{\hat{x}} = A\hat{x} + B\delta_f + L_p(y - \hat{y}) + E\hat{\rho}_{ref} \\ \dot{\hat{\rho}}_{ref} = L_i (y - \hat{y}) \end{cases}$$

The second equation describes the integral loop gain added to the proportional one in the first equation. The matrix gains L_p and L_i are determined in such a way to enable asymptotic convergence to zero of the state and unknown input estimation errors, respectively defined by $e = x - \hat{x}$ and $e_\rho = \rho_{ref} - \hat{\rho}_{ref}$. Error dynamics are given by

$$\begin{bmatrix} \dot{e} \\ \dot{e}_\rho \end{bmatrix} = \begin{bmatrix} A - L_pC & E \\ -L_iC & 0 \end{bmatrix} \begin{bmatrix} e \\ e_\rho \end{bmatrix}$$

The matrix dynamics has to be Hurwitz, it can be rewritten as $\begin{bmatrix} A - L_pC & E \\ -L_iC & 0 \end{bmatrix} = \begin{bmatrix} A & E \\ 0 & 0 \end{bmatrix} - \begin{bmatrix} L_p \\ L_i \end{bmatrix} \begin{bmatrix} C & 0 \end{bmatrix}$, thus an eigenvalue assignment method is applied to obtain the gain $\begin{bmatrix} L_p \\ L_i \end{bmatrix}$.

The estimated road curvature is finally used as an additional feedforward action through a adjusted gain K_ρ. This gain is chosen in order to satisfy nominal steady state gain conditions for disturbance rejection (Figure 3).

IV. Simulation results

In all figures, solid lines correspond to the controlled car responses and dotted ones to the conventional car responses. A first set of simulations addresses yaw dynamics improvement, while the second set concerns lane keeping.

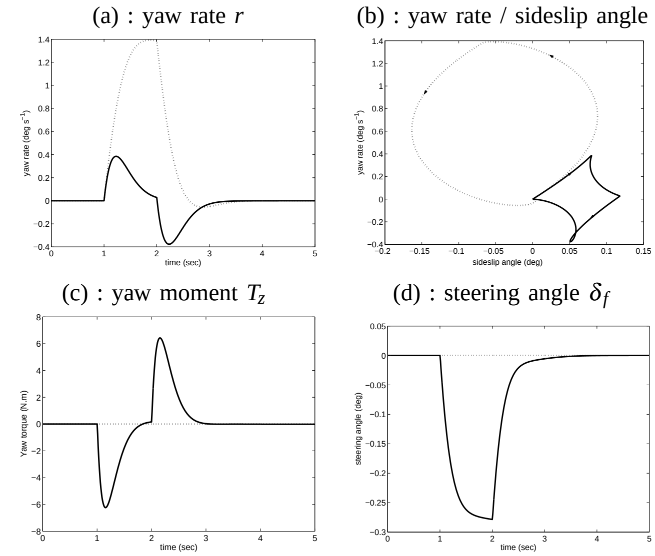

Fig. 4. Wind forces step input rejection for nominal system. Solid : controlled, dotted : conventional.

A. Handling improvement

1) Disturbance rejection: The vehicle is assumed at nominal speed and full road adhesion and is subject to a step disturbance wind force. The wind force appears at time $t_1 = 1$ sec and disappears at $t_2 = 2$ sec. It is assumed that the driver doesn't react to this disturbance. In this case, only controller C_1 is in action. One can note from Figure 4 that the yaw rate is greatly reduced and thus the controlled vehicle will remain closer to road centerline. In addition, the maximum value of yaw rate during the transient phase is smaller than the one of the conventional car and the disturbance is practically rejected within driver reaction time. One can notice that yaw moment quickly vanishes due to limiting effect of the shaping filter W_{22}. It was verified that the controller exhibits good stability and performance robustness.

Responses for $v = 40$ m/s and half adhesion are given in Figure 5. The controller exhibits good stability and performance robustness, in fact wind force disturbance is still well rejected.

2) Lane change maneuver: The handling improvement is now investigated in case of driver steering angle which corresponds to lane change maneuver (Figure 6-c, dotted line). In this case, both controllers C_1 and C_2 are in action. The dashed line corresponds to the response of the reference model. Figure 6 shows results obtained at nominal speed with road adhesion equal to 1. Figure 7 shows results obtained for $v = 40$ m/s and nominal adhesion. Robust model matching occurs, and due to the speed scheduling of the gain parameter $\alpha(v)$, we ensure that the controlled vehicle and the conventional one present the same steady state behavior. When the road adhesion is at its nominal value even when the speed varies, the control effort vanishes within driver reaction time which is assumed to be between

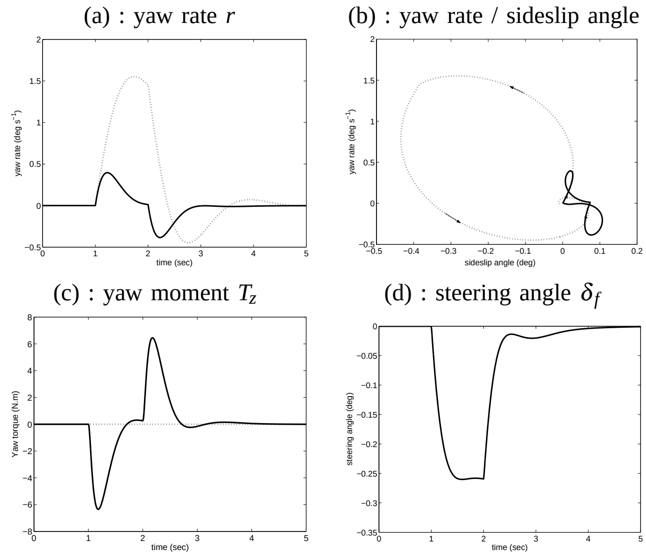

Fig. 5. Wind forces step input rejection for perturbed system (solid : controlled, dotted : conventional)

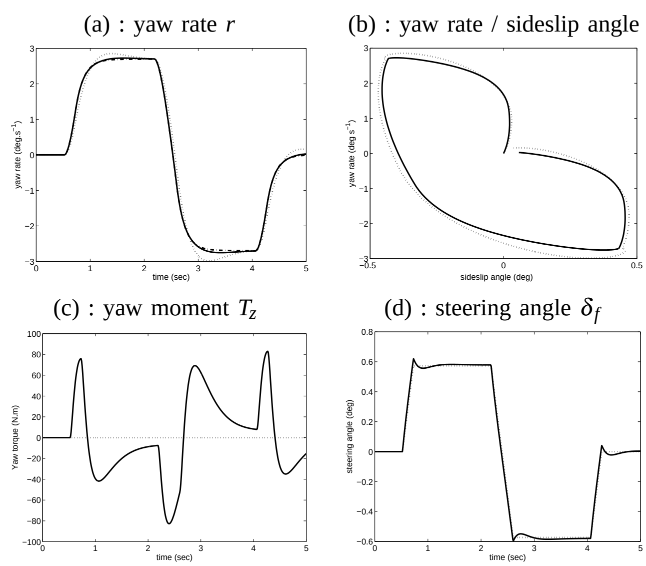

Fig. 6. Lane change maneuver, nominal system. Solid : controlled, dotted : conventional, dashed : ref. model.

0.5 and 1 second. When the road adhesion is decreased, the control actions do not vanish. Lane keeping capabilities are now investigated.

B. Lane keeping improvement

1) Simple lane keeping maneuver: At the beginning of the simulation, the vehicle is on straight road section, at a lateral distance of 0.5 m from the lane centerline. The relative yaw angle is zero. As shown in Figure 8 with dotted line, without any control support, the driver gives a steering angle in order to make the vehicle close to the centerline. The overshot is about -0.2 m and the lateral displacement is near zero 3 sec later. At 5 sec, the vehicle enters a curved road section with 1/500 m^{-1} of road curvature. The lateral

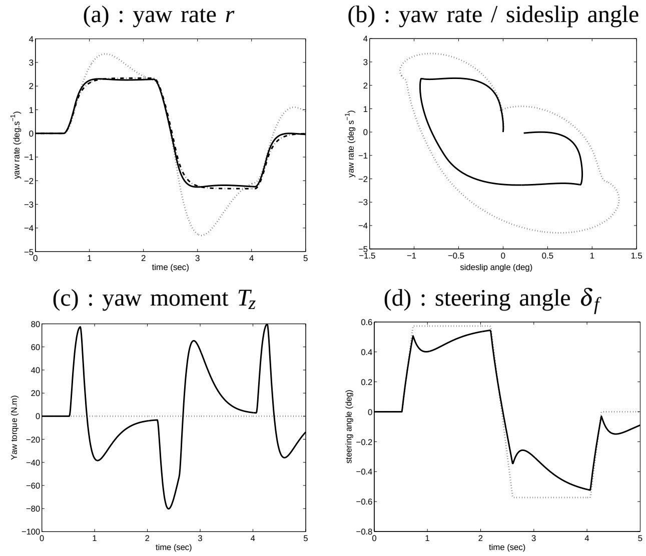

Fig. 7. Lane change maneuver, for nominal road adhesion and speed at 40 m/s (solid : controlled, dotted : conventional, dashed : reference model).

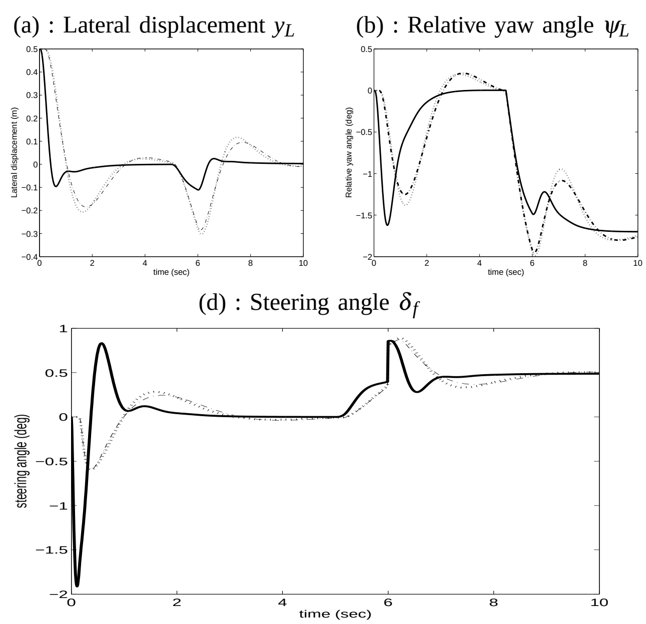

Fig. 8. Simple lane keeping maneuver, nominal system solid : controlled with LK controller, dotted : conventional, dashed : controlled with handling controller.

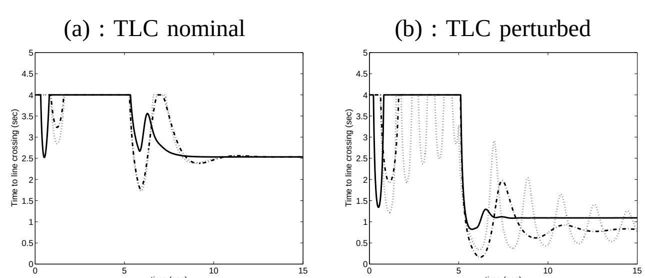

Fig. 9. Time-to-line-crossing in nominal and perturbed cases. Solid : controlled with LK controller, dotted : conventional, dashed : controlled with handling controller.

displacement overshoots again to -0.3m but is less than 0.1 m, 2 sec later. On the same figure, dashed lines show the responses when handling improvement support is activated. Responses are rather the same but small reductions of peak values can be observed. Similarly, solid lines correspond to the vehicle with both handling improvement and lane keeping support. In this case, response time is less than 1 sec and overshoots form centerline are under 0.1 m. However faster and larger amount of steering angle is required. Finally, Figure 9-a and 9-b show the achieved time to line crossing (TLC) by each vehicle when they are first at nominal conditions and then at high speed respectively. The vehicle with both handling and lane keeping support presents the best TLC particularly when entering the curve.

2) Validation on test track: In 1999, INRETS established a test track in Satory, 20Km western Paris. The site is 9Km long with various road profiles including straight lanes, tight bends and squabble (figure 10-a). Lanes markers absolute positions are digitalized each 5cm using differential GPS. Figure 10-b shows the curvature of the test track. It is easy to see that the vehicle will experience high lateral acceleration at the bends and the squabble even at low speeds. The yaw rate measurement is taken from a gyro, while the lateral displacement and relative yaw angle are computed by using RTK DGPS [4]. Sensors and actuator systems are managed using a National Instrument Labview application.

(a) : Test track (b) : Curvature

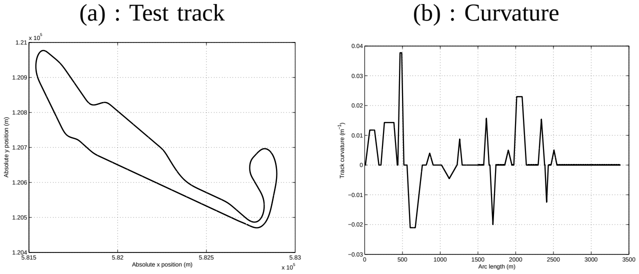

Fig. 10. Digitalized map of the LIVIC test track and corresponding curvature.

First of all, the estimate of the track curvature is shown on figure 11. It is easy to see that the proposed PI observer performs curvature estimation well even on clothoïd sections. Figure 12-a shows the trajectory of the conventional vehicle in dotted line and the trajectory of the vehicle with lane keeping assistance in solid line. The track centerline is in dashed line. The trajectory of the controlled vehicle is always closer than that of the conventional vehicle. The lane keeping is especially enhanced in the squabble.

V. Conclusion

In this paper, some aspects of the combination of active steering and individual wheel braking have been explored. Both handling improvement and lane keeping support are addressed. On the basis of several simulated maneuvers, It has been shown that the controlled vehicle exhibits better yaw damping and enhanced lane keeping.

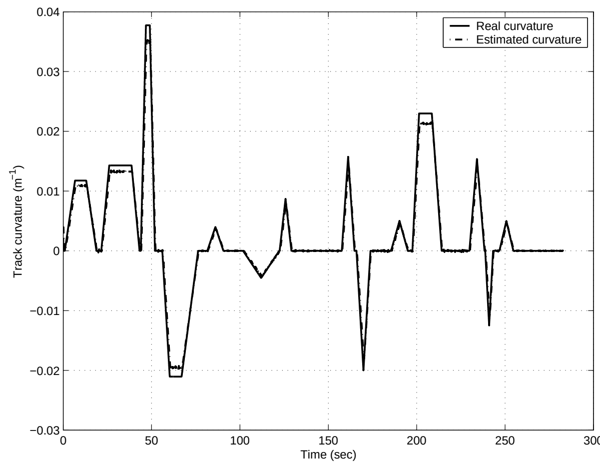

Fig. 11. Real and estimated road curvature.

(a) : Trajectories at the squabble (b) : Zoom on the squabble

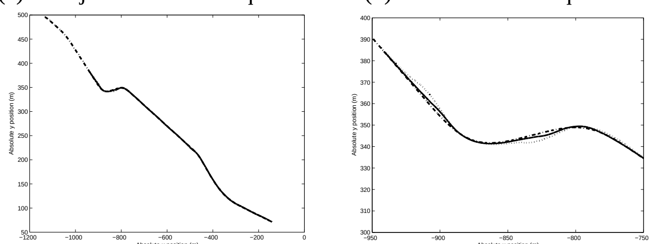

Fig. 12. Lane following ont test track. Solid : controlled, dotted : conventional, dashed : track centerline.

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