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Simulation results on combined automatic lane keeping and driver's maneuvers

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Abstract— **In this paper we address the problem of combining automatic lane keeping and driver's steering for either obstacle avoidance or lane change maneuvers for passing purpose or any other desired maneuvers, through a closed loop control strategy. The automatic lane keeping control loop is never opened, and no on/off switching strategy is used. During the driver's maneuver, the vehicle lateral dynamics is controlled by the driver himself through the vehicle steering system. When there is no driver's steering action, the vehicle center of gravity tracks the center of the travelling lane thanks to the automatic lane keeping system. At the beginning (end) of the maneuver, the lane keeping task is released (resumed) safely and smoothly. The performance of the proposed closed loop structure is shown by means of simulations.**

Index Terms— **Vehicle lateral control, automatic lane keeping, driver's steering.**

I. INTRODUCTION

Intelligent Vehicle Systems (IVS) have recently become an attractive area of research throughout the world. The aim of the research effort is mainly that of enhancing driving safety and reducing driver's workload. In [1] the positive effect of driver assistance systems on driver's physical and mental workload reduction is shown through the measure of some vital reactions such as the electromyographic and the electrodermal activities. In particular, Automated Highway Systems (AHS), extensively studied at the Ohio State University since 1964 ([2]), are receiving renewed attention due to fast developments in hardware/software technology that allow the design of more effective control systems. Since mid-eighties a larger effort is being conducted mainly in the California PATH program. Early attempts of the project were devoted to assess previous knowledge in the field of automatic vehicle control providing the analytical basis for new developments ([3]). In the last years large efforts have been directed to the solution of the highway automatic steering control problem for different type of vehicles and using different control strategies and techniques. Most of the contributions rely on buried magnet or electrified wires placed along the path for the detection of the vehicle lateral position, the so called *look down* sensing scheme. The problem, in the case of passenger cars, was analyzed in this framework by Patwardhan *et al.* in [4] where they show the fundamental control difficulties of this approach. An interesting alternative approach, that avoids the modification of infrastructures, involves the use of vision sensors placed on the vehicle, the so called *look-ahead* sensing scheme. A comparative study of vision-based control strategies was presented by Košeckà *et al.* in [5]. A great deal of remarkable works about application of advanced linear and nonlinear control techniques to the automatic steering control problem were conducted, still in the PATH program, by Tomizuka and coworkers in the case of four wheeled vehicles in [6], heavy-duty vehicles in [7] and tractor-semitrailer combination in [8]. In recent years the problem of steering control has attracted wide interest also outside the PATH program. Relevant contributions in the field of robust steering and vehicle lateral dynamics control were also provided by Ackermann and coworkers (see, e.g., [9], [10]). Preliminary experimental results on robust lateral control conducted by Byrne *et al.* were reported in [11] highlighting several implementation difficulties. Other contributions based on the look-ahead sensing scheme were provided by Hatipoğlu *et al.* in [12], where they use a digital videocamera together with a radar system, by Broggi *et al.* [13] who used a stereo vision system composed by two videocameras, and by Cerone *et al.* [14], [15], [16] where properties of single-input two-output (SITO) systems are exploited. Significant japanese contributions to the development of vision-based intelligent vehicles, given since the mid 1970's to early 1990's are surveyed by the paper of Tsugawa [17].

Some of the advanced maneuvers pertaining to vehicle lateral control are lane change for vehicle passing purpose and obstacle avoidance. The problem of automated lane change maneuvers is widely addressed in the literature. In [12], [18] and [19], for example, the authors consider open loop and closed loop lane change and design time optimal steering controllers with nonlinear constraints. First they generate a particular open loop lane change steering signal which minimizes the period of lane change subject to constraints on the lateral acceleration and jerk. Then, they discuss how to implement those steering commands in the closed loop system using a lane following controller previously published. We point out that autonomous lane change deals with the generation of the appropriate steering signal to have the vehicle accomplish the task without driver assist.

In this paper we address the problem of combining automatic lane keeping and driver's steering for either obstacle avoidance or lane change maneuvers for passing purpose or any other desired maneuvers, through a closed loop control strategy. First, in Section II, we present the physical plant and derive a suitable simplified model focusing on the accordance between simplification hypotheses and experimental context. Then, in Section III, we formulate the control problem.

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Further, in Section IV, we propose a two degree of freedom closed loop control strategy which gives a solution to the formulated problem. The automatic lane keeping control loop is never opened, and no on/off switching strategy is used. During the driver's maneuver, the vehicle lateral dynamics is controlled by the driver himself through the vehicle steering system. When there is no driver's steering action, the vehicle center of gravity tracks the center of the travelling lane thanks to the automatic lane keeping system. At the beginning (end) of the maneuver, the lane keeping task is released (resumed) safely and smoothly. Simulation results obtained with the proposed control structure and the model of a FIAT Brava 1600 ELX, whose numerical parameter values were provided by Centro Ricerche Fiat, are presented and discussed in Section V.

II. PLANT DESCRIPTION AND MODELING

The plant to be controlled, provided by Centro Ricerche Fiat, consists of a Fiat Brava 1600 ELX equipped with a vision system and a steering actuator. The vision system comprises a single CCD videocamera and related image processing algorithms. The steering actuator system is a locally controlled DC brush-less electric motor. Both control and vision algorithms are processed by an INTEL 486 microprocessor based Personal Computer for industrial applications. The sampling time of the whole system is $T_s = 40$ ms. The described given hardware is not supposed to be modifiable by the control designer. The mathematical modeling of such a plant was discussed in detail in paper [14] in which a simplified model, able to describe the vehicle behaviour in highway experimental conditions, was presented. The equations of such a model are here recalled for self-consistency of the paper. The interaction between the vehicle lateral dynamics and the vision system can be modeled by the following state space equations parameterized by the longitudinal velocity v_x

$$\begin{bmatrix} \dot{v}_y \\ \ddot{\psi} \\ \dot{q} \\ \dot{m} \end{bmatrix} = \begin{bmatrix} \frac{-a_1}{m_v v_x} & \frac{-m_v v_x^2 + a_2}{m_v v_x} & 0 & 0 \\ \frac{a_3}{I_\psi v_x} & \frac{-a_4}{I_\psi v_x} & 0 & 0 \\ -1 & 0 & 0 & v_x \\ 0 & -1 & 0 & 0 \end{bmatrix} \begin{bmatrix} v_y \\ \dot{\psi} \\ q \\ m \end{bmatrix} + \begin{bmatrix} b_1 \\ b_2 \\ 0 \\ 0 \end{bmatrix} \delta_v + \begin{bmatrix} 0 \\ 0 \\ -L v_x \\ v_x \end{bmatrix} K_L. \tag{1}$$

where L is the so-called look-ahead distance, taken along the longitudinal axis, from the vehicle center of gravity to a suitable point in the road between 3 and 20 meters ahead the vehicle (see Fig. 1); $K_L(t)$ is the road curvature, i.e., the inverse of the instantaneous curve radius at the look-ahead point; δ_v is the steering-wheel angle; m and q are the measurements supplied by the vision system about the vehicle location on the lane, as shown in Fig. 2. The meaning of the symbols involved in the equations is reported below:

m_v : vehicle mass;
v_x : longitudinal component of center of gravity (CG) velocity;
v_y : transverse component of CG velocity;
ψ : vehicle yaw angle;
I_ψ : inertial vehicle moment around center of gravity referred to the vertical axis;
δ_v : steering-wheel angle;
k : front wheels angle/steering-wheel angle ratio, expressed in rad/degrees;
$\delta_f = \delta_v k$: front wheels angle;
c_r : cornering stiffness of front tires;
c_f : cornering stiffness of rear tires;
l_r : distance between the rear axle and the center of gravity;
l_f : distance between the front axle and the center of gravity;
$l = l_r + l_f$: wheelbase;

$$a_1 = c_f + c_r; \qquad a_4 = l_f^2 c_f + l_r^2 c_r;$$
$$a_2 = c_r l_r - c_f l_f; \qquad a_3 = -l_f c_f + l_r c_r;$$
$$b_1 = \frac{c_f}{m} k; \qquad b_2 = \frac{l_f c_f}{I_\psi} k.$$

The steering actuator, given by *Centro Ricerche Fiat* is described by the following transfer function:

$$G_a = \frac{\delta_v(s)}{\theta(s)} = \frac{1580}{s^2 + 75.5s + 1580} \tag{2}$$

where θ is the reference steering angle due to the sum of θ' (provided by the lateral dynamics controller) and θ'' (i.e, the effect of the torque τ applied by the driver on the steer), as can be seen in Fig. 3. The transfer function between the torque τ applied by the driver on the steer and the reference steering angle is denoted by G_d (see Fig. 3). The bandwidth of $G_d(j\omega)$ is supposed to be wider than the bandwidth of the control system to be designed; from available data, it can be assumed that, in the working frequency range, a static gain $\overline{G}_d = 1/3$ can be used to describe the behaviour of G_d. The transfer function of the nominal plant to be controlled is $G_p \triangleq y(s)/\delta_v(s)$ where $y(t) = q(t) + m(t)L$ is the position error of the vehicle with respect to the lane centerline measured at the look-ahead distance L (see [14] for a discussion about the advantages of using such a quantity as feedback signal in lane keeping control systems). Most of the early contributions in the field of steering control were based on the use of reduced order nominal models like the one presented above, sometimes considering the effect of the longitudinal velocity variations (see for example [20]). In recent years, works have been presented where the attention is focused on the model uncertainty (see, e.g., [9], [11], [10], [7]). Looking at the proposed model it can be noted that some unmodeled linear and nonlinear dynamics are present in the actual plant like for example roll, yaw and heave effects neglected by the single track model, steering gear backlash or actuator voltage command saturation. Moreover, the system has a time varying nature due to parametric dependence on the longitudinal velocity v_x and, finally, some parameters of the simplified model can't be exactly known. Thus, all these sources of uncertainty should be taken into account in the controller design. However, it can be noted that unmodeled dynamics are little excited along highway paths, which are characterized by bends with large radius requiring slow steering actions, while highway longitudinal velocity variations are typically slow. Thus, all considered, the following parametric uncertainties have been considered. The vehicle mass m_v can take values from 1226 K_g (nominal value) to 1626 K_g, corresponding to the mass of the vehicle with 5 passengers, each one weighting 80 K_g, on board. The inertial moment I_ψ can vary coherently from 1900 $K_g m^2$ to 2520 $K_g m^2$. The cornering stiffness coefficients c_f and c_r range respectively in $[51000, 69000]$ and in $[81600, 110400]$. Finally, the velocity is handled as an uncertain parameter whose value belongs to $[60, 120]$ km/h. The nominal values of the parameters $\gamma_{nom} = [m_{v_0}\ I_{\psi_0}\ c_{r_0}\ c_{f_0}\ v_{x_0}\ ]$ and the corresponding parameter uncertainty intervals $PUI_i \triangleq [\gamma_i^{min},\ \gamma_i^{max}]$ are summarized in Table I.

TABLE I

VALUES OF MODEL PARAMETERS

	Parameter	γ_{nom}	**PUI**
γ_1	m_v (kg)	1226	$[1226, 1626]$
γ_2	I_ψ (kgm^2)	1900	$[1900, 2520]$
γ_3	c_r (N/rad)	96000	$[81600, 110400]$
γ_4	c_f (N/rad)	60000	$[51000, 69000]$
	l_r (m)	$1,506$	–
	l_f (m)	$1,034$	–
γ_5	v_x (km/h)	95	$[60, 120]$
	k (rad/degrees)	$\pi/(180 \cdot 18)$	–

The uncertain system generated by the given parameter uncertainty intervals can be represented by the following model set:

$$\mathcal{M}_P = \{G_p(s, \gamma) \in \mathcal{C} : \gamma \in \Gamma\} \tag{3}$$

where $\gamma = [m_v\ I_\psi\ c_r\ c_f\ v_x\ ]$ is the vector of the uncertain parameters and $\Gamma = \{\gamma \in \Re^5 : \gamma_i \in PUI_i, i = 1, 2, 3, 4, 5]$.

III. CONTROL PROBLEM FORMULATION

The problem we are dealing with in this paper is the design of a closed loop control system able to keep the vehicle inside the lane along typical highway paths and to permit any intervention on the side of the driver in order to override the automatic lane keeping system and obtain complete control of the vehicle lateral dynamics. In other words, we aim at a combined automatic lane keeping and driver's steering through a closed loop control strategy. We assume that the automatic lane keeping system is available (see, e.g., [14], [15], [16]). In the paper, by *driver's steering* or *driver's maneuver* we mean any intervention of the driver on the vehicle steering system in order to obtain a desired behaviour of the vehicle (for example, lane change for passing purpose or obstacle avoidance) and, in general, when it is desired to override the automatic lane keeping and obtain complete control of the vehicle lateral dynamics.

Control problem description — The following specifications (S1-S5) concur in the definition of the control problem under consideration, i.e., the design of a closed loop control strategy for combined automatic lane keeping and driver's steering.

(S1) Before the driver's steering action, the vehicle center of gravity (CG) tracks the center of the travelling lane.

(S2) At the beginning of the maneuver, the lane keeping task must be released safely and smoothly.

(S3) During the driver's maneuver, the vehicle lateral dynamics is controlled by the driver himself through the vehicle steering system.

(S4) At the end of the driver's maneuver, the CG tracks the center of the lane in which the vehicle is travelling and the lane keeping task must be resumed safely and smoothly.

(S5) A closed loop control strategy is sought which combines the automatic lane keeping and the driver's maneuvers, which means that the automatic lane keeping control loop is never opened, and no on/off switching strategy is used.

IV. COMBINED AUTOMATIC LANE KEEPING AND DRIVER'S STEERING

A. A two DOF structure

It is desired that the automatic lane keeping control loop be never switched off, i.e., the loop control be always active. Furthermore, it is also specified that the transition between the automatic lane keeping mode and the driver's steering mode be actuated without on/off switching strategy. In order to meet the above requirements, we propose the two DOF structure of Fig. 4. The feedback controller C_1 is designed in order to satisfy automatic lane keeping specifications (see, e.g., [14], [15], [16]), while C_2 is designed with the aim of combining the above described two modes in a smoothly way.

B. Design of controller C_2

The guide lines for the design of controller C_2 are derived from specifications (S1)-(S4) which describe the control problem under consideration. More precisely, C_2 must be designed in such a way that the following conditions hold:

Condition (C-1): the vehicle lateral dynamics be controlled by the automatic lane keeping system when the torque τ applied by the driver on the steer is negligible (≈ 0) and

Condition (C-2): when the driver's torque τ on the steer is different from zero, the vehicle behaviour perceived by the driver is as close as possible to the one of the vehicle without automatic lane keeping.

It is easily seen that the two DOF structure shown in Fig. 4 takes care of condition (C-1). As a matter of fact, when $\tau = 0$ the proposed structure simplifies to a one DOF with regulation to a zero reference signal, i.e., the automatic lane keeping control system. In order to take care of condition (C-2), the following result is given:

Result 1 — Condition (C-2) holds iff

$$C_2 = G_d G_a G_p \tag{4}$$

Proof $\Rightarrow$ Condition (C-2) implies that the error signal e be zero, i.e.,

$$e = \tau \frac{C_2}{1 + C_1 G_a G_p} - \tau \frac{G_d G_a G_p}{1 + C_1 G_a G_p} = 0 \qquad (5)$$

which, in turn implies (4).
$\Leftarrow$ The converse is also true. If $C_2 = G_d G_a G_p$, then $e = 0$ and the automatic lane keeping control loop is open.

C. Implementation issues: stability of C_2

As can be seen from Fig. 4, C_2 is a cascade filter, thus it must enjoy the stability property. While G_d is a constant scalar transfer function (t.f.) and G_a is the stable actuator t.f., unfortunately G_p shows two integrators, responsible of instability. Thus C_2 cannot be implemented in the form of (4). We implemented the following stable approximation of C_2:

$$\tilde{C}_2 = G_d G_a \frac{s^2}{(s-\alpha)^2} G_p \qquad (6)$$

As can be seen, the double integrator contained in G_p is replaced by a couple of real poles $s = \alpha$, and α is chosen in order to get an acceptable compromise between low frequency behaviour approximation and settling time of the control system step response. Results presented in the paper are obtained with $\alpha = -0.2513$.
Remark — We point out that if controller C_2 is given by (4), at least in principle a null signal error e is expected when the driver's torque τ on the steer is different from zero. However, there are a couple of reasons which lead to a non perfectly zero e. Firstly, the components of C_2, i.e., G_d, G_a and G_p, are not exactly known and can only be suitably approximated. Secondly, as discussed above, only a stable approximation of G_p can be implemented. In practice, however, the presence of an error signal e slightly different from zero when $\tau \neq 0$ does not compromise the desired performance of the proposed control structure. The only resulting drawback (which indeed might not be considered as such) is the presence of an "opposing" torque "felt" on the steer by the driver, which, actually can be considered as a simulation of the load due to the steering system.

D. Implementation issues: lane change maneuver

We recall that q is the measurement supplied by the vision system about the vehicle location on the lane, as shown in Fig. 2. Thus, during the lane change maneuver, the vision system is subject to a change in the measurement reference system: before the maneuver q is measured with respect to the current lane, at the end of the maneuver q is measured with respect to the new lane. The transition from one lane to the next can be handled by properly re-setting the initial condition of controller C_2 with respect to the new measurement reference system. This can be simply accomplished if one notes that when the vehicle crosses the line between two lanes there is no loss of continuity in the time evolution of the state variables m, $\dot{\psi}$ and v_y, while there is a discontinuity in the state variable q, as large as the width of the lane, due to the change of the measurement reference system. Thus, in order to reset the controller, a realization is sought in the state variables form, four of which are just the physical state variables (m, $\dot{\psi}$, v_y and q) of the plant, whose model G_p is part of C_2. It can be easily shown that the controller reset can be achieved by changing the sign of the state variable of C_2 corresponding to the physical state variable q.

E. Robustness issues: stability

In order to analyze the robustness of the proposed control system, a suitable, although conservative, description of the uncertainty generated by the considered parameters perturbation is provided by the following model set expressed in the so-called input multiplicative form:

$$\begin{aligned}\mathcal{M}_{G_p} = \{ & G_p(s) = \\ & (1 + \Delta_m(s)) G_{p_n}(s) : |\Delta_m(j\omega)| \leq W_2(j\omega), \forall \omega \}\end{aligned} \qquad (7)$$

where G_{p_n} is the nominal plant, $\Delta_m(s)$ is a complex function which represents the unknown modeling error and $W_2(s)$ is a known function bounding the modeling error. A description of $\Delta_m(j\omega)$ obtained gridding the hypercube Γ is shown in Fig. 5 together with the upper bound $W_2(s) = 0.6$. As far as robust stability is concerned, the application of the Small Gain Theorem to the block diagram of Fig. 4 leads to the following condition:

Result 2 — The control system in Fig. 4 with $G_p(s) \in \mathcal{M}_{G_p}$ is robustly stable iff

$$\left\| \frac{G_{p_n} G_a C_1 W_2}{1 + G_{p_n} G_a C_1 G_m} \right\|_\infty < 1 \qquad (8)$$

where $\| \cdot \|_\infty$ is the H_∞ norm of a system.
Result 2 shows that robust stability of the proposed control scheme is not affected by the design of the filter C_2. The lane-keeping controller $C_1(s)$ designed in paper [16] satisfies condition (8).

F. Robustness issues: performance

As far as robust performance is concerned, we are interested in evaluating the effect of the modeling error on the main task of the proposed approach, expressed by condition (2) of subsection IV-B, which is equivalent to the condition $e(s)/\tau(s) = 0$. Let us assume $C_2(s) = G_d G_a G_{p_n}$ and $G_p(s) = G_{p_n}(s) + \Delta_a(s)$ where $\Delta_a(s) = G_{p_n}(s)\Delta_m(s)$ is the additive unstructured modeling error. The transfer function from the torque $\tau(t)$ to the error $e(t)$ is (see Fig. 4):

$$\frac{e(s)}{\tau(s)} = \frac{-\Delta_a G_a G_d}{1 + G_p G_a C_1} = -\Delta_a G_a G_d S \qquad (9)$$

where $S = (1 + G_p G_a C_1)^{-1}$ is the sensitivity function of the lane keeping control loop. Thus, equation (9) shows that the effects of the modeling error Δ_a are attenuated by the feedback loop. A quantitative evaluation of the robustness of the proposed two DOF structure is provided by the simulations presented in Section V.

V. Simulation results and discussion

In this section we report the simulation results obtained with the proposed two DOF control structure of Fig. 4 where the mathematical model of the plant is the one presented in Section II and the lane-keeping controller designed in [16] is used for $C_1(s)$. The test track consists of a straight section followed by a curve with radius $R \approx 1000$ m. A double lane change maneuver has been performed when the car is driving along the straight section. The driver's torque in the simulation is an experimental torque sequence obtained from a double lane change performed by a test driver on an italian highway driving the vehicle provided by FIAT S.p.A. The feedback output y and the signal $\overline{y}$ are compared in Fig. 7 while Figg. 6 and 8 show the driver's torque τ and the steering angle δ_v respectively. From time $t = 0$ s to time $t \approx 55$ s the vehicle is on a straight section. At time $t = 4.5$ s the driver applies a torque on the steer (see Fig. 6) to perform a double lane change which ends at time $t \approx 20$ s; as can be seen from Fig. 7, during this maneuver the signal $\overline{y}$ is quite close to the output signal y which means that the driver has full control of the vehicle during this stage (specification S3). At the end of the double lane change maneuver the driver's torque τ rapidly decrease to zero and the signal $\overline{y}$, i.e., the output of the designed filter C_2, goes to zero trough a smooth transient; during such a transient, the lane keeping controller C_1 drives the vehicle to the centerline of the lane through the tracking of the signal $\overline{y}$ and, at the end of the transient, the lane keeping task is resumed smoothly (specification S4). From time $t \approx 55$ s to time $t \approx 88$ s the vehicle is driven by the lane keeping system along a curve of radius $R \approx 1000$ m. During the curve the driver does not operate on the steer, thus, as expected, the signal $\overline{y}$ approaches the value zero and the output y of the system, which is the position error with respect to the centerline of the lane measured at the look-ahead distance L, is suitably regulated by the controller C_1 (specification S1). The above simulations have been performed at the velocity $v_x = 90$ km/h. The effect of the parameters uncertainty on the performance of the proposed two DOF control system is analyzed in Fig. 10. Such a figure shows the comparison between the feedback output y and the signal $\overline{y}$ during a double lane change performed at the velocity $v_x = 90$ km/h for a number of different values of the parameters m_v, I_ψ, c_r and c_f. It must be noted that in all the simulations it was assumed that, coherently with the actual maneuver performed by the test driver on the real vehicle, the lane changes are always performed at time $t = 8$ s and $t = 22$ s respectively. In such a way, for each value of the uncertain parameters the simulation study provides a different value to which the state variable q has to be reset. Due to this fact, for each value of the uncertain parameters, we have a different signal $\overline{y}$ although the filter C_2 has been designed on the basis of the nominal plant G_{p_n}. Details of this simulation are shown in Fig. 11. As can be seen from such a figure, the presence of uncertainty in the parameters causes only a slight degradation of the performance. Similar results have been obtained for double lane changes performed at velocities $v_x = 60$ km/h and $v_x = 120$ km/h (see Figg. 9 and 12 respectively).

VI. Conclusions

In this paper we addressed the problem of combining automatic lane keeping and driver's steering for either obstacle avoidance or lane change maneuvers for passing purpose or any other desired maneuvers, through a closed loop control strategy. The automatic lane keeping control loop is never opened, and no on/off switching strategy is used. During the driver's maneuver, the vehicle lateral dynamics is controlled by the driver himself through the vehicle steering system. When there is no driver's steering action, the vehicle center of gravity tracks the center of the travelling lane thanks to the automatic lane keeping system. At the beginning (end) of the maneuver, the lane keeping task is released (resumed) safely and smoothly. Simulation results obtained with the proposed control structure and the model of a FIAT Brava 1600 ELX, whose numerical parameter values were provided by Centro Ricerche Fiat, showed the fulfillment of given specifications.

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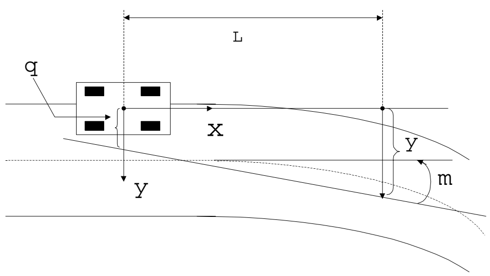

Fig. 1. Feedback signal y.

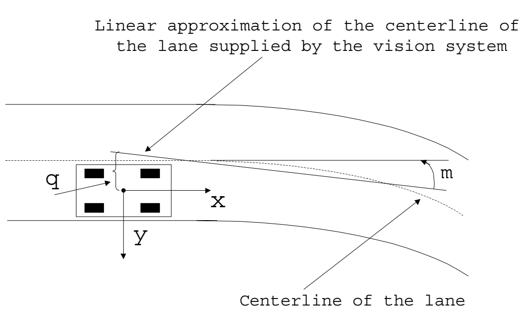

Fig. 2. Centerline linear approximation supplied by the vision system.

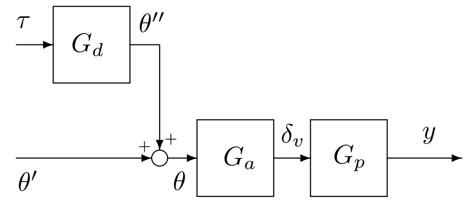

Fig. 3. Plant block diagram.

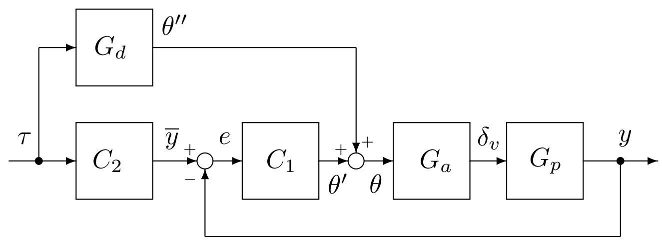

Fig. 4. Combined automatic lane keeping and driver's steering through a two degree of freedom structure.

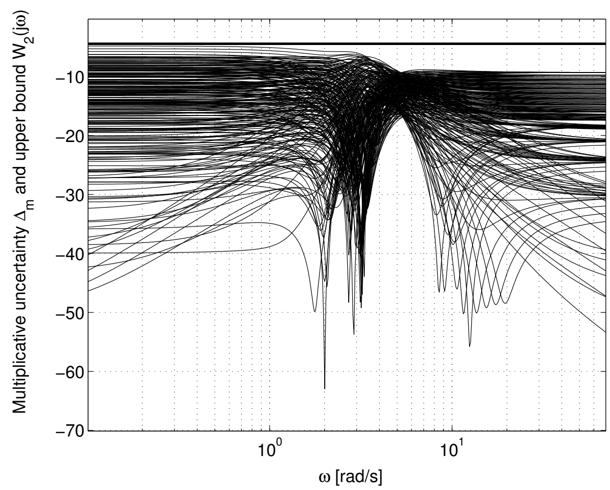

Fig. 5. Uncertainty Δ_m and weighting function W_2 (solid).

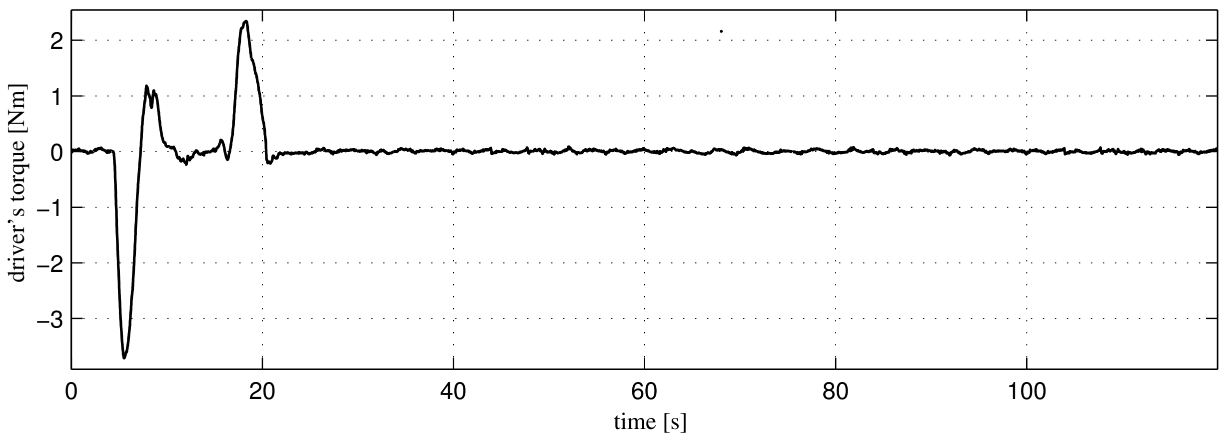

Fig. 6. Simulation results: driver's torque τ on the steer.

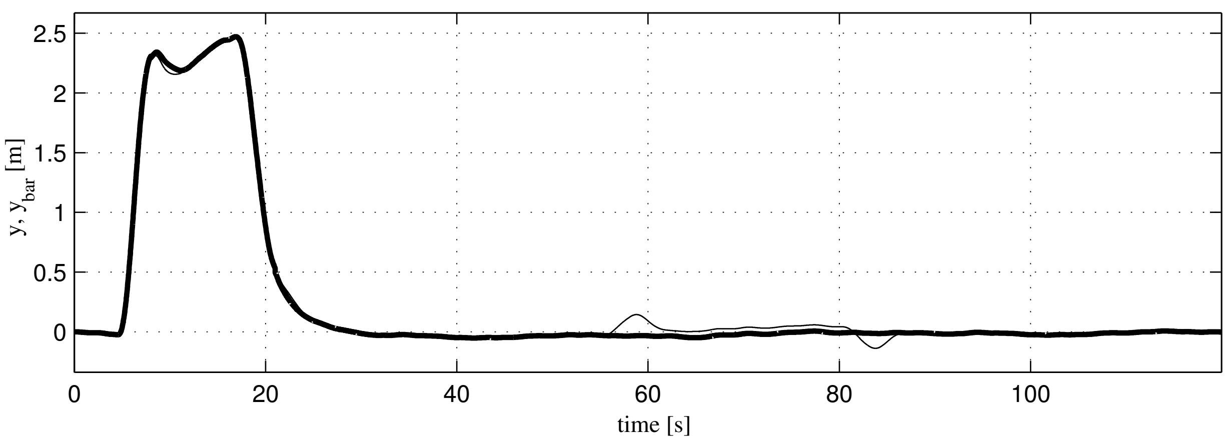

Fig. 7. Simulation results: $\overline{y}$ (thick) and y (thin).

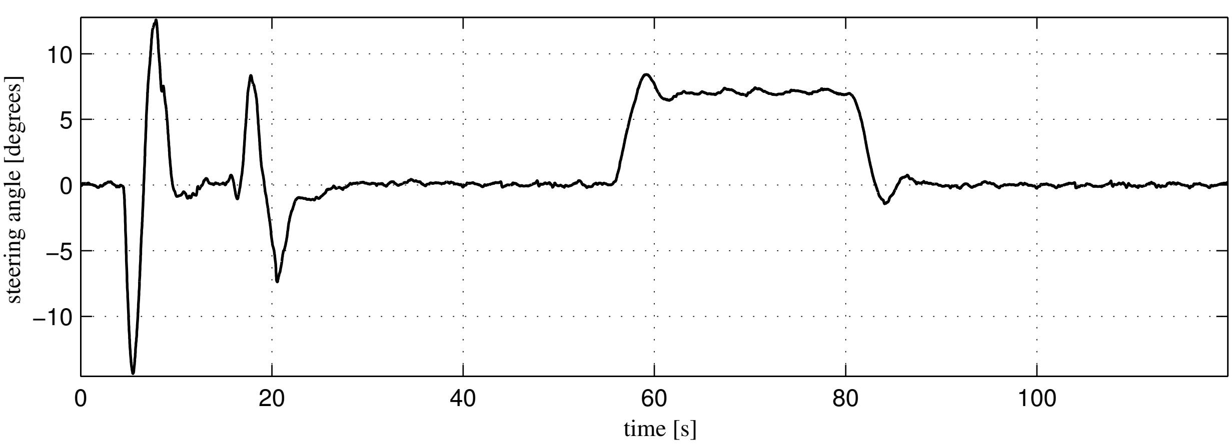

Fig. 8. Simulation results: steering angle.

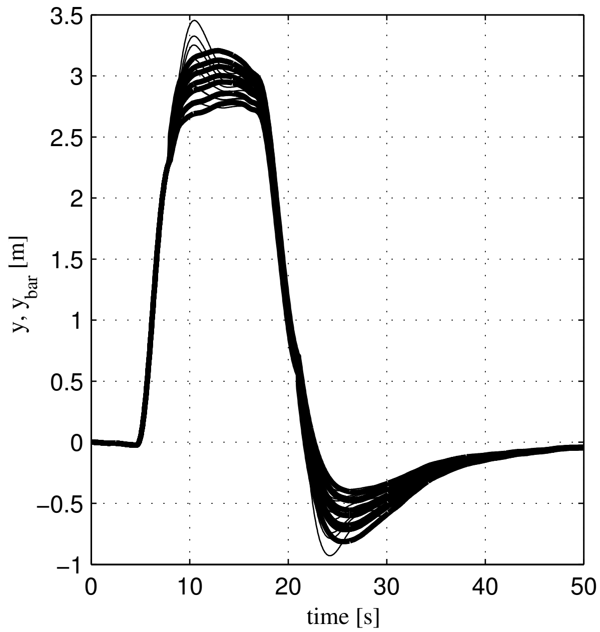

Fig. 9. Simulation results: $\overline{y}$ (thick) and y (thin) for some values of $\gamma \in \Gamma$ and $v_x = 60$ km/h.

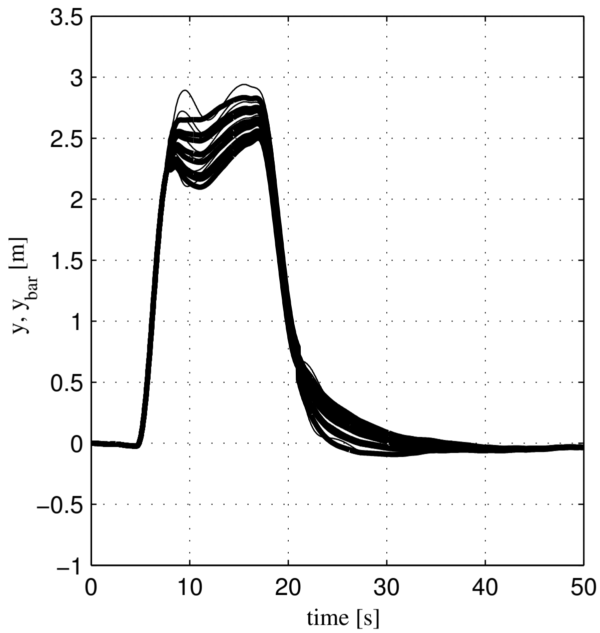

Fig. 10. Simulation results: $\overline{y}$ (thick) and y (thin) for some values of $\gamma \in \Gamma$ and $v_x = 90$ km/h.

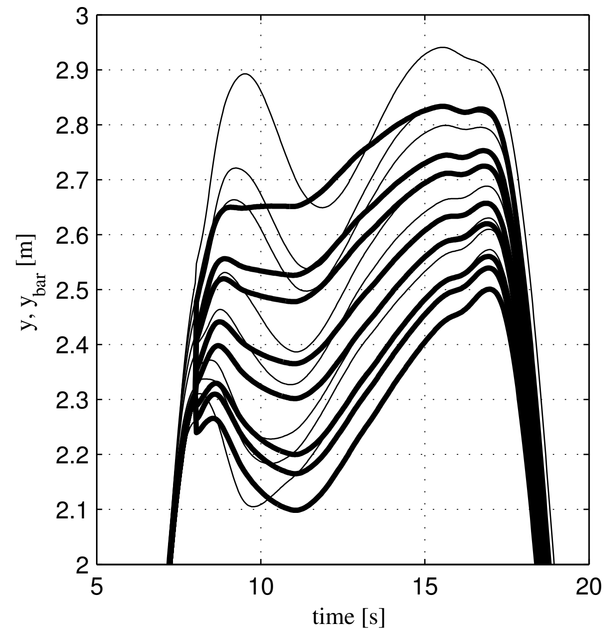

Fig. 11. Simulation results: $\overline{y}$ (thick) and y (thin) for some values of $\gamma \in \Gamma$ and $v_x = 90$ km/h.

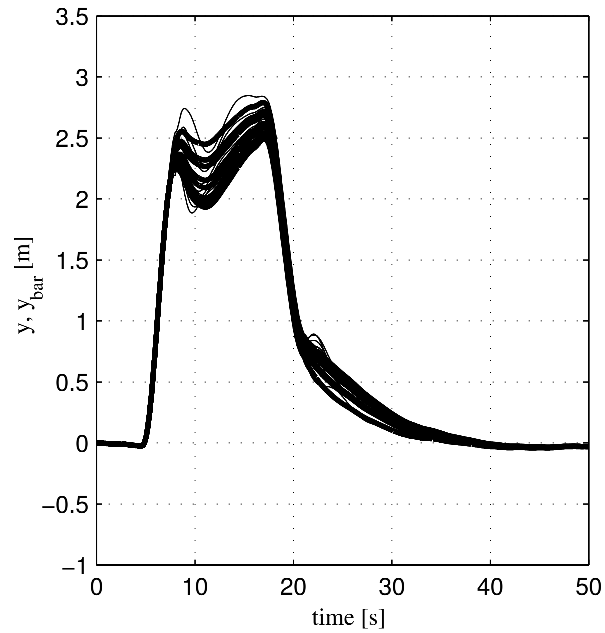

Fig. 12. Simulation results: $\overline{y}$ (thick) and y (thin) for some values of $\gamma \in \Gamma$ and $v_x = 120$ km/h.