Robust Automatic Steering with Cautious Parameter Adaptation

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Abstract

This paper presents an investigation of the robust predictive control with suitable parameter adaptation for the design of an automatic steering system. The involved vehicle is a full-scale nonlinear model of a BMW520i and the feedback is only on the lateral displacement unlike in the available studies where an additional feedback on the yaw rate is commonly used to improve the performance. The underlying control design is performed using a sensitivity function shaping oriented procedure that heavily borrows from the robust control culture. Intensive simulation studies are carried out to emphasize the performance of the proposed automatic steering system.
Keywords: Automotive control, automatic steering, adaptive control, predictive control design, robustness.

1 Introduction

Advanced control of automotive vehicle systems is more and more discussed in the open literature and widely developed in industry (Ackermann *et al.* 1995, Peng and Tomizuka 1993). International Workshops (Kiencke 1995) and Journal special issues have been dedicated to automotive control (Fenton 1991, Rizzoni 1995). It has been pointed out that automatic steering is of fundamental interest for urban transport vehicles and automated highway traffics of the next century (Shladover 1993).

The main purpose of automatic steering consists in performing a robust tracking of a road curvature in spite of the uncertain operating conditions due to large variations in velocity, contact characteristics between tyres and road surface and vehicle mass. The underlying control systems usually involve feedbacks of both the lateral displacement and the yaw rate and use the front steering angle as a manipulated variable. The reference may consist of permanent magnets in the road or the magnetic field of an electrically supplied wire. The involved tracking error is measured by a displacement sensor mounted in the center of the front end of the vehicle while the yaw rate measurement is performed by a gyro.

The involved control problem represents a challenging opportunity to investigate advanced control techniques that have reached a reasonable level of maturity. This is mainly a result of many years of effort devoted to the understanding of the control theory as well as the tremendous progress in the computer technology that makes the implementation of control systems simpler and cheaper.

In the present paper, the authors aim at investigating the applicability of an adaptive robust predictive control for automatic steering using only lateral displacement measurement unlike in earlier design studies (Ackermann *et al.* 1995). A realistic simulation framework involving a full-scale nonlinear model of a BMW520i vehicle is used to this end. This simulator has been developed at the Institute for Industrial Information Techniques of the Technical University Karlsruhe to investigate the automotive control problems (Majjad and Kiencke 1996). More specifically,

- The underlying control design is carried out using an appropriate predictive control approach that has been developed from the generalized predictive control (Clarke *et al.* 1987) in the spirit of the robust control theory (Limbeer and Green 1995, Zhou *et al.* 1996).
- The control model is obtained from appropriate closed-loop identification experiments and updated using a cautious parameter adaptation algorithm incorporating all those robustness features that have been emphasized and addressed in the adaptive control theory (Ioannou and Sun 1996, M'Saad *et al.* 1993, Ortega and Tang 1989).
- Besides the standard design specifications, namely maintaining the lateral displacement as small as possible for typical manoeuvres taking into account the actuator constraints and the passenger comfort, suitable shapings of the usual sensitivity functions have to be achieved. This allows to deal with those ubiquitous variations in velocity, road adhesion factor

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and vehicle mass. The usual sensitivity functions are used as suitable quantifiers for both nominal performance and stability robustness (Limbeer and Green 1995, Zhou *et al.* 1996). This requires a good know-how in system identification as the control system and its usual sensitivity functions are determined from control models that should be identified over the domain of possible operating conditions (Ljung 1987, Gevers 1993).

The involved experimental evaluation is carried out using the advanced control software package SIMART (M'Saad 1994) which allows to perform all the steps involved in any genuine control system design, namely the performance specification, the plant control model identification and its validation, the control design, the stability and performance robustness analysis and the appropriate control design. A suitable parameter adaptation capability can be incorporated into the control algorithm when needed, thanks to available results concerning the robust adaptive control theory. Last but not least, the tracking performance can be improved using the partial state reference model control concept (M'Saad and Sanchez 1992).

2 Modelling of Vehicle Dynamics

A realistic simulator of the vehicle dynamics has been developed at the Institute for Industrial Information Techniques of the Technical University Karlsruhe to investigate the automotive control problems (Majjad and Kiencke 1996, Kiencke 1993). This consists of several sub-models of the chassis, the traction/braking, the suspension and the tyres. The global inputs for the involved simulator are the front steering angle, the traction or braking torques for each wheel, the initial velocity of the vehicle, the weather and road conditions. All those important nonlinearities have been taken into account. The data have been adapted to a BMW520i and proved to demonstrate a realistic behaviour during various driving conditions.

In the following, the components of each sub-model will be briefly described.

2.1 Chassis model

The chassis is described by a six degree of freedom model. The balance of forces in the three directions is given by

$$m\begin{bmatrix}\dot{v}_x\\ \dot{v}_y\\ \dot{v}_z\end{bmatrix}_S = \mathbf{Tus}\begin{bmatrix}[F_{xf1}+F_{xf2}+F_{xr1}+F_{xr2}+F_{wx}+F_{gx}+F_r]\\ [F_{yf1}+F_{yf2}+F_{yr1}+F_{yr2}+F_{wy}+F_{gy}]\\ [F_{zf1}+F_{zf2}+F_{zr1}+F_{zr2}+F_{wz}+F_{gz}]\end{bmatrix}_U$$

where

- m is the total mass of the vehicle
- $\mathbf{v}^T = [v_x \; v_y \; v_z]$ represents the velocity in the inertial system S,
- $\mathbf{F_{f1}}^T = [F_{xf1} \; F_{yf1} \; F_{zf1}]$ denotes the force acting on the front-left tyre.
- $\mathbf{F_{f2}}^T = [F_{xf2} \; F_{yf2} \; F_{zf2}]$ denotes the force acting on the front-right tyre.
- $\mathbf{F_{r1}}^T = [F_{xr1} \; F_{yr1} \; F_{zr1}]$ denotes the force acting on the rear-left tyre.
- $\mathbf{F_{r2}}^T = [F_{xr2} \; F_{yr2} \; F_{zr2}]$ denotes the force acting on the rear-right tyre.
- $\mathbf{F_w}^T = [F_{wx} \; F_{wy} \; F_{wz}]$ represents the aerodynamic force due to wind disturbances
- $\mathbf{F_g}^T = [F_{gx} \; F_{gy} \; F_{gz}]$ represents the gravity force.
- $\mathbf{F_r}^T = [F_r \; 0 \; 0]$ represents the rolling force.
- $\mathbf{Tus}$ is the transformation matrix from the coordinate system of the vehicle U into the inertial system S.

More specifically, a complex nonlinear tyre model is used to generate the longitudinal and lateral forces. The normal forces are generated from the suspension model presented below. The rolling force is primarily caused by the hysteresis in the tyre materials due to the deflection of the carcass while rolling. This force is provided from an empirical formula.

The balance of moments around the vehicle axis is given by the following equation

$$\mathbf{J}\begin{bmatrix}\ddot{\xi}\\ \ddot{\phi}\\ \ddot{\psi}\end{bmatrix}_S = \begin{bmatrix}[(F_{zf1}-F_{zf2})\frac{b_f}{2} + (F_{zr1}-F_{zr2})\frac{b_r}{2} + m\dot{v}_y h]\\ [(F_{zf2}-F_{zf1})l_f + (F_{zr1}+F_{zr2})l_r - m\dot{v}_x h]\\ [(F_{yf1}+F_{yf2})l_f - (F_{yr1}+F_{yr2})l_r + (F_{xf2}-F_{xf1})\frac{b_f}{2}\\ +(F_{xr2}-F_{xr1})\frac{b_r}{2}]\end{bmatrix}$$

with

$$\mathbf{J} = \begin{bmatrix}J_x & 0 & 0\\ 0 & J_y & 0\\ 0 & 0 & J_z\end{bmatrix}$$

where $\ddot{\xi}$, $\ddot{\phi}$ and $\ddot{\psi}$ denote the second derivative of roll, pitch and yaw angles, respectively. J_x, J_y and J_z denote the moments of inertia of the vehicle in the inertial coordinate system, respectively. b_r, b_f, l_f and l_r represent the front track, the rear track, the distance from the gravity center to the front axle and the distance from the gravity center to the rear axle, respectively.

2.2 Traction/Braking model

The resulting balance of moments in the wheels is given by

$$J_r\begin{bmatrix}\dot{\omega}_{f1}\\ \dot{\omega}_{f2}\\ \dot{\omega}_{r1}\\ \dot{\omega}_{r2}\end{bmatrix} = \begin{bmatrix}M_{xf1}\\ M_{xf2}\\ M_{xr1}\\ M_{xr2}\end{bmatrix} - r\begin{bmatrix}F_{xf1}\\ F_{xf2}\\ F_{xr1}\\ F_{xr2}\end{bmatrix}_W$$

where r and J_r denote the radius dynamic of the wheel and moment of inertia of the wheel, respectively. The components of the vector $\omega(t) = [\omega_{f1}\ \omega_{f2}\ \omega_{r1}\ \omega_{r2}]^T$ represent the angular velocities of the wheels. The vector $\mathbf{u_{tb}}(t) = [M_{xf1}\ \ M_{xf2}\ \ M_{xr1}\ \ M_{xr2}]^T$ represent respectively the front-left, front-right, rear-left and rear-right applied traction or braking torques and W denotes the coordinate system of the wheel.

The above model generates the angular accelerations of the wheels as a function of the longitudinal tyre forces and the moment of braking or acceleration. The former can be obtained by a nonlinear tyre model while the latter is assumed to be known.

2.3 Suspension model

The considered suspension model consists in four independent two-degree-of-freedom quarter-vehicle models as shown in figure 1. Though such a model is relatively simple, its captures the most relevant dynamic features regarding the vertical dynamic behaviour.
m_{us} and m_s are the masses of the tyre mass and the

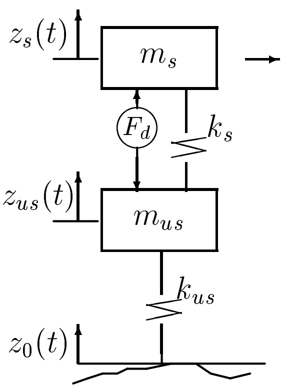

Figure 1: Two-degree-of-freedom quarter-vehicle model

quarter of the vehicle body mass, k_s and k_{us} denote respectively the stiffness of the suspension and tyre springs which are assumed to be linear. $F_d(t)$ is a non linear characteristic that models the damper.
The suspension model can be then given by the following state representation

$$\begin{bmatrix} \dot{x}_1(t) \\ \dot{x}_2(t) \\ \dot{x}_3(t) \\ \dot{x}_4(t) \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -\frac{k_{us}}{m_{us}} & 0 & \frac{k_s}{m_{us}} & 0 \\ 0 & -1 & 0 & 1 \\ 0 & 0 & -\frac{k_s}{m_s} & 0 \end{bmatrix} \begin{bmatrix} x_1(t) \\ x_2(t) \\ x_3(t) \\ x_4(t) \end{bmatrix} + \begin{bmatrix} -1 \\ \frac{c_{us}}{m_{us}} \\ 0 \\ 0 \end{bmatrix} w(t) + \begin{bmatrix} 0 \\ \frac{1}{m_{us}} \\ 0 \\ -\frac{1}{m_s} \end{bmatrix} F_d(t)$$

$$F_z(t) = k_s x_3(t) + F_d(t) + mg\frac{l_r}{l_f + l_r}$$

where the state variables are the tyre deflection $x_1(t) = z_{us}(t) - z_0(t)$, the unsprung mass velocity $x_2(t) = \dot{z}_{us}(t)$, the suspension stroke $x_3(t) = z_s(t) - z_{us}(t)$ and the sprung mass velocity $x_4(t) = \dot{z}_s(t)$, the damping force $F_d(t)$ is assumed to be characterized by a four-order polynomial relation with respect to the suspension stroke variations, $w(t)$ denotes the disturbance due to the ground profile, g represents the gravity constant and $F_z(t)$ is the normal force.

3 The predictive Control Approach

The predictive control design is based on the assumption that input-output behaviour of the plant to be controlled can be appropriately approximated by the following backward shift operator (q^{-1}) model

$$\begin{aligned} A(q^{-1})y(t) &= B(q^{-1})u(t-d-1) + v(t) \\ D(q^{-1})v(t) &= C(q^{-1})\gamma(t) \end{aligned}$$

with

$$\begin{aligned} A(q^{-1}) &= 1 + a_1 q^{-1} + \ldots + a_{na} q^{-na} \\ B(q^{-1}) &= b_o + b_1 q^{-1} + \ldots + b_{nb} q^{-nb} \\ C(q^{-1}) &= 1 + c_1 q^{-1} + \ldots + c_{nc} q^{-nc} \\ D(q^{-1}) &= 1 + d_1 q^{-1} + \ldots + d_{nd} q^{-nd} \end{aligned}$$

where $u(t)$ is the control variable, $y(t)$ is the measured plant output, d denotes the minimum plant model delay in sampling periods, $v(t)$ represents the external disturbances and $\{\gamma(t)\}$ is assumed to be a sequence of widely spread pulses of unknown magnitude or independent random variables with zero mean values and finite variances.

The predictive control objective consists in minimizing, in a receding horizon sense with respect to the vector input $U_f(t+ch-1) = [u_f(t)\ldots u_f(t+ch-1)]^T$, the following linear quadratic cost function

$$E\left\{\sum_{j=sh}^{ph} (y_f(t+j))^2 + \rho(u_f(t+j-sh))^2\right\} \qquad (1)$$

subject to

$$u_f(t+i) = 0 \quad \text{for} \quad ch \leq i < ph \qquad (2)$$

with

$$\begin{aligned} W_{yd}(q^{-1})y_f(t) &= W_{yn}(q^{-1})y(t) \\ W_{ud}(q^{-1})u_f(t) &= D(q^{-1})W_{un}(q^{-1})u(t) \end{aligned}$$

where $E\{\cdot\}$ denotes the mathematical expectation, sh, ph and ch are the starting, prediction and control horizons according to the long range predictive control culture, ρ is a positive scalar and $\mathcal{W}_u(z^{-1}) = \frac{D(z^{-1})W_{un}(z^{-1})}{W_{ud}(z^{-1})}$ and $\mathcal{W}_y(z^{-1}) = \frac{W_{yn}(z^{-1})}{W_{yd}(z^{-1})}$ denote user specified input and output frequency weightings, respectively. The frequency weightings are mainly motivated by stability and performance robustness considerations and are such that all polynomials $W_{xx}(q^{-1})$ are Hurwitz.

The underlying control problem will be handled using the generalized predictive control approach proposed in (Clarke *et al.* 1987) from the following plant reparametrization:

$$\bar{A}(q^{-1})y_f(t) = \bar{B}(q^{-1})u_f(t-d-1) + \bar{C}(q^{-1})\gamma(t) \quad (3)$$

with

$$\begin{aligned} \bar{A}(q^{-1}) &= A(q^{-1})D(q^{-1})W_{yd}(q^{-1})W_{un}(q^{-1}) \\ \bar{B}(q^{-1}) &= B(q^{-1})W_{yn}(q^{-1})W_{ud}(q^{-1}) \\ \bar{C}(q^{-1}) &= C(q^{-1})W_{yn}(q^{-1})W_{un}(q^{-1}) \end{aligned}$$

The resulting controller may be given the following linear form:

$$S(q^{-1})D(q^{-1})u(t) + R(q^{-1})y(t) = 0$$

with

$$\begin{aligned} S(q^{-1}) &= \bar{S}(q^{-1})W_{yd}(q^{-1})W_{un}(q^{-1}) \\ R(q^{-1}) &= \bar{R}(q^{-1})W_{yn}(q^{-1})W_{ud}(q^{-1}) \end{aligned}$$

where the polynomials $\bar{S}(q^{-1})$ and $\bar{R}(q^{-1})$ depend on the plant model as well as the design parameters.

The involved nominal control system may be represented as shown in figure 2 where $v_u(t)$ and $v_y(t)$ denote respectively the input and output disturbances, and $\nu_u(t)$ and $\nu_y(t)$ are the input and output noise measurements. The corresponding input-output behavior is described by

$$\begin{aligned} P_c(q^{-1})y(t) &= q^{-d-1}B(q^{-1})S(q^{-1})D(q^{-1})v_u(t) \\ &+ q^{-d-1}B(q^{-1})S(q^{-1})D(q^{-1})\nu_u(t) \\ &+ A(q^{-1})S(q^{-1})D(q^{-1})v_y(t) \\ &- q^{-d-1}B(q^{-1})R(q^{-1})\nu_y(t) \end{aligned}$$

$$\begin{aligned} P_c(q^{-1})u(t) &= A(q^{-1})S(q^{-1})D(q^{-1})v_u(t) \\ &- q^{-d-1}B(q^{-1})R(q^{-1})\nu_u(t) \\ &- A(q^{-1})R(q^{-1})[v_y(t) + \nu_y(t)] \end{aligned}$$

where $P_c(q^{-1})$ is the characteristic polynomial which can be factored as follows

$$P_c(q^{-1}) = P_f(q^{-1})P_o(,q^{-1}) \quad (4)$$

where $P_f(q^{-1})$ and $P_0(q^{-1})$ denote the characteristic polynomial of the underlying receding horizon linear quadratic control and state predictor, respectively. Notice that the predictor dynamics could be chosen bearing in mind the optimal estimation theory, namely $P_o(q^{-1}) = C(q^{-1})$.

Of particular importance, the control system is asymptotically stable if and only if the characteristic polynomial is Hurwitz.

$$P_c(q^{-1}) = 0 \quad \Rightarrow \quad |q| < 1 \quad (5)$$

The control system performance may be represented as shown in figure 3 and figure 4, where $\mathcal{P}(z^{-1})$ and

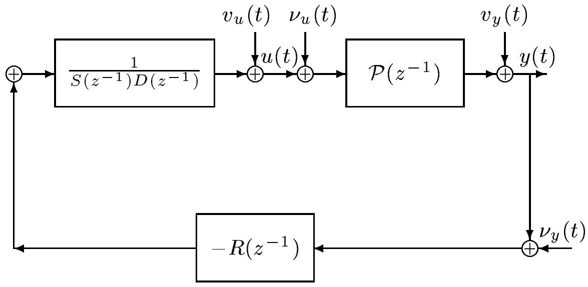

Figure 2: Nominal control system

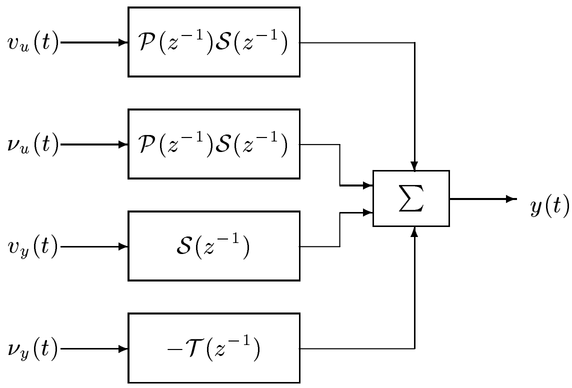

Figure 3: Nominal output performance

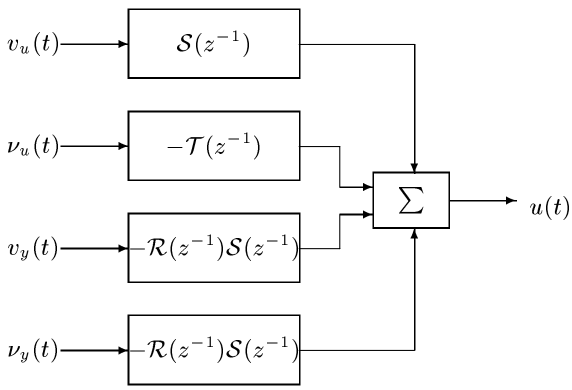

Figure 4: Nominal input performance

$\mathcal{R}(z^{-1})$ denote the control design model and its underlying regulator respectively, and $\mathcal{S}(z^{-1})$ and $\mathcal{T}(z^{-1})$ are respectively the sensitivity and complementary sensitivity functions, i.e.

$$\mathcal{S}(z^{-1}) = \frac{A(z^{-1})S(z^{-1})D(z^{-1})}{P_c(q^{-1})}$$

and

$$\mathcal{T}(z^{-1}) = \frac{z^{-d-1}B(z^{-1})R(z^{-1})}{P_c(q^{-1})}$$

The nominal performances of the control system as well as the stability robustness can be evaluated from its usual sensitivity functions relating the exogenous input to the plant input and output, respectively (M'Saad *et al.* 1996). The shapes of the sensitivity functions may be refined by properly specifying the involved design parameters. To do so, an iterative procedure is needed and hence a useful CACSD software package.

3.1 The Adaptive Controller

A remarkable research activity has been devoted to the question of designing adaptive controllers that would perform well in the presence of state disturbances, plant model parameter variations and unmodelled dynamics (Ioannou and Sun 1996, M'Saad *et al.* 1993, Ortega and Tang 1989). The key issues to get a robust adaptive controller are suitable signal processing and a robust parameter adaptation algorithm. The former can be simply performed from the plant model by filtering and normalizing the data, while the latter has to incorporate an adequate parameter adaptation alertness and freezing. These robustness features can be achieved by the regularized constant trace algorithm with conditional parameter adaptation proposed in (M'Saad *et al.* 1990). More specifically, the parameter adaptation is frozen whenever the available information is not likely to improve the parameter estimation process.

The adaptive control law is obtained by simply invoking the certainty equivalence principle. This consists in replacing the plant model parameters θ by their admissible estimates $\theta_a(t)$ when deriving the predictive control law. The admissible estimated model is obtained from the involved parameter adaptation algorithm bearing in mind the usual stability properties of the adaptive control systems, (M'Saad *et al.* 1993). More specifically, the adaptive control system is implemented as follows:

1. Wait for the clock pulse and sample the plant output.
2. Update the plant model parameters using the considered parameter estimator.
3. Construct the admissible estimated plant model as follows:

$$\theta_a(t) = \begin{cases} \hat{\theta}(t) & \text{if } \hat{\theta}(t) \text{ is stabilizable} \\ \theta_a(t-1) & \text{otherwise} \end{cases}$$

   where $\hat{\theta}(t)$ is the estimated plant model provided by the considered parameter adaptation algorithm.
4. Evaluate the adaptive control law using the admissible estimated plant model $\theta_a(t)$.
5. Implement the control signal and go to 1.

4 Experimental Evaluation

This section is devoted to the investigation of the applicability of the proposed predictive control approach in a realistic simulation framework involving a full-scale nonlinear model of a BMW520i vehicle (Majjad and Kiencke 1996). Recall that the main purpose of automatic steering is to track a given road curvature. The error between the guideline and the vehicle $y_s(t)$ is commonly measured by a displacement sensor mounted in the front end of the vehicle, and the control variable is the front steering angle $\delta(t)$.

The control objective consists in maintaining the lateral displacement as small as possible in spite of variations in road curvature (M'Saad *et al.* 1996). Of particular importance, the controller should perform well in the presence of variations in mass, velocity and contact between tyre and road surface.

The design specifications have been adapted from those considered in earlier design studies (Ackermann *et al.* 1995, Guldner *et al.* 1995, Smith *et al.* 1978), they are primarily given in terms of actuator constraints and passenger comfort as follows:

- The steering angle and its rate should satisfy $|\delta(t)| \leq$ 40 deg and $|\dot{\delta}(t)| \leq 45$ deg/s.
- The displacement from the guideline must not exceed 15 cm in transient behaviour and 2 cm in steady state behaviour.
- The lateral acceleration should satisfy

$$|a_y(t)| \leq \frac{v(t)^2}{\rho(t)} \pm 0.1g$$

  where $v(t)$, $\rho(t)$ and g denote the velocity, the curvature radius and the gravity constant, respectively.

The design parameters involved in the predictive control objective have been specified to meet the above design specifications while ensuring appropriate shaping of the usual sensitivity functions as shown in (Mueller *et al.* 1996).

The control model parameters have been initialized by those of the nominal plant model used to design the underlying robust predictive controller and updated using robust parameter adaptation algorithm incorporating an appropriate signal filtering and normalization, the available prior knowledge, an adequate adaptation gain regularization and a cautious adaptation freezing. All these features have been widely discussed in (M'Saad and Hejda 1994).

The performance of the automatic steering system with suitable parameter adaptation has been demonstrated through two simulation experiments. More specifically,

- In the first simulation experiment, the involving road curvature consists in four successive curves and straight lines where the corresponding curvature radius is 400 m and the vehicle accelerates from 5 m/s to 15 m/s.

Figure 5 shows the road curvature, the vehicle speed, the lateral displacement and the steering angle during the first simulation. Notice that the considered cautious parameter adaptation has actually allowed to improve the performance of the automatic steering system. More specifically, the parameter adaptation has enhanced the performance of the robust controller which has been used to initialize the automatic steering system as clearly shown along the distance intervals $[0, 50m)$ and $[50m, 110m)$. Furthermore, the parameter adaptation has allowed to accomodate the considered speed variations as clearly shown along the distance intervals $[110m, 175m)$ and $[175m, 250m]$.

- In the second simulation experiment, the involving road curvature consists also in four successive curves and straight lines where the corresponding curvature radius is 400 m. The vehicle slows down from 15 m/s to 5 m/s.

The road curvature, the vehicle speed, the lateral displacement and the steering angle during the second simulation are shown in figure 6. Once again the cautious parameter adaptation has allowed to improve the performance of the automatic steering system. More specifically, the performance of the robust controller has been enhanced as clearly shown along the distance intervals $[0, 50m)$ and $[50m, 110m)$. Furthermore, the speed deceleration has been accomodated as clearly shown along the distance intervals $[115m, 190m)$ and $[190m, 250m]$.

It is worth mentionning that the input-output signals were not sufficiently exciting on straight roads, the parameter adaptation was automatically frozen as the incoming information was not likely to improve neither the accurancy of the control model nor its interaction with the underlying control design.

5 Conclusion

The main motivation of this paper was to investigate the applicability of a suitable adaptive robust predictive control approach for the design of an automatic steering system. The standard safety and passenger comfort specifications have been achieved. Three fundamental design features of the proposed control approach are worth to be pointed out, namely offset-free performance, adaptation alertness, stability robustness and implementation simplicity.

A great attention has been paid to the identification of the control design model and its parameter adaptation. A comprehensive iterative procedure has been developed to specify the design parameters of the adaptive predictive controller. The CACSD software package SIMART has been revealed be a powerfull tool to derive such a procedure.

Further research is required to develop an advanced control approach for the design of automatic steering control systems.

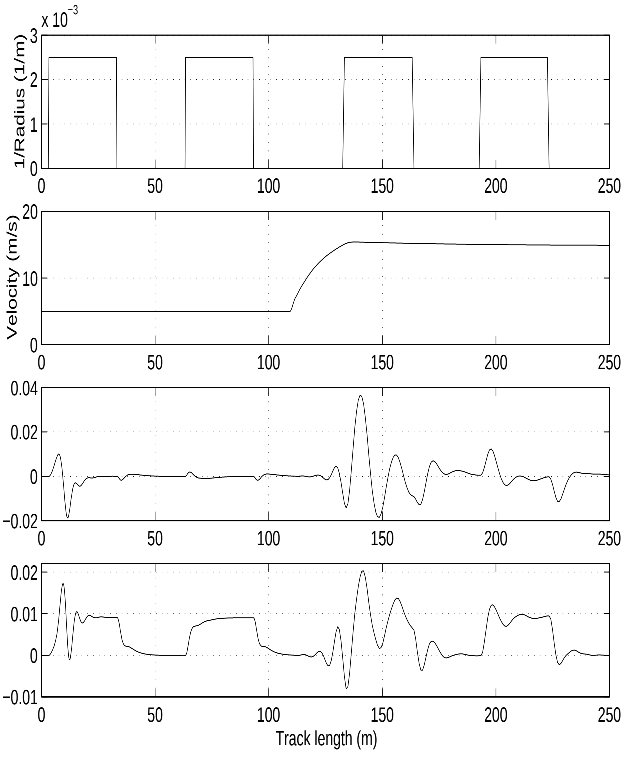

Figure 5: Road curvature, vehicle speed, lateral displacement and steering angle during the first simulation

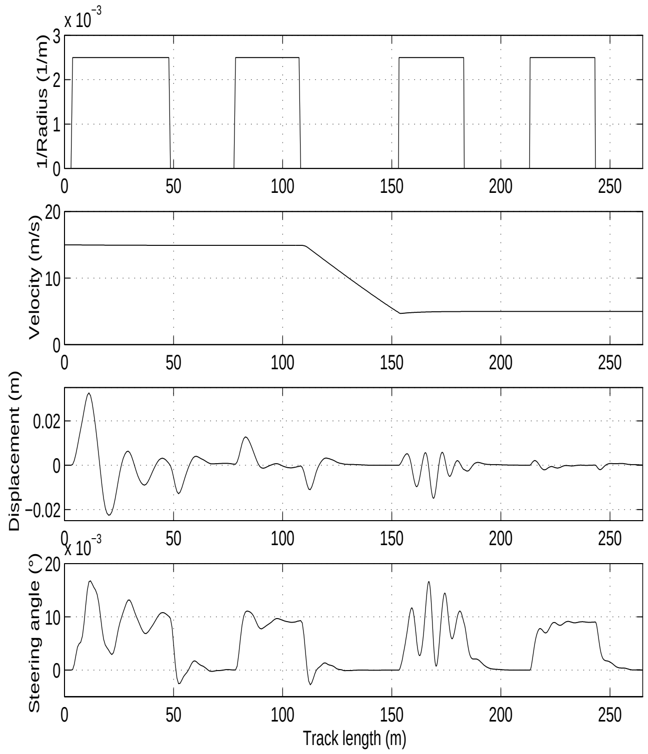

Figure 6: Road curvature, vehicle speed, lateral displacement and steering angle during the second simulation

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