Feedforward and Feedback Control for Driving Assistance and Vehicle Handling Improvement by Active Steering

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Abstract

This paper discusses some aspects of lateral driver assistance using active steering. Vehicle handling improvement and lane keeping assistance are both addressed. A formal control concept is developed for both EAS and steer-by-wire steering mechanism. Finally an example is presented with application for vehicle handling improvement.

1 Introduction

Vehicle handling improvement and lateral driving assistance have received much attention from the research community and car manufacturers during the last decade. One can refer to [1] and the references therein. Active steering has attractive benefits with regard to vehicle handling improvement. For example, front wheels steering leads to additional lateral forces which can be used in order to reject yaw and roll torque disturbances that rise from μ split, asymmetric braking or wind forces and this even on decreased road adhesion conditions. Active steering has also application in rollover avoidance for vehicles with elevated center of gravity.

However, some architecture and control aspects are still confusing. In fact, in previous works on vehicles with classical steering mechanism, the driver action is not always taken into account or the control sharing is generally restricted to a simple variable gain between the control output and the driver's action. On the other hand, the emerging steer-by-wire systems generate other control problems such as passivity and transparency of the steering system including the driver, the steering mechanism, the controller and the environment. These problems are well known in teleoperation community [4].

In this paper, we first discuss some architecture aspects of active steering for both vehicle handling and lane keeping assistance. A combination of feedforward and feedback control procedure is then presented. This procedure is able to take into account in a single framework several control setup and performance specifications.

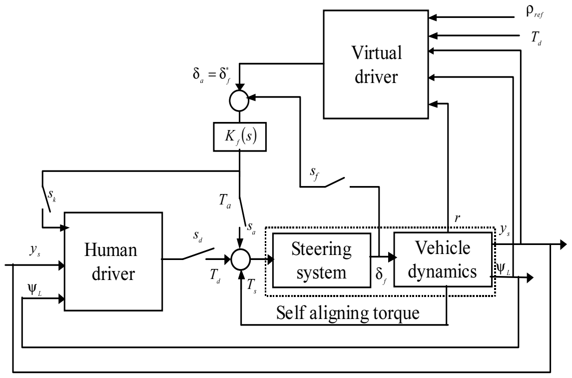

Figure 1: General control setup with the virtual driver concept

The paper is organized as follows: section 2 introduces the concept of virtual driver and the possible active steering schemes and design objectives. Section 3 gives some remarks on human driver models while section 4 is dedicated to the steering mechanisms modeling including classical and steer by wire systems. In section 5, we propose some formal way for control design using the concept of two degrees of freedom control.

2 Virtual driver concept

Concept of virtual driver has been used in several works and is well suitable for systems where the human interacts with the controlled system [2]. A virtual driver is a controller whose inputs are one or several sensor data including human actions perception. It outputs a tire steering angle or a steering torque. In Figure 1, the virtual driver output is primary a steering angle, this angle is achieved by an internal tracking loop. A virtual driver may be attached to any vehicle, however when manual steering is considered, the virtual driver output is not used. In driver assistance configuration, the input to vehicle dynamics is a combination of the human and virtual driver actions. The virtual driver generally acts as a feedback controller but may also use feedforward information such as driver input or sensed perturbations such as road curvature. The concept of virtual driver can be used for different control architecture detailed below.

- Using internal sensors measuring vehicle speed, yaw rate and steering angle or torque, one can improve vehicle handling. The controller adds steering angle or torque of limited value during the driver reaction time in order to achieve steady state rejection of step input disturbances. As for example, these disturbances are side wind force or yaw torque disturbance. In addition, it may enhance vehicle responses damping at all operational domain.
- If road sensors are added, one can consider more general driver assistance such as lane departure avoidance. In this case the road has to be sensed so that the controller can assist the driver during lane keeping maneuver and when a lane departure is expected. In this case an anticipation time is needed such that the controller has sufficient time to take a recovering maneuver. When lane keeping maneuver assistance is considered the virtual driver primary objective may take a more general form than lateral error cancellation. Some kind of admissible trajectory tracking can be included.
- Automatic steering is the ultimate form of the previous case.

	Virtual driver objectives
Manual driving	no
Vehicle handling	- External forces rejection - External torques rejection - Better yaw damping - Driver friendly interaction
Lane keeping assistance	- Maneuver enhancement - Lane departure prevention - Driver friendly interaction
Automatic steering	- Attenuation of displacement at a look-ahead distance - Relative yaw angle

Table 1. Virtual driver objectives

	V. D.	Road	D. I.
Manual driving	no	no	no
Vehicle handling	no	yes	yes
Lane keeping assist.	yes	yes	yes
Automatic steering	not necessary	yes	no

Table 2. Virtual driver needed data [1]

[1] V. D. : Vehicle dynamics, D. I. : Driver input

Tables 1 and 2 summarize the objective of the virtual driver for different steering configurations and the needed information. In all cases, driver comfort and security preservation are primary objectives. It has to be noted that driver comfort must include some friendly interaction with the virtual driver.

3 Human driver modeling

Modeling of human driver is a difficult task, however, several components can be identified [8]. The first one is called structural model. It is constituted by a time delay representing inherent human processing time and neuromotor dynamics. This component represents the high frequency driver compensation component and is generally modeled by a dead time τ_p and a second order low pass filter with damping factor ξ_n and natural frequency ω_n. Typical values for the parameters are $(0.15s \leq \tau_p \leq 0.2s)$, $\xi_n = 0.707$, $\omega_n = 10 rad.s^{-1}$. Some feedback elements may be added to the neuromotor system in order to reproduce some proprioceptive feedback from motion of human limbs and muscle tissue.

The second component corresponds to the driver lead and predictive actions. It is generally modeled by first order lead filter, where the time constant is representative of the driver mental load.

The third component is generally a simple gain representing the proportional action of the driver face to the perceived vehicle positioning relative to the driving environment. It must be noted that all these parameters are not constant and are only valid for a restricted vehicle configuration.

This model is fed by the driver perception of the driving environment which is generally reduced here to lateral displacement or angular error at some lookahead distance.

Finally, the driver model presents a third input represented by the kinesthesic torque applied by the virtual driver which is sensed by a proprioceptor in muscle tissue and processed reflectively in the spinal cord level. When all these components are embedded, driver may be represented by a generalized impedance with speed (or position input) and a torque as output. An admittance representation is also possible.

As proposed in [3], the driver performance task may be evaluated by the energy of the lateral displacement and the physical workload by the energy of the driver applied torque.

4 Steering system equations

Vehicle steering systems may be classified according to the presence or not of a mechanical linkage between the steering wheel and the tire steering mechanism. Steering systems with this linkage will be called conventional and the ones without, are called steer by wire. However conventional hydraulic power steering and now increasingly replaced by electrohydraulic or electromechanical servo actuators. These systems are called Electronic Active Steering (EAS) they permit the addition of a correcting steering angle to the one intended by the driver through the steering wheel.

4.1 Conventional steering systems

4.1.1 General equations: Considering a general steering system setup, differential equation is given by

$$I_s\ddot{\delta}_d = -T_s - B_s\dot{\delta}_d + T_a + T_d \qquad (1)$$

The self aligning torque T_s is obtained from the king pin torque using the steering gear ratio R_s and the power steering assist coefficient K_p. The king pin torque is function of the tire lateral forces and trail length.

Conventional vehicle steering system may be viewed as a dual port. In fact, when the driver inputs a steering angle, vehicle dynamics are used in order to determine the associated steering torque. On the other hand, when the driver inputs steering torque, a steering angle is obtained. Several particular setup are now considered.

4.1.2 Manual steering: In this case, there is no actuator torque $(T_a = 0)$, the self aligning torque is fully transmitted to the steering wheel. During steady state cornering $\left(\ddot{\delta}_d = \dot{\delta}_d = 0\right)$, the driver has to contreact exactly the self aligning torque $(T_d = T_s)$.

4.1.3 Power steering assist: The driver steering torque is sensed and the motor torque is computed such that the driver steering torque does not excess a fixed bound of generally $4N.m$. The hydraulic actuator or electrical motor torque is generally function of the forward speed $T_a = f(T_d, v)$.

4.1.4 Lateral Driver assistance: This case differs from the previous one from the fact that other vehicle measurements are used in order to enhance vehicle handling or lane keeping maneuvers. The controller changes the tire steering angles and a torque feedback is sensed by the driver.

4.1.5 Automatic steering: Full automatic steering may be considered with conventional steering systems. During normal operation, all driver input commands have to be ignored, so a gear system can be used in order to mechanically detach the steering wheel from the steering mechanism. In this case, $T_d = 0$ and δ_d is a free value independent from tire steering angle, and equation 1 has to be written in the tire coordinates

$$I'_s\ddot{\delta}_f = -R_sT_s - B_s\frac{1}{R_s}\dot{\delta}_f + K_tR_aT_a \qquad (2)$$

where I'_s is the steering mechanism inertia without the steering wheel. It is also possible to consider the case where the steering wheel is still acted during automatic steering for safety reason for example.

All the considered conventional steering systems characteristics are summarized on Table 3 [2].

	Swithes	Torques	Angles
M. S.	$s_d = 1$ $s_a = 0$ $s_k = 0$ $s_f = 0$	$T_a = 0$ $T_d = T_s$	$\delta_f = R_s^{-1}\delta_s$
P. S. A.	$s_d = 1$ $s_a = 0$ $s_k = 0$ $s_f = 0$	$T_d \leq 4N.m$ $T_a = f(T_d, v)$	$\delta_f = R_s^{-1}\delta_s$
L. D. A.	$s_d = 1$ $s_a = 1$ $s_k = 1$ $s_f = 1$	T_a obtained by feedback and feedforward	$\delta_f = R_s^{-1}\delta_s$
A. S.	$s_d = 0$ $s_a = 1$ $s_k = 0$ $s_f = 1$	$T_d = 0$	$\delta_d = R_s\delta_f$ or $\delta_d = 0$

Table 3. Possible combinations with the steering torque input in conventional steering system[3]

4.2 Steer by wire system

In steer by wire systems there is no direct connection between the steering wheel and the tire steering mechanism. The tire steering angle or steering torque is computed from the sensed driver steering angle and other sensor data according to the considered application. A wheel steering torque is feedback to the driver as a function of the self aligning torque, vehicle environment or other design objectives. It is obvious that steer by wire system offers the supplementary degree-of-freedom which makes possible realtime switching between manual, assistance and automatic steering but also the inclusion of some stability and performance objectives specifications. A possible simple interpretation of the triplet (driver, virtual driver, tire-road contact) may lead to the two following variable gain implementation of the virtual driver which are obtained

[2] Actual steering systems are longitudinal speed depended. This dependence is not explicitly put in the previous table.

[3] M. S. : Manual Steering, P. S. A. : Power Steering Assistance, L. D. S. : Lateral Driving Assistance and A. S. : Automatic Steering

by weighting the desired steering angle by that of the driver and that by the virtual driver

- If the virtual driver outputs a steering angle

$$\left\{ \begin{array}{l} \delta_f^* = \gamma \frac{1}{R_s} \delta_s + (1-\gamma)\,\delta_a \\ T_d^* = \gamma T_s + (1-\gamma)\,K_d T_a \end{array} \right. \quad (3)$$

  where δ_a is the angle computed by the virtual driver and δ_f^* is the effective desired tire steering angle and T_d^* is the desired torque that the driver needs to compensate (torque fed to the driver). It must be noted that when $\gamma = 0$, automatic steering system is obtained and when $\gamma = 1$, the system becomes a manual driving. Parameter K_d is actuator to driver torque converter

- If the virtual driver outputs a steering torque, the previous equations become

$$\left\{ \begin{array}{l} T_a^* = \gamma K \left(\frac{\delta_s}{R_s} - \delta_f \right) + (1-\gamma)\,T_a \\ T_d^* = \gamma T_s + (1-\gamma)\,K_d T_a^* \end{array} \right. \quad (4)$$

  However a more general control setup of steer by wire systems may be considered. In fact, as a replacement of the actual mechanically linked steering systems, a controller (virtual driver) can be designed. It would receive, from the driver, the input torque and the steering wheel position or speed, and, from the tires, the self aligning torque and the wheel steering angle or rate. It would also output two steering torques : one for the steering wheel and one for the tires. In this case, stability and transparency are the primary objectives for the design of this 4 inputs 2 outputs controller.

In the following, we consider in details some control aspects including a robust reference model tracking using two degree of freedom control concept in order to improve vehicle handling. The procedure is thus only based on proprioceptive sensors.

5 Active steering design example

5.1 Problem setup

In the following, it is considered that in the vehicle active steering configuration, the tire steering angle ($u = \delta_f$) is set in part by the driver, as in the equation

$$u = u_d + \delta_c \quad (5)$$

where $u_d = \frac{\delta_d}{R_s}$. It has to be noted that the gear ratio can be constant but can also be made as a function of the longitudinal speed. The tire steering angle form of equation 5 suggests that the controller add or substract a steering angle to the one set by the controller δ_c.

In the results presented below, only the unilateral effect of the active steering on the vehicle dynamics is

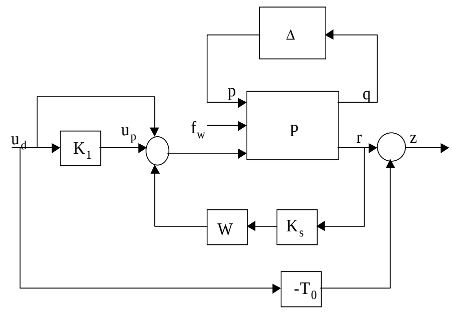

Figure 2: Feedforward control with reference model

examined. The effect on the driver is not taken into account.

Here, the controller is used to combine a feedforward K_1 and a feedback part K_2, its output takes the following form

$$\delta_c = K_1 u_d + K_2 r \quad (6)$$

Controller K_1 acts as a prefilter of the reference signal by adding the feedforward action ($u_p = K_1 u_d$) while the feedback controller K_2 ensures robust stability and damping between the reference signal δ_d and the output r (figure 2).

Suppose now that a two stage approach is adopted so the controller K_2 is first designed. Control objectives concerning the synthesis of K_2 may be robust stability requirement, damping response enhancement and disturbance rejection, such as, wind forces or road curvature during lane keeping. All these requirements have to be insensitive (robust) face to vehicle parameters variations. Afterwards, when one close the loop with the synthesized controller, as the disturbances are assumed to be rejected, the only remaining system input is driver steering angle. The feedforward part K_1 can now be synthesized for different objectives in mind

- For example, one can minimize the side slip angle of the vehicle when driver inputs a steering angle
- Another interesting aspect when vehicle handling is considered is to use the concept of robust model matching defined in [5]. If T_0 is the desired transfer function between u_d and r, the open loop part K_1 of the controller has to keep small the error between the actual vehicle yaw rate and the model response for an entire family of perturbed plants. The error signal z is computed from $(z = r - T_0 u_d)$.
- The same concept may be used for lane keeping support or lane change support.

Following, we give an example of the developed architecture for vehicle handling improvement with robust

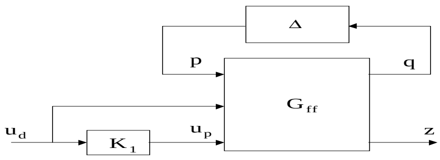

Figure 3: Feedforward control

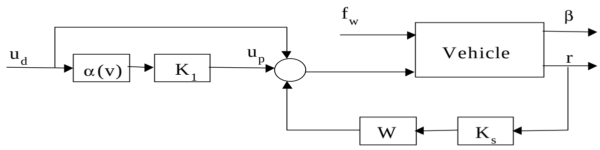

Figure 4: Active steering controller implementation

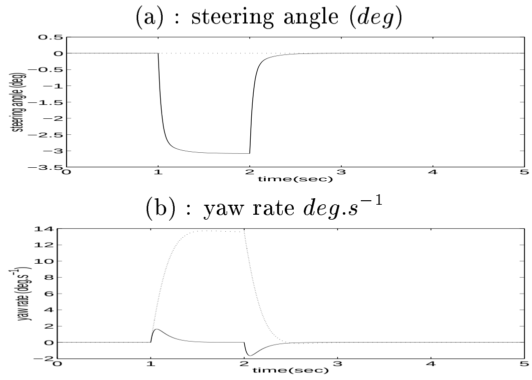

Figure 5: Wind forces step input rejection (solid : controlled, dotted : conventional)

reference model matching [6]. The feedback part K_2 of the controller is computed using the loop shaping coprime synthesis of [7].

5.2 Feedback part synthesis

Let $G(s)$ be the nominal transfer function from the steering angle to the yaw rate. In order to reject a constant step input perturbation on the yaw rate, a weighting compensator of the form of a high DC gain lag filter is introduced according to the loop shaping design methodology [7] [6]. Let now G_s be the shaped plant ($G_s = WG$), using normalized left coprime description of G_s, the stabilizing controller K_s is directly computed using the procedure in [7]. The final feedback controller is thus $K_2 = WK_s$.

5.3 Feedforward part synthesis

Since the wind force is rejected by the closed loop part of the controller, it will not be further considered for the feedforward part synthesis. The loop is closed by controller K_s and system input reduces to $[p, u_d, u_p]^T$. Let now T_0 be the desired transfer function between u_d and r. The open loop part K_1 of the controller has to keep the error signal z small for an entire family of the linear fractional transformation (LFT) modeled perturbed plants. The error signal z is computed from $(z = r - T_0 u_d)$. Vehicle parameter variations are in the LFT uncertainty bloc Δ

The reference model T_0 is chosen as first order transfer function with the same steady state gain as the conventional car in the equation

$$T_0 = \frac{G(v_0)}{0.2s + 1} \tag{7}$$

This model will ensure the same steady state value for the controlled and the conventional car. The settling time is about $0.8(sec)$. The reference model is chosen as a first order in order to avoid any overshot on vehicle responses. System of Figure 2 is thus updated to the one of figure 3 where G_{ff} includes the reference model and is the transfer matrix from $[p, u_d, u_p]^T$ to $[q, z]^T$. The control $u_p = K_1 u_d$ is then designed such as the H_∞ robust performance level γ_f is ensured [6]

$$\sup_{\Delta \in \mathbf{B}_\Delta} \|T_{zu_d}\|_{L_2 \longrightarrow L_2} < \gamma_f \tag{8}$$

5.4 Controller implementation

The controller is finally implemented as shown on Figure 4. The weighting filter W is included in the feedback controller K_2. Disturbances are thus filtered. However, in order to obtain a similar steady state behavior for the conventional and the controlled car, a speed scheduled gain $\alpha(v)$ has to be added to the prefilter part K_1. The controller implementation needs three type of measurement : the yaw rate by a gyro, the speed by an odometer and the steering wheel angle by an encoder.

5.5 Simulation results

All the simulations are conducted on the nonlinear model. In all figures, solid lines correspond to the controlled car responses and dashed ones to the conventional car responses.

5.5.1 Disturbance rejection: Firstly a step disturbance wind force of $5000N$ is applied to the vehicle at nominal speed and full road adhesion. The wind force appears at time $t_1 = 1\,\mathrm{sec}$ and disappears at $t_2 = 2\,\mathrm{sec}$. It is assumed that the driver doesn't react to this disturbance ($\delta_w = 0$). One can note from Figure 5 that the yaw rate goes to zero in steady state, i.e. the yaw angle remains constant in steady state. In comparison, the conventional car presents a non zero steady state yaw rate. This means that the yaw angle presents a ramp increasing without driver reaction. 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 rejected within driver reaction time.

5.5.2 Lane change maneuver: The controllers are now investigated in case of driver steering angle which corresponds to lane change maneuver (Figure 6-a, dashed line). The dash-dot line corresponds to

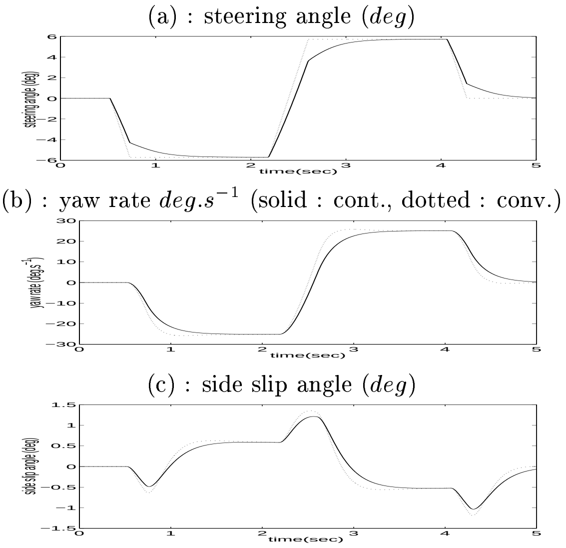

Figure 6: Lane change maneuver, nominal system

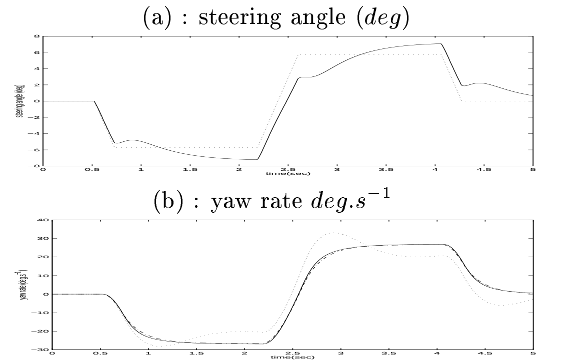

Figure 7: Lane change maneuver for $v = 30m.s^{-1}$ and $\mu = 0.5$ (solid : cont., dotted : conv.)

the response of the reference model. Figure 6 shows results obtained at nominal speed with nominal road adhesion. 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. On the other hand, the controlled yaw rate response of the controlled car closely follows the response of the reference model T_0. Responses are degraded for high speed and low road adhesion values, however, they practically do not present overshot (Figure 7). The robust model matching of the previous section makes the controlled vehicle robustly follow the specified first order reference model T_0. One can notice that the control effort is limited (Figures 6-a and 7). The controller subtracts or adds a steering angle to the driver command. When the road adhesion is at its nominal value even when the speed varies, the control effort vanishes within driver reaction time. When the road adhesion is decreased, there is a remaining steering angle.

6 Conclusion

In this paper, some general aspects of vehicle active steering have been presented. The presentation includes conventional, EAS and steer by wire systems. A two degree of freedom control procedure which combines feedforward and feedback control is developped and some results are outlined.

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Nomenclature

δ_d	steering wheel angle as set by the driver
R_s	Steering gear ratio
δ_f	Front wheel steering angle ($\delta_d = R_s\delta_f$)
T	Steering torque (control input)
T_s	Self aligning torque of front wheels
T_d	Disturbance torque
B_s	Steering system damping coefficient
I_s	Inertial moment of steering mechanism
r	Yaw rate
β	Vehicle side slip angle
ψ_L	Vehicle yaw angle relative to the road
y_s	Lateral displacement at lookahead distance
v	Longitudinal speed
ρ_{ref}	Road curvature