ACTIVE FRONT STEERING FOR PASSENGER CARS: SYSTEM MODELLING AND FUNCTIONS

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Abstract: Active Front Steering is a newly developed technology for passenger cars that realises an electronically controlled superposition of an angle to the hand steering wheel angle that is prescribed by the driver. It enables functionalities such as (vehicle velocity-) variable steering ratio, steering lead, as well as it provides an interface to support vehicle dynamics control systems. This paper focuses on system modelling and derivation of both steering assistance functions and a safety/diagnostics functions. All functions described are model based. Their models are derived and their accuracy is demonstrated.

Keywords: Active steering systems for passenger cars, Steering assistance functions, Safety functions, Model based design.

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1. INTRODUCTION AND MOTIVATION

Active Front Steering (AFS) is a newly developed technology for passenger cars that realises an electronically controlled superposition of an angle to the hand steering wheel angle that is prescribed by the driver. However, the permanent mechanical connection between steering wheel and road wheels remains.

The *steering assistance functions* that can be immediately experienced by the driver are for instance a variable steering ratio and a steering lead. Moreover, AFS provides an interface to support vehicle dynamics control systems. Here, the variable steering ratio is described and some measurements are presented to show the benefit of these functions.

Although AFS is not a steer-by-wire system, a great deal of measures is required in order to ensure the overall safety of the system, which has been recognised widely in the literature by (Harter et.al. 2000), (Knoop et.al. 1999). In fact, the electronic components, the actuator dynamics and the signals used to achieve AFS functionality are continuously monitored in order to ensure correct and safe operation of the system. Many types of so-called safety and monitoring functions are implemented. However, this paper will focus on the *application dependent safety and monitoring algorithms*, such as plausibility check of the angles involved (hand steering wheel angle, road wheel angle, motor angle), plausibility check of (open and closed loop) dynamics etc. Since all of the above mentioned algorithms are model based, their models are derived first. Most of these models base on vehicle or actuator physics or geometry, possibly accompanied by black box error terms that account for variations arising from mass production of the components. Some examples are given to illustrate the detection of sensor failures and to demonstrate the accuracy that is currently achieved.

2. SYSTEM DESCRIPTION AND NOTATION

Fig. 1 shows the Active Front Steering principle: The driver controls the vehicles course via the hand steering wheel; the resulting steering wheel angle is denoted by δ_S. AFS actuates an additional angle δ_M using its electric motor. Both angles result in an pinion angle δ_G down at the steering rack. All three angles relate as given in eqn.(1) below (also accounting for the respective ratios i_M, i_D). Fig. 2 shows the AFS system including actuator (motor and planetary gear box),

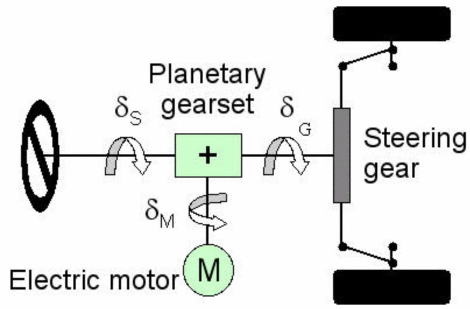

Fig. 1. Schematic view of AFS system. The electric motor superimposes an angle δ_M to the hand steering wheel angle δ_S. The result is the steering gear's pinion angle δ_G.

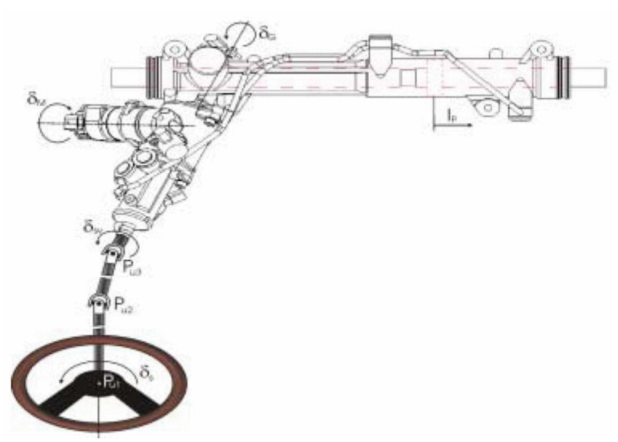

Fig. 2. AFS system including (top-down) steering rack, actuator (motor and planetary gear), steering column and hand steering wheel.

steering column, steering rack and hand steering wheel. The resulting (average) road wheel angle can then be calculated via the pinion angle and a static nonlinearity$F_{SG}(\cdot)$ that accounts for the relation between pinion angle and rack displacement as well as for the steering geometry, cf. (2). Finally, the overall ratio between hand wheel to road wheel is defined in (3).

$$\delta_G(t) = \frac{1}{i_M} \cdot \delta_M(t) + \frac{1}{i_D} \cdot \delta_S(t) \tag{1}$$

$$\delta_G(t) = F_{SG}\left(\delta_F(t)\right) \tag{2}$$

$$\delta_F(t) = \frac{1}{i_V} \cdot \delta_S(t) \tag{3}$$

Having this basic framework at hand, we can start looking at functions that manipulate the motor angle $\delta_M(t)$ in order to e.g. achieve a desired overall steering ratio i_V. This desired motor angle $\delta_{Md}(t)$ will then be passed to the motor's feedback control algorithm.

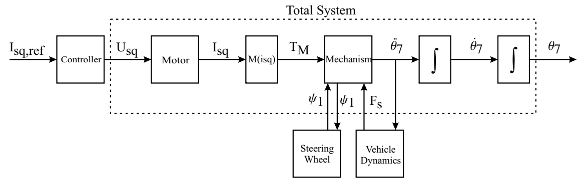

Fig. 3. An overview of the system structure used for mathematical modelling. The motor voltage u_{sq} is input and the angle $\theta_7 = \delta_M$ is output.

3. SYSTEM MODELLING AND PARAMETER ESTIMATION

3.1 *Setup of equations*

The complete system setup including mathematical modelling and parameter estimation is described in great detail in (Klier and Reinelt 2004), (Lundquist and Rothstrand 2003).

Fig. 3 depicts the overall structure of the Active Front Steering system, as is has been used for mathematical modelling. It follows the mechatronic approach in seperating the electrical part from the mechanical part, the torque being the link between these two parts. As can be seen in Fig. 3, the motor voltage u_{sq} is regarded as input to the system, which is then converted by the motor into an electric torque T_M. The mechanical sub-system then converts this torque into a rotation of the motor δ_M. The applied force acting on the rack F_S and the angle of the sun coupled to the steering colum ψ_1 are viewed as noises.

The complete system can be described by a differential equation. The state variables are defined as:

$$\begin{aligned} x_1 &:= i_{sq}, \\ x_2 &:= \delta_M, \\ x_3 &:= \dot{\delta}_M. \end{aligned}$$

The input signal is $u := u_{sq}$ and the applied force acting on the rack F_S and the angle of the sun coupled to the steering colum ψ_1 are modelled as noises:

$$\begin{aligned} z_1 &:= \psi_1 \\ z_2 &:= F_S. \end{aligned}$$

As shown in (Lundquist and Rothstrand 2003), the differentail equation for the electrical part and for the mechanical part can be expressed in one differential equation:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \end{bmatrix} = \begin{bmatrix} -\frac{R_s}{L_s} \cdot x_1 - \frac{\psi_R}{L_s} \cdot x_3 + \frac{1}{L_s} \cdot u \\ x_3 \\ \frac{f_1(x_1, x_2, x_3, z_1, \dot{z}_1, z_2) - \ddot{z}_1 m_{12}}{m_{11}} \end{bmatrix} \tag{4}$$

This is the resulting differential equation describing the actuator. The components f_1, m_{11} and m_{12} (non-linear function and elements of the generalised mass matrix respectively) are elaborated further in (Lundquist and Rothstrand 2003).

3.2 *Parameter estimation and validation*

Having established the equations we are left with parameter estimation. Measurement data have been used to identify the paramenters. For the electrical (linear) part prediction error methods, see (Ljung 1999) or (Reinelt et.al. 2002)., have been used to establish the parameters. For the mechanical (non-linear) part, nonlinear optimisation routines such as Levenberg-Marquardt algorithms have been used to minimise the residual error. In both cases, initial values for the parameters have been chosen using data-sheets or CAD drawings. Fig. 4 and Fig. 5 show sample results of the validation process.

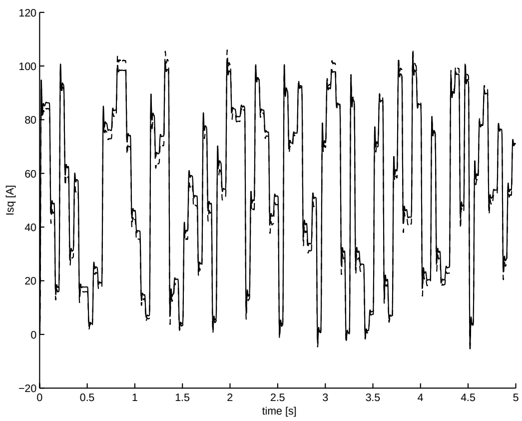

Fig. 4. Validation of the electrical parameters: comparing the modelled (dashed) and measured (solid) motor current.

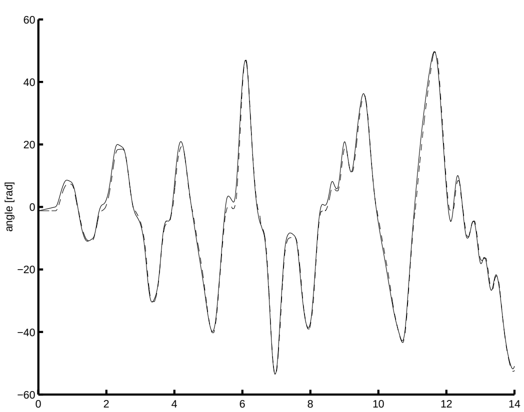

Fig. 5. Validation of the mechanical parameters: comparing the modelled (dashed) and measured (solid) motor angle.

3.3 *Usage of the Model*

The validated model derived above builds the basis for deriving functions the are implemented in the Active Front Steering ECU. One purpose is to use it as simumlation environment for already developed steering assistance functions as for example Variable Steering Ratio. Another purpose is to re-use parts of it within the so-called safety functions. The purpose of the safety functions is to monitor and check the signals that are used by the assistance functions on the one hand and tho monitor the dynamics of the overlal system ans well as the actuator on the other hand. This is done in order to ensure the overall safety of the Active Front Steering system during runtime. More details on the technical safety concept are given in (Reinelt et.al. 2004). General concepts of signal monitoring, change detection in particular, is discussed in (Gustafsson 2000).

4. VARIABLE STEERING RATIO

4.1 *Function Description and Design*

The purpose of the variable steering ratio (VSR) is to adapt the overall ratio between hand wheel angle $\delta_S(t)$ and (averaged) road wheel angle $\delta_F(t)$, cf. (3) to the current driving situation. The driving situation can, for instance, be determined by the vehicle's velocity $v_X(t)$ and pinion angle $\delta_G(t)$. Hence, i_V in (3) becomes a function of velocity and pinion angle. The point in varying the ratio dependent on velocity is to decrease it (compared to the mechanical ratio) when driving at very low velocities, which is of particular use for example during parking maneuvers. Then the driver only has a small steering effort (in terms of turning the hand wheel) due to this direct steering ratio in order to maneuver the car smoothly into the parking space. At higher vehicle velocities the steering ratio becomes increasingly indirect up to the level of a conventional steering system (or even beyond). The principle of varying the overall ratio over the vehicle velocity is shown in Fig. 6.Moreover, AFS can be used to vary the ratio with respect to pinion angle, so that the functionality of a (mechanical) variable rack can be achieved. The desired overall ratio can be specified quite conveniently by the vehicle manifacturer for example in terms of a look-up table. An example for such a look-up table is given in Fig. 6. The means of realising the above described functionality is the control the position of the AFS motor in order to achieve the desired ratio. We therefore insert (2,3) into (1), which yields the motor angle:

$$\delta_M(t) = i_M \cdot F_{SG}\left(\frac{\delta_S(t)}{i_V}\right) - \frac{i_M}{i_D} \cdot \delta_S(t) \qquad (5)$$

Now, *given* the desired ratio at a certain time instant:

$$i_V = i_V(v_X, \delta_G) \qquad (6)$$

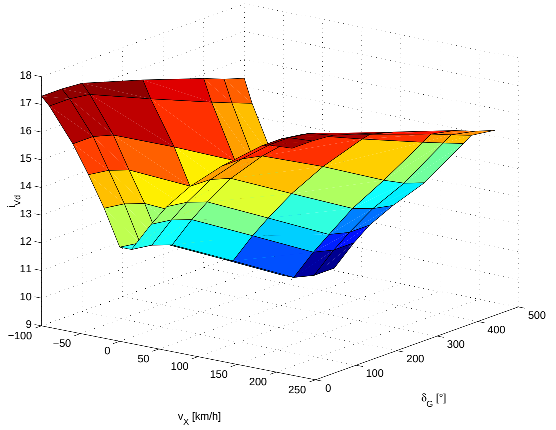

Fig. 6. Example for velocity and pinion angle dependent ratio $i_V(v_X, \delta_G)$.

the respective motor angle can be calculated using (5) and will then be passed on to the motor's feedback controller as reference signal.

4.2 *Results from a Testdrive*

Fig. 7 shows the overall ratio i_V that has been applied during a circle drive accelerating first and the braking. Quite obviously, other ratios have been applied for braking and for accelerating. This is due to the fact that the ratio is filtered during braking actions, which is a comfort feature for the driver. It is generally not appreciated when the ratio decreases as fast during (hard) braking as it increases when accelerating quickly.

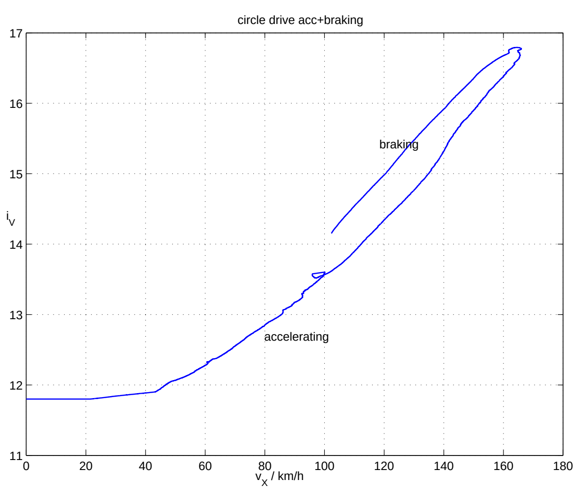

Fig. 7. Example for velocity dependent ratio $i_V(v_X, \cdot)$ driving a circle accelerating and braking. Different rations being applied when accelerating and braking.

5. PINION ANGLE PLAUSIBILITY CHECK

5.1 *Function Description and Design*

The purpose of the Pinion Angle Plausibility Check (PAP) is to monitor the pinion angle sensor signal for possible failures using the road wheel revolutions. The following analytic expression can be easily derived from the steering geometry, see (Wong 2001, ch.5) and Fig. 8:

$$\tan \delta_i = \frac{-I(\omega_i^2 - \omega_o^2)}{S_L \omega_i^2 + \sqrt{S_L^2 \omega_i^2 \omega_o^2 - I^2(\omega_i^2 - \omega_o^2)^2}} \qquad (7)$$

where I and S_L denote the vehicle's wheelbase and track respectively, δ_i the angle of the inner (with respect to the curve) road wheel, and finally ω_i, ω_o the revolutions of the inner and outer road wheel. δ_i is then mapped onto the pinion angle δ_G exploiting (2)[1]. Hence, the model used for estimation of the pinion angle uses the two front wheel revolutions as input signals and the steering geometry (represented by the parameters I, S_L in (7) and the static nonlinearity $F_{SG}(\cdot)$ in (2)). The parameter identification basically follows a two stage procedure. Firstly, the vehicle's wheelbase and track are measured. Secondly, the static non-linearity $F_{SG}(\cdot)$ is estimated, this step essentially being the estimation of a Wiener model. For methods of estimating these and assessing their quality, see (Bauer and Ninness 2002).

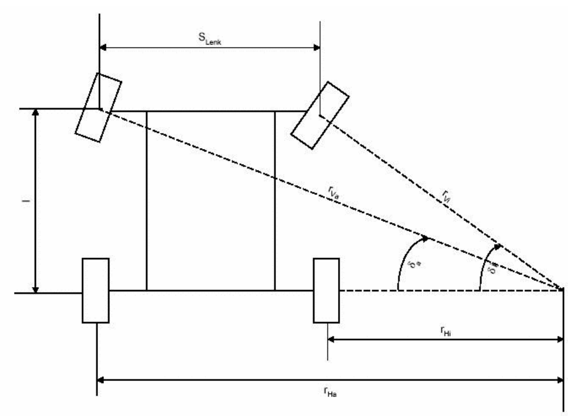

Fig. 8. Vehicle geometry using the front wheel revolutions ω_i, ω_o to calculate the average road wheel angle δ_F.

5.2 *Results from a Testdrive*

Fig. 9 shows sample results from a handling course maneuver for the pinion angle estimation, when no sensor failure is present. One should, however, take into account that a decision whether the pinion sensor signal is faulty or not cannot be made on the result of this safety function alone; the basic equation derived above may not be valid because of particular driving situations such as sliding, swimming or brake interventions by an ESP system an one wheel. To assess the state of the pinion angle sensor signal, the full information

[1] The static non-linearity F_{SG} has to be adapted in a straightforward way to account for the angle of the inner wheel instead of the averaged front wheel angle.

of all safety funcions (electronic dependent and application dependent) is neccessary.

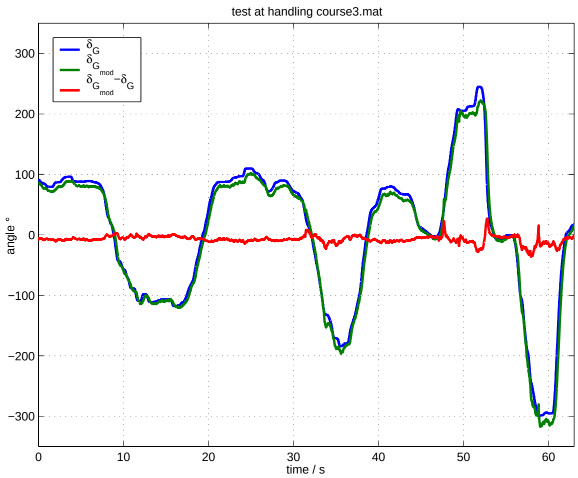

Fig. 9. Data collected at a handling course maneuver for the pinion angle estimation, when no sensor failure is present: estimated angle (green), measured angle (blue), and difference between estimated and measured angle (red).

6. ANGLE MONITORING ALGORITHM

6.1 *Function Description and Design*

The purpose of the Angle Monitoring Algorithm (AMA) is to monitor the kinematic constraint (1) between the three angles in order to detect possible sensor failures or mechanical failures. Although (1) looks quite simple, preliminary investigations show that it cannot be modelled in an acceptable accuracy as a linear black box model. The basic problem that prevents us from this approach is that the hand steering wheel sensor, measuring $\delta_S(t)$ is located in the top of the steering column, while the motor and pinion angle sensors are situated in the lower part of the steering column, see Fig. 2. A full multi-body model of the actuator is derived in (Klier and Reinelt 2004) and could contribute to overcome this problem. However, this model is not suited for implementation on an ECU. Consequently, a simple solution has to be looked for: effects such as torsion of the steering column (which can be modelled in a linear fashion quite accurately) and effects of the universial joints in the steering column (which must be modelled using non-linear models) have been taken into account. All this leads us to a linear black box model and a trailing static nonlinearity, i.e. the following Wiener model:

$$\dot{x}(t) = Ax(t) + b(\delta_G^{\#}(t) - \delta_M^{\#}(t));\ x(0) = x_0 \quad (8)$$
$$\hat{\delta}_S(t) = J\left(Cx(t)\right) \quad (9)$$

where $\delta_G^{\#}(t), \delta_M^{\#}(t)$ are pinion and motor angle respectively, normalised to the hand steering wheel angle using (1). The LTI system (A, B, C) accounts for the linear torsion dynamics in (1) (basically using the steering velocity of the driver), and the static non-linearity $J(\cdot)$ accounts for possible effects given by universial joints in the steering column. Although an analytical expression for $J(\cdot)$ is at hand, it will not be possible to run is on the ECU (from a computational point of view). Hence, a simplified version has been applied.

As in the case of the pinion angle plausibility check, the linear and the non-linear part are estimated in two stages, based on data collected in the vehicle. The data, however, has to cover driving situations where both effects (torsion and universal joints) are present to ensure sufficiently much excitiation in the inputs signals.

6.2 *Results from a Testdrive*

Fig. 10 shows the estimation of the steering wheel angle during a vehicle dynamics maneuver, a circle drive with changing vehicle velocity. A drift in the pinion angle sensor has been injected (starting at $3.7s$) which results in an increasing deviation between estimated and measured steering wheel angle. How much deviation is allowed before a failure is assumed, is an applicable parameters that usually varies with the vehicle in question. Figs. 11 and 12 show the estimation of the steering wheel angle during a slalom drive with constant vehicle velocity. An impulse in the pinion angle sensor has been injected (at $5.14s$ of the experiment) which results in a "punctual" deviation between estimated and measured steering wheel angle.

Note, however, that a decision which of the three sensor signals is faulty cannot be made on the results alone; to assess this, the full information of all safety funcions (electronic dependent and application dependent) is neccessary.

7. CONCLUSIONS AND FUTURE WORK

The Active Front Steering (AFS) System has been described, from a plain functional point of view, though (i.e. a electronically controlled superposition of an angle to the hand steering wheel angle that is prescribed by the driver). The need for installing model based safety functions, additionallly to the basis sensor diagnostics and range and gradient checks, has been motivated and some of the application dependent safety functions have been described, in particular the pinion angle plausibility check and the monitoring of the kinematic relation between the three angles, important for AFS. This list is far from being

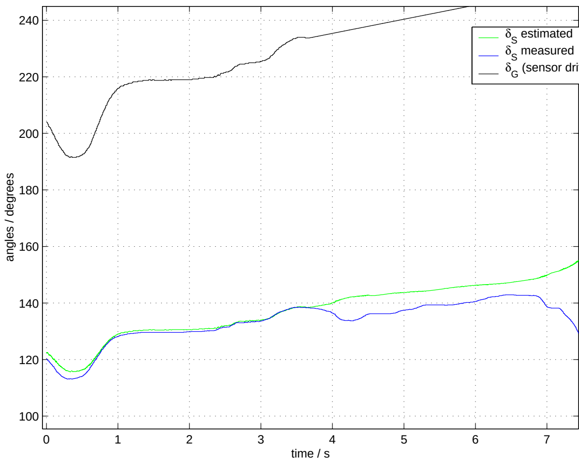

Fig. 10. Data collected during a maneuver on the vehicle dynamics area demonstrating the angle monitoring algorithm dealing with a drift in pinion angle sensor, starting at $3.7s$. Measured steering wheel angle (blue), estimated steering wheel angle (green) and drifting pinion angle signal (black).

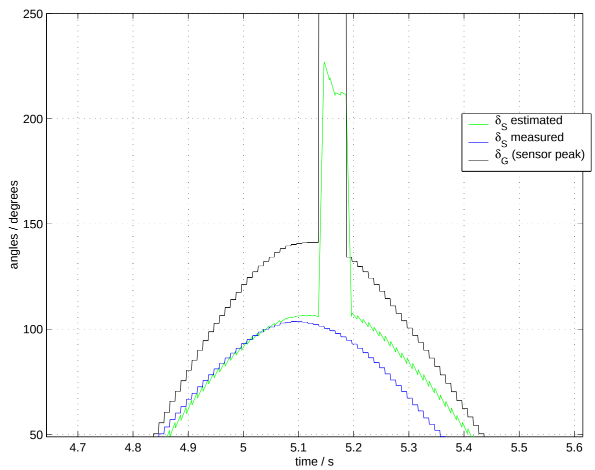

Fig. 11. Data collected during a maneuver on the vehicle dynamics area demonstrating the angle monitoring algorithm dealing with a peak in the pinion angle sensor signal around $5.14s$. Measured steering wheel angle (blue), estimated steering wheel angle (green) and pinion angle signal with peak (black). Zoomed version of Fig. 12.

complete. A function, that can be directly experienced by the driver has been described as well: the (speed and pinion angle) variable steering ratio. All descriptions have been accompanied by measurements from AFS vehicles. Another function that can be experienced by the driver is the so-called steering lead, that essentially enhances the vehicle reaction during quick steering maneuvers. This function will be described in future works, also some more details on particular model based safety functions. Future work with respect to the safety functions will focus on more advanced monitoring strategies, e.g. observer based approaches.

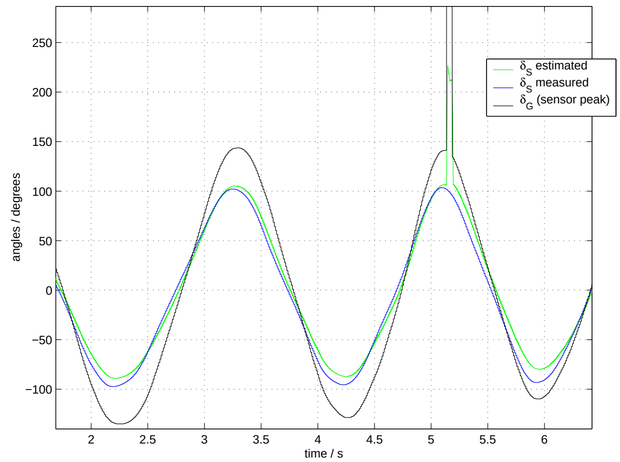

Fig. 12. Data collected during a maneuver on the vehicle dynamics area demonstrating the angle monitoring algorithm dealing with a peak in the pinion angle sensor signal around $5.14s$. Measured steering wheel angle (blue), estimated steering wheel angle (green) and pinion angle signal with peak (black).

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