ESTIMATION OF VEHICLE ROLL ANGLE

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Abstract: Vehicle roll angle is one of the most important values for lateral vehicle dynamics. Electronic lateral dynamic control system operates only as desired during all driving maneuvers on all road surfaces, if the vehicle roll angle is known. Due to economical reasons, the vehicle roll angle is not measurable in series-produced vehicles. Not only more precision of vehicle roll angle estimation, but also evaluation of estimation error are demanded by automobile industry. In this contribution a new concept, which allows a specification of the estimated angle statistically, is presented. The performance of this concept is tested by car tests. *Copyright 2005 IFAC*

Keywords: automotive control, vehicle dynamics, state observers, error analysis and evaluation.

1 INTRODUCTION

The electronic stability program (ESP) is an essential driving dynamic control system (Ding *et al.*, 2004a), (Ghoneim *et al.*, 2000), (Van Zanten *et al.*, 1998). The use of ESP is to provide active safety during car driving. The activation of the lateral dynamic control of ESP relies on two ESP-sensor signals, namely the yaw rate sensor signal and the lateral accelerometer signal. However, both lateral and roll dynamics are involved in this lateral acceleration measurement. This can lead to false activation of ESP on a banked road. Since neither the vehicle sideslip angle nor the vehicle roll angle are directly measurable in series-produced vehicles due to economical reasons, it is difficult to decouple the effect of lateral and roll dynamics in the lateral acceleration measurement. In the last years lots of investigations have been made in this area. The main research results can be found in the literatures listed in this paper. One group of these works attempts to estimate the vehicle sideslip angle. In (Ding *et al.*, 2002) three static models for vehicle sideslip angle estimation are derived on the assumption that driving maneuvers are steady, where vehicle roll angle is considered. A further suitable work for estimating vehicle sideslip angle is described in (Fukada, 1999). However, the vehicle sideslip angle is estimated on the condition that all lateral tire forces have to be measured, which cannot be measured on-line in common vehicles produced in series. Another concept for vehicle sideslip angle estimation is designed by an adaptive observer under the consideration of the change of the tire side stiffness coefficients (Ding *et al.*, 2003), where the vehicle roll angle is not taken into account. The other group of the works try estimating the vehicle roll angle. In (Tseng, 2001) decoupling the effect of vehicle sideslip angle and vehicle roll angle is investigated on the basis of lateral dynamics. This concept is promising either only during steady driving maneuvers or only during driving on even roads. The decoupling cannot be guaranteed, if the car drives on a banked low-μ road dynamically.

2 PROBLEM DESCRIPTION

Summarizing these existing works, the problems to decouple the effect of lateral and roll dynamics can be formulated as follows:

- During car driving the road bank angle changes permanently. This influences directly vehicle lateral dynamics. Without the knowledge of the vehicle roll angle, the vehicle sideslip angle cannot be estimated correctly.
- Apart from that, the road friction coefficient changes also permanently. This leads to changing important vehicle parameters, namely the tire side stiffness coefficients. This parameter uncertainty makes the vehicle sideslip angle estimation worse.

- On the other side, the vehicle roll angle estimation relies on lateral dynamics. As mentioned before the tire side stiffness coefficients and the vehicle sideslip angle are unknown. It is clear that the vehicle roll angle cannot be estimated correctly in all driving situations, too.

Facing these problems described above, we try finding a solution through the other way.

An important component of the passive safety system in vehicles is the so-called Rollover Sensing System (ROSE), in which a roll rate sensor is used for detecting the rollover danger of vehicles (Chen and Peng, 2001), (Ding *et al.*, 2004b). The roll rate sensor delivers important information on roll dynamics, namely the differentiation of the vehicle roll angle. Now, one can consider to obtain the vehicle roll angle by integrating the roll rate sensor signal. This simple method fails in practical applications, since the standard measurement errors, for example, offset tolerance, sensitivity tolerance and unknown disturbances, exist and furthermore the initial value of the vehicle roll angle is unknown. But, we can use this roll rate sensor signal as additional information to evaluate the performance of the roll angle estimation. By means of this evaluation, information on the state of the roll angle estimation, in other words, a specification of the estimated roll angle in form of a so-called confidence interval can be obtained permanently. So, the task of the roll angle estimation is completed. Though the vehicle roll angle cannot be estimated perfectly at all times. The electronic controller receives the confidence interval and uses this information to make its decision on control correctly. To achieve the aim, an estimation concept is designed by using a linear observer.

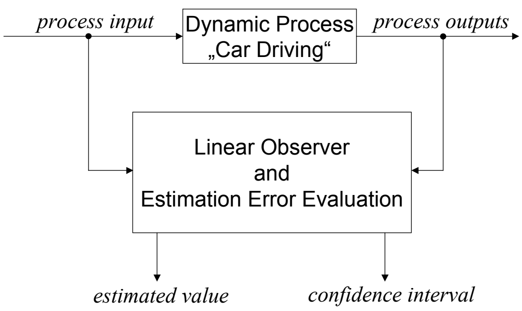

Fig. 1: Structure of the estimation concept

Fig. 1 shows the structure of the estimation concept. The core of the estimation concept is to estimate the vehicle roll angle, to evaluate the estimation error and to generate the confidence interval. The idea to use a linear observer consists not only in the construction of the physical value to be estimated, but also in the evaluation of the estimation error with the help of the observer system matrix by using the existing sensor signals. The input values of the concept are the lateral accelerometer signal, the yaw rate sensor signal, the roll rate sensor signal and the vehicle velocity estimated by ESP. The output values are the estimated value and the confidence interval as Fig. 1 shows. The following chapter presents this observer design.

3 OBSERVER DESIGN

3.1 State space model

Fig. 2 shows the simplified model for the vehicle roll motion on a banked road, where:

g: acceleration of gravity,
χ: road bank angle,
a_y: lateral acceleration of the vehicle body center,
ϕ: vehicle body roll angle in relation to the road,
m_R: sprung mass of the vehicle,
h: height of the center of gravity of the vehicle body in relation to the roll axis.

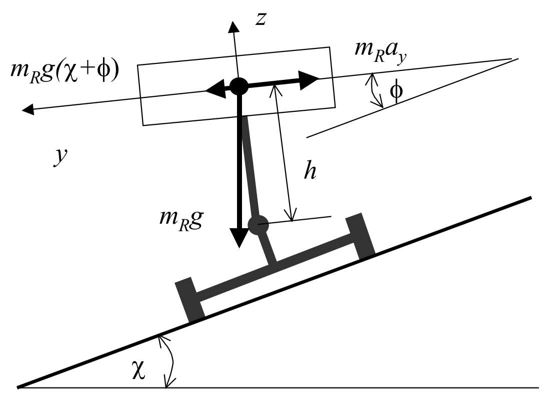

Fig. 2: Schematic of the roll motion

Denote $(\chi + \phi)$ with $\varphi_{x,M}$, where $\varphi_{x,M}$ signifies the vehicle body roll angle in relation to the earth-fixed coordinate system. Then, we have:

$$\phi = \varphi_{x,M} - \chi \, . \qquad (1)$$

The derivative of $\varphi_{x,M}$ corresponds to the roll rate sensor signal $\omega_{x,S}$:

$$\dot{\varphi}_{x,M} = \omega_{x,M} \mathrel{\hat{=}} \omega_{x,S} \, .$$

The signal, which the lateral accelerometer senses, is formulated by the following equation according to the accelerometer functionality:

$$a_{y,S} = a_y + g \sin(\chi + \phi) \, . \qquad (2)$$

In steady driving maneuvers the differentiation of the vehicle sideslip angle β (Ryu *et al.*, 2002) can be assumed as zero. Only in this situation the lateral acceleration a_y can be calculated exactly by using the yaw rate sensor signal $\omega_{z,S}$ and the vehicle velocity v:

$$a_y = v(\omega_{z,S} + \dot{\beta}) \cong v\omega_{z,S} \, . \qquad (3)$$

Inserting Equation (3) into Equation (2), the following equation holds:

$$\sin(\chi + \phi) = \frac{a_{y,S} - v\omega_{z,S}}{g} \, . \qquad (4)$$

Using Equation (4), the vehicle roll angle $(\chi + \phi)$ in relation to the earth-fixed coordinate system can be estimated correctly just in steady driving maneuvers. To achieve the aim mentioned above the roll dynamics is studied in the following.

According to the torque balance in the roll axis (see Fig. 2), the roll dynamics of the vehicle body can be described by the following differential equation (Ding et al., 2004b):

$$\begin{aligned} & I_{xx}\ddot{\phi} + C_R\dot{\phi} + K_R\phi \\ & = m_R a_y h + m_R g h \sin(\chi + \phi), \end{aligned} \tag{5}$$

where the new symbols used in Equation (5) are defined as follows:

I_{xx}: moment of inertia in the roll axis,
C_R: damping coefficient of the roll motion system of the vehicle,
K_R: spring coefficient of the roll motion system of the vehicle,
$\dot{\phi} = \omega_x$: roll rate in relation to the road,
$\ddot{\phi}$: roll acceleration in relation to the road.

Applying Equation (2) to Equation (5), the following equation is achieved:

$$I_{xx}\ddot{\phi} + C_R\dot{\phi} + K_R\phi = m_R a_{y,S} h. \tag{6}$$

Now, we insert Equation (1) into Equation (6) and obtain the new equation:

$$\begin{aligned} & I_{xx}\ddot{\varphi}_{x,M} + C_R\dot{\varphi}_{x,M} + K_R\varphi_{x,M} \\ & = m_R a_{y,S} h + I_{xx}\ddot{\chi} + C_R\dot{\chi} + K_R\chi. \end{aligned} \tag{7}$$

It is assumed again, that the first derivative and the second derivative of the road bank angle χ are very small, so that $\ddot{\chi} \approx \dot{\chi} \approx 0$. Then, we have:

$$\begin{aligned} & I_{xx}\ddot{\varphi}_{x,M} + C_R\dot{\varphi}_{x,M} + K_R\varphi_{x,M} \\ & = m_R a_{y,S} h + K_R\chi. \end{aligned} \tag{8}$$

We define the state variables:

$$x_1 = \varphi_{x,M}$$
$$\dot{x}_1 = x_2 = \dot{\varphi}_{x,M}$$
$$\dot{x}_2 = \ddot{\varphi}_{x,M} = \frac{1}{I_{xx}}\left(-K_R x_1 - C_R x_2 + K_R\chi + m_R h a_{y,S}\right)$$

and formulate the above three equations in a state space model:

$$\begin{bmatrix}\dot{x}_1\\ \dot{x}_2\end{bmatrix} = \begin{bmatrix} 0 & 1 \\ \frac{-K_R}{I_{xx}} & \frac{-C_R}{I_{xx}} \end{bmatrix}\begin{bmatrix}x_1\\ x_2\end{bmatrix} + \begin{bmatrix} 0 & 0 \\ \frac{K_R}{I_{xx}} & \frac{m_R h}{I_{xx}} \end{bmatrix}\begin{bmatrix}\chi\\ a_{y,S}\end{bmatrix}$$
$$y = \begin{bmatrix}0 & 1\end{bmatrix}\begin{bmatrix}x_1\\ x_2\end{bmatrix},$$

whose state variables are the roll angle and the roll rate, inputs are the road bank angle as well as the lateral accelerometer signal and output is the roll rate. The state space model can be reformulated by the standard form of linear state space descriptions:

$$\begin{aligned} & \dot{\underline{x}}(t) = \underline{A}\,\underline{x}(t) + \underline{B}\,\underline{u}(t) \\ & y(t) = \underline{c}^T\,\underline{x}(t). \end{aligned} \tag{9}$$

3.2 Preparation of the first input signal

The observer design is based on the pre-condition, that all of the input signals are known. However, the first input signal χ of the state space model is unknown. This value can be computed by using Equation (4):

$$\chi = \arcsin\frac{a_{y,S} - v\omega_{z,S}}{g} - \phi. \tag{10}$$

The unknown angle ϕ in Equation (10) can be determined as the solution of Equation (6) or with the help of the torque balance in steady cornering state:

$$\phi = \frac{m_R a_{y,S} h}{K_R}. \tag{11}$$

3.3 Linear observer

Based on the state space model developed above, a linear observer can be designed according to the well-known structure:

$$\begin{aligned} & \dot{\hat{\underline{x}}}(t) = \underline{A}\hat{\underline{x}}(t) + \underline{B}\,\underline{u}(t) + \underline{k}\left(y(t) - \hat{y}(t)\right) \\ & \hat{y}(t) = \underline{c}^T\,\hat{\underline{x}}(t), \end{aligned} \tag{12}$$

where:

$\hat{\underline{x}}(t)$: estimated state vector,
$\hat{y}(t)$: estimated output value,
$\underline{k}^T = [k_1 \quad k_2]^T$: feedback gain vector.

Subtracting Equation (12) from Equation (9), the error dynamics of the observer system results:

$$\dot{\underline{x}}(t) - \dot{\hat{\underline{x}}}(t) = \left(\underline{A} - \underline{k}\,\underline{c}^T\right)\left(\underline{x}(t) - \hat{\underline{x}}(t)\right). \tag{13}$$

This observer system is asymptotically stable, if and only if the real components of all eigenvalues of the system matrix $\left(\underline{A} - \underline{k}\,\underline{c}^T\right)$ are negative. This means, that we have to select the vector $\underline{k}$ in such a way, so that the stability condition is fulfilled.

The vector $\underline{k}$ can be selected by using pole placement method. In our case, we can influence specifically the dynamic behavior of the observer system by pole placement.

4 CONFIDENCE INTERVAL

As mentioned in Chapter 1, the vehicle roll angle is not measured in series-produced vehicles and has to be estimated here. Therefore, the error of the roll angle estimation $\Delta x_1(t)$ cannot be obtained so simply like: $\Delta x_1(t) = x_1(t) - \hat{x}_1(t)$, if the true value $x_1(t)$, namely the physical roll angle, is unknown. In this chapter we use the existing observer system dynamics Equation (13) to evaluate the estimation error for the specification of the estimated roll angle.

Define $\Delta\underline{x}(t) = \underline{x}(t) - \hat{\underline{x}}(t)$ as state error vector, $\Delta y(t) = y(t) - \hat{y}(t) = \Delta x_2(t)$ as output error and apply them in Equation (13), we obtain:

$$\begin{pmatrix} \Delta\dot{x}_1(t) \\ \Delta\dot{y}(t) \end{pmatrix} = \begin{pmatrix} a_{11} & a_{12}-k_1 \\ a_{21} & a_{22}-k_2 \end{pmatrix} \begin{pmatrix} \Delta x_1(t) \\ \Delta y(t) \end{pmatrix} = \begin{pmatrix} f_{11} & f_{12} \\ f_{21} & f_{22} \end{pmatrix} \begin{pmatrix} \Delta x_1(t) \\ \Delta y(t) \end{pmatrix} . \quad (14)$$

Since $\Delta y(t) = y(t) - \hat{y}(t)$ is known, the unique unknown variable $\Delta x_1(t)$ can simply be computed by using the second row of the matrix equation (14):

$$\Delta x_1(t) = \frac{\Delta\dot{y}(t) - f_{22}\Delta y(t)}{f_{21}} . \quad (15)$$

From Equation (15) one can see, that the error of the roll angle estimation depends on both the output error and the pole placement. The fact can be ascribed to the linear system theory.

As well-known noise is normally involved in measurement. All the sensor signals used here are not excepted and even strongly affected by lots of unknown disturbances during car driving. Therefore, it is reasonable to use the Standard Deviation (StDe) to evaluate the estimation error. Thereby, the arithmetic average of the estimation error is assumed as zero. Forming the absolute value on both sides of Equation (15), the following equation yields:

$$|\Delta x_1(t)| = \frac{|\Delta\dot{y}(t) - f_{22}\Delta y(t)|}{|f_{21}|} \le \frac{|\Delta\dot{y}(t)| + |f_{22}|\cdot|\Delta y(t)|}{|f_{21}|} . \quad (16)$$

The quadratic mean value of the estimation error can be evaluated as follows:

$$E\{\Delta x_1^{\,2}(t)\} \le E\left\{ \left(\frac{|\Delta\dot{y}(t)| + |f_{22}|\cdot|\Delta y(t)|}{|f_{21}|} \right)^2 \right\} . \quad (17)$$

The quadratic root of the mathematical formulation of the right side of Equation (17) is the confidence interval of the roll angle estimation, while the quadratic root of the left side is the StDe, which cannot be delivered in series-produced vehicles.

5 CAR TEST RESULTS

The new concept is tested in lots of different driving situations. As mentioned before the vehicle roll angle can be reconstructed precisely by using the vehicle velocity, the yaw rate sensor and the lateral accelerometer during steady driving maneuvers. But, the challenge for the vehicle roll angle estimation concept is to be able to operate correctly during all driving maneuvers on all road surfaces. It is exciting to know, how this concept developed here works during highly dynamic driving both on high-μ and on low-μ roads with varying bank and especially during driving on low-μ roads, while the vehicle sideslip angle changes extremely. For the presentation in this paper, just three driving maneuvers on *high-μ roads*:

a) circular driving on a dynamic car test area,
b) lane change on a banked road,
c) handling curse on a wet asphalt road

and three ones on *low-μ roads* with large vehicle sideslip angle:

a) circular driving on ice,
b) driving on ice with step-shaped steer angle signal,
c) handling curse on ice with variable velocity

are chosen. For the car tests, important state values, which cannot be measured in series-produced vehicles, for example, the vehicle sideslip angle and the vehicle roll angle, are measured by additional measurement instruments. It has to be noticed, that these measured angles are used just as reference signals and cannot be regarded as absolutely correct signals. Each figure in the following subchapters contains four diagrams. The first diagram shows the vehicle velocity, while the second one contains the yaw rate sensor (×1), the lateral accelerometer (×2), the sideslip angle (×2) and the steer angle (×2) signal. They are used to illustrate the dynamic behavior. The third diagram presents the estimated vehicle roll angle in comparison with the so-called real value, which is measured by an additional instrument only for this test. The fourth one shows the generated confidence interval and the StDe between the estimated and the reference roll angle.

5.1 Car test on high-μ roads

Fig. 3 shows the result of test a). One can see that the StDe is very small (< 0.5°) during straight forward driving. While the circular driving ($a_y \cong$ 8m/s²), the StDe is about 1°. The StDe is larger, if the sideslip angle changes. In any case the StDe is located in the generated confidence interval as described in Equation (17) .

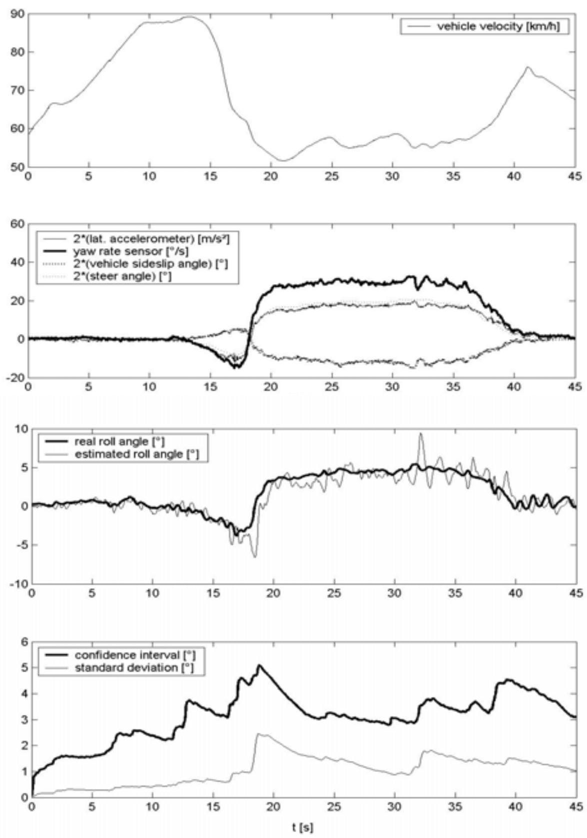

Fig. 3: Circular driving on a dynamic car test area

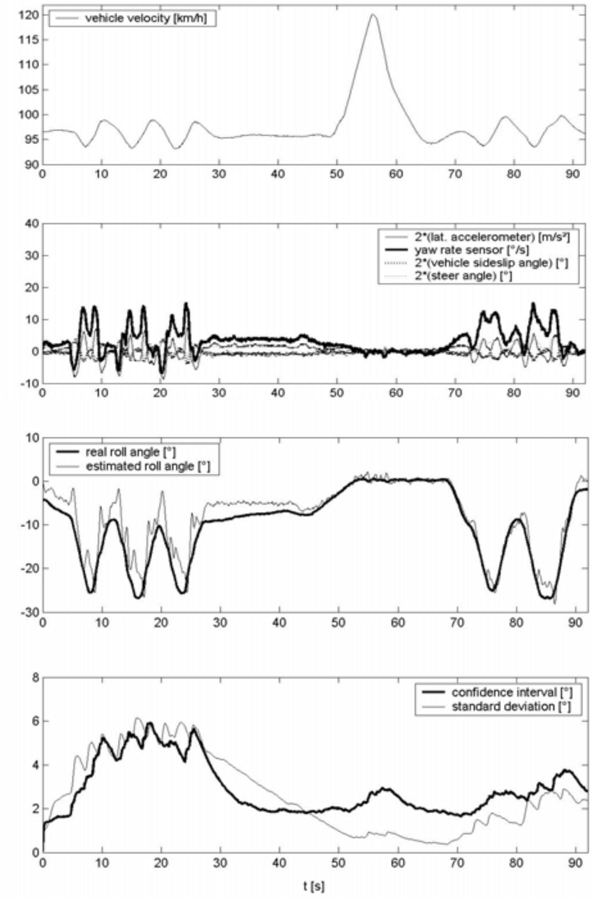

Fig. 4: Lane change on a banked road

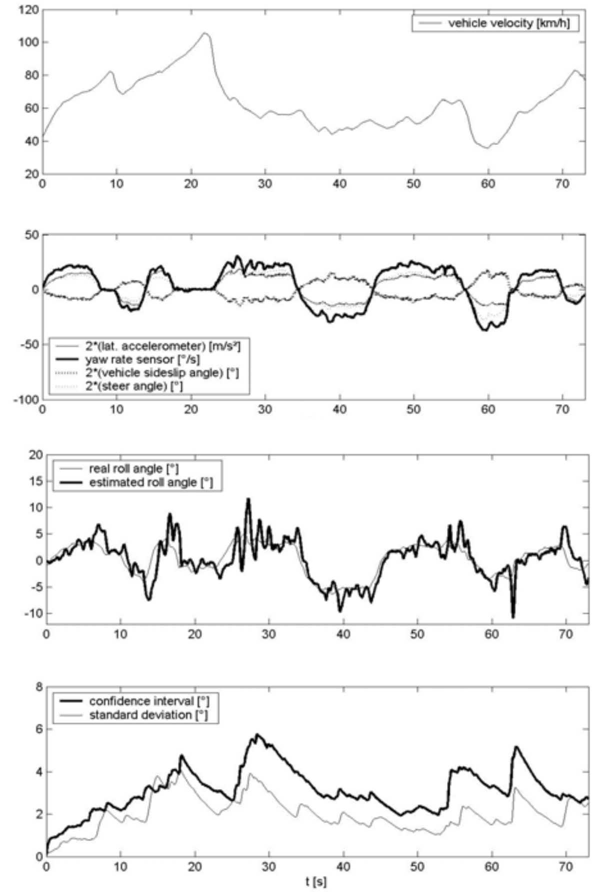

Fig. 5: Handling curse on a wet asphalt road

Fig. 4 illustrates the result of test b). During the lane change on the banked road the road bank angle varies. The StDe increases during the dynamic driving. One can see that the confidence interval can represent the StDe satisfyingly. The concept operates also correctly in car test c), whose result is shown in Fig. 5.

5.2 Car test on low-μ roads

The results of the car tests on low-μ roads a), b) and c) are presented in Fig. 6, Fig. 7 and Fig. 8 respectively. From these test results one can see that the vehicle sideslip angle is much larger and simultaneously changes more strongly on low-μ roads than on high-μ roads during dynamic driving. Therefore, the StDe here is generally increased. One notices that the confidence interval is also able to specify the StDe satisfyingly on low-μ roads.

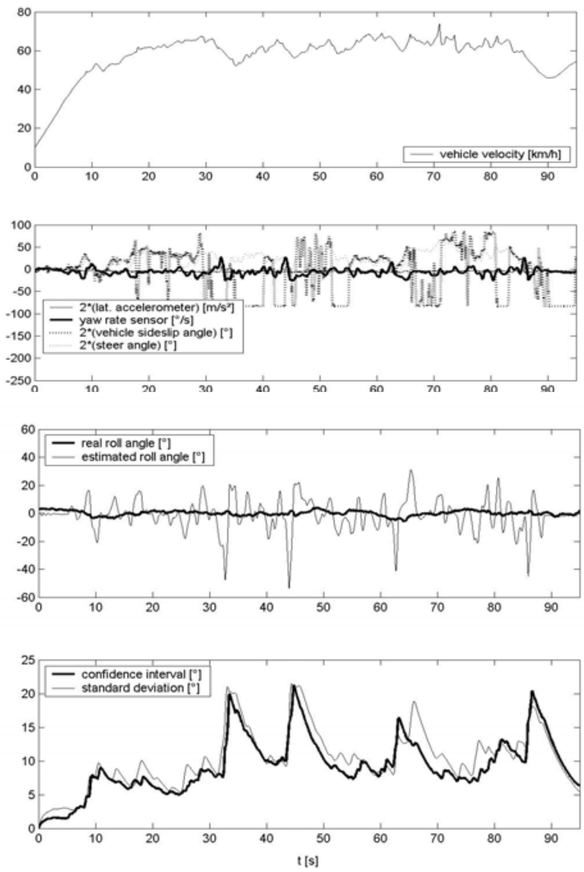

Fig. 6: Circular driving on ice

6 CONCLUSION

In this report a new concept for estimating the vehicle roll angle during car driving is described. A state space model is derived, which is able to describe the vehicle roll dynamics not only on even roads, but also on banked roads. Based on this model, a linear observer is designed for estimating the vehicle roll angle. With the help of the system dynamics of the observer a confidence interval for evaluating the vehicle roll angle estimation can be generated. The estimation concept is tested by lots of driving situations.

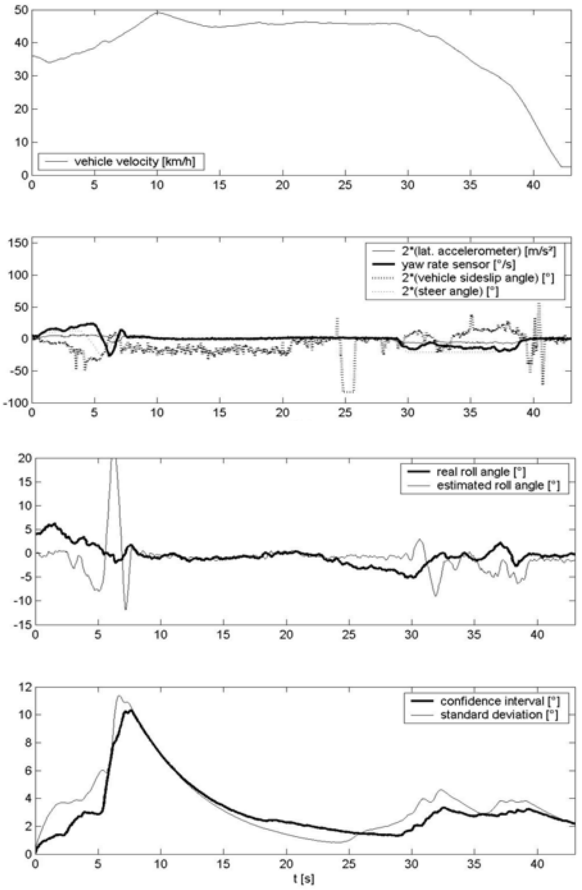

Fig. 7: On ice with step-shaped steer angle change

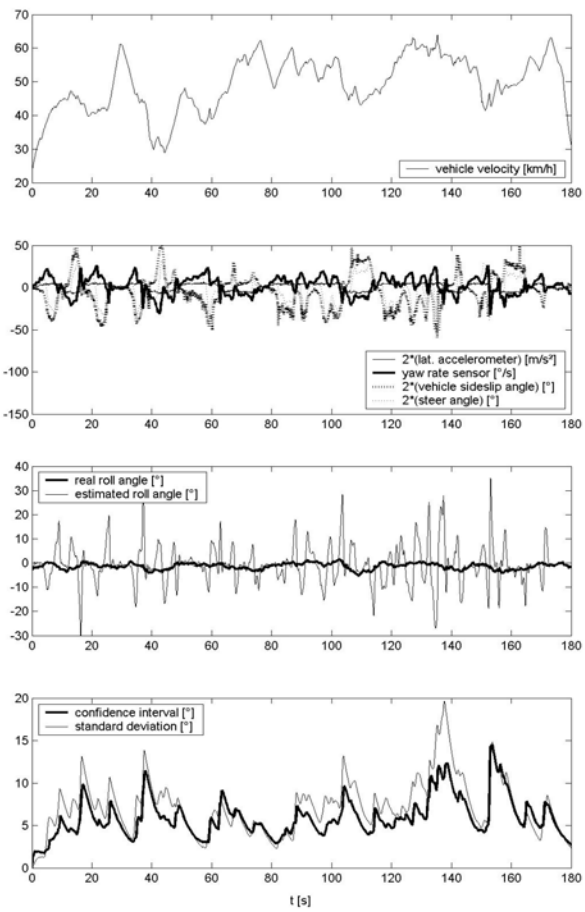

Fig. 8: Handling curse on ice

Though the concept does not estimate the vehicle roll angle in all driving situations exactly, reliable information on the actual state of the estimation can be delivered on-line only by using the existing sensors, so that the ESP-controller can exclude false interventions by using this important information.

7 ACKNOWLEDGEMENT

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