SIDESLIP ANGLE ESTIMATION BASED COMMERCIAL VEHICLE STABILITY CONTROL

Booklet of Ph.D. Theses
written by
Zoltán Hankovszki

Supervisor: László Palkovics, D.Sc.
External advisor: Roland Kovács

Kandó Kálmán Doctoral School for Multidisciplinary Sciences
Science of Vehicles and Mobile Machines

Budapest University of Technology and Economics
Faculty of Transportation Engineering and Vehicle Engineering
Department of Automobiles and Vehicle Manufacturing
Budapest, Hungary

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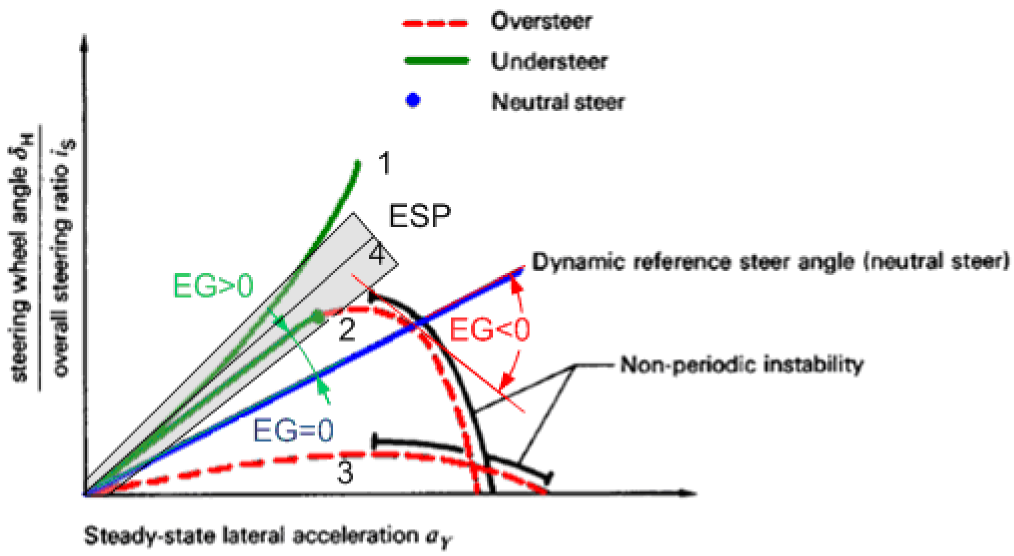
1 Motivation of the research

The currently used vehicle stability control systems use mainly a yaw reference signal to decide that the actual vehicle state (based on the measured yaw rate) is acceptable or not [1], [2], [3]. This acceptance does not mean that the vehicle is stable or not. Vehicle stability and vehicle steerabilty must be separately handled. As it can be seen under in the figure [4] if a yaw rate (in case of steady state lateral acceleration) reference zone is used then stable vehicle situations ("Understeered" vehicle behaviour in the picture) can be corrected and possibly instable vehicle situations ("Oversteer" vehicle behaviour in the picture) can be for a while not corrected. It is because the currently used Electronic Stability Program named systems are in fact not for stability control but for steerabilty control – they do not detect directly vehicle instability, but they maintain the vehicle's yaw rate that results lateral acceleration that is identified by the driver as the vehicle's answer to the steering wheel angle. These systems should be named Electronic Steerabilty Program.



The above mentioned things reveal that the currently used ESP systems are suboptimal from the viewpoint of vehicle stability detection. Furthermore these systems use braking units for interventions – for steerabilty control it is also a suboptimal solution, because this unit can not realize continuous and fine tuneable control signal [5], [6], [7]. To summarize: the meaning of vehicle stability and steerabilty must be separated and the possible main intervention units (braking units, active steering mechanism) must be used according to these separated functions.

To realize this it is a great possibility to use the vehicle sideslip angle – with this signal it is possible to unambiguously define vehicle instability. A problem with vehicle sideslip angle is that it is measureable only with expensive sensors that are not accepted for commercial vehicles due to their costs. Thus the estimation of this signal can be a solution [8], [9], [10], [11]. To estimate vehicle sideslip angle a mathematical indeterminate structure must be solved, furthermore nonlinear vehicle behaviours and tyre characteristics must be also considered (the typical lateral vehicle description is linear). To estimate vehicle sideslip angle in case of commercial vehicles some further problems must be solved: vehicle frame can have even more than 30° roll angle (that easily results even 50% lateral acceleration measurement error), vehicle load (vehicle mass) and centre of gravity can vary in a wide range and due to cost reasons possibly only the currently used sensors can be used. These latter requirements are not considered by the publications in the literature.

If vehicle sideslip angle is also known, then lateral vehicle dynamics can be unambiguously identified – vehicle sideslip angle and yaw rate are the state variables of lateral vehicle dynamics description. With both of the state variables it is possible to define separately and unambiguously the status of vehicle stability and to control vehicle steerabilty. Earlier can be exactly qualified: it can be stable or not. Latter can be just compared to the required vehicle state.

1.1 Dedicated aims

My aims regarding this work can be listed as the followings:

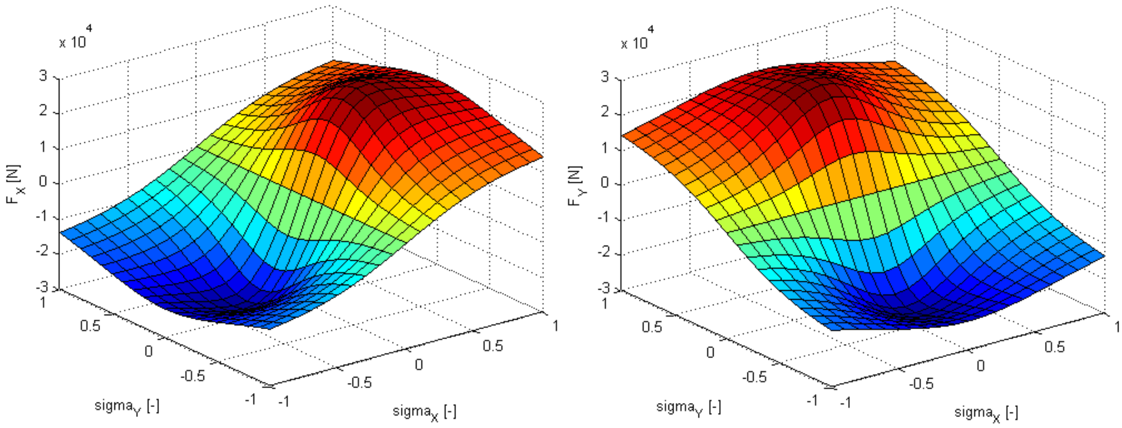
1. Setup more accurate danger recognition conditions for commercial vehicles than nowadays used solutions (definition of vehicle stability). With respect to reference and measured yaw rate values a simple difference can be calculated, and it is not an exact thing to decide that the actual difference should be decreased with intervention or not. Vehicle sideslip angle based stability definition can result exact and absolute conditions regarding safe vehicle state. Oversteered situations could be also earlier recognized sometimes with the observation of vehicle sideslip angle.
2. Development of a lateral vehicle state estimation method that can automatically realize and adapt varying vehicle and environment properties' effects (vehicle load, tyre types, adhesion coefficient, road bank, etc.). The estimation method should contain vehicle sideslip angle estimation, too. This value can't be measured right now with low cost sensors, and the estimation in case of commercial vehicles was not solved until now.
3. With using of all of the above defined techniques development of a brake and steering based electronic stability control. Steering intervention could be realized with steer-by-wire system or active servo engine. Furthermore the control should use the above defined lateral vehicle state estimation method's results to realize an environment-adaptive control signal. The control should use at most the recently used low cost sensors (wheel speed, steering wheel angle, yaw rate and lateral acceleration sensors). It is also an aim for the control that it must operate according the above mentioned definition of vehicle stability. It has to separate the control of vehicle's steerabilty and stability.

To realize these points I decided to use simulation technologies and vehicle measurements together. For simulations I needed to create a good enough and validated vehicle model. So the definition and performing of necessary vehicle measurements had to be done to identify the simulation parameters and verify the realized simulation model. For development and testing I used both real vehicle measurements and simulation technologies.

2 Applied tools and methods

2.1 Complex model and parameter identification

For my work I used simulation technologies [HZ 3] and real measurements together [HZ 1], [HZ 2]. A complex multi-body [12] vehicle model was created [HZ 5] that only required simple definition of bodies, joints and forces between them. For parameter identification and for estimation method and controller validation I used real vehicle measurements. For steering wheel geometry identifying low velocity (about 3m/s) steering wheel angle increasing and decreasing was used. During the tests yaw rate and vehicle sideslip angle was measured with steering wheel angle, vehicle velocity and lateral acceleration. First two ensured that full vehicle state was observable. Tire characteristic identifying was done with circle tests (constant steering wheel angle or steering radius with velocity increasing). Sine steering wheel tests showed how the tyre elasticity can damp the steering wheel's effect. Axle load, vehicle mass, spring characteristics and brake torques were simply measured.



Important part of a vehicle model is the tyre model. I decided to use a brush-model, which is valid up to 10Hz excitations [13], [14]. For a commercial vehicle this frequency is a pneumatic ABS system's upper operating limit, and higher frequencies can not be realized. With this kind of tyre model connection between the several tyre characteristic can be reached – see the figure above for a resulted calculated tyre characteristic.

2.2 Simplified models

For my theoretical work I used linearized, two-axle bicycle models: one for lateral vehicle state estimation and one for vehicle control. The base of these models is the same: they are originated from planar behaviour, yaw rate and vehicle sideslip angle are the state variables.

For lateral vehicle state estimation the main aim was to use as few parameters as possible. Thus e.g. I simplified with vehicle mass; mass specific cornering stiffness parameters and vertical vehicle inertia was used. For vehicle control the main aim was to establish a linear model that can use the general control theories. To make it possible to use directly the estimation model's results in case of the control model, former had to be also linear – but at the same time the estimation method had to estimate nonlinear effects, too.

2.3 Developed controllers and estimation method

The main aim of this work was not to present new control theory methods or very accurate simulation tool with such a control function that performs strictly defined control requirements. The main aim was to find a lateral vehicle state estimation method (mainly vehicle sideslip angle estimation method) that deals with commercial vehicles' special requirements. Further aim was to present a vehicle stability control structure that considers not just control theory but commercial vehicle mechanics. Thus the presented controllers (linear quadratic and adaptive feedforward) do not represent the state of the art of the control theory, but they are built according to the above mentioned vehicle stability control structure.

Regarding the developed estimation method I present "only" the developed estimation method. The development of this estimation method has result a lot of not useable functions that are mentioned during the description of the method (for example explicit or implicit numerical solving of the integration function). The main idea (double circle iteration) can be treated as an axiom; I can not present further background theories for this part. The definition of the used signals and their properties are derived from commercial vehicle mechanics, this part also does not have deeper theoretical background, and it can be handled as a given thing.

3 New scientific results

3.1 Thesis 1: Unambiguous danger detection based on vehicle sideslip angle

The thesis: In case of a two-axle, linearized, lateral bicycle model of a vehicle, BIBO stability criterion can be established by vehicle sideslip angle and its derivative, yaw rate and its derivative, brake system resulted vertical turning torque, front axle cornering stiffness parameter, maximum possible rear axle cornering stiffness parameter, vertical vehicle inertia and vehicle mass and axle distances from centre of gravity.

Consider a lateral vehicle model that can be defined with the following differential equations. Vehicle state variables are vehicle sideslip angle and yaw rate (definitions see under in the table).

$$\begin{cases} \dot{\beta} = -\left(\dfrac{\frac{c_1}{m}+\frac{c_2}{m}}{v_X}\right)\cdot\beta + \left(\dfrac{\frac{c_2}{m}\cdot l_2 - \frac{c_1}{m}\cdot l_1}{v_X^2} - 1\right)\cdot\dot{\psi} + \dfrac{\frac{c_1}{m}}{v_X}\cdot\delta_1 \\ \ddot{\psi} = \left(\dfrac{\frac{c_2}{m}\cdot l_2 - \frac{c_1}{m}\cdot l_1}{I_{ZM}}\right)\cdot\beta - \left(\dfrac{\frac{c_1}{m}\cdot l_1^2 + \frac{c_2}{m}\cdot l_2^2}{I_{ZM}\cdot v_X}\right)\cdot\dot{\psi} + \dfrac{\frac{c_1}{m}\cdot l_1}{I_{ZM}}\cdot\delta_1 + \dfrac{M_Z}{m\cdot I_{ZM}} \end{cases}$$

The state space form of the differential equations can be written as follows.

$$A = \begin{bmatrix} \dfrac{-\frac{c_1}{m}-\frac{c_2}{m}}{v_X} & \dfrac{\frac{c_2}{m}\cdot l_2 - \frac{c_1}{m}\cdot l_1}{v_X^2} - 1 \\ \dfrac{\frac{c_2}{m}\cdot l_2 - \frac{c_1}{m}\cdot l_1}{I_{ZM}} & -\dfrac{\frac{c_1}{m}\cdot l_1^2 + \frac{c_2}{m}\cdot l_2^2}{I_{ZM}\cdot v_X} \end{bmatrix} \qquad x = \begin{bmatrix} \beta \\ \dot{\psi} \end{bmatrix}$$

$$B = \begin{bmatrix} \dfrac{\frac{c_1}{m}}{v_X} & 0 \\ \dfrac{\frac{c_1}{m}\cdot l_1}{I_{ZM}} & \dfrac{1}{I_{ZM}} \end{bmatrix} \qquad u = \begin{bmatrix} \delta_1 \\ \dfrac{M_Z}{m} \end{bmatrix}$$

BIBO stability [15] is performed when linearized bicycle model's state space form's state matrix's roots are in the complex number plane's left half plane.

$$\det(\lambda\cdot I - A) = \begin{vmatrix} \lambda + \dfrac{\frac{c_2}{m}+\frac{c_1}{m}}{v_X} & \dfrac{\frac{c_1}{m}\cdot l_1 - \frac{c_2}{m}\cdot l_2}{v_X^2} + 1 \\ \dfrac{\frac{c_1}{m}\cdot l_1 - \frac{c_2}{m}\cdot l_2}{I_{ZM}} & \lambda + \dfrac{\frac{c_2}{m}\cdot l_2^2 + \frac{c_1}{m}\cdot l_1^2}{I_{ZM}\cdot v_X} \end{vmatrix} = 0$$

This criterion is fulfilled when rear axle's (mass specific) cornering stiffness parameter is proportionally greater than front axle's (mass specific) cornering stiffness parameter [HZ 14].

$$\frac{l_1}{l_2} \cdot \frac{c_1}{m} < \frac{c_2}{m}$$

Based on above criterion and the vehicle description equations, a minimum vehicle sideslip angle value can be calculated for a BIBO stable vehicle (based on (mass specific) front axle cornering stiffness parameter). Furthermore with the using of the maximum possible (mass specific) rear axle cornering stiffness parameter (that fulfils previous criterion) and the vehicle description equations, a maximum possible vehicle sideslip angle value can be calculated for a BIBO stable vehicle.

$$\begin{cases} \beta^{stab1} = \dfrac{\dot{\psi} \cdot l_2}{v_X} - \dfrac{v_X \cdot \left(\dot{\beta} + \dot{\psi}\right) \cdot l_1 - I_{ZM} \cdot \ddot{\psi} + \dfrac{M_Z}{m}}{\dfrac{l_1}{l_2} \cdot \dfrac{c_1}{m} \cdot \left(l_1 + l_2\right)} \\ \beta^{stab2} = \dfrac{\dot{\psi} \cdot l_2}{v_X} - \dfrac{v_X \cdot \left(\dot{\beta} + \dot{\psi}\right) \cdot l_1 - I_{ZM} \cdot \ddot{\psi} + \dfrac{M_Z}{m}}{\dfrac{c_2^{max}}{m} \cdot \left(l_1 + l_2\right)} \end{cases}$$

If the actual vehicle sideslip angle is between the calculated maximum and minimum vehicle sideslip angle values, then the vehicle is BIBO stable.

$$\max\left(\beta^{stab1}, \beta^{stab2}\right) > \beta > \min\left(\beta^{stab1}, \beta^{stab2}\right)$$

Marking	Meaning	Unit
A	linear bicycle model's state space's state matrix	abstract
B	linear bicycle model's state space's state input matrix	abstract
c^{max}_2	rear maximal axle cornering stiffness parameter	N/rad
$c_{1/2}$	front/rear axle cornering stiffness parameter	N/rad
I	identity matrix	-
I_{ZM}	mass specific vertical vehicle inertia	m^2
$l_{1/2}$	front/rear axle distance from centre of gravity	m
m	vehicle mass	kg
M_Z	vertical turning moment	Nm
u	linear bicycle model's state space's input vector	abstract
$v_{X/Y}$	longitudinal/lateral vehicle velocity	m/s
x	linear bicycle model's state space's state vector	abstract
β	vehicle sideslip angle	rad
β^{stab1}	BIBO stable vehicle sideslip angle limit 1	rad
β^{stab2}	BIBO stable vehicle sideslip angle limit 2	rad
δ_1	front axle steered wheel angle	rad
λ	state matrix's root	abstract
Ψ	yaw angle	rad

3.2 Thesis 2: A vehicle sideslip angle estimation method

The thesis: From longitudinal vehicle velocity, steered wheel angle, yaw rate, yaw acceleration and mass specific brake system resulted vertical turning torque as measured signals and axle distances from centre of gravity as parameters, vehicle sideslip angle can be estimated in case a two-axle, front axle steered vehicle.

Consider a lateral vehicle model that can be defined with the following differential equations – definitions see under in the table. Vehicle sideslip angle, its derivative, front axle's mass specific cornering stiffness parameter and rear axle's mass specific cornering stiffness parameter are unknown. Yaw rate, yaw acceleration, front axle's steered wheel angle, longitudinal vehicle velocity and vertical turning moment are measured. Axle distances from centre of the gravity, vertical vehicle inertia and vehicle mass are previously defined (based on nominal vehicle properties, independently from actual vehicle properties).

$$v_X \cdot \left(\dot{\beta}^{est} + \dot{\psi}\right) = \alpha_1^{est} \cdot \left(\frac{c_1}{m}\right)^{est} + \alpha_2^{est} \cdot \left(\frac{c_2}{m}\right)^{est}$$

$$\text{where} \quad \begin{bmatrix}\alpha_1^{est} & \alpha_2^{est}\end{bmatrix} = \begin{bmatrix}\delta_1 - \beta^{est} - \dfrac{\dot{\psi} \cdot l_1^{ob}}{v_X} & -\beta^{est} + \dfrac{\dot{\psi} \cdot l_2^{ob}}{v_X}\end{bmatrix}$$

$$I_{ZM}^{ob} \cdot \ddot{\psi} = \alpha_1^{est} \cdot \left(\frac{c_1}{m}\right)^{est} \cdot l_1^{ob} - \alpha_2^{est} \cdot \left(\frac{c_2}{m}\right)^{est} \cdot l_2^{ob} + \frac{M_Z}{m^{ob}}$$

$$\text{where} \quad \frac{M_Z}{m^{ob}} = \frac{M_{Z1}}{m^{ob}} + \frac{M_{Z2}}{m^{ob}}$$

The vehicle sideslip angle estimation method has four independent main parts:

1. Calculation of vehicle sideslip angle's derivative
2. Saturation of estimated vehicle sideslip angle [HZ 9]
3. Calculation of mass specific front axle cornering stiffness parameter [HZ 1], [HZ 2], [HZ 4], [HZ 10], [HZ 11]
4. Modification of mass specific front axle cornering stiffness parameter

The estimation method operates in a discrete time environment, and vehicle mass is eliminated from the basic equations [HZ 13]:

- instead of cornering stiffness parameters mass specific cornering stiffness parameters are used,
- instead of vertical vehicle inertia mass specific vertical vehicle inertia is used,
- instead of brake system resulted vertical turning torque mass specific brake system resulted vertical turning torque is used and
- instead of vehicle mass a unit is used.

Vehicle description equations are summarized, rear axle's mass specific cornering stiffness parameter is eliminated. With the using of following functions, vehicle sideslip angle can be estimated.

1. The vehicle sideslip derivative's calculation is based on the vehicle description equation as follows in discrete form.

$$\dot{\beta}^{est}[k+1]=\frac{\left(\frac{c_1}{m}\right)^{est}[k]\cdot\left(\delta_1[k]-\beta^{est}[k]-\frac{\dot{\psi}[k]\cdot l_1^{ob}}{v_X[k]}\right)\cdot 1-\ddot{\psi}[k]\cdot I_{ZM}^{ob}+\frac{M_Z[k]}{m^{ob}}}{v_X[k]\cdot l_2^{ob}}-\dot{\psi}[k]$$

where $\beta^{est}[k]=\dot{\beta}^{est}[k]\cdot\Delta t+\beta^{est}[k-1]$

2. Vehicle sideslip angle calculated by the above equation can be limited with the following functions.
   a. Maximum and minimum possible mass specific front and rear axle cornering stiffness parameters can be approximated.

$$\begin{bmatrix}\left(\frac{c_1}{m}\right)^{max} & \left(\frac{c_1}{m}\right)^{min}\\ \left(\frac{c_2}{m}\right)^{max} & \left(\frac{c_2}{m}\right)^{min}\end{bmatrix}=\frac{G}{l\cdot 0.1}\cdot\begin{bmatrix}0.9\cdot l_2^{ob} & 0.1\cdot l_2^{ob}\\ 0.9\cdot l_1^{ob} & 0.1\cdot l_1^{ob}\end{bmatrix}=\frac{9.81}{l}\cdot\begin{bmatrix}9\cdot l_2^{ob} & 1\cdot l_2^{ob}\\ 9\cdot l_1^{ob} & 1\cdot l_1^{ob}\end{bmatrix}$$

   b. With the using of above defined mass specific cornering stiffness limit values and the vehicle model equations, limit vehicle sideslip angles can be calculated in discrete form.

$$\dot{\beta}^{F\,max}[k]=\frac{\delta_1[k]-\frac{\dot{\psi}[k]\cdot l_1^{ob}}{v_X[k]}-\frac{v_X[k]\cdot\dot{\psi}[k]+\frac{I_{ZM}^{ob}\cdot\ddot{\psi}[k]-\frac{M_Z[k]}{m^{ob}}}{l_2^{ob}}}{\frac{l}{l_2^{ob}}\cdot\left(\frac{c_1}{m}\right)^{max}}-\beta^{F\,max}[k-1]}{\Delta t+\frac{v_X[k]}{\left(\frac{c_1}{m}\right)^{max}}\cdot\frac{l_2^{ob}}{l}}$$

where $\beta^{F\,max}[k]=\beta^{F\,max}[k-1]+\dot{\beta}^{F\,max}[k]\cdot\Delta t$

$$\dot{\beta}^{F\,min}[k]=\frac{\delta_1[k]-\frac{\dot{\psi}[k]\cdot l_1^{ob}}{v_X[k]}-\frac{v_X[k]\cdot\dot{\psi}[k]+\frac{I_{ZM}^{ob}\cdot\ddot{\psi}[k]-\frac{M_Z[k]}{m^{ob}}}{l_2^{ob}}}{\frac{l}{l_2^{ob}}\cdot\left(\frac{c_1}{m}\right)^{min}}-\beta^{F\,min}[k-1]}{\Delta t+\frac{v_X[k]}{\left(\frac{c_1}{m}\right)^{min}}\cdot\frac{l_2^{ob}}{l}}$$

where $\beta^{F\,min}[k]=\beta^{F\,min}[k-1]+\dot{\beta}^{F\,min}[k]\cdot\Delta t$

$$\dot{\beta}^{R\max}[k]=\frac{\dfrac{\dot{\psi}[k]\cdot l_2^{ob}}{v_X[k]}-\dfrac{v_X[k]\cdot\dot{\psi}[k]+\dfrac{\dfrac{M_Z[k]}{m^{ob}}-I_{ZM}^{ob}\cdot\ddot{\psi}[k]}{l_2^{ob}}}{\left(\dfrac{c_2}{m}\right)^{\max}\cdot\dfrac{1}{l_1^{ob}}}-\beta^{R\max}[k-1]}{\Delta t+\dfrac{v_X[k]\cdot l_1^{ob}}{\left(\dfrac{c_2}{m}\right)^{\max}\cdot l}}$$

where $\beta^{R\max}[k]=\beta^{R\max}[k-1]+\dot{\beta}^{R\max}[k]\cdot\Delta t$

$$\dot{\beta}^{R\min}[k]=\frac{\dfrac{\dot{\psi}[k]\cdot l_2^{ob}}{v_X[k]}-\dfrac{v_X[k]\cdot\dot{\psi}[k]+\dfrac{\dfrac{M_Z[k]}{m^{ob}}-I_{ZM}^{ob}\cdot\ddot{\psi}[k]}{l_2^{ob}}}{\left(\dfrac{c_2}{m}\right)^{\min}\cdot\dfrac{1}{l_1^{ob}}}-\beta^{R\min}[k-1]}{\Delta t+\dfrac{v_X[k]\cdot l_1^{ob}}{\left(\dfrac{c_2}{m}\right)^{\min}\cdot l}}$$

where $\beta^{R\min}[k]=\beta^{R\min}[k-1]+\dot{\beta}^{R\min}[k]\cdot\Delta t$

c. The calculated vehicle sideslip limits must include the estimated vehicle sideslip angle – the following functions ensure this functionality.

$$\beta^{\max 1}=\min\left(\max\left(\beta^{F\max}[k],\beta^{F\min}[k]\right),\max\left(\beta^{R\max}[k],\beta^{R\min}[k]\right)\right)$$

$$\beta^{\min 1}=\max\left(\min\left(\beta^{F\max}[k],\beta^{F\min}[k]\right),\min\left(\beta^{R\max}[k],\beta^{R\min}[k]\right)\right)$$

$$\beta^{\max 2}[k]=\max\left(\beta^{\max 1}[k],\beta^{\min 1}[k]\right))$$

$$\beta^{\min 2}[k]=\min\left(\beta^{\max 1}[k],\beta^{\min 1}[k]\right)$$

$$\beta^{est}[k+1]:=\max\left(\beta^{\min 2}[k],\min\left(\beta^{\max 2}[k],\beta^{est}[k+1]\right)\right)$$

3. The mass specific front axle cornering stiffness parameter calculation uses the vehicle sideslip angle's (k+1)th value instead of its kth value as follows.

$$\left(\frac{c_1}{m}\right)^{est}[k+1]=\frac{\left(\dot{\psi}[k]+\dot{\beta}^{est}[k]\right)\cdot v_X[k]\cdot l_2+\ddot{\psi}[k]\cdot I_{ZM}-\dfrac{M_Z[k]}{m}}{\left(\delta_1[k]-\beta^{est}[k+1]-\dfrac{\dot{\psi}[k]\cdot l_1}{v_X[k]}\right)\cdot l}$$

a. With the using of above defined vehicle sideslip angle and mass specific front axle cornering stiffness parameter estimation functions, it can be proved that when the actual mass specific cornering stiffness parameter is over a defined limit, then it will be increased, else it will be decreased.

$$
\begin{aligned}
&\text{if} && \frac{\dot{\psi}[k]\cdot v_X[k]\cdot l_2^{ob} + \ddot{\psi}[k]\cdot I_{ZM}^{ob} - \dfrac{M_Z[k]}{m^{ob}}}{\left(\delta_1[k] - \beta^{est}[k] - \dfrac{\dot{\psi}[k]\cdot l_1^{ob}}{v_X[k]}\right)\cdot l} < \left(\frac{c_1}{m}\right)^{est}[k] \\
&\text{then} && \left(\frac{c_1}{m}\right)^{est}[k] < \left(\frac{c_1}{m}\right)^{est}[k+1] \\
&\text{else} && \left(\frac{c_1}{m}\right)^{est}[k] \geq \left(\frac{c_1}{m}\right)^{est}[k+1]
\end{aligned}
$$

4. The mass specific front axle cornering stiffness parameter calculation resulted value is simply multiplied with an empirical gain that is bigger than one – thus the estimated mass specific front axle cornering stiffness parameter will be pushed always in the direction of a well griping front axle, because cornering stiffness parameters are assumed to be low only locally.

$$
\begin{aligned}
&\text{if} && \frac{\dot{\psi}[k]\cdot v_X[k]\cdot l_2^{ob} + \ddot{\psi}[k]\cdot I_{ZM}^{ob} - \dfrac{M_Z[k]}{m^{ob}}}{\left(\delta_1[k] - \beta^{est}[k] - \dfrac{\dot{\psi}[k]\cdot l_1^{ob}}{v_X[k]}\right)\cdot l} < G^{MAFAC}\cdot\left(\frac{c_1}{m}\right)^{est}[k] \\
&\text{then} && G^{MAFAC}\cdot\left(\frac{c_1}{m}\right)^{est}[k] < \left(\frac{c_1}{m}\right)^{est}[k+1] \\
&\text{else} && G^{MAFAC}\cdot\left(\frac{c_1}{m}\right)^{est}[k] \geq \left(\frac{c_1}{m}\right)^{est}[k+1]
\end{aligned}
$$

Marking	Meaning	Unit
$c_{1/2}$	front/rear axle cornering stiffness parameter	N/rad
$(c_1/m)^{est}$	front axle estimated mass specific cornering stiffness	N/rad/kg
$(c_1/m)^{max}$	front axle maximum mass specific cornering stiffness	N/rad/kg
$(c_2/m)^{max}$	rear axle maximum mass specific cornering stiffness	N/rad/kg
$(c_1/m)^{min}$	front axle minimum mass specific cornering stiffness	N/rad
$(c_2/m)^{min}$	rear axle minimum mass specific cornering stiffness	N/rad
G	gravitational acceleration	m/s^2
G^{MAFAC}	mass specific front axle cornering stiffness param. gain	-
I^{ob}_{ZM}	observer mass specific vertical vehicle inertia	m^2
k	step number of discrete time system	-
l	axle distance	m
$l^{ob}_{1/2}$	observer front/rear axle distance from centre of gravity	m
m^{ob}	observer vehicle mass	kg
M_Z	vertical turning moment	Nm
$v_{X/Y}$	longitudinal/lateral vehicle velocity	m/s
β^{est}	estimated vehicle sideslip angle	rad
β^{Fmax}	maximum mass. spec. front axle corn. stiff. related sideslip angle	rad
β^{Fmin}	minimum mass. spec. front axle corn. stiff. related sideslip angle	rad
β^{min1}	minimum of greatest axle sideslip angles	rad
β^{min2}	minimum axle sideslip angle	rad
β^{max1}	maximum of smallest sideslip angles	rad
β^{max2}	maximum axle sideslip angle	rad

Marking	Meaning	Unit
β^{Rmin}	maximum mass. spec. rear axle corn. stiff. related sideslip angle	rad
β^{Rmax}	minimum mass. spec. rear axle corn. stiff. related sideslip angle	rad
Δt	step time	s
δ_1	front axle steered wheel angle	rad
Ψ	yaw angle	rad

3.3 Thesis 3: Lateral vehicle control structure

The thesis: I set up a vehicle stability and steerabilty control system for two-axle, front axle steered vehicles where vehicle stability, vehicle steerabilty and their control are separated in interest of most effective control. Vehicle stability is observed via vehicle sideslip angle and it is controlled with brake system resulted vertical turning torque. Vehicle steerabilty is observed via yaw rate and it is controlled with steering intervention.

To prevent vehicle's self-exciting breakout, a fast and effective intervention is needed – control signal's accuracy is not a primary requirement [HZ 6], [HZ 7], [HZ 8]. At the same time this intervention does not require a full time activity – vehicles are built to be safe and possibly not to realize breakout. Breakout can happen mainly only in forced situations or at local road adhesion loss.

To prevent a vehicle breakout situation where vehicle rotates around its vertical axle, BIBO stability requirements must be ensured. By definition of BIBO stability, bounded inputs must result bounded outputs – in case of a vehicle breakout, a fixed steered wheel angle can result continuously increasing vehicle sideslip angle. To ensure BIBO stability, it can be proved that vehicle sideslip angle must be between two defined limit values – Thesis 1.

$$\begin{cases} \beta^{stab\,min\,1} = \dfrac{\dot{\psi}\cdot l_2}{v_X} - \dfrac{v_X\cdot\left(\dot{\beta}+\dot{\psi}\right)\cdot l_1 - I_{ZM}\cdot\ddot{\psi} + \dfrac{M_Z}{m}}{\dfrac{l_1}{l_2}\cdot\left(\dfrac{c_1}{m}\right)\cdot\left(l_1+l_2\right)} \\ \beta^{stab\,max\,1} = \dfrac{\dot{\psi}\cdot l_2}{v_X} - \dfrac{v_X\cdot\left(\dot{\beta}+\dot{\psi}\right)\cdot l_1 - I_{ZM}\cdot\ddot{\psi} + \dfrac{M_Z}{m}}{\left(\dfrac{c_2}{m}\right)^{max}\cdot\left(l_1+l_2\right)} \end{cases}$$

Furthermore considering vehicle sideslip angle derivatives' physical calculation, it can be seen that brake system resulted vertical turning torque influences vehicle sideslip angle's derivative and vehicle sideslip angle stability limits in reverse directions (practically in the direction of each other). Brake system resulted vertical turning torque must be realized on front axle of the vehicle: braking force on any tyre decreases its lateral capability (front axle's cornering stiffness parameter will be so decreased) and so will be the vertical turning torque's effect in the vehicle sideslip limit calculation gained.

$$\dot{\beta}^{est}[k] = \frac{\left(\dfrac{c_1}{m}\right)^{est}[k]\cdot\left(\delta_1[k]-\beta^{est}[k]-\dfrac{\dot{\psi}[k]\cdot l_1}{v_X[k]}\right)\cdot 1 - \ddot{\psi}[k]\cdot I_{ZM} + \dfrac{M_Z[k]}{m}}{v_X[k]\cdot l_2} - \dot{\psi}[k]$$

A further effect of the brake system is that it decelerates the vehicle. In case of emergency situations after a brake based intervention the driver can perform the same track with lower tyre forces due to the lower vehicle velocity. Lower tyre forces mean lower chance for further vehicle instabilities.

Due to these reasons I built a linear quadratic control that realizes vehicle sideslip angle based stability control with brake units. From the above defined sideslip angle stability limits (see details in Thesis 1) a free play zone can be defined for vehicle sideslip angle: if the actual vehicle sideslip angle is in the zone, then no intervention is realized. If the actual vehicle sideslip angle is outside the zone, then brake system based intervention will be realized. To set up the vehicle sideslip angle stability limits and to get the vehicle sideslip angle's actual value, the estimation method of Thesis 2 is used. This method produces the estimated value of the actual vehicle sideslip angle and it produces the estimated front axle mass specific cornering stiffness parameter that is necessary for the calculation of the sideslip angle limits.

When vehicle sideslip angle is observed to ensure BIBO stable behaviour of the vehicle, then this state variable can be independently from lateral vehicle model's other state (yaw rate) handled. Nevertheless a further control function can control vehicle's yaw rate independently from its vehicle sideslip angle.

To support the driver in the vehicle's driving, it must be ensured that the vehicle a linear and constant behaviour performs. To realize this kind of behaviour, a continuous and accurate intervention is needed – to do this steering system is a great choice. Human body can feel accelerations – but it can not feel velocity. Thus the driver is able to feel lateral acceleration, and not able to feel vehicle sideslip angle. Lateral acceleration depends on vehicle velocity, yaw rate and vehicle sideslip angle's derivative. Latter can be neglected in case of everyday situations – nevertheless it must be kept under control with the stability control. Thus yaw rate is the most relevant at a given vehicle velocity. To summarize: vehicle's steerabilty can be best influenced with steering system and yaw rate control.

Thus I built an adaptive feedforward control for the steering system's control. It uses the results of Thesis 2's estimation method: the estimated mass specific cornering stiffness parameters and the estimated vehicle sideslip angle are considered. With the using of the mass specific cornering stiffness parameters actual tyre properties and adhesion coefficient are also considered. A reference model produces the ideal vehicle state variables. With the using of these state variables and estimated mass specific cornering stiffness parameters, a feedforward control is realized. It produces an ideal steered wheel angle value – considering this and the driver's steered wheel angle, an additive steered wheel angle can be calculated for the active steering mechanism.

Marking	Meaning	Unit
$(c_1/m)^{est}$	front axle estimated mass specific cornering stiffness	N/rad/kg
$(c_2/m)^{max}$	rear axle maximum mass specific cornering stiffness	N/rad/kg
I_{ZM}	mass specific vertical inertia	m^2
k	step number of discrete time system	-
l	axle distance	m
$l_{1/2}$	front/rear axle distance from centre of gravity	m
m	vehicle mass	kg
M_Z	vertical turning moment	Nm
$v_{X/Y}$	longitudinal/lateral vehicle velocity	m/s

Marking	Meaning	Unit
β^{est}	estimated vehicle sideslip angle	rad
$\beta^{stabmin1}$	proportional MAFAC related sideslip stability limit	rad
$\beta^{stabmax1}$	maximum MARAC related sideslip stability limit	rad
δ_1	front axle steered wheel angle	rad
Ψ	yaw angle	rad

4 Future work

The hereby presented stability control system was tested with a real vehicle in real environment. It can be seen that the system is able to control the vehicle's behaviour, but at the same time this ESP is not enough to control a vehicle's dynamic behaviour – tyre behaviour must be also controlled. To reach this aim, two ways are possible:

1. integrate the hereby presented stability control system into an existing ABS system,
2. or develop an own ABS software.

Lateral vehicle stability control software tightly cooperates with anti-lock braking system and it is a mandatory [16] to use ABS [17], [18], [19]. This cooperation can happen in both directions: it is possible that the ESP system requires a brake torque that can not be realized on the given surface and in this case an ABS-active braking must be performed. Or in case of a mu-split situation (when the left and right side of the vehicle run on different adhesion coefficient surface) the ESP can correct the brake force difference resulted vertical turning torque. Thus for vehicle stability control both systems are necessary.

In case of commercial vehicles, ABS systems have also special conditions – the reasons are similar as it was mentioned in case of the vehicle sideslip angle estimation process. For example the same rear axle can have 2ton and 11,5ton vertical load, and this load is not every time known. The simplest commercial vehicles have only 4 wheel speed sensors with 10ms sample rate and that is all we have. It means no brake pressure sensor and no known vehicle velocity. Furthermore with several axle loads the tires' rolling radius change and this can result even 10% velocity measurement error – cca. 10% slip is the ideal braking slip. The height of the centre of the gravity is also unknown and the spring stiffness of the suspension can be variable in case of air suspensions. Thus the pitch dynamic of the commercial vehicles is also almost fully unknown – but it influences the vertical tyre load during braking.

Two things can be seen: ABS and ESP systems must cooperate and commercial vehicles require special solutions because their operation conditions do not let us to use the systems that are derived from passenger vehicles. My future aim is to develop an ABS software that cooperates with the here presented ESP software and deals with commercial vehicle's special requirements. To realize this, I would like to use simulation tools [HZ 3], [HZ 12], too. Thus my further aim is to improve the hereby presented vehicle simulation model [HZ 5], especially the tyre model. Thus the presented stability control software can cooperate with such an ABS software that can be developed especially for this ESP (and not just cooperates with a general ABS software).

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