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Modeling the Driver Behavior during Critical Traffic Scenarios by means of the Theory of Elastic Bands

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Abstract: In the last two decades, the automotive industry has focused on the improvement of vehicle's and occupant's safety. Therefore, a variety of safety systems has been developed. For the development of such safety systems, the computer simulation has become meanwhile more and more important. For the computer simulation the vehicle dynamics, the sensors and the controllers of safety systems have to be reproduced. Furthermore, the active input of the driver has to be modeled. In this paper, the modeling of the driver's behavior and his active intervention during critical traffic scenarios is presented. Therefore, his active tasks, like steering, accelerating and braking, have to be mapped. The path planning of the evasion maneuvers is realized by the theory of elastic bands where obstacles are modeled by repulsive potentials. Additionally, the necessary vehicle velocity has to be calculated. To determine the required driver's intervention while following the planned path, the vehicle dynamics has to be considered. This is modeled with help of multibody systems in the three dimensional vehicle dynamic simulation environment FASIM_C++. *Finally, simulation results for a typical critical traffic situation are presented and discussed.*

Keywords: collision free path planning, path control, safety systems, vehicle dynamics, multibody systems

NOMENCLATURE

$\underline{b}$ = generalized gyroscopic forces, N or Nm
c_{ext} = spring constant for repulsion, $\mathrm{kg\,s^{-2}}$
c_{int} = spring constant for contraction, $\mathrm{kg\,s^{-2}}$
$\underline{d}_k$ = distance vector between obstacle and node μ_k, m
f = number of degrees of freedom, dimensionless
$\underline{F}_{ext_k}$ = spring force between obstacle and node μ_k, N
$\underline{F}_i$ = applied force, N
$\underline{F}_{int_{k-1,k}}$ = spring force between nodes μ_{k-1} and μ_k, N
i_{steer} = transmission ratio of steering mechanism, dimensionless
m = number of obstacles, dimensionless
m_i = mass of body i, kg
$\underline{M}$ = generalized mass matrix, kg or $\mathrm{kg\,m^2}$
n = number of nodes, dimensionless
$\underline{q}$ = vector of generalized coordinates, m or deg
$\underline{Q}$ = generalized applied forces, N or Nm
r_0 = safety radius around an obstacle, m
$\hat{\underline{\dot{r}}}_i^{(j)}$ = pseudo velocity, $\mathrm{m\,s^{-1}}$
$\hat{\underline{\ddot{r}}}_i$ = pseudo acceleration, $\mathrm{m\,s^{-2}}$
$\underline{r}_k$ = position vector of node μ_k, m
$\underline{r}_{obst}$ = position vector of obstacle, m
$s_{evasion}$ = length of evasion trajectory, m
s_{obst} = margin of obstacle relative to vehicle, m
$s_{original}$ = length of original trajectory, m
t_{end} = end time of evasion maneuver, s
t_{hit} = time of collision, s
$\underline{T}_{S_i}$ = applied torque, Nm
t_{start} = start time of evasion maneuver, s
$\underline{u}_k$ = displacement of node μ_k, m
$v_\emptyset$ = effective velocity of vehicle, $\mathrm{m\,s^{-1}}$
$v_{current}$ = current velocity of vehicle, $\mathrm{m\,s^{-1}}$
v_{obst} = velocity of obstacle, $\mathrm{m\,s^{-1}}$
v_{target} = target velocity of vehicle, $\mathrm{m\,s^{-1}}$

Greek Symbols

δ = steering angle, deg
δ_{acc} = position of acceleration pedal, deg
δ_{brake} = position of braking pedal, deg
δ_f = steering angle of front wheel, deg
$\delta\underline{\phi}_{S_i}$ = virtual angular displacement, deg
$\delta\underline{r}_{S_i}$ = virtual linear displacement, m
μ_k = node of trajectory, dimensionless
$\hat{\underline{\omega}}_i^{(j)}$ = pseudo velocity, $\mathrm{m\,s^{-1}}$
$\hat{\underline{\dot{\omega}}}_i$ = pseudo acceleration, $\mathrm{m\,s^{-2}}$
$\underline{\Theta}_{S_i}$ = inertia tensor of body i, $\mathrm{kg\,m^2}$

Subscripts

E = relative to inertial system $\mathscr{K}_E$
V = relative to vehicle system $\mathscr{K}_V$

INTRODUCTION

In the last two decades, the automotive industry has focused on the improvement of vehicle's and occupant's safety. Therefore, a variety of safety systems has been developed. The active safety systems assist the driver in critical situations. One example is the Electronic Stability Program (ESP), which stabilizes the car by braking wheels individually. Active safety systems are specified to prevent from an accident. On the other hand, there are the passive safety systems, which should reduce the consequences of an accident for occupants. The occupant restraint systems, like airbags and belt pretensioners, protect the occupants against severe injuries. In order to further increase the safety on roads the existing systems are continuously improved, and new assistant and occupant protection systems are developed. This trend will certainly lead in the future towards vehicles with versatile and powerful driver assistant systems and as a consequence to semi autonomous vehicles on the long term.

But this trend poses some problems. The interaction of the numerous safety systems has to be well organized and coordinated. Therefore, the influences of safety systems on the vehicle dynamics are to be investigated, especially in critical driving maneuvers, like a rollover maneuver or an evasion maneuver. As a consequence, the analysis of a large number of different maneuvers is required.

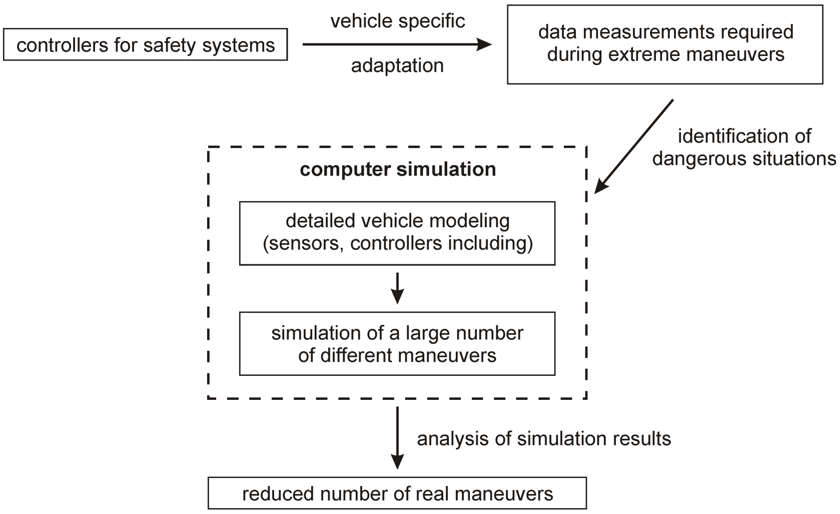

Figure 1 – Vehicle dynamic simulations to support the controller development.

These investigations are only feasible by using sophisticated and efficient simulation tool. Computer simulation has therefore become meanwhile more and more important, since it provides a cost efficient and comfortable tool in order to reduce the number of expensive and complex experiments (Fig. 1). For the investigation of different driving maneuvers, a complex simulation environment is needed (Fig. 2), which is able to reproduce the vehicle dynamics, the sensors and the controllers of active and passive safety systems. Furthermore, the active input of the driver and passive occupant's behavior can be modeled.

In this paper, a simulation environment is presented. It combines three existing simulation tools: the multibody system library MOBILE(Kecskeméthy and Hiller, 1994), the vehicle dynamic simulation environment FASIM_C++ (Pichler, 1999) and the commercial simulation tool MATLAB/SIMULINK® (The MathWorks, 2000). This simulation environment enables the user to simulate and analyze the dynamic behavior of both: vehicle and occupants. Thus, he gets a general idea of the overall system behavior for the development of safety systems.

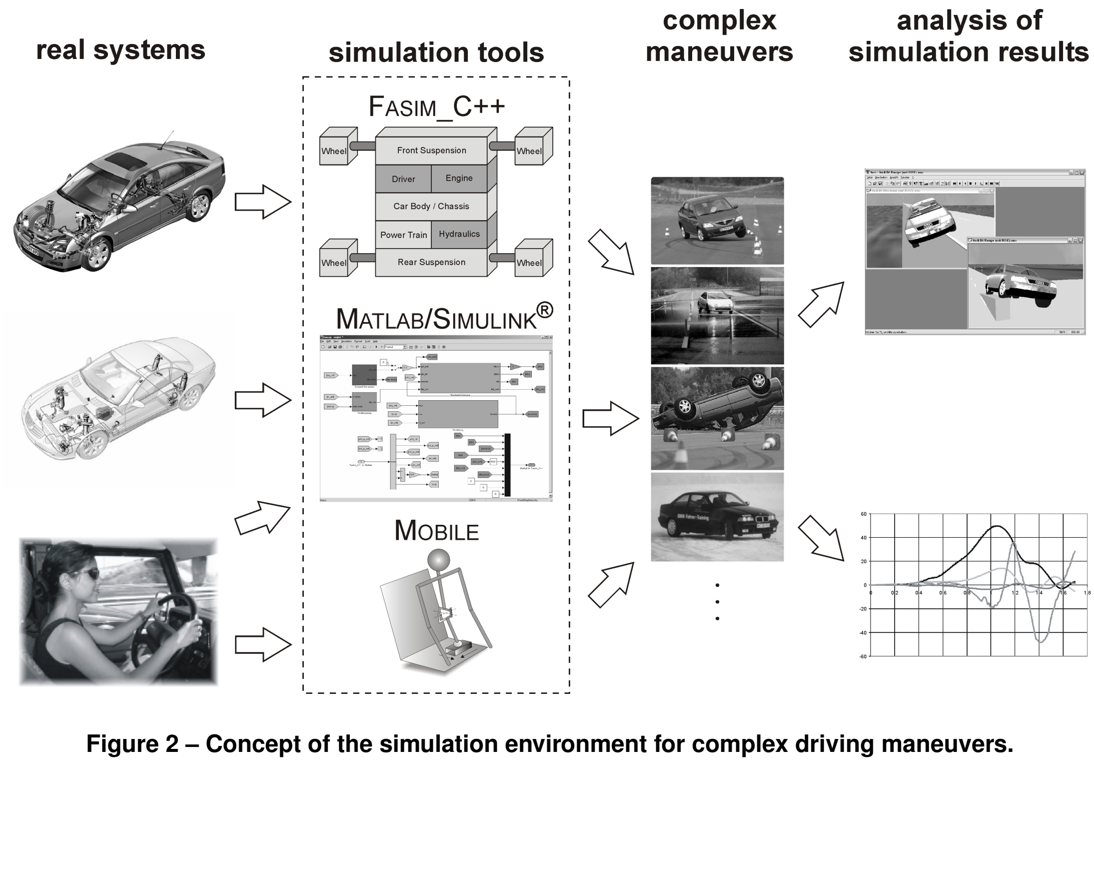

Figure 2 – Concept of the simulation environment for complex driving maneuvers.

For the simulation the vehicle dynamics has to be modeled on the computer. Here, the mechanical part of the vehicle is modeled as a complex multibody system (Fig. 7), which is completed by additional non-mechanical components, like hydraulics, driver and environment, to an overall mechatronical vehicle model by means of the three-dimensional vehicle dynamic simulation environment FASIM_C++. The implementation in the object-oriented programming language C++ provides for mapping the modular vehicle structure on the simulation model. In this way, an easy model adaptation is possible for the investigation of different vehicle types.

The sensors and controllers of the active safety systems, for example the ESP, and of the passive safety systems, like the belt pretensioners, are implemented in MATLAB/SIMULINK®. There is a permanent co-simulation between them and the vehicle model in FASIM_C++. The required state variables of the vehicle, for example yaw rate and slip angle of the vehicle for the ESP controller, are transmitted from the vehicle model to the controller models, which calculates the necessary control variables, in this case a braking torque for one of the wheels. The control variables are then sent back to the vehicle model.

This simulation environment enables the investigation of complex driving maneuvers with regard to the vehicle's and occupants' behavior. One possible application is to use it for the analysis of accident scenarios. Thus, a large spectrum of different accidents can be simulated, for example skidding maneuvers, deviating from the carriageway, vehicle rollovers (Hirsch, Schramm and Hiller, 2006) and evasion maneuvers (Hirsch et al., 2005). The presented simulation environment is characterized by the ability to simulate maneuvers, which are a combination of simple maneuvers considering the driver input. Furthermore, the simulation of the safety systems allows the investigation of their influence on the sequences of accidents events. Consequently, the user is able to oppose the accident effects without and with intervention of one or more safety systems. In addition, the motion of the bodies of all occupants can be determined, so that the risk of injury can be analyzed, what is especially important for the development of occupant restraint systems.

COLLISION FREE PATH PLANNING

The planning of collision free paths is based on the theory of elastic bands which was originally used to determine collision free paths for robots. Based on this method the trajectory of the controlled vehicle is determined. In addition, the timing velocity run has to be given for describing a driving maneuver (Lenthaparambil, 2006).

Theory of elastic Bands

The principle of the path planning with elastic bands is illustrated in Fig. 3. The path is planned in the inertial coordinate system $\mathscr{K}_E$. The original trajectory of the vehicle is characterized by $n+1$ equidistant nodes $\mu_0, \ldots, \mu_n$ connected by n springs (spring constant c_{int}) which account for the contraction of the elastic band. In this position, the springs are

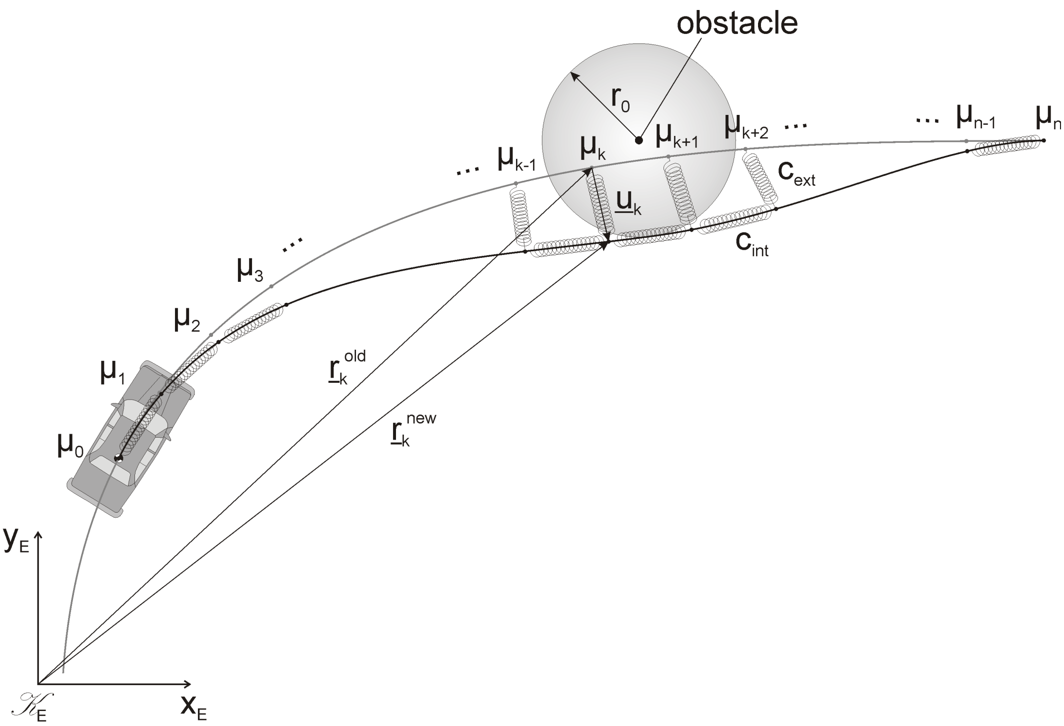

Figure 3 – Principle of the path planning with elastic bands.

not stressed, that means the spring forces $\underline{F}_{int_{0,1}}, \ldots \underline{F}_{int_{n-1,n}}$ are zero. Otherwise, the spring forces are given by:

$$\underline{F}_{int_{k-1,k}} = c_{int}\,(\underline{u}_{k-1} - \underline{u}_k)\ , \quad k = 1, \ldots, n\ , \tag{1}$$

where the vector $\underline{u}_k$ is a possible displacement of the node μ_k.

An obstacle is mapped by a repulsive potential field realized by a spring potential (spring constant c_{ext}). Consequently, a node μ_k, which lies in the defined safety radius r_0 around the obstacle, is shifted by the spring force $\underline{F}_{ext_k}$ to the outside of the safety circle. Because of the inner contraction of the elastic band, all nodes except the both fixed boundary points (assumption $\underline{u}_0 = \underline{u}_n = 0$) are re-positioned. The displacement vectors $\underline{u}_1, \ldots, \underline{u}_{n-1}$ of the nodes $\mu_1, \ldots, \mu_{n-1}$ are then calculated as:

$$c_{int} \begin{bmatrix} 2 & -1 & & & \\ -1 & 2 & -1 & & \\ & \ddots & \ddots & \ddots & \\ & & -1 & 2 & -1 \\ & & & -1 & 2 \end{bmatrix} \begin{bmatrix} \underline{u}_1^T \\ \underline{u}_2^T \\ \vdots \\ \underline{u}_{n-2}^T \\ \underline{u}_{n-1}^T \end{bmatrix} = - \begin{bmatrix} \underline{F}_{ext_1}^T \\ \underline{F}_{ext_2}^T \\ \vdots \\ \underline{F}_{ext_{n-2}}^T \\ \underline{F}_{ext_{n-1}}^T \end{bmatrix} . \tag{2}$$

The repulsive spring forces are given by:

$$\underline{F}_{ext_k} = \begin{cases} -c_{ext}\left(r_0 \frac{\underline{d}_k}{\|\underline{d}_k\|_2} - \underline{d}_k\right) , & \|\underline{d}_k\|_2 < r_0 \\ \underline{0} , & \|\underline{d}_k\|_2 \geq r_0 \end{cases} , \quad k = 1, \ldots, n-1\ , \tag{3}$$

where $\underline{d}_k := \underline{r}_k - \underline{r}_{obst}$ describes the distance between the obstacle and the node μ_k. The linear equation system of Eq. (2) can be solved with help of the GAUSS elimination including a LU decomposition. There, the compute time can be reduced, if the tridiagonal shape of the coefficient matrix is considered.

The solutions $\underline{u}_k$, $k = 1, \ldots, n-1$, of the linear equation system and the assumption $\underline{u}_0 = \underline{u}_n = 0$ yield the new position vectors of all nodes:

$$\underline{r}_k^{new} := \underline{r}_k^{old} + \underline{u}_k\ , \quad k = 0, \ldots, n\ . \tag{4}$$

Optimization of the Trajectory

The following figure shows two examples for trajectories planned with the theory of elastic bands. The original path is a straight line presented by the dashed line. The other both plots present elastic bands which should avoid four obstacles with different safety radii. The dash-dot line is calculated with the spring constants:

$$c_{int} = 500\frac{\mathrm{N}}{\mathrm{m}}\ , \quad c_{ext_1} = c_{ext_2} = c_{ext_3} = c_{ext_4} = 0.02\frac{\mathrm{N}}{\mathrm{m}}\ . \tag{5}$$

This obvious that the path is not collision free. The dash-dot line intersects the first safety radius. At the three other obstacles, the trajectory lies far away from the safety radii.

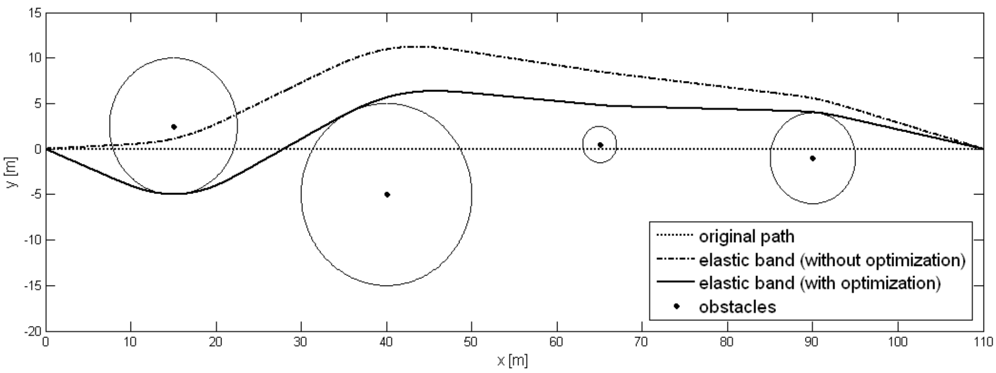

Figure 4 – Trajectories generated with elastic bands.

The optimization of the path solves these problems. Therefore, the length of the elastic band should be minimized while the safety radii are not pass through. That should be achieved by varying all spring constants c_{int} and c_{ext} – in this example these are c_{int} and $c_{ext_1}, c_{ext_2}, c_{ext_3}, c_{ext_4}$. So, the quality function

$$Q_{path}(c_{int}, c_{ext_1}, \ldots, c_{ext_m}) = \sum_{k=1}^{n} \|\underline{r}_k - \underline{r}_{k-1}\|_2 \tag{6}$$

has to be minimized. Thereby, the index m is the number of obstacles. In addition, the boundary conditions

$$g_i(c_{int}, c_{ext_1}, \ldots, c_{ext_m}) = d_{0_j} - \|\underline{r}_k - \underline{r}_{obst_j}\|_2 \leq 0\,, \quad i = 1, \ldots, m(n-1)\,, \quad j = 1, \ldots, m\,, \quad k = 1, \ldots, n-1\,, \tag{7}$$

prevent that the elastic band contracts to a straight line.

The problem is solved with help of LAGRANGE multipliers $\underline{\lambda} \in \mathbb{R}^{m(n-1)}$. For the LAGRANGE function

$$\mathscr{L}(\underline{c}, \underline{\lambda}) = Q_{path}(\underline{c}) + \underline{g}^T \cdot \underline{\lambda} \tag{8}$$

with $\underline{c} := [c_{int}, c_{ext_1}, \ldots, c_{ext_m}]^T$ and $\underline{g} := [g_1(\underline{c}), \ldots, g_{m(n-1)}(\underline{c})]^T$, a minimum is detected with help of the Sequential Quadratic Programming (SQP) – a quasi NEWTON method (Luenberger, 1937). A minimum $(\underline{c}^*, \underline{\lambda}^*)$ of the LAGRANGE function in Eq. (8) meets the necessary conditions

$$\nabla_{\underline{c}} \mathscr{L}(\underline{c}^*, \underline{\lambda}^*) \;=\; \nabla_{\underline{c}} Q_{path}(\underline{c}^*, \underline{\lambda}^*) - \left(\frac{\partial \underline{g}}{\partial \underline{c}^*}\right)^T \underline{\lambda} \;=\; 0\,, \tag{9}$$

$$\nabla_{\underline{\lambda}} \mathscr{L}(\underline{c}^*, \underline{\lambda}^*) \;=\; \underline{g}(\underline{c}^*) \;=\; \underline{0} \tag{10}$$

and the sufficient condition

$$\underline{v}^T \underline{H}(\underline{c}^*, \underline{\lambda}^*) \underline{v} > 0\,, \tag{11}$$

where $\underline{H}$ presents the HESSE matrix of the LAGRANGE function $\mathscr{L}$.

With the SQP method, the elastic band can be optimized. The optimization provides the following spring constants for the inner contraction and the external repulsion:

$$c_{int} = 499.9918\frac{\mathrm{N}}{\mathrm{m}}\,, \quad c_{ext_1} = 0.0456\frac{\mathrm{N}}{\mathrm{m}}\,, \quad c_{ext_2} = 0.0203\frac{\mathrm{N}}{\mathrm{m}}\,, \quad c_{ext_3} = 0.0420\frac{\mathrm{N}}{\mathrm{m}}\,, \quad c_{ext_4} = 0.0211\frac{\mathrm{N}}{\mathrm{m}}\,. \tag{12}$$

From this spring constants, the collision free trajectory results, shown in Fig. 4 as solid line. This new trajectory drives round all obstacles and safety radii.

Modeling of a moving Obstacle

For the planning of evasion maneuvers considering the vehicle dynamics, it is expedient, vehicles, which act as obstacles, cannot be represented with one repulsive potential only. An adaptation to the geometry of the vehicle by using four repulsive potentials is preferred. Figure 5 shows such a model of a vehicle as an obstacle.

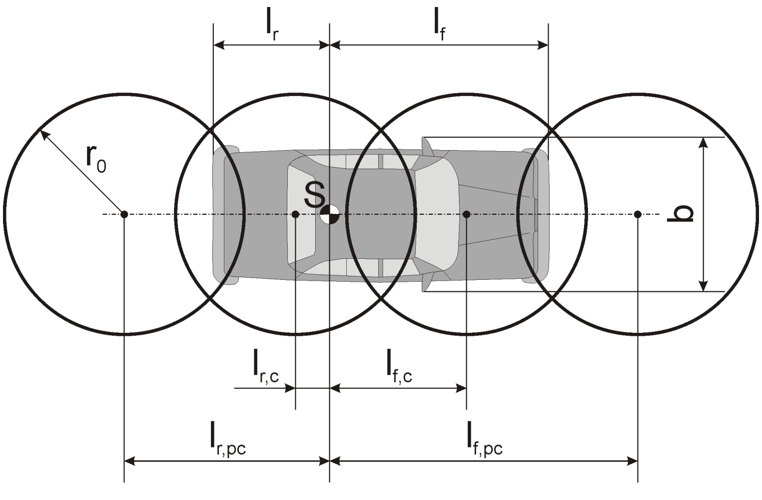

Figure 5 – Modeling an obstacle vehicle.

The length of the obstacle is described by the width b and the length $l = l_f + l_r$, where l_f is the distance between the center of mass S of the vehicle and the vehicle front side and l_r the distance between S and the vehicle rear end.

The two potentials in the middle represent the crash zone. Their center points are positioned on the longitudinal axle of the vehicle with the distances $l_{f,c}$ and $l_{r,c}$ from the center of mass S of the vehicle. The whole vehicle has to be covered by this two inner potentials. The outer potentials act as pre-crash zones. Their center points are positioned in front of the vehicle ($l_{f,pc}$) and behind it ($l_{r,pc}$), respectively, on the longitudinal axle of the vehicle. The diameter $2r_0$ of the safety circle of the repulsive potentials must be at least as large as the width b of the obstacle plus the half width of the controlled vehicle.

In addition, the velocity of the obstacle v_{obst} and its position $\underline{r}_{obst}$ to the controlled vehicle are given, for example by sensors. Is is assumed, that the velocity v_{obst} of the obstacle is constant.

Planning Method for Evasion Maneuvers

At least two objects are involved in an evasion maneuver: the controlled vehicle and the obstacle, for example a second vehicle. With help of the trajectory planned with an elastic band the controlled vehicle is able to pass the obstacle without collision. Based on the known values: original path of the controlled vehicle $\underline{r}_k$, $k = 1, \ldots, n-1$, its current velocity $v_{current}$, the position of the obstacle $\underline{r}_{obst}$ and its constant velocity v_{obst}, the collision free evasion maneuver including trajectory and velocity can be calculated in nine steps:

1. Determination of the time of collision:
$$t_{hit} = t_{start} + \frac{s_{obst}}{v_{current} - v_{obst}} \,. \tag{13}$$
where s_{obst} is the margin of the obstacle relative to the controlled vehicle.
2. Calculation of the end of the evasion maneuver:
$$t_{end} = 2t_{hit} - t_{start} \,, \tag{14}$$
that means the obstacle should be passed at the midterm of the evasion maneuver.
3. Discretization of the original path:
the original path between t_{start} and t_{end} is substituted by $n+1$ equidistant nodes μ_k, $k = 0, \ldots, n$, located on the original path.
4. Modeling of the obstacle:
positioning of the four safety radii around the obstacle vehicle (Fig. 5).
5. Planning of the evasion trajectory with the theory of elastic bands:
calculating the position of the elastic band at the time of collision t_{hit} and optimizing the path length by varying the spring constants $c_{ext_1}, \ldots, c_{ext_4}$ and c_{int}.
6. Planning of the vehicle velocity:
in general, the evasion trajectory is longer than the original one, so the velocity has to be varied to finish the evasion maneuver at the time t_{end}.

Optimization of the Vehicle Velocity

The goal of the optimization of the vehicle velocity is to retain the maneuver time. The controlled vehicle should require the same time for the possibly longer evasion trajectory as for the original one. Hence, it is incidental a higher effective velocity during the evasion maneuver than before.

For the calculation of the velocity, the position of the nodes μ_k, $k = 0, \ldots, n$, has to be modified at the points of time t_k, $k = 0, \ldots, n$ (with $t_0 := t_{start}$ and $t_n := t_{end}$). First, the length of the evasion trajectory has to be calculated:
$$s_{evasion} = \sum_{k=1}^{n} \|\underline{r}_{k-1}^{new} - \underline{r}_k^{new}\|_2 \,. \tag{15}$$
Based on this, the effective velocity is given by:
$$v_\emptyset = \frac{s_{evasion}}{t_n - t_0} \,, \tag{16}$$
where the start time is assumed as $t_0 = 0\,\mathrm{s}$ for simplification and the wanted end time is detected with help of the current velocity v_{curent} of the vehicle at the time of detection of the obstacle and the length of the original trajectory $s_{original}$:
$$t_{end} = \frac{s_{original}}{v_{current}} \,. \tag{17}$$

The required target velocity v_{target} has to be continuously differentiable. Here, the preset for velocity is created with the GAUSS error distribution curve:
$$v_{gauss}(t_k) = \mathrm{e}^{-\frac{(t_k - t_{hit})^2}{2\sigma^2}} - \mathrm{e}^{-\frac{(t_0 - t_{hit})^2}{2\sigma^2}} \,, \quad k = 0, \ldots, n \,. \tag{18}$$
The maximum of the velocity v_{gauss} is given at the time t_{hit}. The parameter σ denotes the inflection points of the curve. Hence, the velocity of the planned evasion maneuver is given by:
$$v_{target}(t_k) = \frac{v_\emptyset}{\frac{1}{t_n - t_0} \sum_{i=0}^{n} v_{gauss}(t_i)} v_{gauss}(t_k) \,, \quad k = 0, \ldots, n \,. \tag{19}$$

Based on the calculated velocity vector $\underline{v}_{target} = [v_{target}(t_0), \ldots, v_{target}(t_n)]^T$ and the given equidistant time vector $\underline{t} = [t_0, \ldots, t_n]^T$, the nodes $\underline{\mu} = [\mu_0, \ldots, \mu_n]^T$ of the elastic band have to be replaced with help of a spline interpolation by the previously calculated new positions $\underline{r}_k^{new}$, $k = 0, \ldots, n$, from Eq. (4).

VEHICLE DYNAMIC SIMULATION

Development of vehicle controllers requires an appropriate model of the vehicle dynamics built into a versatile simulation environment (Bertram et al., 2003). This simulation environment has to be able to simulate different vehicle types or models without any recompilation. The vehicle model has to have a modular form so that single component of the vehicle may be exchanged, depending on the simulation task. Thus, models of the vehicle dynamics with differing levels of complexity can be defined covering correspondent physical effects with the desired accuracy. The modular structure of a vehicle model in FASIM_C++ is shown in Fig. 6 using the example of a passenger car.

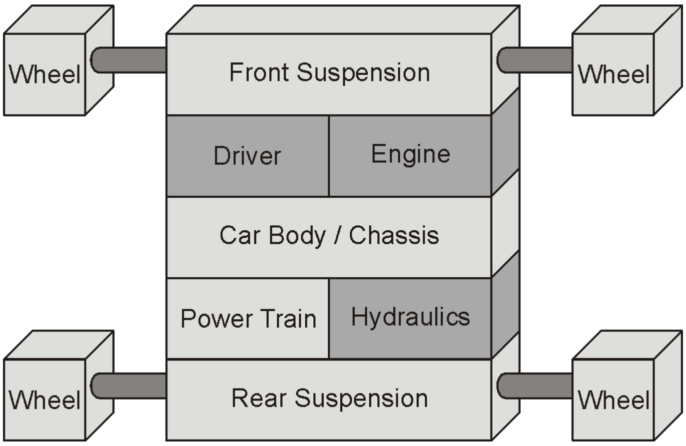

Figure 6 – Modular structure of a passenger car in FASIM_C++.

The structure presented does not show the design details of the modules, e. g. which kind of front suspension is used. During initialization this is not important, because the required information for generating the equations of motion is part of the modules and only at the beginning of simulation is it evaluated. For example, the topology of an all-wheel driven upper class passenger car, which describes the kinematic topology of the individual modules, is shown in Fig. 7. For reasons of clarity the modules engine hydraulics (braking system), driver and environment are not shown. Using this modeling technique it is possible to decide during runtime which configuration of a vehicle is used without any recompilation of the program.

FASIM_C++ contains a large library of different vehicle modules such as suspensions, tire models, drive trains, engines, engine mounts, controllers, sensors, elasticities, a rigid or flexible car body, several hydraulic braking systems, a driver and an environment model. The structure of the modules makes it easy to expand the library by adding new modules. The

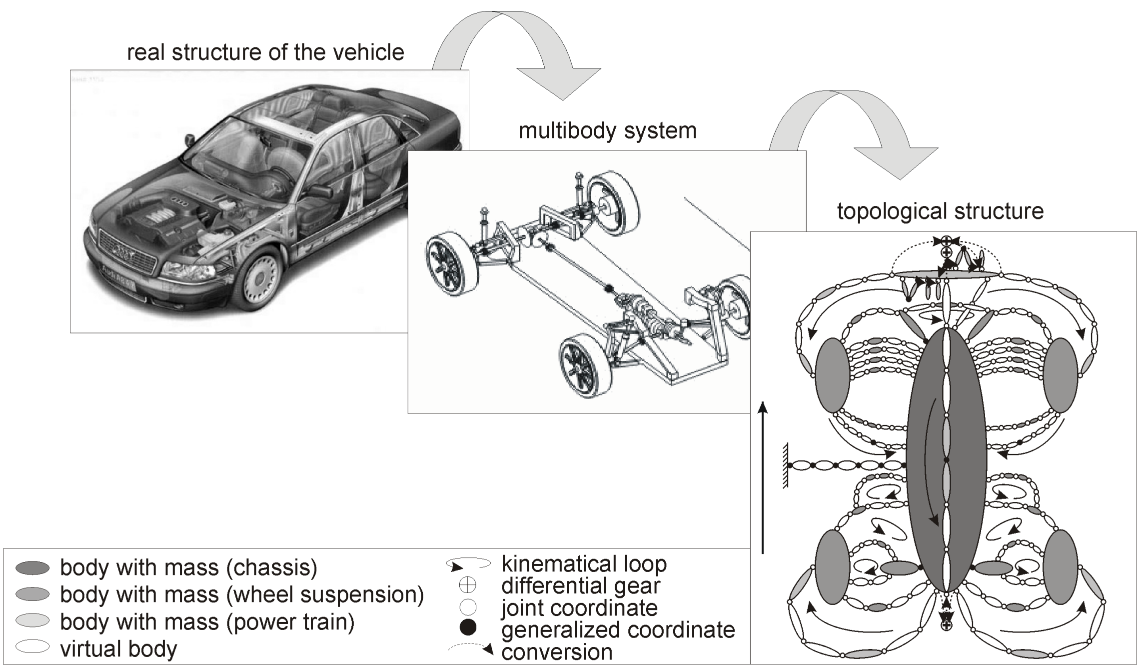

Figure 7 – Kinematic topology of an all-wheel driven upper class passenger car.

equations of motion are based on D'ALEMBERT's principle:

$$\sum_{i=1}^{n_B}(m_i\ddot{\underline{r}}_{S_i}-\underline{F}_i)\cdot\delta\underline{r}_{S_i}+(\underline{\Theta}_{S_i}\dot{\underline{\omega}}_i+\underline{\omega}_i\times\underline{\Theta}_{S_i}\dot{\underline{\omega}}_i-\underline{T}_{S_i})\cdot\delta\underline{\varphi}_i=0\,, \tag{20}$$

where n_B is the number of mass-endowed bodies, m_i, $\underline{\Theta}_{S_i}$ are the mass and inertia tensor of body i, $\ddot{\underline{r}}_{S_i}$ the acceleration of center of gravitation, $\underline{F}_i$, $\underline{T}_{S_i}$ the applied force and torque, and $\delta\underline{r}_{S_i}$, $\delta\underline{\varphi}_i$ are the virtual linear and angular displacement.

Due to the constraints in the system, the virtual displacements are not independent. To generate the equations of motion in minimal coordinates the choice of f independent generalized coordinates $q_1,q_2,\ldots,q_f$, is necessary, corresponding to the number of degrees of freedom in the system. The equations of motion of the mechanical system in minimal coordinates can then be written as:

$$\underline{M}(\underline{q})\ddot{\underline{q}}+\underline{b}(\underline{q},\dot{\underline{q}})=\underline{Q}(\underline{q},\dot{\underline{q}},t)\,, \tag{21}$$

where $\underline{M}$ is the generalized mass matrix, $\underline{b}$ the generalized gyroscopic forces, $\underline{q}$ the generalized coordinates, and $\underline{Q}$ the generalized applied forces.

Applying the principle of kinematic differentials, the elements of the equations of motion are calculated expressing partial derivatives using kinematic terms (Hiller and Kecskeméthy, 1989). Due to the modular structure of the matrices and vectors, their elements can easily be calculated from the corresponding modules. For this reason they are subdivided into an inner sum, inside the module l considering all its bodies n_B and in an outer sum considering all modules n_M.

$$\begin{aligned}
\underline{M}_{j,k} &= \sum_{l=1}^{n_M}\sum_{i\in I_l}\left[m_i\hat{\underline{r}}_i^{(j)}\cdot\hat{\underline{r}}_i^{(k)}+\hat{\underline{\omega}}_i^{(j)}\cdot(\underline{\Theta}_i\hat{\underline{\omega}}_i^{(k)})\right]\,,\\
\underline{b}_j &= \sum_{l=1}^{n_M}\sum_{i\in I_l}\left[m_i\hat{\underline{r}}_i^{(j)}\cdot\hat{\hat{\underline{r}}}_i+\hat{\underline{\omega}}_i^{(j)}\cdot(\underline{\Theta}_i\hat{\hat{\underline{\omega}}}_i+\underline{\omega}_i\times\underline{\Theta}_i\underline{\omega}_i)\right]\,,\\
\underline{Q}_j &= \sum_{l=1}^{n_M}\sum_{i\in I_l}\left[\hat{\underline{r}}_i^{(j)}\cdot\underline{F}_i+\hat{\underline{\omega}}_i^{(j)}\cdot\underline{T}_{S_i}\right]\,.
\end{aligned} \tag{22}$$

The pseudo velocities $\hat{\underline{r}}_i^{(j)}$, $\hat{\underline{\omega}}_i^{(j)}$ and pseudo accelerations $\hat{\hat{\underline{r}}}_i$, $\hat{\hat{\underline{\omega}}}_i$ are defined as follows:

$$\begin{aligned}
\hat{\underline{r}}_i^{(j)} &= \frac{\partial\underline{r}_i}{\partial q_j}\,, \quad \hat{\hat{\underline{r}}}_i = \sum_{j=1}^{f}\sum_{k=1}^{f}\frac{\partial^2\underline{r}_i}{\partial q_j\partial q_k}\dot{q}_j\dot{q}_k\,,\\
\hat{\underline{\omega}}_i^{(j)} &= \frac{\partial\underline{\omega}_i}{\partial\dot{q}_j}\,, \quad \hat{\hat{\underline{\omega}}}_i = \sum_{j=1}^{f}\frac{\partial\underline{J}_{\omega i}}{\partial q_j}\dot{\underline{q}}\dot{q}_j\,, \quad \underline{J}_{\omega i} = \frac{\partial\underline{\omega}_i}{\partial\dot{\underline{q}}}\,.
\end{aligned} \tag{23}$$

PATH CONTROL

For the planning of a driving maneuver, along with the trajectory and the velocity also the active inputs of the driver are required. This includes the steering angle δ and the position of the acceleration pedal δ_{acc}. Both can be determined with help of a path control for the complex vehicle model in FASIM_C++ (Jörißen, 2006).

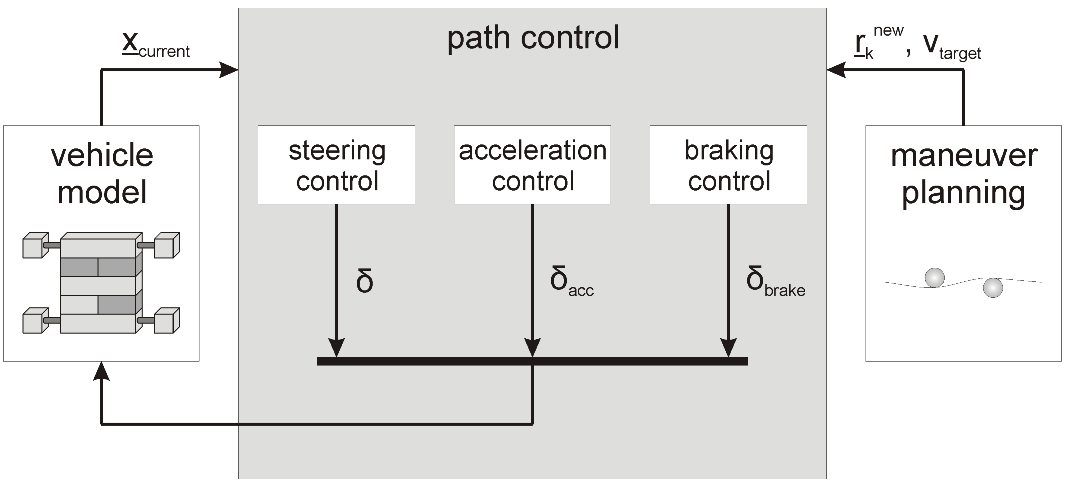

Figure 8 – Concept of the path control.

For the steering control, a lead point is chosen. Its distance to the vehicle front side is depend on the velocity. The faster the velocity is, the farther the lead point lies ahead. The chosen steering angle δ_f for the front wheels is calculated with the difference between the current track angle and the angle between the current position of vehicle and the position of the lead point. Finally, the steering angle δ of the driver arises from the steering angle δ_f for the front wheels and the transmission ratio i_{steer}:

$$\delta = i_{steer} \cdot \delta_f \ . \tag{24}$$

For the acceleration control, the lead point is used, too. It is about a control of the longitudinal deviation controlling the acceleration pedal position δ_{acc}. The control of the lateral vehicle dynamics is realized by the control of the braking pedal position δ_{brake} and the brake pressures for the four wheels, respectively.

SIMULATION RESULTS

In this chapter, the simulation results of a critical driving maneuver are presented. Therefore, a overtaking maneuver during a right rolling turn is investigated. The radius of the curve is $r_{curve} = 100\,\mathrm{m}$. The controlled vehicle drives with a velocity of $v_{current} = 40\,\frac{\mathrm{m}}{\mathrm{s}}$. During the turn maneuver, the sensors detect a vehicle ahead on the maneuver path which drives too slow. The obstacle is located $s_{obst} = 40\,\mathrm{m}$ forward and its velocity is only $v_{obst} = 30\,\frac{\mathrm{m}}{\mathrm{s}}$.

First, a collision free trajectory is planned based on the theory of elastic bands. Assuming the detection time is the start time $t_{start} = t_0 = 3\,\mathrm{s}$ for the evasion maneuver, the collision time is calculated: $t_{hit} = 4\,\mathrm{s}$. Therewith, the evasion maneuver should be finished at $t_{end} = t_n = 5\,\mathrm{s}$.

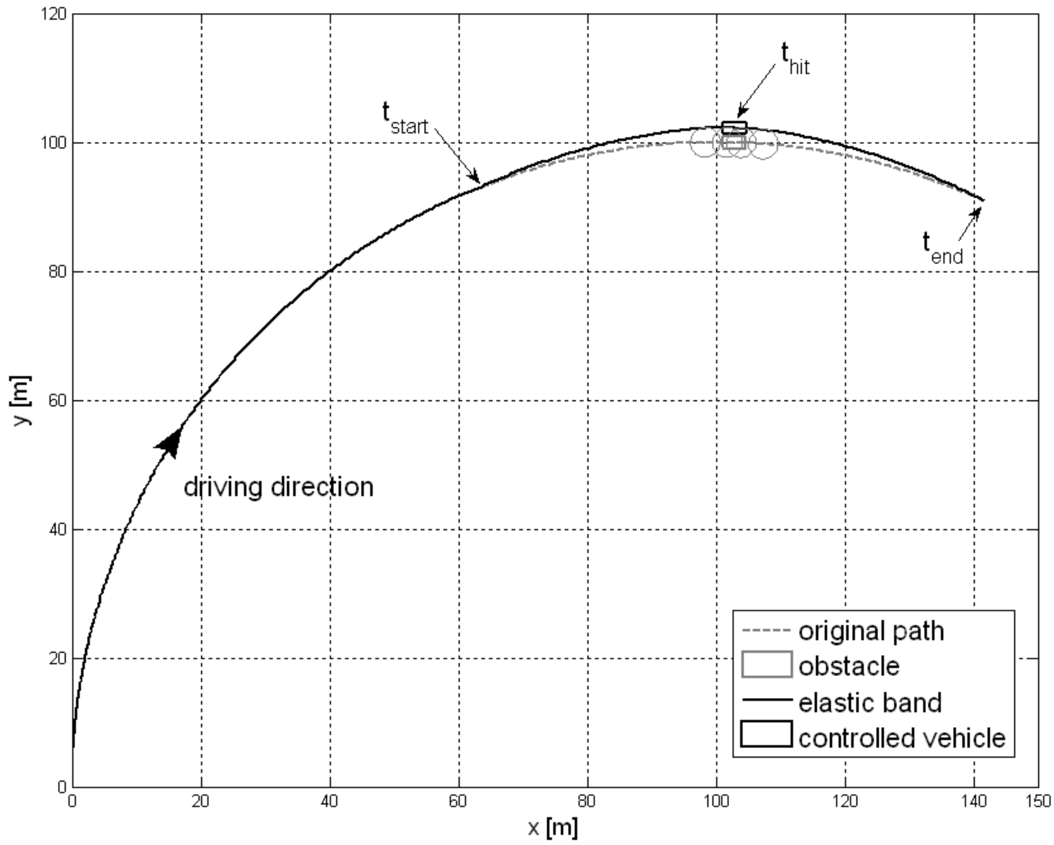

Figure 9 – Evasion maneuver during a right turn.

Based on the theory of the elastic bands, the trajectory of the evasion maneuver is determined. The optimization of the elastic band provides the spring constants

$$c_{int} = 10.367\frac{\mathrm{N}}{\mathrm{m}}\,, \quad c_{ext_1} = 0.127\frac{\mathrm{N}}{\mathrm{m}}\,, \quad c_{ext_2} = 0.001\frac{\mathrm{N}}{\mathrm{m}}\,, \quad c_{ext_3} = 0.006\frac{\mathrm{N}}{\mathrm{m}}\,, \quad c_{ext_4} = 0.125\frac{\mathrm{N}}{\mathrm{m}}\,. \tag{25}$$

A snap-shot of this maneuver is shown in Fig. 9. It shows the moment when the controlled vehicle passes the obstacle with the four safety circles. The controlled vehicle follows the solid evasion trajectory, and so it does not hit the obstacle. The corresponding velocity calculated with the GAUSS error distribution curve is shown in Fig. 10.

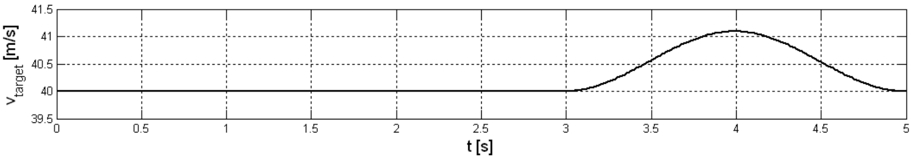

Figure 10 – Velocity of the controlled vehicle for the evasion maneuver.

The planned trajectory and the calculated velocity present the input for the path control for the complex vehicle model in FASIM_C++. The path control achieves the necessary steering angle δ and the acceleration pedal position δ_{acc} for driving the giving maneuver. These represent the driver's inputs for such a maneuver (Fig. 11).

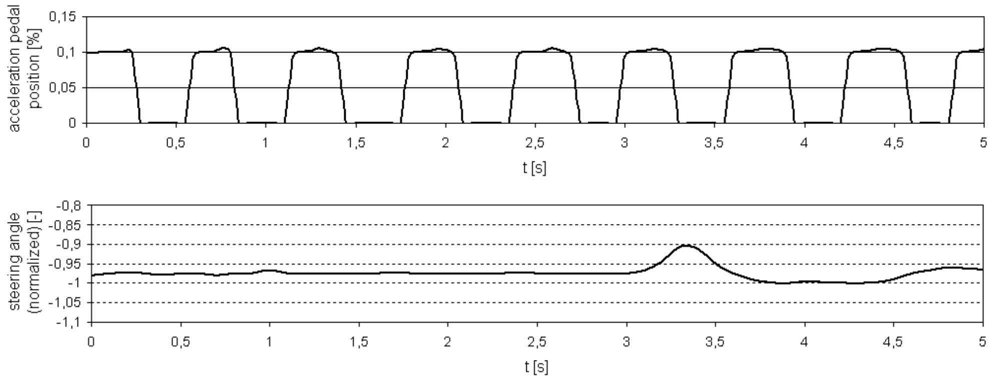

Figure 11 – Driver input for the evasion maneuver.

CONCLUSIONS AND OUTLOOK

This paper presents a method for planning collision free paths for vehicles. The trajectory of evasion maneuvers are determined based on the theory of elastic bands optimizing the path by varying the spring constants. The corresponding velocity is calculated with the GAUSS error distribution curve. For determination the active input of the driver, a path control for a complex vehicle model in FASIM_C++ is used. This achieves the steering angle and the position of the acceleration pedal. By means of an example of a right turn, the results are discussed.

In the future, the presented method will be tested with a vehicular model. Therefore, the algorithm for the maneuver planning has to be able to run in real-time. The vehicular model has to be equipped with sensors and a microcontroller. Also, the path control has to be arranged for the new vehicle.

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