ANALYSIS OF A SITUATIONAL ADAPTIVE CHASSIS WITH RESPECT TO MANEUVERABILITY AND FOOTPRINT

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ABSTRACT

The specifications of ground vehicles considering requirements like maneuverability and footprint are mainly caused by the target traffic. Future population distribution is subject to rural exodus resulting in megacities with the corresponding rise of traffic density and immense parking-space shortage. The problems caused by the compromise between a small footprint due to traffic requirements and a large one due to safety reasons are well known by the automotive industry as urban concept cars from different automobile exhibitions attest [1] [2] [3] [4] [5] [6] [7]. One of the known approaches is active tilt as it seems to be promising to reduce the vehicle’s dimension while coping with lateral acceleration demand and roll-over safety [4] [7]. But, active tilt represents only one solution for a situational adaptive chassis.

The proposed paper analyzes two alternative concepts, a variable track width and a variable height of the CG, and compares them to the known active tilt principle [8]. By using the exemplary maneuver of “steady-state cornering” the potential of each function concerning an increase of maneuverability and a decrease of the footprint is investigated. The results are promising and justify further investigation on the subject as for example dynamic simulation which has not been part of the study. The presented analytical examination of the influence on the yaw velocity gain and roll-over safety may open the discussion about the chances and risks of variable chassis design.

Keywords: Urban Suburban Traffic, Situational Adaption, Active Tilt, Narrow Track Vehicles, City Cars, Personal Electric Vehicle (PEV)

MOTIVATION

Studies of the United Nations forecast that by 2050, 67.2% (85.9% in developed countries) of the world population will live in urban and suburban areas [9]. The intense agglomeration results in mega cities with a rising demand for individual mobility. Characteristic mobility chains bring out the requirements of urban-tailored vehicles: High maneuverability, low cornering radiiuses and “easy parking”, rounded off by the demand of suburbanite people and commuters to travel into cities at an appropriate driving velocity using motorways and highways.

The motivation for the investigation of alternative chassis concepts can be found in these two, from a vehicle dynamic point of view, contrary requirements. Basically, a vehicle built for urban transportation shall have a high maneuverability providing extreme agility (quickly turning) and a small footprint to navigate through nose-to-tail traffic and
reduce the required parking space. Estimating two minutes per parking activity (leaving, searching for and entering a parking lot) [8], assuming 13400 km driven distance per vehicle and year as well as an average trip of 5 km [10] brings out that the common vehicle user spends nine days per year for parking activities. In contrary, the demand of commuters and suburbanite people to travel with an appropriate velocity on motorways and country roads require a comparable large footprint to provide driving stability and safety at higher speeds. To find an improved chassis design it is of high interest to combine and fulfill these two demands in one and the same vehicle. Several car manufacturers brought vehicles to market, tailored to urban use. Developed in a conservative way, they always made a compromise considering footprint and maneuverability. Thus the main goal of an adaptive chassis is to reduce the footprint to disburden traffic density and parking problems and increase maneuverability due to safety, dynamics and comfort by assimilating the vehicle to each maneuver situationally.

**STATE OF THE ART IN CITY-VEHICLE DESIGN**

In the following section well known and partly established city cars are introduced. The intention is to point out the compromise made by each concept with the according disadvantages.

**smart fortwo**

The **smart fortwo** is one of the most established city cars worldwide. Its small wheelbase makes parking perpendicular to common parking directions possible if permitted by law. It provides comfort and safety comparable to other middle or compact vehicles, assuring its suitability for motorway passages [11].

Even though the **smart fortwo** is amongst the most popular city cars due to its relatively small dimensions a further decrease of the footprint is imaginable, desirable and, as it will be investigated, possible. Conclusively the **smart fortwo** has a disadvantage considering lane width demand and footprint, Table 2.

**Renault Twizy**

The **Renault Twizy** is a personal electric vehicle for urban use. The different available configurations are all based on the same chassis platform. It is amongst the smallest four-wheeled vehicles available. Respecting the compromise between footprint and safe maneuverability the **Renault Twizy**’s maximal pace is limited to 80 km/h [5]. Tailored to urban traffic it has a disadvantage on motorways in comparison to vehicles that travel with the average speed of 130 km/h (recommended speed on german motorways). Considering its footprint, even the **Renault Twizy**’s small dimensions bear potential for diminution, Table 2.

**MIT CityCar / Hiriko**

The **MIT CityCar** was developed by different research groups at the **Massachusetts Institute of Technology** and will be published commercially by **Hiriko Driving Mobility** under the name **Hiriko** with the intention of car sharing projects in different European cities. This all electric vehicle has two interesting features: The vehicle is “foldable”, decreasing the footprint for parking; all wheels can be steered independently allowing the vehicle to rotate about its own z-axis on the spot. The **MIT CityCar** represents an intelligent attempt to avoid the compromise between footprint and safe maneuverability. Its wheelbase is able to shrink about 25% minimizing its parking-space demand [6]. Still, especially regarding its width which is comparable to the **smart fortwo**, it bears a high potential for diminishment on the search of a smaller footprint, Table 2.

**TTW Personal Commuting Vehicle**

The **TTW Personal Commuting Vehicle (PCV)** is a three wheeled parallel hybrid vehicle. It trusts in a front-axle active-camber steering combined with an active tilt system to balance lateral forces, facilitating the design of an extremely narrow track. While tilting, all three wheels and the body stay parallel, providing equal camber angles at each wheel. An electric actuator together with a planetary gearbox initializes the torque for tilting and assures a proper contact patch between tires and road. It promises the agile maneuverability and driving excitement of a motorcycle, combined with the safety and comfort of regular cars [7]. Further concept studies, as for example the **Toyota iRoad** (exhibited during the **Geneva International Auto Show 2013**) [4], that use active tilt system exist. The present work compares the active tilt principle as it is used in the **TTW PCV** to other adaptive functions (variable height of the center of gravity (CG), variable track width) that assimilate the vehicle to different lateral accelerations, Table 2.

**GENERAL REQUIREMENTS TO FOOTPRINT AND MANEUVERABILITY**

In the following section the differences between demands deriving from city traffic and motorway passages shall be specified with the goal of refining the purpose of an adaptive chassis [8]. The main difference between these two traffic surroundings can be found in the subject areas which determine the major requirements. Whereon for city traffic the demands to the footprint and maneuverability mainly derive from boundary conditions as parking space and route characteristics, on motorway passages the demands follow vehicle dynamic principles considering stability and safety.

**Footprint and Maneuverability**

City routes can hardly be generalized in their construction. However the difference to motorways can be pointed out. Maneuverability in terms of maneuvering through the city means turning with small radiuses, a frequent occurrence of consecutive maneuvers with a change of direction up to 180° demanding a high yaw velocity gain. The traffic itself can be
described as heavy and unpredictable; conclusively a high seating position of the driver to provide a good overview is desirable. Considering the footprint, a small one helps to disburden traffic density and especially alleviates the search and entering of parking lots.

On motorway passages the footprint shall follow vehicle dynamic considerations assuring not only roll-over-safety but improve dynamic performance by providing a reasonable ratio of the height of the CG and the track width [12]. Maneuverability in terms of maneuvering on the motorway means above all keeping and changing lane with the corresponding demand for a low yaw velocity gain to increase the vehicle’s stability.

**Lateral Accelerations**

Further demands to the chassis due to the traffic surroundings can be differentiated according to the occurring accelerations. A study by HACKENBERG [13] found out that lateral accelerations in city traffic vary from 0.36 g (95% of the test subjects) to 0.51 g whereon lateral accelerations on motorway interchanges are above varying from 0.44 g (95% of the test subjects) to 0.63 g [13].

**Minimum Footprint with Respect to Anthropometrical Data**

Especially the customers’ demand for an easy parking arouses the question for the smallest footprint possible. The answer can be found in anthropometrical considerations. To define an ideal footprint (ideal in terms of footprint diminution), the anthropometrical data of a male of the 95th percentile [14] combined with average seating positions [15] yield the corresponding chassis dimensions. By positioning the wheels around the seated driver a vehicle determined by ergonomic limitations can be designed, Figure 1. This “ideal” configuration will be used for the comparison of different functions of adaption in the accordingly headed section. The CG is assumed to be positioned in the driver’s hip-point (H-point). The seating height has been chosen to compete with common seating heights in personal vehicles. Even though a lower position of the CG is more reasonable, this seating height provides an overview comparable to established city cars as the smart fortwo, which is desirable especially regarding the narrow and unpredictable characteristics of city traffic. Therefore the defined chassis dimensions and seating height (CG) depict a significant illustration of how far a footprint diminishment can go, emphasizing the effectiveness of a situational adaptive chassis.

**SAFETY LIMITATIONS FOR NARROW TRACK VEHICLES**

MITSCHKE [12] compared the maximal lateral acceleration, defined by the force closure limit, of a vehicle without and with dynamic wheel load changes (WLC). As a result, the bearable lateral acceleration of the considered vehicle, limited by skidding, decreased from 0.96 g (without WLC) to 0.8 g (with WLC). To reduce the wheel load changes the height of the CG needs to be held low relative to the track width. For narrow track vehicles with a large ratio of the height of the CG and the track width, not skidding but roll-over limits the maximum lateral accelerations. Roll-over happens as soon as the dynamic wheel loads on both outer wheels become zero. Figure 2 clarifies that this state is reached as soon as the resulting of centripetal force and weight points through the connection line between the contact patches of the outer wheels [12], hence the moment about this line changes its augury [8]. For an easier understanding, a vehicle with rigid suspensions is considered. To find out the minimum track widths to corner with a certain lateral acceleration Figure 3 displays the geometric correlations.
A circle with the radius
\[ r_y = \frac{w}{2} = h_{CG} \cdot \frac{a_y}{g} \]  
represents the border for roll-over: All wheels have to be gathered around it in a way that the connection line between the contact patches of each side proceeds tangential to the bordering circle. This configuration provides roll-over safety for omni-directionally equal accelerations.

\[ (1) \]

**Figure 2: Roll-over-arising forces [8] [12]**

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**ADAPTIVE FUNCTIONS TO AVOID ROLL-OVER**

Common vehicles provide roll-over-safety up to the force closure limit, hence lateral accelerations above 1 g, for any driving situation. The purpose of an adaptive chassis is to provide roll-over-safety situationally only, resulting in the smallest footprint possible for each driving situation [8].

![Adaptive Functions Diagram](https://via.placeholder.com/150)

**Figure 4: Lateral movement of the CG [8]**

Considering Figure 3, Figure 4 and Figure 5 the following adaptive functions are promising to assimilate the chassis to different lateral accelerations:

1. The CG can be moved laterally about \( \Delta y \) into the curve center, resulting in a movement of the bordering circle for roll over, Figure 4.
2. The CG can be moved vertical about \( \Delta z \) to a lower position, resulting in a decrease of the bordering circle for roll over, Figure 5.
3. The track widths can be enlarged about \( \Delta w \) to enclose the bordering circle for roll over, Figure 3.

**Figure 5: Vertical movement of the CG [8]**
Active tilt represents a combination of mainly 1 and marginally 2 coupling a lateral and vertical movement by a rotational movement with an active roll angle $\phi_{active}$; 2 and 3 represent further functions of adaption. Based on an analytical investigation [8], the present work compares those three principles by evaluation of the following properties:

- Footprint for parking (no lateral acceleration)
- Lane width demand width for steady-state cornering (lateral acceleration of 0.51 g)
- Influence on maneuverability during steady-state cornering (yaw velocity gain).

### COMPARISON OF ADAPTIVE FUNCTIONS’ INFLUENCE ON FOOTPRINT, LANE WIDTH DEMAND AND MANEUVERABILITY

Fundament for the following comparison is the chassis defined in Figure 1. The considered maneuver is steady-state cornering with the maximal lateral acceleration of city traffic according to HACKENBERG [13]. The relevant parameters are listed in Table 1.

#### Table 1: Parameters of the investigated maneuver

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Track width front/rear (“ideal”)</td>
<td>0.545 m</td>
</tr>
<tr>
<td>Height of the CG (“ideal”)</td>
<td>0.7 m</td>
</tr>
<tr>
<td>Height of the vehicle</td>
<td>1.55 m</td>
</tr>
<tr>
<td>Height of the roll center front/rear (worst case assumption)</td>
<td>0 m</td>
</tr>
<tr>
<td>Passive roll angle due to roll stiffness (worst case assumption)</td>
<td>$\phi_{passive} = 8^\circ \cdot \frac{a_y}{g}$</td>
</tr>
<tr>
<td>Lateral acceleration [13]</td>
<td>0.51 g</td>
</tr>
</tbody>
</table>

**Footprint**

Parking is presupposed to be a maneuver which is performed without accelerations. Conclusively the chassis may have the dimensions as defined in Figure 1 during parking, leaving safety limitations disregarded. None of the considered functions stands in conflict with this suggested chassis design. For driving with a certain lateral accelerations, as soon as the parking process is completed an adaptive chassis needs to switch into “driving mode” assuring roll-over-safety in the first place and if desired reduce the dynamic wheel load changes. As it will be presented in the following subsection, the three investigated functions react as listed below:

- Active tilt: Assures roll-over-safety by providing an active roll angle
- Variable height of the CG: Assures roll-over-safety by lowering the position of the CG
- Variable track width: Assures roll-over-safety by enlarging the tracks.

Conclusively all functions allow the design of an adaptive chassis which facilitates parking with the defined minimum footprint, minimizing the parking-space demand.

**Lane Width Demand**

Due to the different principles of the adaptive functions they result in different lane width demands regarding cornering with a certain lateral acceleration as it will be compared in the following subsection.

**Active Tilt:** The principle of active tilt provides the active roll angle that is necessary to prevent roll-over due to the situational lateral accelerations. Vehicles that rely on this adaptive function tilt the body until the dynamic equilibrium about the roll axis is reached [7]; hence the centripetal force and the weight balance each other. This active roll angle can be calculated by [8]

$$\phi_{active} = \tan^{-1}\left(\frac{a_y}{g}\right)$$  

Considering Figure 6, the dynamic lane width demand follows equation (3). The body is presupposed to be equally wide as the track.

$$LD_{dynamic} = \frac{w}{2} \cdot \left(1 + \cos \phi_{active}\right) + h_{vehicle} \cdot \sin \phi_{active}$$

![Figure 6: Lane width demand of an active tilting vehicle [8]](image)

The lane width demand for cornering with the defined parameters and using the described active tilt principle can be found in Table 2.
Variable Height of the CG: Varying the height of the CG can be performed actively (analogous to active tilt) or passively by defining a certain height of the CG for different “driving mode”. Either way the position of the CG needs to be lowered to provide roll-over-safety. Extending the considerations made by Figure 2 and equation (1) by respecting a passive roll angle (common roll angle aroused by centripetal force with a lever arm to the roll axis) due to suspension kinematics and roll stiffness as defined in Table 1, the maximal height of the CG to corner with the defined parameters can be calculated by

\[
h_{CG,max} = \frac{w_{ideal}}{2 \cdot \left(\frac{a_y}{g} \cdot \cos \varphi_{passive} + \sin \varphi_{passive}\right)} \quad (4)
\]

The necessary variation of the height of the CG can be found in Table 2 as the difference between the defined “ideal” height (Table 1) and the maximum height calculated in equation (4). The lane width demand is defined by the used track width \(w_{ideal}\).

Variable Track Width: Analogue to a variable height of the CG, varying the track width can be used by two different strategies. Either, the track widths change actively during cornering to provide roll-over-safety or the track widths change passively in dependency of the traffic surroundings and the accordingly anticipated lateral accelerations to provide roll-over-safety.

Either way the track widths have to be enlarged. The differentiation is important due to maneuverability considerations as it will be explained in the accordingly headed subsection. Taking a passive roll angle into account, the minimum track width to provide roll-over-safety for cornering with the defined parameters can be calculated by

\[
w_{min} = 2 \cdot h_{CG} \cdot \left(\frac{a_y}{g} \cdot \cos \varphi_{passive} + \sin \varphi_{passive}\right) \quad (5)
\]

The passive roll angle has been considered as it influences the minimum track width and maximal height of the CG to provide roll-over-safety. Since these passive roll angles can be assumed to be small (compare Table 1) they are not taken into account for calculating the lane width demand as it has been done for the active roll angle. Hence the minimum track width to corner with the defined parameters as calculated in equation (5) defines the lane width demand and can be found in Table 2.

Compared to the dimensions of the smart fortwo, all of the presented functions represent an improvement considering a small footprint and lane width demand. Alleviating parking, the functions are on the same level presupposing the anthropometrically determined chassis design. Considering the dynamic lane width demand the functions which facilitate a variable height of the CG and variable track widths are outstanding. It has to be mentioned that the compared vehicles (smart fortwo and Renault Twizy) are able to withstand higher lateral accelerations as well which justifies their comparable large lane width demand and in case of the Renault Twizy its low seating position. An adaptive chassis is capable of the same by using the corresponding values of adaption which would bring the different lane width demands even closer together.

Table 2: Comparison of space demand for parking and lane width demand for steady-state cornering

<table>
<thead>
<tr>
<th>Vehicle</th>
<th>Space Demand (y) for parking ((a_y = 0))</th>
<th>Lane width demand for cornering ((a_y = 0.51 \text{ g}))</th>
<th>Adaption to lateral acceleration</th>
</tr>
</thead>
<tbody>
<tr>
<td>smart fortwo [11]</td>
<td>1.345 m</td>
<td>1.345 m</td>
<td>-</td>
</tr>
<tr>
<td>Renault Twizy [5]</td>
<td>1.094 m</td>
<td>1.094 m</td>
<td>-</td>
</tr>
<tr>
<td>Active tilt</td>
<td>0.545 m</td>
<td>1.22 m</td>
<td>(\varphi_{active} = 27^\circ)</td>
</tr>
<tr>
<td>Variable height of CG</td>
<td>0.545 m</td>
<td>0.545 m</td>
<td>(\Delta z = 0.23 \text{ m})</td>
</tr>
<tr>
<td>Variable track width</td>
<td>0.545 m</td>
<td>0.82 m</td>
<td>(\Delta w = 0.275 \text{ m})</td>
</tr>
</tbody>
</table>

Maneuverability

Besides influencing the footprint, the adaptive functions can be used to influence maneuverability, or more precisely vary the self-steering behavior thus the yaw velocity gain.

Active Tilt: To create the active tilt angle a torque is necessary to overcome the arising centripetal force. The active roll angle according to equation (2) has to be reached within the peak response time (period of time in which the centripetal force arises) yielding the angular acceleration

\[
\dot{\varphi}_{active} = 2 \cdot \frac{a_y, steady-state}{g \cdot T_{\varphi, max}} \quad (6)
\]

With the centripetal force as a function of time the active torque follows equation (7) until the stationary state is reached.

\[
T_{\varphi, active} = J_k \cdot \ddot{\varphi} + F_c(t) \cdot h_{CG} \cdot \cos(\varphi(t)) - m \cdot g \cdot \sin(\varphi(t)) \quad \forall 0 \leq t \leq T_{\varphi, max} \quad (7)
\]

This torque has to be supported by the wheels resulting in dynamic wheel load changes (equation (8)), thus leading to a decrease of the axles’ slip angle stiffness of front and rear axle (non-linear tire characteristics).

\[
\Delta F_{z,f} = \frac{D_f \cdot T_{\varphi, active}}{w_f} \quad (8)
\]

\[
\Delta F_{z,r} = \frac{D_r \cdot T_{\varphi, active}}{w_r}
\]

Through arbitrary distribution \((D_f, D_r)\) of the active torque between the axles, the self-steering behavior can be set up. The effect follows the same principle as common roll moment distribution which is either defined passively by the setup of suspension stiffness using anti-roll bars or actively by systems like active roll stabilization as e.g. the PDCC implemented into the Porsche 911 [16].
Variable Height of the CG: Varying the height of the CG is not expected to have instant influence on the self-steering behavior. The wheel load changes aroused by the reaction forces of the vertical acceleration of the CG depend on the design layout of the function and can be set up to be equal on both axles. The basic layout of the self-steering behavior, determined by longitudinal position of the CG, roll moment distribution (slip angle stiffness of the axles) and driven axle, stays untouched.

Variable Track Width: A variable track width can be used in two different ways to vary the yaw velocity gain. The simple one is to distribute the roll moment between the axles by providing different track widths at front and rear axle, resulting in different wheel load changes and conclusively in different slip angle stiffness on the axles (non-linear tire characteristics). The innovative possibility comes along with the principle of a variable track width [8]: To vary the track, the wheels have to be moved along the axles adding velocity in \( y \)-direction to each wheel.

For steady state cornering as shown in Figure 7, the wheels velocity can be expressed, exemplarily for the outer front wheel, by

\[
v_{x,W,f,o} = v_o \cdot \cos \beta + \dot{\psi} \cdot w_{f,o} = v_o \cdot \cos \beta + \frac{v_o}{R} \cdot w_{f,o} \tag{9}
\]

\[
v_{y,W,f,o} = v_o \cdot \sin \beta + \dot{\psi} \cdot l_f = v_o \cdot \sin \beta + \frac{v_o}{R} \cdot l_f \tag{10}
\]

yielding the corresponding slip angle of [17]

\[
\alpha_{f,o} = \delta_{f,o} - \tan^{-1} \left( \frac{v_{y,W,f,o}}{v_{x,W,f,o}} \right) \tag{11}
\]

If the track width is varied during steady-state cornering the slip angles change due to the additional velocity in \( y \)-direction [8], Figure 8. The track widths are assumed to change with a constant velocity yielding the variable track width in dependency of time

\[
w_{f,o}(t) = w_{f,o} + \Delta w_{f,o}(t) = w_{f,o} + \Delta w_{f,o} \cdot t \tag{12}
\]

Assuming small side slip angles \( \beta \) as well as a small ratio of \( \Delta w_{f,o}/R \ll 1 \) leads to the wheels velocities (outer front wheel) as expressed in equation (13) and (14).

\[
v_{x,W,f,o} = v_o \cdot \cos \beta + \dot{\psi} \cdot w_{f,o}(t) \approx v_o \cdot \left( 1 + \frac{w_{f,o}}{R} \right) \tag{13}
\]

\[
v_{y,W,f,o} = v_o \cdot \sin \beta + \dot{\psi} \cdot l_f - \frac{v_o \Delta w_{f,o}}{v_o} \approx v_o \cdot \left( \beta + \frac{l_f}{R} - \frac{\Delta w_{f,o}}{v_o} \right) \tag{14}
\]

Applying equation (13) and (14) onto (11) yields the resulting slip angle

\[
\alpha_{f,o} = \delta_{f,o} - \beta + \frac{l_f}{R} - \frac{\Delta w_{f,o}}{v_o} \tag{15}
\]

Comparing equation (15), (11) and Figure 8 reveals that the slip angles of the outer front wheel increases during an enlargement of the track width. Equivalent calculations for the remaining wheels yield that the slip angles on the outer wheels
increase whereon the slip angles on the inner wheels decrease during an enlargement of the track widths.

The effect on the corner behavior is similar to a slight toe in on the axles. Conclusively, enlarging the track width during cornering raises the slip angle stiffness of the according axle.

Equation (15) carves out that the increase and decrease of slip angles depends on the ratio of the velocity the track width variation is performed with and the driven velocity.

Considering different velocities $v_{\Delta w}$ on front and rear axle three possibilities can be differentiated to set up the self-steering behavior:

$v_{\Delta w,f} = v_{\Delta w,r}$: Both axles change their track width in the same velocity. The slip angle stiffness of both axles rises, hence the centripetal acceleration rises. Considering constant velocity and steering angle, a smaller corner radius results.

$v_{\Delta w,f} > v_{\Delta w,r}$: The slip angle stiffness on the front axle rise more than on the rear axle. This raises the yaw velocity, meaning that the vehicle turns more into the curve than commonly.

$v_{\Delta w,f} < v_{\Delta w,r}$: The slip angle stiffness on the rear axle rise more than on the front axle. This lowers the yaw velocity, meaning that the vehicle turns less into the curve than commonly.

However the use of a variable track width to influence the self-steering behavior permanently has to be considered critically. The track width cannot be enlarged permanently. To gain an effective variation of the slip angles a certain ratio of driven velocity and track width changing velocity has to be reached. Conclusively the necessary track width changing velocity rises with the driven velocity.

During a long turn on the motorway for example, the influence on the self-steering behavior by changing the track width could only cover a fraction of this maneuver, hence the driving behavior would change during its performance.

Assuming several turns in a row, the maximal track width will be reached in time as well; hence the self-steering behavior will change.

Therefore it is recommendable to use a variable track width passively as described in the according subsection. Influencing the yaw velocity gain actively by changing the track might be more useful for temporary path corrections to regain stability, similar to e.g. ESP intervention.

Table 3 summarizes the information of the previous subsection to emphasize the differences between common cars and the three investigated adaptive functions concerning maneuverability thus the possibilities of setting up the yaw velocity gain. Differentiated have been a passive intervention which influences the yaw velocity gain permanently (recommended for setup due to traffic surroundings [8]) and an active intervention which influences the yaw velocity gain temporarily only (recommended for path corrections [8]). Common systems with influence on the yaw velocity gain like four wheel steering, active roll stabilization, ESP or torque vectoring are not listed since they are compatible with each chassis design.

**Table 3: Passive and active influence on the yaw velocity gain (YVG)**

<table>
<thead>
<tr>
<th>Chassis design</th>
<th>Passive variation of YVG (permanently)</th>
<th>Active variation of YVG (temporarily)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conservative</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Active tilt</td>
<td>-</td>
<td>Active-roll-torque distribution</td>
</tr>
<tr>
<td>Variable height of the CG</td>
<td>Depends on basic layout</td>
<td>Depends on basic layout</td>
</tr>
<tr>
<td>Variable track width</td>
<td>Different track widths on front and rear axle</td>
<td>Variation of the track widths (front/rear) independently and actively</td>
</tr>
</tbody>
</table>

**CONCLUSION AND OUTLOOK**

**Conclusion**

Despite the positive effects of an adaptive chassis, its use comes along with a safety risk that has to be considered. The diminution of the footprint results in a vehicle that moves close to its limits regarding stability, implicitly safety. The anthropometrical limit, as it has been used, therefore only represents a comparison base to display the margin of footprint-decrease and is not to be seen as an optimal solution.

For the first time different concepts of a situational adaptive chassis have been investigated and compared concerning a footprint decrease as well as an active and passive interference with the self-steering behavior. It has been shown that, besides the principle of active tilt, further functions bear a high potential to improve future urban mobility with respect to maneuverability, footprint and safety.

This paper shall emphasize the capability of the developed functions and initiate the discussion about the potentials and risks of variable chassis design in comparison to evolutionary justified chassis design.

**Outlook**

Analytical investigations brought out tendencies which have been documented [8]. The results are promising and justify the effort of further investigation by e.g. building up dynamic simulation. Therefore, continuing this work necessarily needs to start with verifying and refining the present...
results. Dynamic simulations by using a valid simulation model of an adequate vehicle which is extended by the physics of the adaptive functions need to be executed. Of special interest is the dynamic influence on the self-steering behavior, thus approaches considering a coupling between steering and variable footprint parameters need to be assessed. Furthermore a reasonable footprint to gain a satisfying performance for each driving situation needs to be determined. By using this simulation, the potential of each function can be detailed and the strategy of use refined which needs to be extended by estimations about power demand and the effort for technical implementation.

SYMBOLS

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>α</td>
<td>°/rad</td>
<td>Slip angle</td>
</tr>
<tr>
<td>β</td>
<td>°/rad</td>
<td>Side slip angle</td>
</tr>
<tr>
<td>ϕ</td>
<td>°/rad</td>
<td>Roll angle</td>
</tr>
<tr>
<td>ψ</td>
<td>°/rad</td>
<td>Yaw angle</td>
</tr>
<tr>
<td>a</td>
<td>m/s²</td>
<td>Acceleration</td>
</tr>
<tr>
<td>D</td>
<td>/</td>
<td>Distribution factor</td>
</tr>
<tr>
<td>F</td>
<td>N</td>
<td>Force</td>
</tr>
<tr>
<td>h</td>
<td>m</td>
<td>Height</td>
</tr>
<tr>
<td>J</td>
<td>kgm²</td>
<td>Mass moment of inertia</td>
</tr>
<tr>
<td>l</td>
<td>m</td>
<td>Wheelbase</td>
</tr>
<tr>
<td>LD</td>
<td>m</td>
<td>Lane width demand</td>
</tr>
<tr>
<td>T</td>
<td>s</td>
<td>Peak response time</td>
</tr>
<tr>
<td>v</td>
<td>m/s</td>
<td>Velocity</td>
</tr>
<tr>
<td>w</td>
<td>m</td>
<td>Track width</td>
</tr>
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</table>

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