Design of a nonlinear trailer steering controller

Gunter Nitzsche¹, Klaus Röbenack², Sebastian Wagner¹ and Stephan Zipser¹

Abstract—In this paper the authors present a design approach of a nonlinear controller for steered semi-trailers of heavy commercial vehicles, which is used to improve their maneuverability. The proposed control structure uses the exact input-output linearization. Additionally, a sliding mode control approach (SMC) is used to overcome model uncertainties as well as control errors and increase robustness. The practicability of the control structure is shown with simulation results of a nonlinear three-dimensional multi-body model.

I. INTRODUCTION

Beside improving the efficiency of transport, the European Commission announces in its White Paper for Transporta- tion [1] that the number of fatalities in road traffic accidents should be halved by 2020 and reduced to almost zero by 2050. This makes traffic safety an important topic of current research activities. Achieving this objective is going to be more complex, considering the predicted traffic growth in the next years. Especially the growth in goods transportation leads to new challenges. As the capacity of railroad goods transportation is almost exceeded, more and more cargo is going to be transported on roads with truck-trailer combinations.

The study [2] shows that accidents with trucks cause 50% more seriously injured persons and 550% more fatalities than car accidents. Furthermore, the average physical damage quantified with 70,000 ₹ per accident is approximately seven times higher. Therefore, an improvement of traffic safety of heavy commercial vehicles has a great potential for reaching the goal announced by the European Commission. Easing the driving task for the driver helps to increase the concentration, which can be spent on the surrounding of the vehicle and accident avoidance. In this article the authors will sketch a nonlinear control design for an advanced driver assistance system using one steerable trailer axle. It is used to reduce the tractive force of the trailer, but could be used for vehicle dynamics control as well.

The most important driver assistance system, when it comes to accident avoidance, is the Electronic Stability Program (ESP). These braking-based yaw rate controllers are proven to reduce the number and severity of real life car accidents [3]. Starting from the basic system proposed in [4], current developments focus on the combination with electronic steering systems [5]. Especially for articulated heavy commercial vehicles steering the trailer axles is taken into account, e.g. [6], [7], [8]. In the former mentioned articles linear controllers are used to improve the vehicle dynamics and the maneuverability. In [9] and [10] the maneuverability of double articulated vehicles is enhanced based on an estimation of the first module’s driven track.

The above stated references use either linear controllers or rather complex algorithms. Furthermore, usually all trailer axles are assumed to be steerable. This is a costly solution as steering axles are expensive. Therefore, the authors will show in this paper how nonlinear controller design techniques can be used to improve the maneuverability with just one steering axle in the trailer. Such steering axles with fixed steering schemes are current state of the art for enhancing the maneuverability of semi-trailers, cf. [11].

Figure 1 shows the block diagram of the control structure to be derived in the next section. The reference generator estimates a desired vehicle movement based on the driver steering input $\delta_f$. This reference movement is used by the second order sliding mode controller (SMC). This controller drives the new input $v_{\text{EA}}$ provided by the exact input-output linearization. The vehicle block will be replaced by a suitable simulation model in this paper.

The proposed controller structure is based on the following idea. A kinematic vehicle behavior is easily predictable for the driver, as tire slip does not have to be taken into account. This holds for both maneuverability and dynamic stability. If the real vehicle acted like a kinematic vehicle it would follow the driver’s steering input without skidding. Thus, a robust controller is designed, which eliminates the deviations between the actual vehicle behavior and the reference generated by a suitable kinematic model. Such deviations are caused for example by tire slip.

The final controller will consist of an exact input-output linearization stabilized with a second order sliding mode feedback controller. The control laws will be derived in Section III. Preliminarily, Section II gives an introduction to the vehicle models used during controller design and simulation. The simulation results can be found in Section IV.

¹Fraunhofer Institute for Transportation and Infrastructure Systems IVI, Dresden, Germany
²Technische Universität Dresden, Institute of Control Theory, Dresden, Germany
The variables $x_1$ and $y_1$ give the planar coordinates of the truck’s center of gravity. The yaw angle $\psi_1$ gives the orientation of the truck. The orientation of the trailer relative to the truck is given by the articulation angle $\psi_{12} = \psi_2 - \psi_1$.

The derivatives of the joint coordinates are expressed by joint velocities $\eta$. These are chosen in the truck’s reference frame $C_1$, cf. Fig. 3. Therefore a transformation matrix $H$ is necessary for the calculation of $\beta$.

$$
\eta = \begin{pmatrix}
\dot{x}_1 \\
\dot{y}_1 \\
\dot{\psi}_1 \\
\dot{\psi}_{12}
\end{pmatrix}
$$

(2)

$$
\beta = H \eta
$$

(3)

$$
H = \begin{pmatrix}
\cos \psi_1 & -\sin \psi_1 & 0 & 0 \\
\sin \psi_1 & \cos \psi_1 & 0 & 0 \\
0 & 0 & 1 & 0 \\
0 & 0 & 0 & 1
\end{pmatrix}
$$

(4)

In (2) the left index denotes the coordinate system in which the component is given, here $C_1$.

The full state vector $x$ consists of $\beta$ and $\eta$.

$$
x = \begin{pmatrix}
x_1 \\
y_1 \\
\psi_1 \\
\psi_{12} \\
\dot{x}_1 \\
\dot{y}_1 \\
\dot{\psi}_1 \\
\dot{\psi}_{12}
\end{pmatrix}
$$

(5)

For the vehicle dynamics just the last five states are important.

The state space model is given by:

$$
\dot{x} = \begin{pmatrix}
\dot{\beta} \\
\dot{\eta}
\end{pmatrix} = \begin{pmatrix}
H \eta \\
M^{-1} (k_e + k_c)
\end{pmatrix}.
$$

(6)

The simplicity of this equation may be misleading. The mass matrix $M$ and the centrifugal and Coriolis forces $k_c$ are complex equations, whose derivation is omitted for the sake of brevity. The variable $k_e$ stands for the external forces, namely the tire forces longitudinal ($F_x$) and lateral ($F_y$) with respect to the according vehicle module. The tire forces will be replaced by a tire model later. Introducing the dependencies in (6) leads to:

$$
\dot{x} = \begin{pmatrix}
H(\psi_1) \eta \\
M^{-1}(\psi_{12}) (k_e(\eta, \psi_{12}) + k_c(\psi_{12}, F(\eta, \delta, \psi_{12})))
\end{pmatrix}
$$

(7)

If this model is reckoned as controller design model, the steering angles are disturbances rather than inputs. The control input $u$ will be the lateral force at the last trailer axle. This will give an affine state space model, easing the controller design.

To obtain the standard form of a state space system,

$$
\dot{x} = f(x) + g(x)u,
$$

(8)
equation (7) needs to be transformed. The driver steering angle, driving/braking forces are included in $f(x)$ and $g(x)$.
As $k_c$ depends linearly on the tire forces, the second line of (7) can be divided in parts.

$$\dot{\eta} = M^{-1}k_c + M^{-1} \sum_{i,j} \left( g_{x_{i,j}}F_{x_{i,j}} + g_{y_{i,j}}F_{y_{i,j}} \right)$$

(9)

$$g_{x_{i,j}} = \frac{\partial k_c}{\partial (F_{x_{i,j}})}$$

(10)

$$g_{y_{i,j}} = \frac{\partial k_c}{\partial (F_{y_{i,j}})}$$

(11)

The index $i = 1, 2$ stands for the module number. Index $j$ indicates the wheel number at the according module, cf. Fig. 3.

Equation (9) confirms the decision to steer the last trailer axle. The vector

$$g_{y_{2,j}} = \begin{pmatrix} -\sin \psi_{12} \\ \cos \psi_{12} - c_1 \cos \psi_{12} - d_2 + 2x_{wheel_{2,j}} \\ -d_2 + 2x_{wheel_{2,j}} \end{pmatrix}$$

(12)

describes the impact of the lateral tire forces of the trailer axles. The measurements $c_1$ and $d_2$ are shown in Fig. 3. The variable $2x_{wheel_{2,j}}$ denotes the longitudinal coordinate of the $j$-th trailer wheel in the coordinate frame $C_2$. So the trailer axle which is farthest from the kingpin has the greatest impact on the vehicle dynamics.

Combining the wheels which are not steered or steered by the driver in a set $L_p$, provides a possibility to write the vector field $f(x)$ in a compact form. The set $L_a$ of the controlled wheels will be helpful for the derivation of $g(x)$.

$$L_p = \{(i,j) \mid \text{passive or steered by driver}\}$$

(13)

$$L_a = \{(i,j) \mid \text{controlled wheels}\}$$

(14)

$$f(x) = \begin{pmatrix} \dot{H}\eta \\ M^{-1} \left( k_c + \sum_{(i,j) \in L_p} g_{y_{i,j}}F_{y_{i,j}} \right) \\ \cdots + \sum_{(i,j) \in L_a} g_{x_{i,j}}F_{x_{i,j}} \end{pmatrix}$$

(15)

The parts of (7) and (9) respectively which are not part of $f(x)$ form the input vector field $g(x)$. The first line of (7) is fully represented by $f(x)$. For the second line, shown in (9), the influence of the lateral forces of the controlled wheels $L_a$ is missing in $f(x)$. This part is given by:

$$M^{-1} \sum_{(i,j) \in L_a} g_{y_{i,j}}F_{y_{i,j}}$$

(16)

Both wheels at the controlled trailer axle have the same $x$-coordinate, hence $g_{y_{2,5}} = g_{y_{2,6}}$. So this sum can be simplified to:

$$M^{-1} \sum_{(i,j) \in L_a} g_{y_{i,j}}F_{y_{i,j}} = M^{-1} \left( g_{y_{2,5}}2F_{y_{2,5}} + g_{y_{2,6}}2F_{y_{2,6}} \right)$$

$$= M^{-1}g_{y_{2,5}} \left( 2F_{y_{2,5}} + 2F_{y_{2,6}} \right)$$

$$= M^{-1}g_{y_{2,5}}2F_{y_{u}}.$$ 

(17)

The complete lateral force at the last trailer axle is given by $2F_{y_u}$. As this will be the control input $u$, the input vector field $g(x)$ can be read from (17), keeping in mind that the input $u$ has no direct influence on the first line of (7).

$$g(x) = \begin{pmatrix} 0 \\ M^{-1}g_{y_{2,5}} \end{pmatrix}$$

(18)

$$u = 2F_{y_u}$$

(19)

With (15), (18) and (19) the parts of the standard state space system equation (8) are given.

Leaving out the tire model in the derivation above has two advantages:

- the state space model (8) is affine in its inputs,
- the tire model can be replaced later without great effort as the model is independent of the tire model.

For a complete model of the vehicle a tire model is necessary. The relation of the tire forces is shown in Fig. 4(a). The forces acting in longitudinal ($F_{x_{i,j}}$) and lateral ($F_{y_{i,j}}$) direction with respect to the tire can be transformed into the forces used in the state space model by:

$$\begin{pmatrix} F_{x_{i,j}} \\ F_{y_{i,j}} \end{pmatrix} = \begin{pmatrix} \cos \delta_{i,j} & -\sin \delta_{i,j} \\ \sin \delta_{i,j} & \cos \delta_{i,j} \end{pmatrix} \begin{pmatrix} F_{x_{i,j}} \end{pmatrix}$$

(20)

The longitudinal forces $F_{x_{i,j}}$ are given by either driving or braking of the truck. The lateral forces $F_{y_{i,j}}$ are calculated with respect to the slip angle $\alpha$, cf. Fig. 4(b). A linear tire model is assumed.

$$F_y = c_\alpha \alpha$$

(21)

On the one hand this tire model is used to complete the state space model. On the other hand it is used as inverse tire model to gain the necessary slip and the corresponding steering angle from the desired lateral force at the controlled trailer axle.

C. Kinematic Model

The kinematic model described in this section is going to be used for the reference generator in Fig. 1. This type of model is usually used to describe the maneuver behavior of
the vehicle, neglecting tire slip. For this model it is assumed that every tire rolls without slip along its center plane. As it is not possible to describe the actual semi-trailer with this type of model, it is necessary to lump the axles to a single wheel, cf. Fig. 5. It is assumed that the actual trailer turns approximately about its middle axle. Thus, the single wheel of the kinematic model represents the same position.

Neglecting side slip of the tires reduces the number of local degrees of freedom to three.

\[ x_{kin} = (v_1 \quad \psi_1 \quad \psi_{12})^T \]  

(22)

After short calculations the state space model of the kinematic vehicle behavior can be obtained.

\[ \dot{x}_1 = u_1 \]  
\[ \dot{x}_2 = \frac{x_1}{l_1} \tan \delta_f \]  
\[ \dot{x}_3 = \frac{x_1}{l_2} \left( \frac{\Delta \psi L}{l_2} \frac{\cos (\delta_t + \psi_{12})}{\cos \delta_t} - 1 \right) \frac{l_2}{l_1} \tan \delta_f \ldots \]  
\[ \ldots - \frac{\sin (\delta_t + \psi_{12})}{\cos \delta_t} \]  

(23) (24) (25)

This nonlinear model is used to generate the reference vehicle behavior, tracked by the controller proposed in the following section.

III. CONTROLLER DESIGN

As already mentioned in section I, the task of the proposed controller is to make the nonlinear vehicle act like the kinematic one as good as possible. This is going to reduce the risk of slip caused by the nonlinearities of the tires for high slip angles. The controller structure in Fig. 1 is divided into an exact input-output linearization, compensating the nonlinearities, and a robust second order sliding mode controller (2-SMC), coping with modeling uncertainties. These uncertainties are inevitable, as the simulation model cannot be used as controller design model, due to its complexity. Thus, a robust feedback is necessary.

Before starting the controller design, a suitable control variable has to be chosen. The variable should provide the possibility to:

- improve the maneuverability,
- avoid overshoot of the trailer’s lateral acceleration during obstacle avoidance maneuvers and
- avoid excessive tire wear.

The most promising control variable is the articulation angle \( \psi_{12} \) and its derivatives, respectively. With the control of the articulation angle rate the vehicle movement can be influenced. A superposed control of the articulation angle itself ensures a better maneuverability.

A. Exact Input-Output Linearization

The feedforward control is based on the exact input-output linearization by [13] and [14]. But instead of using it as a feedback linearization, it is used as a feedforward version similar to [15]. As the reference generator does not provide all necessary states needed for the exact input-output linearization, the missing states have to be fed back from the vehicle or its simulation representation.

The exact input-output linearization is designed with the articulation angle as output.

\[ y_{EA} = h(x) = \psi_{12} \]  

(26)

Starting with the state space model, the relative degree can be found to be two, as the input \( u \) occurs in the second derivative of the output \( y_{EA} \).

\[ y_{EA} = x_4 \]  
\[ \dot{y}_{EA} = \dot{x}_4 = x_8 = L_f h(x) \]  
\[ \ddot{y}_{EA} = f_8(x) + g_8(x)u_{EA} \]  
\[ \dddot{y}_{EA} = L_f^2 h(x) + L_g L_f h(x)u_{EA} \]  

(27) (28) (29) (30)

In (29) \( f_8 \) denotes the eighth element of \( f(x) \) and \( g_8 \) the eighth element of \( g(x) \). For feasible vehicle parameters and articulation angle \( \psi_{12} < 90^\circ \), the relative degree is well-defined. In (28) \( L_f h(x) \) stands for the Lie-Derivatives along the according vector field. The Lie-Derivatives \( L_f^2 h(x) \) and \( L_g L_f h(x) \) follow accordingly.

\[ L_f h(x) = \frac{\partial h(x)}{\partial x} f(x) \]  

(31)

With the definition of the new input

\[ v_{EA} := \dddot{y}_{EA} = \dddot{\psi}_{12} \]  

(32)

the necessary lateral force at the steered trailer axle can be calculated by

\[ u_{EA} = \frac{2 F_{yu}}{\dddot{y}_{EA} - \frac{L_f^2 h(x)}{L_g L_f h(x)}} \]  

(33)

Applying (33) in (8) yields a second order external dynamic, leaving an uncontrollable sixth order internal dynamic.

\[ \begin{pmatrix} \dddot{\psi}_{12} \\ \dddot{\psi}_{12} \end{pmatrix} = \begin{pmatrix} 0 & 1 \\ 0 & 0 \end{pmatrix} \begin{pmatrix} \dot{\psi}_{12} \\ \dot{\psi}_{12} \end{pmatrix} + \begin{pmatrix} 0 \\ 1 \end{pmatrix} v_{EA} \]  

(34)

The internal dynamic must be stable itself, as it is not controllable. For the sake of brevity the stability analysis of the internal dynamic is omitted. As long as the vehicle acts within the stable region of the nonlinear tire model used in the simulation model, the internal dynamic is stable.

Equation (33) calculates the force at the steered trailer axle lateral to \( C_2 \). Transforming this force with (20) in the tire fixed coordinate frame and applying the inverse of the tire model (21) leads to the according steering angle of the axle. The slip angle, which is calculated as an intermediate result, is limited to the stable region of the nonlinear tire characteristic of the vehicle model. This prevents excessive tire slip.

The control law (33) needs a stabilizing feedback. According to [13], usually a desired dynamic is impressed. In the next section the authors will show a stabilizing feedback by a 2-SMC resulting in a higher robustness and better tracking performance than the standard approach.
B. Second Order Sliding Mode Control

The standard stabilizing feedback usually used for the exact input-output linearization lacks robustness. Therefore a robust 2-SMC feedback is used, generating smooth control inputs rather than switching inputs as a standard SMC would. The 2-SMC could be used without the exact input-output linearization, but linearizing the controlled system eases the design process of the SMC.

An introduction to higher order sliding modes can be found in [16], [17]. For the sake of brevity just the basic idea is outlined in the present paper. For a standard SMC a sliding surface \( \sigma \) needs to be defined with relative degree one. This means, that the first derivative of the distance from the sliding surface \( \sigma \) can be influenced by the switching control input. The deviations from the sliding surface will be eliminated by the control input, yielding \( \sigma = 0 \). For a 2-SMC \( \dot{\sigma} = 0 \) should be achieved as well, improving robustness and tracking performance, cf. [16]. Furthermore, smooth control inputs will be generated, which is mandatory for a steering controller.

The 2-SMC is applied to the linearized system (34). Thus, the controlled output is the articulation angle and the control input will be its second derivative.

\[
y_{\text{SMC}} := \dot{\psi}_{12} \tag{35}
\]

\[
u_{\text{SMC}} := \ddot{\psi}_{12} \tag{36}
\]

The 2-SMC is designed as tracking controller, whose desired trajectory \( y_d \) is given by the reference generator of section II-C.

A sliding surface, fulfilling the requirements stated above, and its first derivative are given by:

\[
\sigma = T(\dot{y}_{\text{SMC}} - \dot{y}_d) + y_{\text{SMC}} - y_d \tag{37}
\]

\[
\dot{\sigma} = T(\ddot{y}_{\text{SMC}} - \ddot{y}_d) + \dot{y}_{\text{SMC}} - \dot{y}_d \tag{38}
\]

For the s2-SMC also the second derivative of the sliding surface is necessary.

\[
\ddot{\sigma} = T(\dddot{y}_{\text{SMC}} - y_d^{(3)}) + \ddot{y}_{\text{SMC}} - \ddot{y}_d \tag{39}
\]

For the controller the Super Twisting Algorithm is used, cf. [16].

\[
u_{\text{SMC}} = u_1 + u_2 \tag{40}
\]

\[
\dot{u}_1 = -\alpha \text{sgn} \sigma \tag{41}
\]

\[
\dot{u}_2 = -\lambda |\sigma|^{p} \text{sgn} \sigma , \quad 0 < p < 1 \tag{42}
\]

Applying (40) to (39) yields for \( \sigma \neq 0 \):

\[
\ddot{\sigma} = -T \left( \alpha \text{sgn} \sigma + \lambda \left[ 1 - \frac{1}{|\sigma|^{p}} \right] \dot{\sigma} - y_d^{(3)} \right) + \ddot{y}_{\text{SMC}} - \ddot{y}_d \tag{43}
\]

The controller parts in (43) must dominate the rest of the equation. Assume that

\[
T \dot{y}_d^{(3)} + \ddot{y}_{\text{SMC}} - \ddot{y}_d < C \tag{44}
\]

holds. Then \( \alpha \) and \( \lambda \) need to be sufficiently large, according to [16]. A detailed description how to choose the parameters can be found in [18], [19].

C. Combination

The exact input-output linearization is combined with the 2-SMC in serial order, cf. Fig. 1. Thus the control value \( u_{\text{SMC}} \) calculated by the 2-SMC is connected to the new input \( v_{EA} \) of the exact input-output linearization.

\[
v_{EA} = u_{\text{SMC}} \tag{45}
\]

D. Stability Analysis

The stability analysis can be divided in two steps. The first one is to ensure a stable external dynamic. This is fulfilled via the 2-SMC with correctly chosen parameters of the sliding surface and the Super Twisting Algorithm. The second step is to check the residual internal dynamic for stability. Due to the complexity of the vehicle model, an analytical stability analysis is not possible. Recall that the control design model is of eighth order. The results shown in Fig. 6 indicate that the controlled system is stable unless the slip angle limit is met. If the limit is met, the controller, especially the stabilizing feedback, is not active anymore. Thus the unstable behavior indicated by the fast increase of \( \sigma \) is not surprising.

IV. Simulation Results

In this section simulation results are presented, which were obtained with the controller described in section III applied to the multi-body simulation model from subsection II-A.

The truck is driven with \( v = 5 \frac{m}{s} \). At \( t = 1 \text{ s} \) the virtual driver performs a step steering input of \( \delta_f = 20^\circ \) at the truck’s front wheels, filtered by a second order time lag with \( T_1 = T_2 = 0.1 \text{ s} \). Accordingly the truck goes from straight driving to circular driving with a radius of \( 11 \text{ m} \) and a lateral acceleration of \( a_y = 2.3 \frac{m}{s^2} \). For heavy commercial vehicles this is a relatively high lateral acceleration. State of the art rollover mitigation systems already start to intervene at \( 2.5 \frac{m}{s^2} \). An empty truck is simulated with a total mass of about \( 15 \text{ t} \).

For comparison four settings of the reference generator are investigated, in which the trailer wheel base \( l_2 \) is changed, cf. Fig. 5. The nominal distance between the kingpin and the second trailer axle is \( 7.7 \text{ m} \). From this starting point the wheel base \( l_2 \) is changed by \(+1, -1 \text{ and } -2 \text{ m} \). The results are shown in Fig. 6.

The curve for \( l_2 = 7.7 \text{ m} \) starts with rather small steering angles. This is not surprising as this is the nominal trailer wheel base and thus just a small control effort is necessary to keep \( \sigma \) small. At \( t \approx 3 \text{ s} \) the slip angle at the axle would become too high and the controller starts to avoid excessive slip angles. Hence, the buckle in the steering angle curve. As the controller becomes inactive, \( \sigma \) increases rapidly.

When the reference wheel base is changed to \( l_2 = 8.7 \text{ m} \) the controller tries to push the trailer in the direction of the curve center, as expected. Already at \( t \approx 2 \text{ s} \) the slip angle limit is reached and the controller becomes inactive. Like stated in the former case, \( \sigma \) increases rapidly from this point on.
For the cases $l_2 = 6.7\, \text{m}$ and $l_2 = 5.7\, \text{m}$ the slip angle limit is not reached. Throughout the whole driving maneuver the sliding condition can be fulfilled, resulting in low $\sigma$. These cases are more controller friendly than extending the reference wheel base, due to the vehicle kinematics.

All steering outputs produced by the controller are smooth as a result of the 2-SMC. Just the residual noise in $\sigma$ is due to the switching in the proximity of the sliding surface.

In Table I the maximum offtracking of the trailer axles with respect to the path of the tractor's rear axle are shown. The maximum offtracking usually occurs at the first trailer axle. As a reference the uncontrolled trailer is shown, too. The controller with $l_2 = 8.7\, \text{m}$ was expected to increase the offtracking. But due to the slip angle limitation the controller is switched off rather early to avoid excessive slip angles. This causes a reduction of the offtracking. The same holds for $l_2 = 7.7\, \text{m}$. Even the same offtracking is produced, as the trailer steering angle is the same as in the former case after $t = 3\, \text{s}$. The cases with a further reduction of $l_2$ decrease the offtracking, as expected. For $l_2 = 5.7\, \text{m}$ the offtracking is more than halved in contrast to the unsteered trailer. As the trailer axle is actively controlled with feedback, these results can also be achieved under disturbances in contrast to state of the art feedforward control systems with fixed steering kinematics.

V. Conclusion

In this paper a nonlinear control design approach was shown which reduces the offtracking of a semi-trailer significantly. It is capable of changing the driving behavior of the vehicle towards a defined kinematic reference generator, even when the reference vehicle dynamics differ considerably from the uncontrolled vehicle. The combination of an exact input-output linearization and a second order sliding mode controller provides a robust controller with very good tracking performance.

In the present paper the implementation and practicability of the controller are shown at the example of improved maneuverability. The promising simulation results suggest an extension of the controller to dynamic stabilization, e.g. in obstacle avoidance maneuvers.

References


