Advanced Steer-by-Wire System for Worlds longest Busses*

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Abstract—This article describes a novel automatic steering system for a 36.0m long double articulated bus equipped with four independent steer-by-wire axles. A model-based control design approach is proposed, which uses a kinematic vehicle model to design a train-like guidance system and a force-canceling that avoids excessive mechanical stress in the joints and the chassis.

The control design generalizes the well-known Ackermann-steering system that performs well on available automotive electronic control units (ECU).

Multi-body-system (MBS) simulations show that the proposed approach offers high tracking performance and low mechanical stress in the chassis of the vehicle. Furthermore, road tests with a true scale prototype vehicle confirm the simulation results. Very good test drive results show that the steering system for the first time enables the operation of very long buses even in urban areas.

I. INTRODUCTION

Transport is an important driver for modern and emerging societies. But the steadily and fast increasing demand faces limited resources and results in increased air pollution. Furthermore, higher traffic density requires more sophisticated safety measures. Hence, cleaner and more efficient transport solutions need to be developed in near future.

A promising approach for increased urban public transport needs are long articulated buses as an alternative to light tram systems. These vehicles provide a very high transport capacity which is similar to light rail systems while the life cycle costs are significantly lower. Due to savings for a smaller driver staff the vehicles can be more complex featuring electric or hybrid drive trains, advanced safety features, or even autonomous driving functions.

The key technology for these vehicles is the steering system, which makes them highly maneuverable, enabling a safe and efficient operation in urban areas. The origin of this topic is the AutoTram[®] Extra Grand, a 30,62m long prototype vehicle developed in a German national research project (www.autotram.info). The AutoTram[®] Extra Grand consists of a driver controlled tractor, a full trailer, and a semitrailer. Two suitable steering controllers with reasonable tracking behavior are described in [1].

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Fig. 1. Considered Vehicle Structure



Fig. 2. 3D View of the considered Vehicle

This article introduces a steering system for a new 36m long, double articulated bus. Unlike the AutoTram[®] Extra Grand this vehicle consists of a driver controlled tractor and **two** full trailers (Fig. 1 and Fig. 2) instead of one full and a semi-trailer as investigated in [1]. The main advantage of the two axle trailer is the significantly higher maximum permissible total weight since it is distributed to two axles instead of only one, which is the standard for articulated buses. Furthermore, the whole system is designed for bidirectional operation. The steering system already went through the whole demonstrator development chain starting with controller design and ending with realworld demonstrations.

The vehicle features a unique challenging vehicle structure. More specifically, the two full trailers are attached to its predecessor module by a single joint linkage making the vehicle statically indeterminate. As

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a consequence, an inaccurate steering angle for axle A_3 or A_5 induces unacceptable mechanical stress to important parts of the vehicle structure. According to simulations these forces can easily exceed 20kN [1] for a fully laden vehicle.

To overcome this aspect and to enable a safe and reliable handling of the long vehicle an active steering system needs to be designed. This design can be subdivided into two consecutive tasks to be solved consecutively:

- **Task 1:** Design a steering controller for axles A_4 and A_6 that enables a high maneuverability and keeps a selected design goal like e.g. swept path or interaxle tracking deviations as small as possible.
- **Task 2:** Design a controller for axles A_3 and A_5 that limits mechanical stress to the vehicle's chassis.

This article describes a model-based steering system design based on a kinematic single-track vehicle model and an extension of the well-known Ackermann steering concept. This steering system is designed to perform well on certified but low performant automotive ECUs that can be distributed in the vehicle modules and are interconnected by a CAN network.

A secondary design goal is to minimize the number and costs of required sensors which is achieved by a mixture of feedforward and feedback controllers.

Due to the innovative vehicle structure with two full trailers and single degree of freedom (DOF) connection for the vertical axis directly related work is only partially available. The only aspects considered in literature is related to Task 1 - the high maneuverability and train-like guidance.

Regarding articulated vehicles featuring semitrailers, [2], [3] investigated master-slave guidance systems for slow moving mobile robots. Amongst other, the tramlike guidance of articulated road vehicles was investigated by [4], [5], [6], [7]. However, the developed control systems assumed small steering angles and centripetal forces limiting the applicability especially in urban areas.

In [8], [9] double articulated trucks are investigated, among several vehicle structures. The usage of linear controllers improves the lateral stability and the maneuverability of the vehicles. As a side-effect, the tracking behavior was improved as well. The proposed vehicle structures lack axle A_3 and A_5 which is peculiar to the vehicle considered here.

Other advanced steering systems like e.g. for wheeled loaders (see [10]) are less important for this work. Current publications on the topic are not known.

The article starts with the derivation of both, a kinetic and a kinematic model of the vehicle. Subsequently, the kinematic model is used for controller design. Finally, simulation with the kinetic model and road test results show that the proposed controller performs well and enables a sound operation of the vehicle even in urban areas.



Fig. 3. Single Track Model of the vehicle

II. KINETIC MODEL OF THE VEHICLE

For the steering system design and simulation studies appropriate vehicle models are required. For the steering controller design a kinematic model was selected to keep the overall complexity small enough for a later ECU implementation. The simulations studies have to investigate the influence of centripetal forces and tire characteristics. Hence, a kinetic model is required. Subsequently, such a kinetic vehicle model is derived. The kinematic model is obtained by setting all tire slip angles of the kinetic model to zero. A detailed derivation of the kinematic model is omitted in this article.

Since the steering and articulation angles are typically greater than 40° with respect to the center position both models will remain non-linear in order to fit to real world applications. For public transport centripetal accelerations of around $2m/s^2$, and velocities up to at least 15m/s are required.

Within this chapter, a nonlinear kinetic vehicle model is derived that covers the new vehicle structure including the essential effects of axles A_3 and A_5 . The model is an extension of the single track model for planar motion presented in [11] that particularly neglects nicking, rolling, and translation in *z*-direction. A more detailed description for general *n*-times articulated vehicles is presented in [12].

For nonlinear control design and for modularity and flexibility reasons it is useful to keep the vehicle model linear in its inputs by assuming the tire forces as inputs of the vehicle and using an inverse tire model to calculate the steering angles that result in the desired tire forces. This way, the tire model can be exchanged easily.

A. Vehicle Model

The considered vehicle structure is shown in Fig. 3. The vehicle consists of a tractor with a steerable front axle A_1 and a rigid driven axle A_2 . Attached to the tractor are two full trailers with two steerable axles each.

It is assumed that the tractor and each semitrailer are moving on the $x_{\mathcal{I}}$ - $y_{\mathcal{I}}$ -plane. Together with the one DOF articulation joints G_1 and G_2 this results in $n_h =$



Fig. 4. FORCES ACTING ON THE VEHICLE AT AXLE A_i

13 holonomic constraints. Thus the multi body system (MBS) has $n_f = 5$ DOF. A suitable set of n_f minimal coordinates is

$$\mathbf{z} = \begin{pmatrix} x, & y, & \psi_1, & \psi_2, & \psi_3 \end{pmatrix}$$
(1)

where the coordinates *x* and *y* denote the position of the coordinate system (CS) \mathcal{K}_1 located in the center of gravity (CG) of the tractor and where ψ_i are the yaw angles of each vehicle module.

For MBS with only holonomic constraints, the equations of motion with respect to z can be derived by applying Lagrange's Equation:

$$\frac{\mathrm{d}}{\mathrm{d}t} \left(\frac{\partial \mathrm{E}_{Kin}}{\partial \dot{\mathbf{z}}}\right)^{\mathsf{T}} - \left(\frac{\partial \mathrm{E}_{Kin}}{\partial \mathbf{z}}\right)^{\mathsf{T}} = \mathbf{q}$$
(2)

In this equation, E_{Kin} is the kinetic energy of the vehicle and **q** are the generalized forces acting on it. Following the explanations in [12] the equations of motion for the new vehicle structure can be written as

$$\mathbf{M}(\mathbf{z})\ddot{\mathbf{z}} + \mathbf{G}(\mathbf{z},\dot{\mathbf{z}})\dot{\mathbf{z}} = \mathbf{q}$$
(3)

where $\mathbf{M}(\mathbf{z})$ is the generalized mass matrix, the term $\mathbf{G}(\mathbf{z}, \dot{\mathbf{z}})\dot{\mathbf{z}}$ models the centrifugal forces and the vector \mathbf{q} represents the generalized impressed forces acting on the vehicle.

The generalized forces **q** are caused by tire-road contact forces. To keep the vehicle model linear in its inputs and independent from the steering angles γ_i , the forces \mathbf{f}_{A_i} acting on the tires of the *i*-th axle are represented in the corresponding module fixed CS \mathcal{K}_j (see Fig. 4):

$$\mathbf{f}_{A_i/\mathcal{K}_j} = \begin{pmatrix} x_{f_{A_i/\mathcal{K}_j}} \\ y_{f_{A_i/\mathcal{K}_j}} \end{pmatrix} \quad \text{with } j = \begin{cases} 1 & \text{for } i = 1, 2 \\ 2 & \text{for } i = 3, 4 \\ 3 & \text{for } i = 5, 6 \end{cases}$$
(4)

Using the principle of virtual displacement and virtual work, the effect of these forces with respect to the minimal coordinate z can be calculated by

$$\mathbf{q} = \sum_{i=0}^{n} \left(\frac{\partial \mathbf{r}_{\mathcal{I} \mathcal{A}_{i}}}{\partial \mathbf{z}^{\mathsf{T}}} \right)^{\mathsf{T}} \mathbf{f}_{A_{i}}$$
(5)

where $\mathbf{r}_{\mathcal{I}\mathcal{A}_i}$ is the position vector of the *i*-th axle depending on \mathbf{z} . In order to substitute \mathbf{f}_{A_i} by its representation in \mathcal{K}_j , the transformation

$$\mathbf{A}_{\mathcal{I}\mathcal{K}_j} = \begin{pmatrix} \cos\psi_j & -\sin\psi_j \\ \sin\psi_j & \cos\psi_j \end{pmatrix}$$
(6)

is introduced. With

$$\mathbf{J}_{A_i} = \frac{\partial \mathbf{r}_{\mathcal{I},\mathcal{A}_i}}{\partial \mathbf{z}^{\mathsf{T}}} \tag{7}$$

follows for Eqn. (5)

$$\mathbf{q} = \sum_{i=0}^{n} \mathbf{J}_{A_{i}}^{\mathsf{T}} \mathbf{A}_{\mathcal{I}\mathcal{K}_{j}} \mathbf{f}_{A_{i}/\mathcal{K}_{j}} = \sum_{i=0}^{n} \mathbf{J}_{A_{i}/\mathcal{K}_{j}}^{\mathsf{T}} \mathbf{f}_{A_{i}/\mathcal{K}_{j}}$$
(8)

This equation subsumes the longitudinal forces \mathbf{f}_{dr} and lateral forces \mathbf{f}_{cr} . To separate them the sum in Eqn. (8) is replaced by scalar products yielding

$$\mathbf{q} = \mathbf{J}_{dr} \, \mathbf{f}_{dr} + \mathbf{J}_{cr} \, \mathbf{f}_{cr} \tag{9}$$

with

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$$\mathbf{f}_{dr} = \begin{pmatrix} x f_{A_1/\mathcal{K}_1}, & x f_{A_2/\mathcal{K}_1}, & \dots, & x f_{A_6/\mathcal{K}_3} \end{pmatrix}'_{\mathsf{T}}$$
(10)

$$\mathbf{f}_{cr} = \begin{pmatrix} y_{f_{A_1/\mathcal{K}_{1'}}} & y_{f_{A_2/\mathcal{K}_{1'}}} & \dots & y_{f_{A_6/\mathcal{K}_3}} \end{pmatrix}$$
(11)

$$\mathbf{J}_{dr} = \begin{pmatrix} 1: \mathbf{J}_{A_1/\mathcal{K}_1}, & 1: \mathbf{J}_{A_2/\mathcal{K}_1}, & \dots, & 1: \mathbf{J}_{A_6/\mathcal{K}_3} \end{pmatrix} \quad \text{and}$$
(12)

$$\mathbf{J}_{cr} = \begin{pmatrix} 2: \mathbf{J}_{A_1/\mathcal{K}_1}^{\mathsf{T}}, & 2: \mathbf{J}_{A_2/\mathcal{K}_1}^{\mathsf{T}}, & \dots, & 2: \mathbf{J}_{A_6/\mathcal{K}_3}^{\mathsf{T}} \end{pmatrix}$$
(13)

Substituting **q** in Eqn. (3) by Eqn. (9) yields for the vehicle dynamics in the CS \mathcal{I}

$$\mathbf{M}(\mathbf{z})\ddot{\mathbf{z}} + \mathbf{G}(\mathbf{z}, \dot{\mathbf{z}})\dot{\mathbf{z}} = \mathbf{J}_{dr}(\mathbf{z})\,\mathbf{f}_{dr} + \mathbf{J}_{cr}(\mathbf{z})\,\mathbf{f}_{cr} \qquad (14)$$

where the forces \mathbf{f}_{dr} and \mathbf{f}_{cr} are the inputs of the vehicle model.

For control design the equations of motion are transformed into the vehicle fixed CS \mathcal{K}_1 . Applying

$$\dot{\mathbf{z}} = \begin{pmatrix} \mathbf{A}_{\mathcal{I}\mathcal{K}_1} & \mathbf{0} \\ \mathbf{0} & \mathbf{I} \end{pmatrix} \dot{\mathbf{z}}_{\mathcal{K}_1} = \mathbf{T}_{\mathcal{I}\mathcal{K}_1} \dot{\mathbf{z}}_{\mathcal{K}_1} \quad \text{and} \qquad (15)$$

$$\ddot{\mathbf{z}} = \mathbf{T}_{\mathcal{I}\mathcal{K}_1} \, \ddot{\mathbf{z}}_{\mathcal{K}_1} + \dot{\mathbf{T}}_{\mathcal{I}\mathcal{K}_1} \, \dot{\mathbf{z}}_{\mathcal{K}_1} \tag{16}$$

to Eqn. (14) yields after left multiplication with $\mathbf{T}'_{\mathcal{IK}_1}$

$$\mathbf{M}_{\mathcal{K}_{1}}\ddot{\mathbf{z}}_{\mathcal{K}_{1}} + \mathbf{G}_{\mathcal{K}_{1}}\dot{\mathbf{z}}_{\mathcal{K}_{1}} = \mathbf{J}_{dr/\mathcal{K}_{1}}\mathbf{f}_{dr} + \mathbf{J}_{cr/\mathcal{K}_{1}}\mathbf{f}_{cr}$$
(17)

with

1

$$\mathbf{M}_{\mathcal{K}_1} = \mathbf{T}_{\mathcal{I}\mathcal{K}_1}^{\mathsf{T}} \mathbf{M} \mathbf{T}_{\mathcal{I}\mathcal{K}_1}$$
(18)

$$\mathbf{G}_{\mathcal{K}_{1}} = \mathbf{T}_{\mathcal{I}\mathcal{K}_{1}}^{\mathsf{I}} \left(\mathbf{M} \, \dot{\mathbf{T}}_{\mathcal{I}\mathcal{K}_{1}} + \mathbf{G} \, \mathbf{T}_{\mathcal{I}\mathcal{K}_{1}} \right) \tag{19}$$

$$\mathbf{J}_{dr/\mathcal{K}_1} = \mathbf{T}'_{\mathcal{I}\mathcal{K}_1} \mathbf{J}_{dr} \quad \text{and} \tag{20}$$

$$\mathbf{J}_{cr/\mathcal{K}_1} = \mathbf{T}'_{\mathcal{I}\mathcal{K}_1} \, \mathbf{J}_{cr} \tag{21}$$

The dependencies on $\mathbf{z}_{\mathcal{K}_1}$ and $\dot{\mathbf{z}}_{\mathcal{K}_1}$ have been omitted for brevity.



Fig. 5. TIRE FORCES AND VELOCITY VECTOR AT AXLE A_i

B. Tire Model

The tire model describes the relation between tire forces and steering angles, vehicle state, and tire properties. These relations are formulated in the tire fixed CS A_i as depicted in Fig. 5.

In order to represent the forces $\mathbf{f}_{A_i/\mathcal{K}_j}$ in the corresponding tire CS \mathcal{A}_i , the transformation

$$\mathbf{f}_{A_i/\mathcal{A}_i} = \begin{pmatrix} {}^{x} f_{A_i/\mathcal{A}_i} \\ {}^{y} f_{A_i/\mathcal{A}_i} \end{pmatrix} = \mathbf{A}_{\mathcal{A}_i \mathcal{K}_j} \mathbf{f}_{A_i/\mathcal{K}_j}, \ j = \begin{cases} 1 & \text{for } i = 1, 2 \\ 2 & \text{for } i = 3, 4 \\ 3 & \text{for } i = 5, 6 \end{cases}$$
(22)

is applied with

$$\mathbf{A}_{\mathcal{A}_i \mathcal{K}_j} = \begin{pmatrix} \cos \gamma_i & \sin \gamma_i \\ -\sin \gamma_i & \cos \gamma_i \end{pmatrix}$$
(23)

and γ_i the steering angle of axle A_i (Fig. 4).

In Eqn. (22), ${}^{x}f_{A_{i}/A_{i}}$ is the driving force and ${}^{y}f_{A_{i}/A_{i}}$ is the lateral force at the tires of axle A_{i} . The driving forces are assumed as inputs whereas the lateral forces are modeled using a linear approach depending on the slip angle φ_{i}

$${}^{y}\!f_{A_i/\mathcal{A}_i} = c_i \, \varphi_i \qquad (i = 1, 2, \dots, 6)$$
 (24)

with the cornering stiffness of the tire c_i . This tire model holds for centripetal accelerations up to 4 m/s^2 ([13], [14], [15]).

The tire slip angle is the difference between steering angle γ_i and side slip angle β_i , see Fig. 5:

$$\varphi_i = \gamma_i - \beta_i \tag{25}$$

where the side slip angle is calculated by

$$\beta_i = \arctan \frac{{}^{y} v_{A_i/\mathcal{K}_j}}{{}^{x} v_{A_i/\mathcal{K}_i}} \text{ and }$$
(26)

$$\mathbf{v}_{\mathcal{I}\mathcal{A}_i/\mathcal{K}_j} = \begin{pmatrix} x_{\mathcal{V}_{A_i}/\mathcal{K}_j} \\ y_{\mathcal{V}_{A_i}/\mathcal{K}_j} \end{pmatrix} = \mathbf{J}_{A_i/\mathcal{K}_j} \dot{\mathbf{z}}$$
(27)

With Eqn. (22) and Eqn. (24)-(27) follows for the nonlinear tire model interfacing with the vehicle model derived in section II-A

$$\mathbf{f}_{A_i/\mathcal{K}_j} = \mathbf{A}_{\mathcal{K}_j\mathcal{A}_i} \begin{pmatrix} {}^{x}f_{A_i/\mathcal{A}_i} \\ {}^{y}v_{A_i/\mathcal{K}_j} \\ c_i \left(\gamma_i - \arctan \frac{{}^{y}v_{A_i/\mathcal{K}_j}}{{}^{x}v_{A_i/\mathcal{K}_j}} \right) \end{pmatrix}.$$
(28)

The input of the tire-model is γ_i (i = 1, 2, ..., 6). The overall model makes it easy to configure the vehicle with steerable and non-steerable axles. To make an axle non-steerable the corresponding angel just needs to be zero.

III. CONTROLLER DESIGN

A key requirement for the steering system is the ability to run on automotive ECUs. Hence, the steering control system of the vehicle is based on a simplified kinematic vehicle model instead of the kinetic model derived in section II. The kinetic model is used for simulation studies only.

Even though the kinematic model limits the accuracy especially for high lateral accelerations it is still a very suitable approach for commercial and especially public transport vehicles.

The control design approach follows a two-step procedure. In a first step the steering angles of axles A_4 and A_6 are calculated to achieve a high maneuverability and tracking performance. In a second step the steering angles of axle A_3 and A_5 are calculated with the intention to restrict mechanical stress for the chassis.

The steering control design is based on the idea that the center points of axles A_4 and A_6 need to travel along a path prescribed by a selected point on the first vehicle module - e.g. A_2 . This leads to the following controller design tasks:

- **Task 1 Train-like Guidance:** Feedforward control guiding axles A_4 and A_6 approximately on the path of A_2 .
- **Task 2 Force Minimization:** Feedforward steering of axle A_3 and A_5 in order to minimize the lateral force in hinges G_1 and G_2 .

In order to solve task 1 the well known principle of Ackermann-Steering is applied. The driver-selected steering angle of the first axle and the geometric vehicle properties lead to a circle on which the first vehicle module is currently moving. Usually, this circle is constantly changing according to the drivers steering inputs and only fixed in steady state circular driving situations. The vehicle modules 2 and 3 are intended to always follow the consecutive circles described by the first module in order to achieve exact path tracking.

1) Task 1 - Approximate Train-like Guidance: The calculation of the steering angles of axles A_4 and A_6 is based on Fig. 6. The axles are steered in such a way that the vehicle has one instantaneous center of rotation M. The additional constraint

$$R_2 = R_4 = R_6 \tag{29}$$

forces axles A_4 and A_5 to move on the same circle described by axle A_2 . The radius R_2 is calculated from the steering angle of the first axle

$$R_2 = \frac{l_1}{\tan \gamma_1}.$$
 (30)



Fig. 6. KINEMATICS WITH ONE CENTER OF ROTATION

This leads to the radius R_{g1} on which the first hinge center is moving.

$$R_{g1}^2 = l_g^2 + R_2^2 \tag{31}$$

In the triangle A_4 - G_1 -M the following equation can be found with the constraint defined by Eqn. (29) applied

$$\beta_1 = \arccos\left(\frac{R_{g1}^2 + l_2^2 - R_2^2}{2R_{g1}l_2}\right).$$
 (32)

With

$$p_4 = \gamma_4 + \frac{\pi}{2} \tag{33}$$

and the law of sines in the same triangle, the steering angle of axle A_4 is calculated by

$$\gamma_4 = \arccos\left(\frac{R_{g1}}{R_2}\sin\beta_1\right). \tag{34}$$

With the steering angle γ_4 , the radius of the second hinge's movement can be calculated by

$$R_{g2} = l_{g2}^2 + R_4^2 - 2l_{g2}R_4\cos\left(\frac{\pi}{2} - \gamma_4\right)$$

= $l_{g2}^2 + R_4^2 - 2l_{g2}R_4\sin\gamma_4.$ (35)

Following the same procedure as for the calculation of γ_4 , the steering angle of axle A_6 is calculated by

$$\beta_2 = \arccos\left(\frac{R_{g2}^2 + l_3^2 - R_2^2}{2R_{g2}l_3}\right)$$
(36)

$$\gamma_6 = \arccos\left(\frac{R_{g2}}{R_2}\sin\beta_2\right).$$
 (37)

With the constraint given by Eqn. (29) the calculated steering angles for axles A_4 and A_6 are only valid for steady state circular driving. Therefore, an addition is necessary, coping with transient steering behavior e.g. when entering or leaving a circle.

If the vehicle is going from straight forward driving to steady state circular driving the steering angles of axle A_4 and A_6 have to change from zero to the steady state values calculated above. This transition is performed by first-order lag elements. Instead of the



Fig. 7. KINEMATICS WITH TWO CENTERS OF ROTATION

usual time-based delay, the tracks arc length is used as basis resulting in an arc lag. This allows the vehicle to travel at arbitrary speed without changing the steering behavior, except for any kinetic effects this approach neglects intrinsically.

Transforming a first-order arc lag into a first-order time lag results in a velocity dependent *time constant*

$$T = \frac{S}{v_k}, v_k > 0, \tag{38}$$

where *S* is the *way constant* and v_k the velocity at the timestep *k*.

The well known discrete first-order dynamics

$$T\frac{y_k - y_{k-1}}{\Delta t} + y_k = Ku_k \tag{39}$$

can be rewritten as

$$y_k = \frac{1}{\frac{T}{\Delta t} + 1} \left(K u_k - y_{k-1} \right) + y_{k-1}.$$
 (40)

Inserting Eqn. (38) yields

$$y_k = T_k^* \left(K u_k - y_{k-1} \right) + y_{k-1}, \tag{41}$$

with

$$T_k^* = \frac{v_k \Delta t}{v_k \Delta t + S}.$$
(42)

Although excluded in Eqn. (38) this implementation works for $v_k = 0$, too.

2) Task 2 - Force Minimization: As already mentioned, axles A_3 and A_5 need to be steered in such a way that it does not create unwanted lateral forces. As for axles A_4 and A_6 , a kinematic approach is presented here.

The steering angles of axle A_3 and A_5 depend on the current vehicle state and therefore on the current steering and articulation angles. During transient maneuvers these values significantly differ from the steady state circular driving values, resulting in individual instantaneous centers of motion for each vehicle module. Thus, constraint Eqn. (29) does not hold in transient maneuvers. This situation is depicted in Fig. 7 where γ_1 and R_2 are < 0.



Fig. 8. TRACES OF AXLES A2, A4 AND A6

With Eqn. (30) the angle α_1 can be calculated

$$\alpha_1 = \arctan\left(\frac{R_2}{l_{g1}}\right),\tag{43}$$

which leads to

$$\beta_1 = -\alpha_1 + \Delta \Psi_1. \tag{44}$$

Applying the law of sines in the triangle A_4 - G_1 - M_2 with Eqn. (33) and

$$\xi = \pi - \beta_1 - \varphi_4 \tag{45}$$

results in

$$R_{gM2} = l_2 \frac{\cos \gamma_4}{\sin \xi}.$$
 (46)

The radius R_3 can be calculated using the law of cosines in the triangle A_3 - G_1 - M_2

$$R_3^2 = R_{gM2}^2 + l_3^2 - 2R_{gM2}l_3\cos\beta_1.$$
 (47)

Finally, the steering angle of axle A_3 is given by

$$\gamma_3 = \arccos\left(\frac{R_{gM2}}{R_3}\sin\beta_1\right). \tag{48}$$

The steering angle γ_5 is calculated in the same way.

IV. RESULTS

The driving maneuver used for obtaining the presented results is a standardized test maneuver for commercial vehicles that ensures an appropriate maneuverability for public roads and especially urban infrastructure. Starting from straight forward driving, the outermost front point of the vehicle is guided along a full circle with radius 17.5m. After 360° the vehicle drives straight forward again. Hence, beside steady state driving this maneuver includes transients into and out of the circle as well, which are in general the most challenging parts.

Fig. 8 shows the simulated traces of axles A_2 , A_4 and A_6 for the test maneuver. As the movement of the first module is hardly influenced by the steering of module 2 and 3, just one trace for axle A_2 is shown and considered as the reference. It can be clearly seen



Fig. 9. DEVIATIONS OF AXLES A_4 AND A_6 WITH RESPECT TO THE TRACE OF AXLE A_2



Fig. 10. LATERAL FORCE IN FIRST HINGE

that a large offset occurs if axles A_4 and A_6 are not steered. This can be reduced significantly by the proposed steering system. Figure 9 shows the deviations of axles A_4 and A_6 with respect to the trace of axle A_2 . The maximum deviation of approximately 6m without steering is reduced to less than 1.0m in general. Hence, the simplified steering approach improves the maneuverability and reduces the swept path significantly.

Furthermore, Fig. 10 shows the lateral force in the first joint. The maximum force caused by the simplified steering algorithm of axle A_3 is less than 4kN. This value is small enough not to deteriorate the driving behavior of the vehicle by excessive lateral tire forces at axle A_2 . The joint can withstand this lateral load continuously.

The steering algorithm is implemented on three automotive ECUs - one for each vehicle module. The ECUs are interconnected by CAN. They are reading the steering and articulation angles of their respective vehicle module as well as the velocity and calculate the new steering angels as described above. The systems overall cycle time is 10ms. Driving tests proved that this cycle time is fast enough for the vehicles maximum speed of 60 km/h.

The road tests with a true scale prototype vehicle confirmed the promising simulation results. The swept road area proved to be comparable to the area used by conventional 18.0m single-articulated buses. This is a remarkable result. It potentially enables the operation of ultra long buses in urban areas that are currently serviced by conventional articulated buses.

In addition to the good tracking performance, lanechange, slalom and emergency braking maneuvers verified that the vehicle is stable not only on dry but also on wet roads as well. Furthermore, two directional operation was successfully tested. Due to the symmetric vehicle structure the vehicle handling and the tracking performance is independent of the driving direction.

V. CONCLUSION

The article proposes an automatic steering controller for double-articulated road vehicles whereas each of the three vehicle modules is equipped with two independent steer-by-wire axles. The presented steering control system requires only sensor data like e.g. steering and articulation angels that can be obtained easily and are available in most articulated buses. Additional infrastructure like e.g. (D)GPS, magnetic markers, or road markings are not required. Even though the vehicle is 36m long the driver can easily control it. The vehicle handling is comparable to a conventional 18m bus.

Simulation results as well as road tests showed that the proposed steering control systems provides a very good tracking performance enabling a safe and reliable operation of long multiple articulated vehicles on public roads even in urban areas enabling a wide range of operation scenarios including replacement of light tram systems in medium and big cities.

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