Steering Method for Automatically Guided Tracked Vehicles

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Key words: Tracked Vehicles, Robot Tachymeter, Steering Method

SUMMARY

The objective of this contribution is the design of an alternative steering method for a two-track crawler model, based on a scalable force transmission. Generally, the steering method of a two-track crawler chassis is based on a skid-steer concept (Beetz 2012). Thereby the curve drive is realized by adjusting different track velocities on the individual tracks. The resulting difference between the velocities of the right and left track has a direct influence on the curve radius. The bigger the difference, the smaller is the resulting curve radius. Such steering method is applied for bulldozers, track loaders and excavators (Beetz 2012). Generally, mechanically compounded drives are common for tracked vehicles. Among them, step-less drives or infinitely variable drives are most superior (Gebhardt 2010).

The Institute of Engineering Geodesy (IIGS), University of Stuttgart operates a crawler model, at scale 1:14, which is part of the construction machine simulator, that has been developed to test and evaluate different sensors, sensor combinations, as well as filter and control algorithms (Beetz 2012, Lerke and Schwieger 2015). Further components of the simulator are a robot tachymeter Leica TS30, a control computer, an analogue/digital converter and a remote control. The crawler model, that has been used for the current investigation, has a two-stage continuous electric drive and thus complies with the requirements of the step-less drive functionality.

For the design of the new steering method a mechanical differential steering block has been used as a role model. The key part of such a block is the compensating gear, which balances different velocities of the inner and outer track during the curve drive. The approach is based on the kinematic model for tracked vehicles according to Le (1991), where the equation, which describes the relationship between radius and different velocities for the right and left track, has been modified and solved in a way, that a scaling factor n could be derived. This scaling factor corresponds to the functionality of the compensating wheel of the differential steering block. The factor n scales the driving forces for the two tracks. Left and right track velocities are scaled by this factor and by its reciprocal respectively, in order to perform the corresponding curve drives.

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1. INTRODUCTION

Construction machines play a significant role in construction processes (Kühn 1991). Nowadays automation is finding its way onto construction sites. The construction machines can be categorized according to their degree of automation (Stempfhuber and Ingesand 2008) or under aspects of their application field, e.g. transportation, roadworks or earthworks (Kühn 1991). Earthwork machines can be divided into two main groups, according to their chassis design: wheeled chassis and tracked chassis (Gebhardt 2010). Tracked vehicles have advantages over wheeled vehicles with regard to traction and soil compaction. The disadvantages are the lack of mobility outside the application field as well as lower working velocities.

The objective of this investigation is the design of an alternative steering method for tracked vehicles, which is limited to movements without slippage. The steering method of a two-track crawler chassis is based on a skid-steer concept (Beetz 2012). Thereby the curve drive is achieved by adjusting different track velocities on left and right track. The difference of the velocities from both tracks has a direct influence on the curve radius. The bigger this difference is the smaller is the resulting radius. Such a steering method is applied for bulldozers, track loaders and excavators (Beetz 2012).

Generally, mechanically compounded drives are common for tracked vehicles. Hydrostatic drives are most widespread (Gebhardt 2010). The advantages of hydrostatic drives are low-wear steering, easy reversing operations and possible spot turns. For steering operations however, the hydrostatic drives will not achieve any improvements to the conventional steering systems. Here, step-less drive or infinitely variable drives are still superior. The model crawler, operated by the Institute of Engineering Geodesy (IIGS), fully complies with the requirements of the step-less drive functionality.

2. METHODOLOGY

2.1 Construction Machine Simulator – Hardware In-the-Loop System

The IIGS simulator has been used for testing of the above descripted method. The simulator has been developed to test and evaluate different sensors, sensor combinations, as well as filter and control algorithms (Beetz 2012, Lerke and Schwieger 2015). The simulator system consists of a crawler model at scale 1:14. Further components for the presented purpose are a robot tachymeter Leica TS30, a control computer, an analogue/digital converter and a remote control. The crawler model, used for the current investigation, has a two-stage continuous electric drive and thus complies with the requirements of the step-less drive functionality. Figure 1 depicts the simulator system with its hardware components and the data flow.

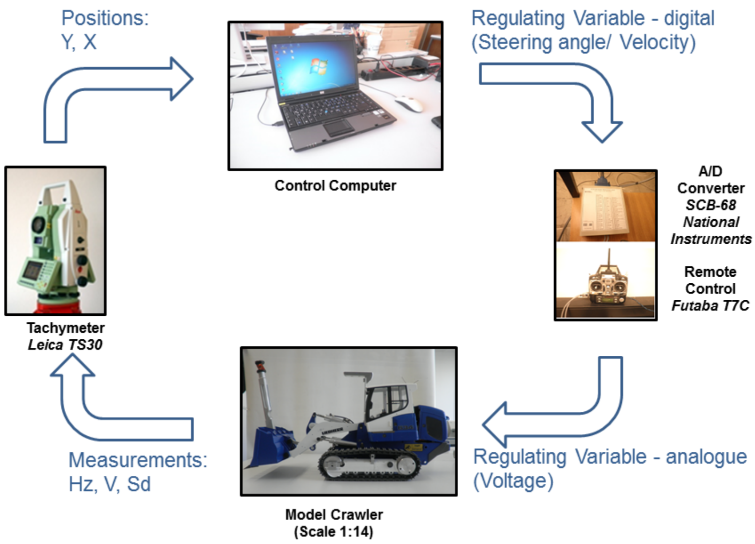

Figure 1: Hardware components of the simulator

The control of the model crawler is realized by a closed-loop-system. The scheme of the closed-loop-system is depicted in figure 2. Table 1 shows the general variables of the closed-loop system and the appropriate simulator items. The loop performs as follows: the tachymeter measures the position of the prism y(t), mounted in the centre of gravity of the crawler model and sends it to the control computer. The computer calculates the perpendicular distance/lateral deviation e(t) between the crawler position and the reference trajectory. Based on this information, the algorithm calculates the steering angle respectively the regulating variable u(t), to get the crawler back on the reference trajectory as fast as possible. The steering angle of the model crawler is directly linked with the driving radius and thus with the velocities of the left and right track. The radius is tightly connected to the ratio between the left and right track velocities. In turn, the factor n, which describes this ratio, is part of the term which scales the driving forces for the two tracks. The loop sequence is executed 8 to 10 times per second. The rate is mainly depending on the kinematic measurement ability of the used robot tachymeter. According to the instrument's data sheet the sampling rate is between 8 and 10 Hertz (Leica 2015).

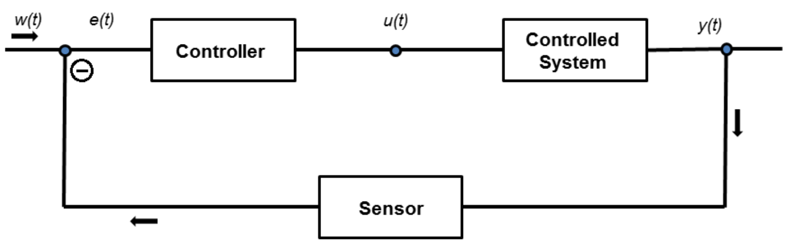

Figure 2: Closed-loop-system

Table 1: Closed-Loop System Variables

Variable	Meaning within Closed-Loop	Appropriate Simulator Item
w(t)	reference variable	reference trajectory
e(t)	control deviation	lateral deviation between reference trajectory and actual position
u(t)	regulating variable	steering ratio
y(t)	controlled variable	position

In order to control the closed-loop system, a PID controller has been chosen. The PID controller output signal can be established as follows (Busch 2012):

$$x_{outPID} = K_P \cdot x_e + K_I \cdot x_e \cdot \Delta t + K_D \cdot \frac{x_e}{\Delta t}. \tag{1}$$

K_P – proportional gain,
K_I – integral gain,
K_D – differential gain,
x_e – input signal.

The output signal can also be described as a function of hold-back time T_v and the reset time T_n (Busch 2012). Thus the following equation for the PID output can be established:

$$x_{outPID} = K_P \cdot \left(x_e + \frac{1}{T_n} \cdot x_e \cdot \Delta t + T_v \cdot \frac{\Delta x_e}{\Delta t}\right) \tag{2}$$

The PID controller combines the advantages of the individual base terms, namely P-term, I-term and D-term. Thus it complies with the requirements of high control speed and high accuracy. However, an optimal control performance can only be achieved by an exact tuning of the three parameters K_P, K_I ,K_D, respectively K_P, T_n, T_v. This tuning can be realized by different methods, as e.g. approximation methods according to Chien, Hrones and Reswick (CHR). The CHR method is applicable if the parameters of the plant being controlled, are known. In case of unknown plant parameters the method of Ziegler and Nichols, which is based on controller stability limit, is better suited (Mann et al. 2005). Within this work an empirical approach for the determination of optimal control parameters has been performed.

2.2 Requirements on the Steering Method

For the design of the new steering method some boundary conditions must be satisfied. For this the following three demands have been defined:
The first aspect concerns the curve velocities. The curve drives performed by a classical skid-steer concept are realized by slowing down the inner curve track. As a consequence the total speed of the machine drops due to the following relationship:

$$v_{total} = (\frac{v_l + v_r}{2}) \tag{3}$$

v_{total} – machine's total speed,
v_l – velocity left track,
v_r – velocity right track.

To overcome this disadvantage the requirement on the presented method was, to keep the total speed stable during curve drives.
The second aspect is the limitation of the tracks rotational velocities in order not to exceed the maximum motor performance and thus not to damage the drive power unit:

$$v_l, v_r \leq v_{limit} \tag{4}$$

The last aspect refers to the calibration procedure and the establishment of a transfer function to describe the relationship between voltages applied on the motors and resulting velocities, indicated in metric sizes as e.g. meter per second, respectively kilometer per hour. The goal in this work was to create a simple and quickly executed calibration procedure.

2.3 Mathematical Approach

The mathematical approach is based on the kinematic model for tracked vehicles according to Le (1991). The following steps explaining the derivation of the scale factor n, introduced as steering ratio in table 1, upon which the steering method is based. The presented technique is limited to movements of tracked vehicles without slippage.

According to figure 3 the following equation results:

$$R = \frac{B(v_l + v_r)}{2(v_l - v_r)} \tag{5}$$

R – radius,
B – gauge of the appropriate vehicle,
v_l and v_r – velocities of left and right track.

Solve equation (1) for v_l and v_r.

$$v_l = v_r \frac{(2R+B)}{(2R-B)} \tag{6}$$

$$v_r = v_l \frac{(2R-B)}{(2R+B)} \tag{7}$$

The term $\frac{(2R+B)}{(2R-B)}$ can be redefined as the ratio n between the velocities of both tracks.

$$n = \frac{(2R+B)}{(2R-B)} \tag{8}$$

Hence the formulas (6) and (7) can be rewritten as follows:

$$v_l = v_r n \tag{9}$$

$$v_r = v_l \left(\frac{1}{n}\right) \tag{10}$$

The factor n now allows to physically scale the driving forces. Originating from a straight drive, the two track velocities are the same: $v_l = v_r$. Thus the following expression can be defined for straight drive: $v_{total} = v_l = v_r$. Utilizing equation (3) and in consideration of equations (9) and (10) the expressions (11) can be stated:

$$v_{total} = \frac{1}{2} \cdot \left((v_r \cdot n) + \left(\frac{v_l}{n}\right) \right) = \frac{v_r \cdot n}{2} + \frac{v_l}{2 \cdot n} \tag{11}$$

In order to scale the driving forces for the two tracks, to perform curve drives, the following steps have been performed:
Solve (11) for v_l and v_r:

$$v_l = (2 \cdot n \cdot v_{total} - v_r \cdot n^2) \tag{12}$$

$$v_r = \frac{2 \cdot v_{total}}{n} - \frac{v_l}{n^2} \tag{13}$$

Substitution of v_r in equation (12) using equation (10) gives the speed for the left track:

$$v_l = v_{total} \cdot \frac{2 \cdot n}{1+n} \tag{14}$$

Substitution of v_l in equation (13) using equation (9) gives the speed for the right track:

$$v_r = v_{total} \cdot \frac{2}{1+n} \tag{15}$$

The expressions $\frac{2 \cdot n}{1+n}$ for the left track and $\frac{2}{1+n}$ for the right track represent scaling terms of the machine's total velocity during curve drives. Both track velocities depend on the factor n, which in turn depends on the curve radius R (see (8)).

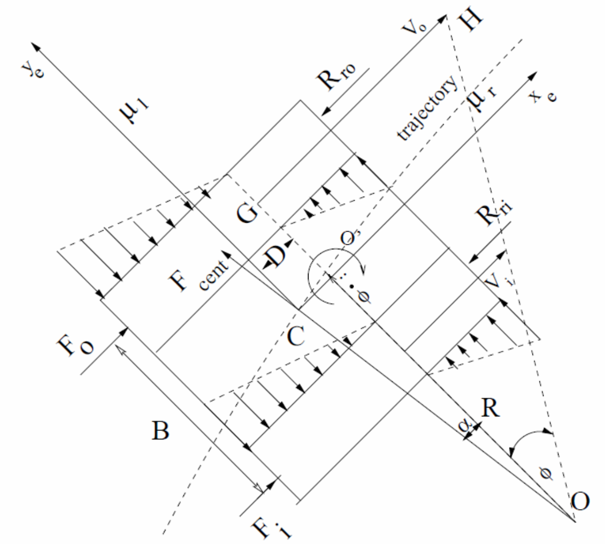

Figure 3: Kinematic motion model for tracked vehicles (Le 1999)

The velocities v_l and v_r have to be within the limits defined in (4). In other cases v_{total} is reduced, whereby n is kept and consequently v_l and v_r have to keep the limiting values. The functionality of the above derived factors allows the compensation and balance of different velocities of the inner and outer track during curve drives.

2.4 Calibration

In this contribution the calibration was limited to forward movements, only. Backward movements or special manoeuvres, like spot turns, have not been inspected. Moreover, the calibration only refers to the transfer function between voltages and velocities, as the relationship between the radius and the scale factor is already specified by equation (8). The two-stage continuous electric drive of the IIGS crawler model is triggered by applying voltage values to the two driving actuators. For this reason the relationship between voltage values and the driving speed of the tracks must be derived. Thus, the objective of the next step is the obtainment of a transfer function between voltages and track velocities. The power characteristic of the actuators is as follows: 0.7 volts is defined as full power state and thus accelerates the tracks to their maximum speed, whereas 1.6 volts state is defined as idle and thus stops the track's rotation completely. The calibration procedure is based on comparison of voltages and resulting velocities. It has been conducted by applying different voltages to the left and right track actuators and subsequently calculating the velocities from coordinate measurements of the tachymeter. For this, a calibration step width of 0.1 volts has been chosen. After each measurement the voltage value has been increased by 0.1 volts and the respective velocity has been calculated subsequently. The procedure covered the complete forward driving voltage range between 1.6 and 0.7 volts. The resulting value pairs have been approximated by polynomial fitting techniques. During the experiment different polynomials approaches have been tested. However, the final decision was made in favour of a polynomial of second order. The transfer function is depicted in figure 4. The appropriate polynomial is represented by equation (16).

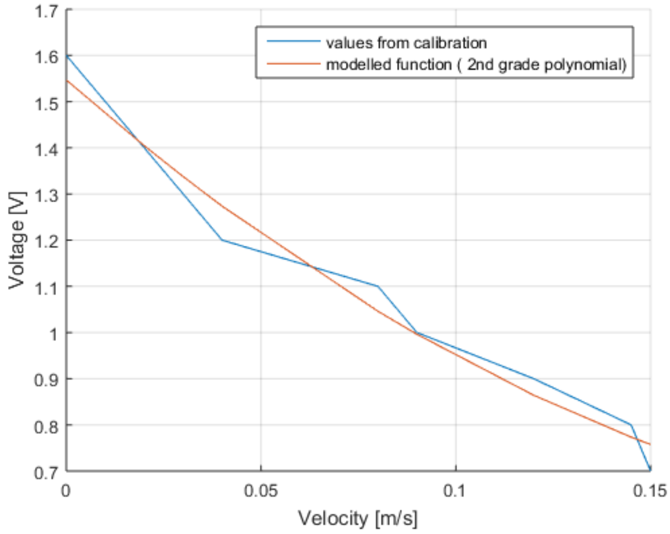

Figure 4: Transfer function between voltage and velocity

A non-linear characteristic of the electric driving actuators of the model crawler is revealed. This is also underlined by the resulting polynomial of second order degree:

$$U = 14.26 \cdot v^2 - 7.40 \cdot v + 1.55 \text{ [V]} \tag{16}$$

This behaviour can be reduced to the non-ideal manufacturing of the used model kit.

3. RESULTS AND ASSESSMENT

To prove the performance of the presented steering method, 3 trajectories have been examined. The trajectories are shaped as oval, eight and kidney and contain the route design elements as clothoids, curves and straight lines. The experimental setup is shown in figure 5.

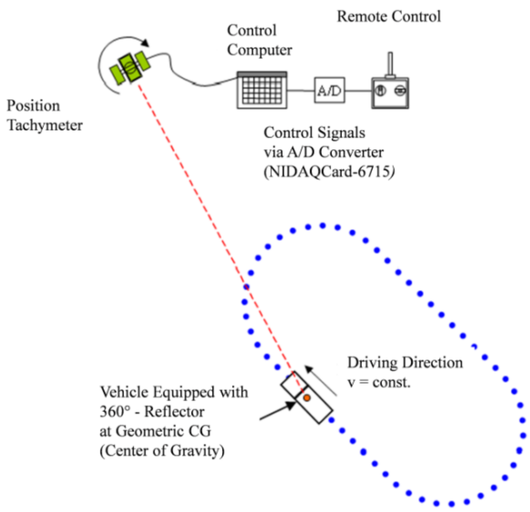

Figure 5: Experimental setup for evaluation (Beetz 2012)

The evaluation has been carried out by comparing the reference trajectory with the effectively driven trajectory and the subsequent calculation of the lateral deviations and their RMS (Beetz 2012).

$$RMS = \sqrt{\frac{\sum_{i=1}^{n} e_i^2}{n}} \tag{17}$$

e_i – lateral deviation
n – number of measurements.

In the following section the results of the conducted test drives are summarized. The depicted RMS values consist of the instrument's measurement accuracy as well as the system's control quality. These values represent the so called "combined measures" (Lerke and Schwieger 2015). In overall, 3 sessions have been performed, in which the 3 trajectories have been examined. For each trajectory, a separately evaluation of the RMS for the 3 route design elements have been made. For optimal identification, the trajectory's sections are colored as follows: green – clothoids, blue – straight lines, red – circle arcs.

Session 1:
Trajectory: Oval
PID controller parameter settings: $K_P = 200, K_I = 0.6, K_D = 1$

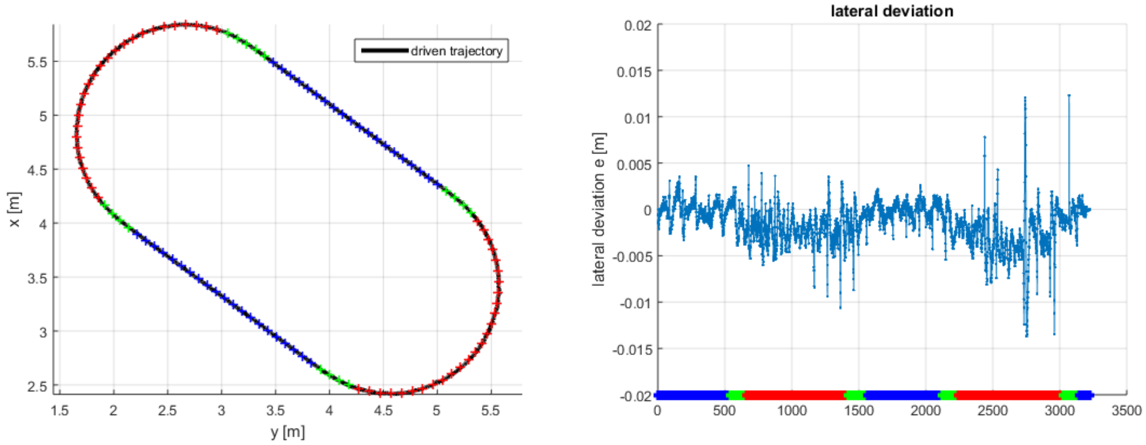

Figure 6: Oval trajectory (left), lateral deviations (right)

Table 2: Results for Oval

	Number	**Individual RMS**	**Average RMS**	**Weighted Total RMS**
Clothoids	1	0.0022 m	0.0021 m	**0.0027 m**
	2	0.0019 m		
	3	0.0021 m		
	4	0.0022 m		
Straight Lines	1	0.0010 m	0.0009 m	
	2	0.0008 m		

Circle Arcs	1	0.0040 m	0.0035 m	
	2	0.0029 m		

<u>Session 2:</u>
Trajectory: Eight
PID controller parameter settings: $K_P = 200, K_I = 0.6, K_D = 1$

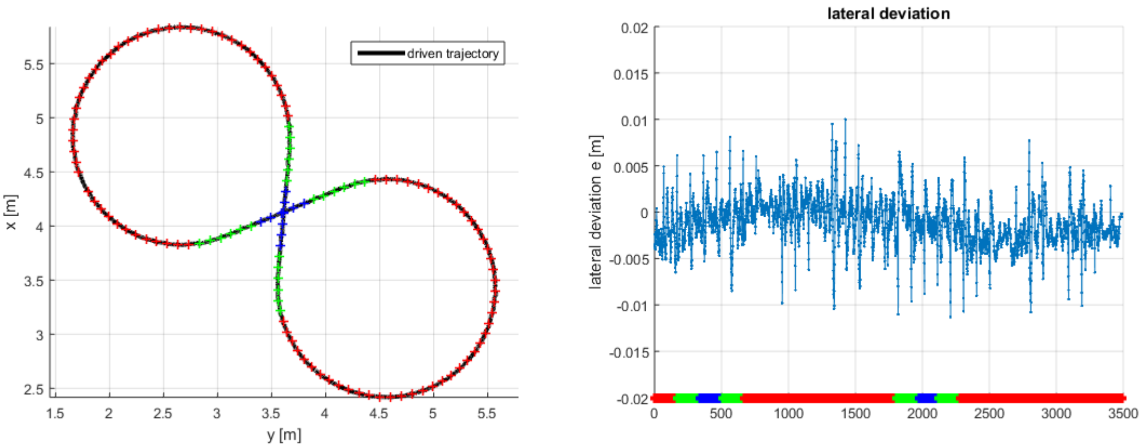

Figure 7: Eight trajectory (left), lateral deviations (right)

Table 3: Results for Eight

	Number	**Individual RMS**	**Average RMS**	**Weighted Total RMS**
Clothoids	1	0,0030 m	0,0027 m	**0,0027 m**
	2	0,0027 m		
	3	0,0024 m		
	4	0,0027 m		
Straight Lines	1	0.0024 m	0.0023 m	
	2	0.0021 m		
Circle Arcs	1	0.0032 m	0.0028 m	
	2	0.0024 m		

<u>Session 3:</u>
Trajectory: Kidney
PID controller parameter settings: $K_P = 200, K_I = 0.6, K_D = 1$

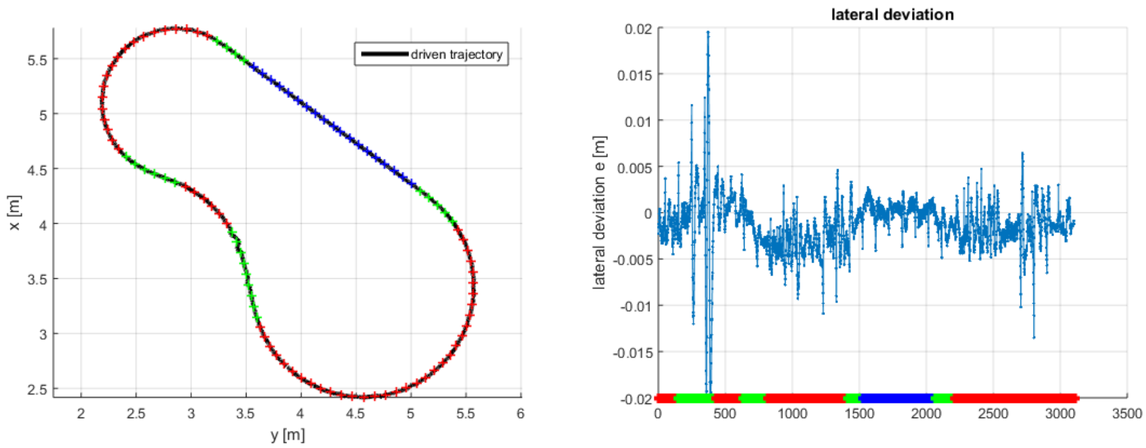

Figure 8: Kidney trajectory (left), lateral deviations (right)

Table 4: Results for Kidney

	Number	Individual RMS	Average RMS	Weighted Total RMS
Clothoids	1	0.0093 m	0.0035 m (0.0023 m)	**0.0033 m (0,002 m)**
	2	0.0039 m		
	3	0.0018 m		
	4	0.0011 m		
	5	0.0029 m		
	6	0.0019 m		
Straight Lines	1	0.0011 m	0.0011 m	
Circle Arcs	1	0.0023 m	0.0026 m	
	2	0.0017 m		
	3	0.0038 m		

In consideration of the kinematic measurement accuracy of the robot tachymeter, which is approx. 3 to 5 mm (Leica 2015), the achieved performance of the presented steering method can be regarded as satisfactory. The straight line sections show the best results. During the drive through the first clothoid of the kidney trajectory, an outlier has been detected. As a consequence, the RMS increased to 9.3 mm. This outlier may probably be caused by a measurement gap of the tachymeter. The outlier can also be identified in figure 8 (first green section). By eliminating that outlier, the results take comparable values to the other trajectories and are lying in the range of 2.3 mm for the clothoid sections and 2 mm for the complete kidney trajectory. In table 4, the RMS values after outlier elimination are depicted in brackets.

4. CONCLUSIONS

A method to steer a two-track crawler model has been developed. The consisting kinematic model for tracked vehicles could be solved and a scaling term, referred as steering ratio, could be derived. This term can be used to scale the track velocities and therefore to perform steering actions. The presented method fully meets the requirements of a constant curve speed, simple calibration procedures and safety limits for the actuators in order not to damage the drive power unit. The evaluation of control and guidance performance by the use of the presented steering method reveals an overall RMS of 2.9 mm, respectively 2.4 mm after outlier elimination, for the 3 different trajectories. Compared to earlier investigations on a different steering method, as e.g. Beetz (2012), the results are in the same range and thus can be regarded as satisfactory. In the future the steering model must be extended by slippage to illustrate more realistic driving scenarios as e.g. loose ground on construction sites.

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BIOGRAPHICAL NOTES

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