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Development of a Driver Assistance System for Long Timber Transportation Trucks to Improve Tracking Behavior

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Abstract— **This paper introduces the newest development of a driver assistance system for log-hauling trucks, which transport long heavy timber from harvesting sites to distant storage yards or sawmills through off-road paths and on urban highway. Using suitable control strategies on the trailer steering unit, the long timber transportation can be more maneuverable and less hazardous. In this paper we proposed two control strategies for different driving conditions. The first one is a feedback control based on the measured articulation angles between the tractor truck and the payload as well as between the payload and the trailer. The simulation results show that the trailer could follow the trajectory of the tractor satisfactorily, if a normal road curvature was applied. On an extremely curvy road with a curvature larger than 0.1 (according to a turning radius of 10 meters), it was difficult to keep the trailer on the track of the tractor and an unstable situation could happen. As a solution to overcome this problem on extremely curvy narrow path in the forest, a feed-forward control strategy based on path prediction was investigated. Both control strategies have been applied on a 3-DOF linear yaw/plane model in computer simulations and presented promising results for the tests on real vehicles.**

I. INTRODUCTION

The transport of long timber is one of the most demanding of all transportation tasks. Logs of lengths of up to 22 m have to be transported from the forest to storage yards or sawmills. Moving long and heavy logs can be extremely difficult and may be hazardous at cornering, especially on roads covered with snow and ice.

For this reason, the forest transport industry has been actively developing vehicle combinations and control strategies specially designed for long timber transportation. In [1] a vehicle combination consisting of tractor, jeep and trailer was introduced for log-hauling in Canada. The jeep is connected to the tractor with a fifth wheel joint, while the jeep is connected with the trailer through a pintle hook joint. Although the payload on the truck is held on turntables which are free to rotate relative to the jeep and trailer units, the trailer itself doesn't have a steerable axle. The main deficiency with such a configuration is the stability of the entire tractor-trailer combination and the increased space requirements in a sharp turn maneuver. In [2] and [3] the stability of the entire tractor-trailer combination was improved using active control such as unilateral brake and torque control in the fifth-wheel joint, however there is no evidence that the trailer can follow the trajectory of the tractor exactly with those control strategies. Because of the long logs, the entire vehicle combination may have up to 8 meters off-tracking at cornering.

In this paper a long timber transportation system with self-steering trailer is introduced. Two control strategies to improve the tracking ability of the trailer along the trajectory of the tractor will be proposed. The first part of this paper concerns with a tractor-trailer configuration with self-steering trailer, which is used to enhance the maneuverability of long timber transportation trucks at cornering. In the second part a simplified three-degree of freedom (DOF) vehicle model is used to describe the dynamic behavior of this vehicle combination and to evaluate the performance of the self-steering trailer. In the third part two control strategies are proposed based upon this 3-DOF model and applied on the trailer steering unit to minimize the tracking differences between the tractor truck and the trailer. In the fourth part the examination and evaluation of these control strategies are conducted using computer simulation. Finally, conclusions will be made in respect to the comparison of simulation results.

II. SELF-STEERING TRAILER

In [4] a self-steering trailer was designed for carrying large construction equipment and other heavy or bulk loads. Similar trailers have been widely used in other commercial trucks for the

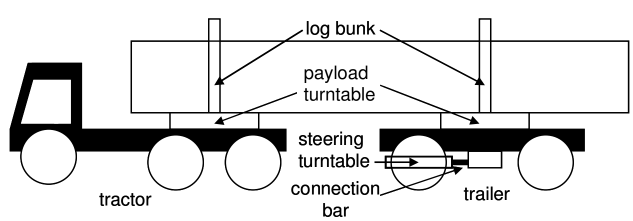

Fig. 1. A tractor-trailer combination for long timber transportation

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transportation of long and heavy objects, mainly due to their simplicity and better directional behavior. Fig. 1 shows a truck-trailer vehicle combination, which consists of a standard tri-axle truck with a turntable as the tractor and a four-wheel dolly with another turntable as the trailer. In this combination the logs are held by log bunks mounted on both turntables, free to rotate relative to the chassis on which they are mounted. To improve the dynamic behavior of the long timber transportation system at cornering, the first axle of the trailer is mounted on a turntable, which is connected with the payload turntable on the dolly, so that the rotation angle of the payload turntable can be translated to a steering angle on the front axle of the trailer.

In our configuration this connection between the steering turntable and the payload turntable is implemented by hydraulic cylinders. That way the steering of the front axle can be conducted using hydraulic valves. In the majority of cases, the hydraulic valves are closed and the hydraulic cylinders are like two stiff bars connecting the turntables. Although this special design feature meets the requirements to steer the trailer on public roads and enhances the maneuverability of the long timber transportation system, this passive control mode still has several drawbacks:

- A lateral pulling of the trailer may take place, if some disturbances act on the vehicle. This problem becomes more remarkable as the trailer brakes. Since different braking forces may act on the left and right wheels of the trailer, a yaw moment can be generated, which accounts for a lateral pulling of the trailer.
- The off-tracking problem still exists as shown in Fig. 2. Without manual steering the width of the swept path of the vehicle combination during a 180-degree turn maneuver could be 7 meters and it will be very hard for the vehicles to get through narrow paths in the forest without any collisions.

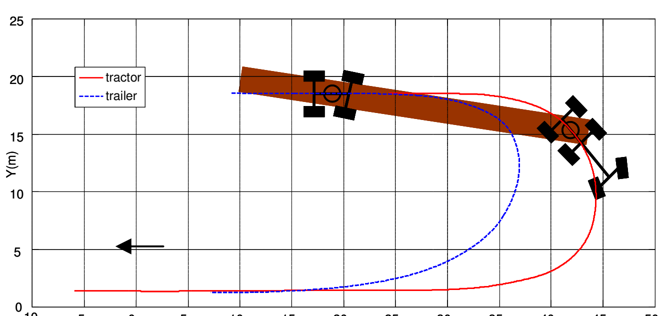

Fig. 2. Trajectories of the turntables center on the trailer (blue doted) and the tractor (red) during a 180-degree turn maneuver.

- Until now the active control of the hydraulic valves to steer the trailer can only be conducted with manual effort. That means the driver in the tractor truck must give a control command manually to the steering unit on the trailer. Such a manual control is extremely demanding because the driver must pay full attention to the position of the trailer in relation to himself through a rear view mirror and the back up cam.

Based on such considerations, we have started our research work on developing a driver assistance system for long timber transportation since 2009. The main objectives of our study are to enhance the stability of the tractor-trailer combination at critical situations and to minimize the swept path width (SPW) of the vehicles, which is subject of this paper.

In order to prevent the trailer from lateral pulling, the steering wheel of the trailer must be controlled actively to generate another yaw moment in the opposite direction which compensates the yaw moment caused by braking forces. In an active control mode the hydraulic valves work as actuators and the cylinder pistons are moved to a desired position to generate a proper steer angle on the front axle of the trailer.

III. Modeling of Tractor-Trailer Combination

In this paper, a simplified 3-DOF model of vehicle has been used for designing various control strategies proposed in our study. Since we investigate only the lateral dynamics of the vehicle, we assume that the velocity of the tractor does not have much influence on the lateral dynamics. In the linear analysis, the forward speed of the tractor is assumed to be constant and the angular displacements occurring during the maneuver are assumed to be small. The experimental vehicle can be approximated using a single track model [5] as shown in Fig. 3.

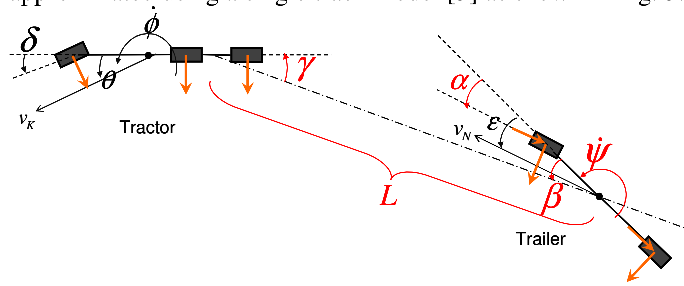

Fig. 3. Single track model of the Tractor-trailer combination

A. *Linear Analysis of the Tractor*

In our study the tractor-trailer combination is separated into three parts: the tractor, the trailer and the payload between them. Fig. 4 shows the vehicle dynamics of the tractor truck with all lateral forces acting on the three axles. Because of the small

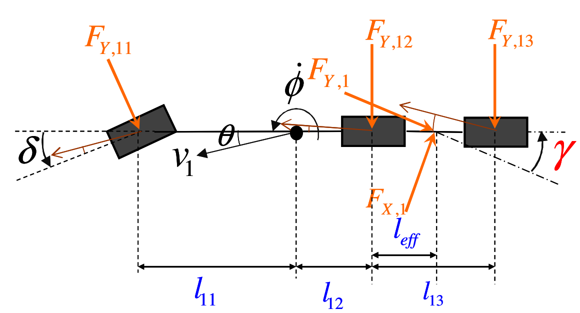

Fig. 4. Model of the tri-axle truck using linear analysis

side-slip angle of the front, middle and rear wheels, the lateral forces on these axles can be calculated approximately as:

$$
\begin{aligned}
F_{Y,11} &= c_{11} \cdot \left(\delta - \vartheta - \frac{l_{11}\dot{\phi}}{v_1} \right) \\
F_{Y,12} &= c_{12} \cdot \left(\frac{l_{12}\dot{\phi}}{v_1} - \vartheta \right) \\
F_{Y,13} &= c_{13} \cdot \left(\frac{l_{13}\dot{\phi}}{v_1} - \vartheta \right)
\end{aligned} \tag{1}
$$

Considering both the yaw movement and the lateral acceleration of the vehicle, the dynamic model of the vehicle can be created using the following relations:

$$
\begin{aligned}
J_1\ddot{\phi} &= l_1 F_{Y,11} - l_2 F_{Y,12} - l_3 F_{Y,13} \\
&\quad + l_{eff} F_{X,1} \cos\gamma - l_{eff} F_{Y,1} \sin\gamma \\
m_1 v_1 \left(\dot{\phi} + \dot{\vartheta}\right) &= F_{Y,11}\cos\delta + F_{Y,12} + F_{Y,13} \\
&\quad + F_{Y,1}\sin\gamma - F_{X,1}\cos\gamma .
\end{aligned} \tag{2}
$$

Combining the equations in (1) and (2) the linear model of the lateral dynamics of the leading truck can be calculated as follows:

$$
\dot{\vartheta} = \frac{1}{m_1 v_1}\begin{bmatrix} c_{11}\left(\delta - \frac{l_{11}\dot{\phi}}{v_1} - \vartheta\right)\cos\delta + c_{12}\left(\frac{l_{12}\dot{\phi}}{v_1} - \vartheta\right) + \\ c_{13}\left(\frac{l_{13}\dot{\phi}}{v_1} - \vartheta\right) + F_{X,1}\sin\gamma - F_{Y,1}\cos\gamma \end{bmatrix} - \dot{\phi} \tag{3}
$$

$$
\ddot{\phi} = \frac{1}{J_1}\begin{bmatrix} c_{11}\left(\delta - \frac{l_{11}\dot{\phi}}{v_1} - \vartheta\right) l_{11}\cos\delta - c_{12}\left(\frac{l_{12}\dot{\phi}}{v_1} - \vartheta\right) l_{12} - \\ c_{H,K}\left(\frac{l_{11}\dot{\phi}}{v_1} - \vartheta\right) l_{11} - F_{X,1} l_{ers}\sin\gamma + F_{Y,1} l_{ers}\cos\gamma \end{bmatrix} \tag{4}
$$

B. Linear Analysis of the Trailer

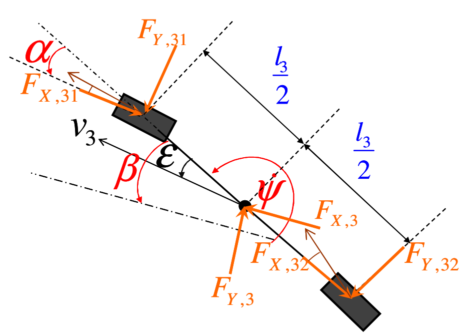

Fig. 5. Model of the self-steering trailer

Similarly we conduct the linear analysis of dynamics for the trailer using the single track model (Fig. 5). In this analysis we take consideration about the longitudinal forces acting on the front and rear axle of the trailer. Because both axles are not drive axles, the longitudinal forces can be calculated from the rolling resistances, which are proportional to the load on each axle as:

$$
\begin{aligned}
F_{X,31} &= r_{31} F_{Z,31} \\
F_{X,32} &= r_{32} F_{Z,32}
\end{aligned} \tag{5}
$$

Equations describing the motion of the trailer can then be obtained analogous to (3) and (4) as:

$$
\dot{\varepsilon} = \frac{1}{m_3 v_3}\begin{bmatrix} c_{31}\left(\alpha - \frac{l_3\dot{\psi}}{2v_3} - \varepsilon\right)\cos\alpha + c_{32}\left(\frac{l_3\dot{\psi}}{2v_3} - \varepsilon\right) \\ - r_{31}F_{Z,31}\sin\alpha - F_{Y,3}\cos\beta + F_{X,3}\sin\beta \end{bmatrix} - \dot{\psi} \tag{6}
$$

$$
\ddot{\psi} = \frac{l_3}{2J_3}\begin{bmatrix} c_{31}\left(\alpha - \frac{l_3\dot{\psi}}{2v_3} - \varepsilon\right)\cos\alpha - c_{32}\left(\frac{l_3\dot{\psi}}{2v_3} - \varepsilon\right) \\ - r_{31}F_{Z,31}\sin\alpha \end{bmatrix} \tag{7}
$$

$$
\dot{v}_3 = \frac{1}{m_3}\begin{bmatrix} F_{X,3}\cos\beta + F_{Y,3}\sin\beta - c_{31}\left(\alpha - \frac{l_3\dot{\psi}}{2v_3} - \varepsilon\right)\sin\alpha \\ - r_{31}F_{Z,31}\cos\alpha - r_{32}F_{Z,32} \end{bmatrix} \tag{8}
$$

The dynamic load on the front axle of the trailer, which depends not only on the total weight of the trailer including the payload on the turntable, but also on the longitudinal acceleration of the vehicle, can be written as:

$$
F_{Z,31} = \left(m_3 + m_{22}\right)\left(\frac{1}{2}g - \frac{h_3}{l_3}a_{X,3}\right), \tag{9}
$$

where m_3 is the mass of the trailer and m_{22} the mass of payload lifted by the trailer.

The longitudinal acceleration of the trailer is obtained from the kinematic relationships as:

$$
a_{X,3} = \dot{v}_3\cos\varepsilon - v_3\left(\dot{\varepsilon} + \dot{\psi}\right)\sin\varepsilon . \tag{10}
$$

Combining the equations (9) and (10), the dynamic load on the front axle of the trailer can be represented as:

$$
F_{Z,31} = \left(m_3 + m_{22}\right)\left(\frac{1}{2}g - \frac{h_3}{l_3}\left(\dot{v}_3\cos\varepsilon - v_3\left(\dot{\varepsilon} + \dot{\psi}\right)\sin\varepsilon\right)\right), \tag{11}
$$

while the dynamic load on the rear axle is:

$$
F_{Z,32} = \left(m_3 + m_{22}\right)\left(\frac{1}{2}g + \frac{h_3}{l_3}\left(\dot{v}_3\cos\varepsilon - v_3\left(\dot{\varepsilon} + \dot{\psi}\right)\sin\varepsilon\right)\right). \tag{12}
$$

Replacing the axle loads in equations (6), (7) and (8), we obtain a linear equation system demonstrating the dynamics of the trailer under the external excitation of steering angle α, the forces $F_{X,3}$ and $F_{Y,3}$ which are exerted on the trailer due to the articulation between payload and turntable.

C. Payload

In equations (6), (7) and (8) the forces $F_{X,3}$ and $F_{Y,3}$ are unknown. To determine these forces, the dynamics of the payload is investigated. The equations of the motion of the payload can be expressed as:

$$\begin{aligned} F_{X,1} - F_{X,3} &= m_2\left(\dot{v}_{2,X} - v_{2,Y}\dot{\omega}\right) \\ F_{Y,1} + F_{Y,3} &= m_2\left(\dot{v}_{2,Y} + v_{2,X}\dot{\omega}\right) \\ F_{Y,1} l_{21} - F_{Y,3} l_{22} &= J_2 \cdot \ddot{\omega} \end{aligned} \tag{13}$$

Combining equations (4) and (13), the forces $F_{X,1}$, $F_{X,3}$, $F_{Y,1}$ and $F_{Y,3}$ can be determined.

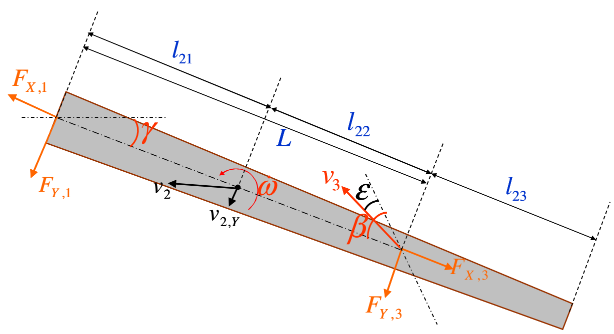

Fig. 6. Model of the payload

D. State-Space Description

First of all, the dynamics of the articulation yaw angle between the payload and the trailer β can be represented using:

$$\dot{\beta} = \frac{v_1 \cos\vartheta \sin\gamma - \left(l_{eff}\,\dot{\phi} - v_1 \sin\vartheta\right)\cos\gamma + v_3 \sin(\beta - \varepsilon)}{L} - \dot{\psi} \tag{14}$$

Combining the equations (6), (7), (8) and (14) and replacing the axle loads and articulating forces with equations (11), (12) and (13), the dynamics of the trailer can be determined using a state-space description as:

$$\underline{\dot{x}} = \underline{A} \cdot \underline{x} + \underline{b} \cdot \alpha + \underline{E} \cdot \underline{z}\,. \tag{15}$$

where the state vector is

$$\underline{x}^T = [\varepsilon\ \dot{\psi}\ \beta\ \mathrm{v}_3],$$

and the measurable disturbance vector is

$$\underline{z} = \left[\dot{\phi}\ \ddot{\phi}\ \gamma\ \dot{\gamma}\ \vartheta\ \mathrm{v}_1\right].$$

IV. Controller Designs

A. Feedback Control of Articulation Angles

When designing a control system, the control objectives must be established. The main objective of our control designs in the present case is to reduce the swept path width of the entire vehicle combination. Therefore we consider only about the planar motion of the trailer. In [6] the authors proposed three different control strategies for semi-trailer: namely self-steering, command steer and pivotal bogie. In a typical command steer system the front axle of the trailer is made to steer in relation to the articulation angle between the tractor and the trailer. In our study we have modified the control strategy with command steer by generating a command α on the steerable axle of trailer in relation to the difference between the articulation angle between the tractor and payload γ and the articulation angle between the payload and trailer β. The control structure for the trailer is shown in Fig. 7 as a cascade feedback control with a disturbances feed-forward.

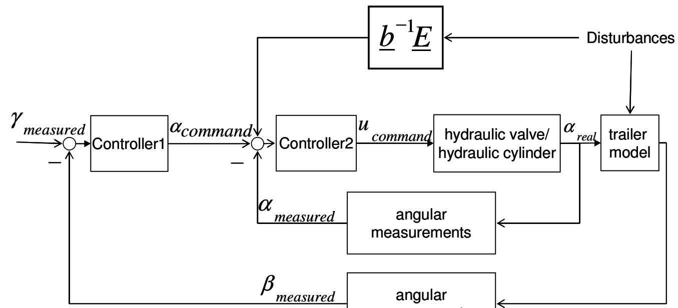

Fig. 7. Cascade control with feedback of angular measurements.

The cascade control consists of an inner and an outer loop. The inner loop controls the difference between the commanded steering angle and the real steering angle and generates a corresponding command signal for the hydraulic valves. The outer loop controls the difference between the measured articulation angles and generates a commanded steering angle. In order to eliminate the influence of measurable disturbances on the angular control, a feed-forward control is applied to calculate the steer command.

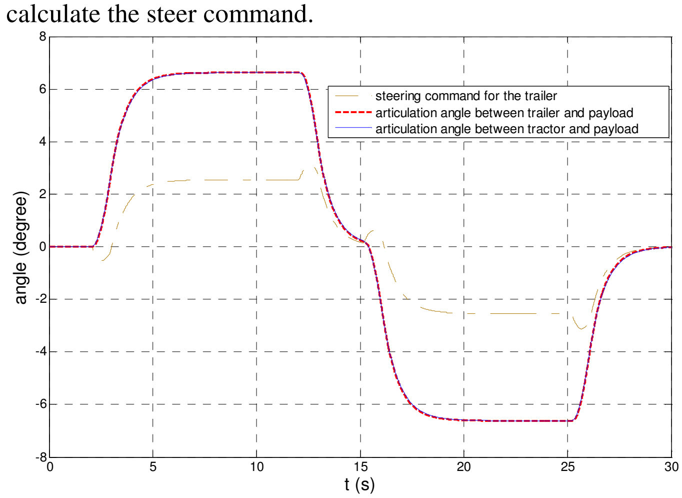

Fig. 8. Results of the angular control during a cornering drive at a speed of 36 km/h.

The change of articulation angles during a synthesized lane change maneuver on a public road is shown in Fig. 8. The solid blue curve represents the articulation angle between the tractor and payload, while the red dashed curve represents the articulation angle between the trailer and payload. The computer simulation results show that difference between both angles can be very little using the articulation angle control strategy.

B. Feed-forward Control Using Path Planning

Another control strategy, which we propose here to reduce the swept path width, is based on a path planning using polynomials. Fig. 9 demonstrates how this control strategy works. At the very beginning the position of the trailer (center of the turntable) $(x_3(t), y_3(t))$ is chosen as the origin of the coordinate system and the position of the tractor (also center of the turntable) $(x_1(t), y_1(t))$ is chosen as a point on the x-axis with a distance L from the origin. A first curve fitting is conducted through the positions $(x_1(t), y_1(t))$ and $(x_3(t), y_3(t))$ with the slopes $\tan\beta(t)$ and $-\tan\gamma(t)$ at both ends, using a cubic polynomial.

This black solid curve is used as the tracking objective in the next time period till $t+\Delta T$. After a time period of ΔT, the trailer moves to the position $(x_3(t+\Delta T), y_3(t+\Delta T))$ and the tractor to the position $(x_1(t+\Delta T), y_1(t+\Delta T))$. A spline is calculated through the points $(x_1(t+\Delta T), y_1(t+\Delta T))$, $(x_1(t), y_1(t))$, $(x_3(t+\Delta T), y_3(t+\Delta T))$ and $(x_3(t), y_3(t))$ using the given slopes. The red dashed curve is used as the new tracking objective for the trailer in the next period. The trailer velocity is integrated over the time to calculate its current position along the tracking objective. According to the curvature of the spline at the current position κ, the steering command of the trailer can be calculated as: $\alpha = \tan^{-1}(l_3\kappa)$.

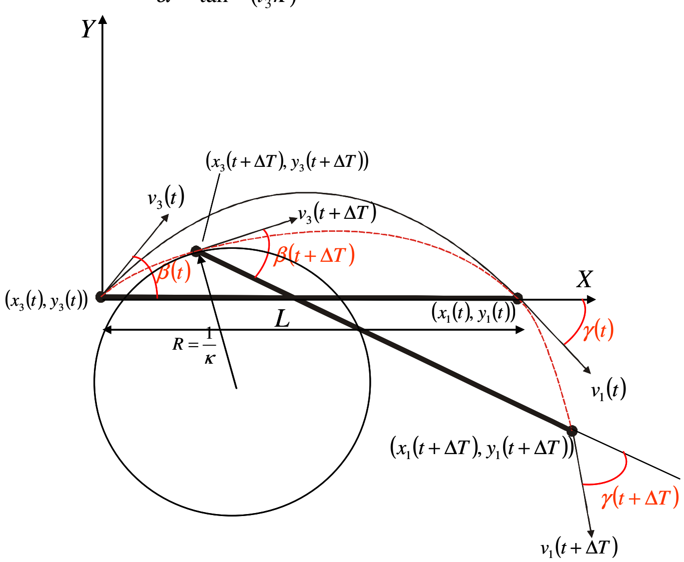

Fig. 9 Path planning with polynomials

This approach is just a feed-forward control approach, since an exact control of the trailer's position is impossible. GNSS based solutions won't work accurately in the forest and the position measurements using gyroscopes may drift after several time periods.

V. Evaluation of Different Control Strategies

The introduced control strategies are investigated and evaluated in computer simulation. Some results are presented here. Firstly, we have conducted the simulation of a cornering maneuver with a curvature change from 1/60 to -1/60. Fig. 10 shows the trajectories of the tractor and trailer during the cornering maneuver (both measured at the center of turntables).

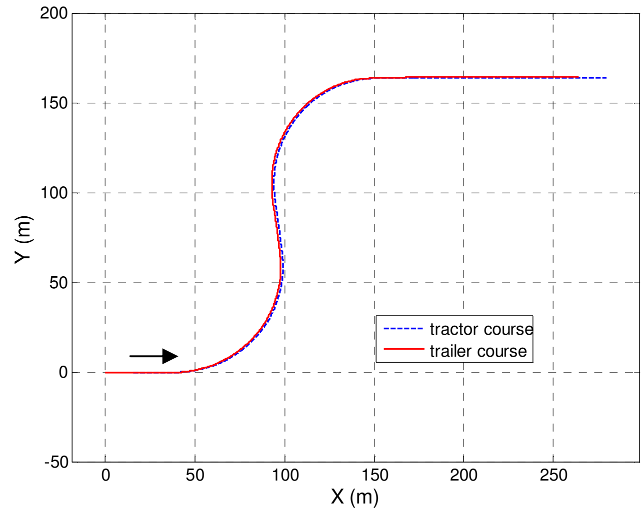

Fig. 10 Tractor and trailer courses at a cornering maneuver with a curvature change from 1/60 to -1/60 using the control of articulation angles.

In this simulation the control of articulation angles was applied. Accordance between the tractor course and trailer course is in evidence. The largest width of the swept path about 1.2 meters occurs at the time, when the curvature changes from the left side to the right side. According to the German legislation StVZO, the permitted maximal SPW for travelling on public roads must be less than 7.2 meters [7]. That means the legislatory regulations can be met with the angular control very comfortably.

The simulation of a sharp turn maneuver at a low speed was conducted using the articulation angle control. The tractor and trailer courses during this maneuver are demonstrated in Fig. 11. The width of the swept path in this case is considerably lager than in Fig. 10.

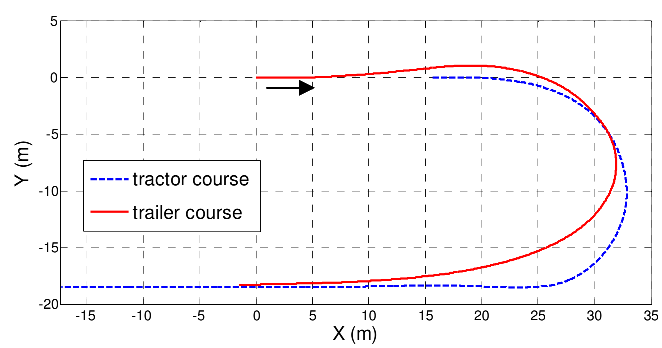

Fig. 11 Tractor and trailer courses during a 180-degree turn maneuver at a low speed of 5 km/h using the control of articulation angles.

In spite of the poorer performance of the angular control in this case, the width of the swept path is still much smaller than in the original configuration shown in Fig. 2. To improve the control results at low speed and large path curvature, we have conducted the same turn maneuver using the path planning based steering control. With this control strategy the width of the swept path can be reduce to less than 1 meter (Fig. 12).

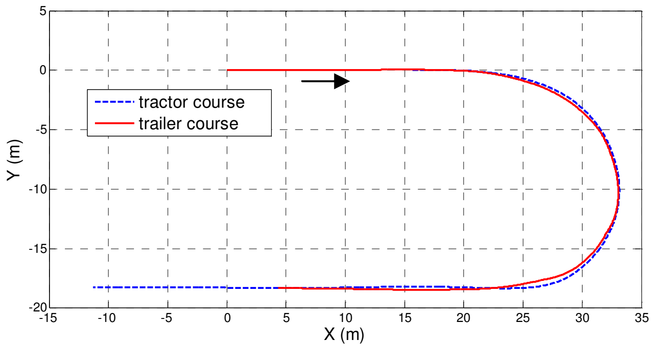

Fig. 12 Tractor and trailer courses during a 180-degree turn maneuver at a low speed of 5 km/h using the path planning based steering control.

VI. CONCLUSION

Two control strategies to reduce the width of swept path (SWP) were proposed and applied on long timber transportation vehicles. In the first approach the articulation angles between the tractor, payload and trailer are measured and difference between these angles are controlled. This control method has shown satisfactory results in respect to reducing the SWP during the drive on normal rural roads. Compared with the system without such an angular control, the SWP can be reduced by up to 80%. On extremely curvy and narrow paths, this approach has a poorer performance and still can reduce the SWP by more than 50%.

The second approach provides better control results than the first one in respect to reducing the SWP at narrow and curvy paths. The deficiency of this approach is the larger calculation cost and the mean robustness against disturbances. Without a possibility to detect the effective trailer position the control loop can't be closed and the system can't react on disturbances. Therefore, this approach can only be applied at low speed, while the first approach can meet the requirements to reduce the SWP at much larger speed.

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NOMENCLATURE

$F_{X,ij}$	Longitudinal tyre forces on the j-th axle of the i-th vehicle (i=1 for tractor, i=3 for trailer)
$F_{Y,ij}$	Lateral tyre forces on the j-th axle of the i-th vehicle part
$F_{X,i}$	Longitudinal forces acting on the turntable of the i-the vehicle
L	Distance between the turntables on tractor and trailer
l_{11}	Distance between the tractor's center of gravity and its front axle
l_{12}	Distance between the tractor's center of gravity and its middle axle
l_{13}	Distance between the tractor's middle and rear axle
l_{21}	Distance between the payload's center of gravity and the turntable on the tractor
l_{22}	Distance between the payload's center of gravity and the turntable on the trailer
l_{23}	Distance between the turntable on the trailer and its rear end
l_3	Distance between the tractor's middle and rear axle
l_{eff}	Distance between the tractor's center of gravity and the effective rear axle
v_i	Speed of the vehicles or payload
$v_{i,X}$	Longitudinal speed of the vehicle or the payload
$v_{i,Y}$	Lateral speed of the vehicle or the payload
α	Steering angle on the trailer
β	Articulation angle between the tractor and payload at the turntable
γ	Articulation angle between the tractor and payload at the turntable
δ	Steering angle on the tractor
ε	Side slip angle of the trailer
ϑ	Side slip angle of the tractor
$\dot{\phi}$	Yaw rate of the tractor
$\dot{\psi}$	Yaw rate of the trailer
$\dot{\omega}$	Yaw rate of the payload