A steering control system for the tractor – semi-trailer combination vehicle with the electromechanical transmission

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Abstract. The article offers an algorithm for the operation of the steering and active drive control system of a saddle-type tractor – trailer combination. The algorithm allows reducing the width of the cornering corridor when maneuvering at low speeds. In the combination vehicle under study, all the wheels of the tractor and semi-trailer are driving and steered. The algorithm of the steering control system takes into account the current values of the folding angle and speed of the combination vehicle. The active drive control is based on the analysis of the forces in the coupling device. The efficiency and effectiveness of the proposed algorithm has been proved using computational experiments.

1. Introduction

Recent years have seen many research works in the field of the automation of road vehicles aimed at the improvement of the efficiency and safety of transportation operations \cite{1-5}. Special-purpose combination vehicles take an important part in the transportation of large-sized and long cargo in different industries: in civil engineering, forestry, oil and gas production, as well as in the Armed Forces \cite{6-10}.

Long combination vehicles have a large number of axles with considerable distances between them, which may lead to their different steering kinematics. The large dimensions of the road vehicles of this type significantly reduce their maneuverability. These features impose a number of restrictions on the use of mechanical links and conventional steering systems.

Thus, we may conclude that improvement of the maneuverability of long vehicles is a complex technical problem that can be solved by the development of special steering systems.

The cornering corridor width of a combination vehicle at steering can be reduced by installing a memory servosystem in the steering system of the trailer unit. The servosystem steers the trailer wheels depending on the steer angles of the tractor wheels but does it with a time lag needed for the trailer wheels to travel the distance to the wheels of the tractor. Drive wheels of the trailer impose additional difficulties on the steering. In the case when the traction force of the trailer becomes higher than the traction force of the tractor, a free joint in the coupling between the units may cause jack-knifing of the combination vehicle.

Use of the automatic control system for steering and active drive can increase the maneuverability of combination vehicles and make the work of the driver more comfortable, and in the future it can be integrated in the high-level systems providing autonomous operation of the combination vehicle.
2. Theory

The object of the study is a combination vehicle with a gross mass of 120 000 kg. The vehicle consists of a four-axle tractor with electromechanical transmission and a three-axle semi-trailer. All the axles are driving and steerable.

Projection of the center of the tractor rotation during cornering on its longitudinal axis lies on its third axle. The minimal width of the required travel path is provided when the center of the rear axle of the semi-trailer moves along the trajectory of the coupling center [11]. Figure 1 shows a schematic diagram for semi-trailer wheel steering angles.

![Figure 1: Schematic diagram of the tractor – semi-trailer combination vehicle at turning](image)

If we assume that the combination vehicle moves without side slip, its folding angle can be expressed using the steer angle of an imaginary steered wheel located at the center of the first axle of the tractor:

\[
\theta_s = \arcsin \left( \frac{l_{35} \cdot \tan \theta_r}{2 \cdot l_{13}} \right),
\]

where \(\theta_s\) is the folding angle of the combination vehicle, rad; \(\theta_r\) is the angle of the imaginary steered wheel, rad; \(l_{35}\) is the distance from the pivot of the coupling device to the third axle of the semi-trailer, m; \(l_{13}\) is the distance from the first axle to the third axle of the tractor, m.

Then, steer angles of the semi-trailer wheels can be calculated as:
\[
\begin{align*}
    l_{sp} &= 0.5 \cdot l_3; \\
    R_{kin2} &= \frac{l_{sp}}{\tan \theta_s}; \\
    \theta_{w2i} &= \arctg \left( \frac{l_{sp} - l_{wi}}{R_{kin2} \pm 0.5 \cdot B_2} \right),
\end{align*}
\]

here \( l_{sp} \) is the distance from the pivot of the coupling device to the projection of the rotation center of the semi-trailer on its longitudinal axis, \( m \); \( R_{kin2} \) is the theoretical turning radius of the semi-trailer, \( m \); \( \theta_{w2i} \) is the steer angle of the \( i \)th wheel of the semi-trailer, rad; \( B_2 \) is the track width of the semi-trailer, \( m \).

The actual turning radius and shift of the rotation center for the tractor can be calculated if we know all the components of the linear velocity of its center of mass and the angular velocity of its rotation (see Fig. 2):

\[
\begin{align*}
    R_{p1} &= \frac{v_{Cz}}{\omega_{z1}}; \\
    R_{C1} &= \frac{v_{C}}{\omega_{z1}}; \\
    l_{Cp} &= \sqrt{R_{C1}^2 - R_{p1}^2},
\end{align*}
\]

here \( R_{p1} \) is the actual turning radius of the tractor, \( m \); \( R_{C1} \) is the turning radius of the tractor center of mass, \( m \); \( v_{Cz} \) – is the longitudinal component of the linear velocity of the tractor, \( m/s \); \( v_{C} \) – is the linear velocity of the tractor, \( m/s \); \( \omega_{z1} \) is the tractor angular velocity, rad/s; \( l_{Cp} \) – is the distance from the projection of the tractor rotation center on its longitudinal axis to its center of mass, \( m \).

**Figure 2.** Schematic diagram of the tractor at turning

The radii of the trajectories of the tractor wheels can be calculated from the following relation:

\[
R_{wi} = \sqrt{\left( l_{Cp} + x_{wi} \right)^2 + \left( R_{p1} - y_{wi} \right)^2},
\]

here \( x_{wi} \) and \( y_{wi} \) are the coordinates of the \( i \)th wheel relative to the tractor center of mass, \( m \).

The projection of the semi-trailer center of rotation on its longitudinal axis and its turning radius can be found in a similar manner.
The time lag of the steer angles of the semi-trailer wheels should be calculated as a function of the combination vehicle speed, which can be estimated by the average value of the tractor wheels rotation speed:

$$v_c = \frac{\omega_{w1} \cdot r_{di}}{n_{w1}},$$  \hspace{1cm} (5)

here $\omega_{w1}$ is the rotational speed of the tractor $i$th wheel, rad/s; $r_{di}$ is the distance from the axle of the $i$th wheel to the road, m; $n_{w1}$ is the number of the tractor wheels.

In order to prevent jack-knifing we offer to limit the traction force on the semi-trailer wheels based on the analysis of the forces in the coupling device. The primary sensor function is performed by the pivot of the coupling device. The sensor reads elastic deflections of the pivot relative to its axis caused by the reaction forces between the tractor and the semi-trailer at different operation modes [12]. The diagram of the primary sensor is shown in Fig. 3.

![Diagram of the primary sensor](image)

**Figure 3.** A sensor for measuring forces in the pivot of the coupling device: a – sensor schematic diagram; b – method of measuring forces

Pivot 2 is anchored in non-deformable base 3. Central pin 1 of the sensing device is also fixed at non-deformable base 3. Between parts 1 and 2 there is a gap, its size being equal to the maximum possible deformation of pin 2. Three mutually perpendicular transmitters 4 are mounted on the end portion of the central pin 1. Four segments of electronic units 6 fixed on the top plane of the end portion of pin 2 contain eight receivers 5. Receivers 5 are mounted in pairs on two sides of the emitters with a standardized gap size. Thus, when the reaction forces act on base 3 and pivot 2, the pivot elastically deflects relative to central pin 1 and changes the standardized gaps $\Delta_x$ and $\Delta_y$ between sensors 4 and receivers 5 of the sensing device. The signals generated by the receivers provide calculation of the components ($P_x$ и $P_y$) of the force in the coupling device and detection of its direction.
Semi-trailer electric drive motor activation and the degree of its use is controlled by the coefficient of the motor power \((h_{st})\), which may vary from 0 (the motor is off) to 1 (the motor operates on its external characteristics).

The relative values of the force components in the coupling device are used as inputs for the control algorithm of the semi-trailer active drive:

\[
\Delta P_x = \frac{P_x - P_{x\text{st}}}{P_x}, \quad \Delta P_y = \frac{P_y - P_{y\text{st}}}{P_y},
\]

(6)

here \(P_{x\text{st}}\) and \(P_{y\text{st}}\) are the limit values of the force for activation of the drive, N.

The proposed control algorithm of the semi-trailer active drive is based on the use of the Fuzzy Logic. For the inputs \((\Delta P_x, \Delta P_y)\) and output \((h_{st})\) signal membership functions (see Fig. 4 and 5) were derived, which describe the following linguistic values:

- \(\Delta P_x\) is negative (\(«-»\)), zero (\(«0»\)), positive (\(«+»\));
- \(\Delta P_y\) is negative (\(«-»\)), zero (\(«0»\)), positive (\(«+»\));
- \(h_{st}\) is going down quickly (\(«\text{quick-}»\)), is going down slowly (\(«\text{slow-}»\)), zero (\(«0»\)), is going up slowly (\(«\text{slow+}»\)), is going up quickly (\(«\text{quick+}»\)).

![Figure 4. Membership functions of the input signals \(\Delta P_x\) (a) and \(\Delta P_y\) (b)](image)

![Figure 5. Membership functions of the output signal \(h_{st}\)](image)

The degree of the electric drive motor use is dependent on the values of \(\Delta P_x\) and \(\Delta P_y\). For instance, if \(\Delta P_x\) and \(\Delta P_y\) are positive, it means that the semi-trailer pushes the tractor forward and the traction...
force on the semi-trailer needs reduction. All possible combinations of $\Delta P_x$ and $\Delta P_y$ are contained in table 1 forming the rule base of the fuzzy logic controller.

### Table 1. Rule base of the fuzzy logic controller

| №  | $\Delta P_x$ | $\Delta P_y$ | $h_{st}$ |
|----|--------------|--------------|---------|
| 1. | –            | –            | quick+  |
| 2. | –            | 0            | slow+   |
| 3. | –            | +            | slow-   |
| 4. | 0            | –            | slow+   |
| 5. | 0            | 0            | 0       |
| 6. | 0            | +            | slow-   |
| 7. | +            | –            | slow+   |
| 8. | +            | 0            | slow-   |
| 9. | +            | +            | quick-  |

### 3. Research results

Mathematical model of the combination vehicle with saddle-type active semi-trailer and electromechanical transmission was implemented in MATLAB Simulink. The description of the mathematical model is given in [13].

Analysis of the algorithm and of its effect on the maneuverability of the combination vehicle was performed by simulation of the following road events: Steering and consecutive driving in a turn with a fixed steering wheel; Lane change. For comparison, the simulation was also done for a combination vehicle the wheels of which are steered depending only on the folding angle (the direct action system). All the events were simulated for the speed 5 m/s. The simulation results are shown in Fig. 6 and 7.

![Figure 6](image_url). Trajectories at turning: a – direct action system; b – automatic steering and active drive control system
4. Conclusions
From the simulation results we can conclude that in the analyzed road events the proposed algorithm provides identical trajectories for the tractor and semi-trailer and prevents jack-knifing.

Further research may be focused on the estimation of the effect of the automatic steering and active drive control system on the stability of the combination vehicle at a higher speed. In the future, the automatic steering and active drive control system may be integrated into high-level systems providing autonomous operation of the combination vehicle.

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