The method of increasing the stability of trailer-trucks in case of emergency braking in a turn and emergency failure of the trailer brake system

M Zhileykin\(^1\) and G Skotnikov\(^1,2\)

\(^1\)Bauman Moscow State Technical University

\(^2\)E-mail: skotnikov.g@yandex.ru

Abstract. One of the most important properties of semi-trailer trucks are stability and steerability. This is because with an increase in driving speeds, these properties largely determine the safety of vehicle operation. Improving the semi-trailer trucks safety is all the more relevant because the consequences of road accidents involving semi-trailer trucks are the most severe and cause enormous damage. One of the most common causes of accidents is the loss of stability during emergency braking in a turn, especially in the event of a failure of trailer braking system. The goal of this research is to develop principles for improving the stability of semi-trailer trucks movement during emergency braking in a turn and failure of trailer braking system. A method has been developed to increase semi-trailer trucks stability during emergency braking in a turn and the failure of trailer braking system, which allows to maintain the trajectory stability of the semi-trailer truck and to avoid accidents with severe consequences. Using simulation methods, the effectiveness of the proposed method is proved.

Introduction

The widespread use of trailer-trucks in the transportation structure is determined by the need for transportation of heavy bulky goods, the need to ensure low specific pressures on the road while increasing the carrying capacity of vehicles and increasing operating speeds [1].

One of the most important properties of semi-trailer trucks are stability and steerability. This is because with an increase in driving speeds, these properties largely determine the safety of vehicle operation. Improving the semi-trailer trucks safety is all the more relevant because the consequences of road accidents involving semi-trailer trucks are the most severe and cause enormous damage.

One of the most common causes of accidents is the loss of stability during emergency braking in a turn, especially in the event of a failure of trailer braking system.

Due to its relevance, a large number of publications are devoted to the issues of ensuring maneuverability, steerability, and stability in the design of trailer trucks [2-11].

Literary sources are mainly devoted to the development of algorithms for the operation of dynamic stabilization systems of separately moving vehicles either by methods of creating stabilizing moments by braking individual wheels [12-15], or by steering [16-19]. As regards the stabilization of the movement of trailers and semi-trailers, it is usually a question of increasing overturning stability [20], as well as equipping trailer-trucks with anti-lock brake systems.

The goal of this research is to develop principles for improving the stability of semi-trailer trucks movement during emergency braking in a turn and failure of trailer braking system.
Substantiation of the principles of stabilization of trailer-trucks movement during emergency braking in a turn and emergency failure of trailer braking system

In case of emergency braking when trailer braking system is failed, a characteristic picture of the loss of trailer-truck stability is the deviation of the truck from a given trajectory due to the fact that heavier trailer push the truck to an adjacent lane. An emergency can be prevented by applying a range of measures.

1. Preventive reduction in the speed of a trailer-truck in a turn if it is close to critical values. This is achieved by reducing the engine power demand and braking the wheels of the trailer.

2. Emergency braking mode leads to a redistribution of normal reactions between the axles of the truck. The front wheels are loaded with additional vertical force. This circumstance, as well as the operation of the anti-lock braking system, which prevents the wheels from locking by the braking torque, leads to an increase in the grip of these wheels. These facts allows to apply a corrective change in the angle of rotation of the steered wheels (steering), which will help to keep the truck on the path set by the driver.

A simplified mathematical model of trailer-truck planar motion will be consider to compose the equations of trailer-truck motion. The model has several assumptions:

a) the slip angles of right and left wheels of each axis are the same;
b) the steering angles of the wheels and wheel slip angles are small, i.e. do not exceed 10°;
c) slip resistance coefficients for all wheels of the axle are the same.

The design diagram of the forces acting on the trailer-truck in curved motion is shown in Fig. 1.

![Fig. 1. Scheme of forces acting on the trailer-truck: $X, Y$ – coordinate axes associated with the truck center of mass; $C_T, C_{TP}$ – centers of mass of the truck and semi-trailer; $S$ – the fifth wheel center; $l_{sc}$ – distance from the center of mass of the truck to the fifth wheel center; $j_o$ – acceleration of the center of mass of the truck; $V_a$ – linear velocity of center of mass of the truck; $o_z$ – the angular velocity of rotation of the truck around a vertical axis passing through its center of mass; $R_{sc}$ – force acting at the fifth wheel center from the side of the semi-trailer to the truck during braking; $X_{1l}, X_{1r}$ – longitudinal forces acting on the left and right wheels of the first axle of the truck from the road; $\Theta_{1l}, \Theta_{1r}$ – steering angles of left and right wheels of the first axle of the truck; $\delta_{1l}, \delta_{1r}$ – slip angles of left and right wheels of the first axle of the truck; $\delta_{2l}, \delta_{2r}$ – slip angles of left and right wheels of the second axle of the truck; $\gamma$ – jackknifing angle of the trailer-truck]
Consider the movement of a truck in braking mode, replacing the effect of the semi-trailer with the force \( R_{sc} \) applied to the fifth wheel center on the truck, acting along the longitudinal axis of the semi-trailer. For a rear drive two-axle vehicle the following differential equations were obtained in [21]:

\[
\dot{\delta}_1 = \frac{V_a}{L} \left( \Theta_{1sr} + \delta_2 - \delta_1 \right) - \frac{K_y}{V_a} \left( \frac{g + L^2}{4J_z} \right) \delta_2 - \frac{K_y}{V_a} \left( \frac{g - L^2}{4J_z} \right) \delta_1 + \frac{j_a}{V_a} \left( \Theta_{1sr} - \delta_1 \right) + \frac{g}{G_\appa V_a} P_y + \frac{L}{2J_z V_a} M_z - \frac{\Theta_{1sr}}{V_a} \left( \frac{g - L^2}{4J_z} \right) X_1;
\]

\[
\dot{\delta}_2 = \frac{V_a}{L} \left( \Theta_{1sr} + \delta_2 - \delta_1 \right) - \frac{K_y}{V_a} \left( \frac{g + L^2}{4J_z} \right) \delta_1 - \frac{K_y}{V_a} \left( \frac{g - L^2}{4J_z} \right) \delta_2 - \frac{-j_a}{V_a} \frac{g}{G_\appa V_a} P_y - \frac{L}{2J_z V_a} M_z - \frac{\Theta_{1sr}}{V_a} \left( \frac{g + L^2}{4J_z} \right) X_1;
\]

\[
M_z = P_{sc} \sin(\gamma) I_{sc}; V_a = \frac{L \omega_z}{\Theta_{1cp} + \delta_2 - \delta_1}; P_y = P_{sc} \sin(\gamma)
\]

\[
\dot{\delta}_A = - \frac{K_y}{V_a} \frac{L^2}{2J_z} + \frac{j_a}{V_a} + P_y \sin(\gamma) I_{sc} \frac{1}{J_z \omega_z} \delta_2 - \left( P_y \sin(\gamma) I_{sc} \frac{1}{J_z \omega_z} - \frac{L^2}{2V_a J_z} X_1 + \frac{j_a}{V_a} \right) \Theta_{1sr},
\]

where \( \delta_A = \delta_2 - \delta_1 \) – difference between rear and front slip angles; \( X_1 = X_{1l} + X_{2r} \) – full force on the front axle of the truck; \( K_y \) – total (for the vehicle axis) coefficient of resistance to tire retraction; \( \Theta_{1sr} = \Theta_{1l} + \Theta_{1r} \) – average steering angle; \( \delta_0 = \frac{\delta_0 + \delta_{ir}}{2}, i = 1, 2 \) – average slip angle of \( i \)-truck axle; \( G_\appa \) – truck weight; \( L \) – truck wheelbase; \( J_z \) – the moment of inertia of the truck relative to the vertical axis passing through its center of mass.

As known, difference between rear and front slip angles \( \delta_A = \delta_2 - \delta_1 \) characterizes its turnability: \( \delta_A = 0 \) – neutral; \( \delta_A > 0 \) – oversteer; \( \delta_A < 0 \) – understeer. In case of loss of stability by a trailer-truck (jackknifing) during braking, the truck acquires oversteer to stabilize the truck, neutral steering is required.

For disturbed motion, equation (1) takes the form

\[
\Delta \dot{\delta}_A = - \frac{K_y}{V_a} \frac{L^2}{2J_z} + \frac{j_a}{V_a} + P_y \sin(\gamma) I_{sc} \frac{1}{J_z \omega_z} \Delta \delta_2 - \left( P_y \sin(\gamma) I_{sc} \frac{1}{J_z \omega_z} - \frac{L^2}{2V_a J_z} X_1 + \frac{j_a}{V_a} \right) \Delta \Theta_{1sr},
\]

where the symbol \( \Delta \) denotes the increment of the value over the time \( \Delta t \).

Equation (2) has the form

\[
\dot{X}(t) = A(t) X(t) + R(t) U(t),
\]

where \( X(t) = \Delta \delta_A \) – state variable; \( U(t) = \Delta \Theta_{1sr} \) – control action;

\[
A(t) = \left\{ \frac{K_y}{V_a} \frac{L^2}{2J_z} + \frac{j_a}{V_a} + P_y \sin(\gamma) I_{sc} \frac{1}{J_z \omega_z} \right\}, \quad R(t) = \left\{ P_y \sin(\gamma) I_{sc} \frac{1}{J_z \omega_z} - \frac{L^2}{2V_a J_z} X_1 + \frac{j_a}{V_a} \right\}
\]

The control \( U(t) \) for obvious physical reasons is limited, i.e. belonging to a closed set:

\[
|U(t)| \leq U_{\text{max}}.
\]
Lyapunov’s studying of the stability of movement, which allows one to judge the properties of perturbed movements, indicates the path to the rational design of regulators. Therefore, the Lyapunov function method is widely used directly for the synthesis of so-called stabilizing controls, i.e. providing stability of the designed dynamic system. The idea of a general approach to finding stabilizing control for objects (3) with a control action \( U(t) \) is as follows.

Let some scalar function \( V(X) \) be given, which can be considered as a measure of the deviation of a moving object (3) from the steady state. In this case, the steady state is determined by a trivial solution: \( X=0 \). In this case, a positive definite Euclidean norm of the vector \( ||X|| \) or the square of its value will be introduced as the Lyapunov function \( V(X) \)

\[
V(X) = \Delta \delta^2
\]  

(4)

The role of the control system is to reduce the distance between the moving object (3) and the steady-state value \( X=0 \).

The direct Lyapunov method reduces the problem of studying the stability of system (3) to studying the properties of the function \( V(X) \) and its first derivative, which is calculated by the formula

\[
\frac{dV(X)}{dt} = \frac{\partial V}{\partial X} AX = -2\Delta \delta^2 \left( \frac{K_y}{V_a} \frac{L^2}{2J_z} + \frac{j_a}{V_a} + P_y \sin(\gamma) J \frac{1}{J_z \omega_z} \right)
\]

(5)

To find the optimal control vector \( U(t) \), we use the method of optimal damping of transients V.I. Zubova [22]:

\[
U(t) = -U_{\max} \text{sign} \left( \frac{\partial V}{\partial X} R(t) \right)
\]

(6)

The magnitude order of the summands of equation (5) and their signs based on the positive directions of vectors adopted in Figure 1 are shown in table 1.

| №  | Summand                                      | Sign | Magnitude order |
|----|----------------------------------------------|------|----------------|
| 1  | \( \frac{K_y}{V_a} \frac{L^2}{2J_z} \)       | >0   | \( 10^3 \ldots 10^4 \) |
| 2  | \( \frac{j_a}{V_a} \)                        | <0   | \( 10^1 \)      |
| 3  | \( P_y \sin(\gamma) J \frac{1}{J_z \omega_z} \) | <0   | \( 10^2 \)      |

Therefore, the expression in brackets of equation (5) is always greater than zero. Then \( \frac{dV(X)}{dt} < 0 \), and the function \( V(X) \) is a Lyapunov function. Then the optimal control (6) takes the form

\[
U(t) = -U_{\max} \text{sign} \left( -2D \Delta \delta \right)
\]

(7)

\[
D = \left( P_y \sin(\gamma) J \frac{1}{J_z \omega_z} - \frac{L^2}{2V_a J_z} X_1 + \frac{j_a}{V_a} \right)
\]

Because \( \frac{L^2}{2V_a J_z} X_1 > 0 \), then \( D < 0 \), and expression (7) can be written

\[
U(t) = -U_{\max} \text{sign} \left( \Delta \delta \right)
\]

(8)
A control restriction is introduced $U_{\text{max}} = |\Delta \gamma(t) = \gamma(t) - \gamma_0|$, where $\gamma(t)$ – actual hitch angle, $\gamma_0$ – value of hitch angle when the driver presses the brake pedal.

Since the difference between slip angels of the axes $\Delta \delta_A$ cannot be measured in practice, it is necessary to replace this variable with another one that has the same sign when jackknifing is occur. As such a variable, the hitch angle $\gamma(t)$ can be used. Then the finally optimal control (8) will have the form

$$U(t) = \Delta \Theta_{sr}(t) = -|\Delta \gamma(t)| \text{sign}[\gamma(t)]$$

(9)

The research of the effectiveness of the proposed algorithm for the stabilization of the trailer-truck with the help of active steering when braking by simulation methods

As a criterion for the effectiveness of the algorithm of the stabilization system of the trailer-truck during braking, the absence of the road train from the lane was taken.

Description of the research object and the conditions of simulation tests

Consider the movement of a trailer-truck with a gross weight of 36,000 kg, consisting of a truck and a semi-trailer. 4x2 truck has double-tire rear wheels.

The simulation model of the trailer-truck takes into account the displacement of the vertical axis of the fifth wheel to the right by 0.03 m with respect to the longitudinal axis of symmetry of the truck. The braking systems of the truck and semi-trailer are equipped with an anti-lock system that prevents the wheels from locking during braking.

Theoretical studies were conducted using simulation methods to confirm the effectiveness of the proposed algorithm of the stabilization system of the trailer-truck. Features of the mathematical model of the trailer-truck motion are considered in [23].

The motion on the “dry asphalt” rigid flat road (tire grip coefficient is $\mu_{\text{max}} = 0.7$) was investigated.

The simulated maneuver consisted of the following parts: entrance to the turn, the movement in the turn with a constant angle of the steered wheels and an increasing heading speed not exceeding critical speed according to the rollover condition, emergency braking. At the same time, the braking torque of 0.7 from the maximum value was realized at the wheels of the left side of the semi-trailer, and 0.1 of the maximum value of the braking moment at the wheels of the right side.

Research of the movement of trailer-truck not equipped with a stabilization system

In Figure 2 shows the final position of the trailer-truck, not equipped with a stabilization system, after stopping. Fig. 3 shows the time-history of the hitch angle $\gamma$.

![Fig. 2. Final position of the trailer-truck, not equipped with a stabilization system, after emergency braking in a turn](image-url)
Figure 2 and 3 show that during emergency braking in a turn during an emergency failure of the braking system of a semi-trailer truck, deviations from a given trajectory are significant. The hitch angle reaches 75°.

Research of the movement of trailer-truck equipped with a stabilization system

In Figure 4 shows the final position of the trailer-truck, equipped with a stabilization system, after stopping. Fig. 5 shows the time-history of the hitch angle $\gamma$.

From the presented simulation results (Fig. 4-5), it is seen that during emergency braking in a turn during an emergency failure of the trailer braking system, the developed method of stabilizing the movement is effective. In no case did the road train leave its lane; the hitch angle is reduced by 49%.
Conclusion
1. A method has been developed to increase the stability of trailer-truck during emergency braking in a turn and failure of the braking system of the trailer, which allows to maintain the trajectory stability of the trailer-truck and to avoid accidents with severe consequences.
2. The effectiveness of the proposed method for increasing trailer-truck stability during emergency braking in a turn and failure of the braking system of trailer has been proved using simulation methods.

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