https://doi.org/10.14669/AM/225308

# MANOEUVRABILITY AND DIRECTIONAL STABILITY OF THREE-LINK ROAD TRAINS TYPE "B-TRIPLE"

**ROMAN MARCHUK[1], VOLODYMYR SAKHNO[2], VIKTOR POLIAKOV[3], MYKOLA MARCHUK[4], NAZAR MARCHUK[5], OLEKSII PALAMARCHUK[6]**

## Abstract

Improving the efficiency of freight road transport, in particular container transport, requires changes to the length and full mass of trailer and semitrailer road trains. Accordingly, for three-link layout schemes, the overall dimensions and weight parameters increase, while it is necessary to comply with the requirements for axle loads.
Since the regulatory frameworks of many countries require compliance with the axle load, this makes it necessary to increase the number of axles and to extend the road train, which adversely affects its turning performance. Conversely, improving turning performance through the use of steered axles (wheels) on trailers and semitrailers reduces straight-line directional stability. Therefore, to enhance the applicability of such road trains, the option of employing a dual steering drive system for the steered bogie axles of semitrailers is considered.
The research methodology involves taking into account folding angles of the road train links, steering drive transmission ratios and accordingly the turning angles of the semitrailers steered axles, which determine the links trajectories displacement and the road train

[1] Department of Automobiles and Automotive industry, National University of Water and Environmental Engineering, Rivne, Ukraine, email: r.m.marchuk@nuwm.edu.ua, ORCID: 0000-0002-9974-8769

[2] Automobiles Department, National Transport University, Kyiv, Ukraine, email: sakhno@ntu.edu.ua, ORCID: 0000-0002-5144-7131

[3] Automobiles Department, National Transport University, Kyiv, Ukraine, email: poljakov_2006@ukr.net, ORCID: 0000-0001-7042-3066

[4] Department of Automobiles and Automotive industry, National University of Water and Environmental Engineering, Rivne, Ukraine, email: m.m.marchuk@nuwm.edu.ua, ORCID: 0000-0003-2538-0882

[5] Department of Automobiles and Automotive industry, National University of Water and Environmental Engineering, Rivne, Ukraine, email: n.m.marchuk@nuwm.edu.ua, ORCID: 0000-0002-7439-7229

[6] Automobiles Department, National Transport University, Kyiv, Ukraine, email: alex-magtrans@ukr.net, ORCID: 0009-0003-9782-1015

swept path width for different locations of the semitrailer with extended-base and for circular movement of the road train.
For a three-link "B-triple" road train with non-steered semitrailers, the swept path width was 11.3 m, which significantly exceeds the permissible value of 7.2 m. Equipping the semitrailers with dual steering drive systems for the rear bogie axles substantially improved manoeuvrability, reducing the swept path width to 9.13 m, however, this value still does not meet regulatory requirements. At the same time, road trains with rear-steered axles (wheels) of the trailer links demonstrate inferior directional stability compared to road trains with front-steered axles (wheels) of the trailer links.
To reach a final conclusion regarding the feasibility of a particular road train configuration, further studies on such road trains' directional stability are required.

**Keywords:** road train; trailer; semitrailer; axle; wheel; link; stability; manoeuvrability

## 1. Introduction

Ensuring the sustainable and stable development of the national economy necessitates the implementation of intermodal transport systems. One of the most promising and widespread among them is container transportation. The container is the most promising and widespread multimodal means of transport. They are now transported by almost all modes of transport, owing to the mobility of the packaging [1]. The widespread implementation of container transport is based on the use of heavy goods vehicles and justifies energy savings and a reduction in environmental pollution due to the limited number of vehicles and drivers, necessary for the transport of large quantities of goods and people. As a consequence, two- and three-link truckloads are increasingly used, whose articulated joints make long vehicles versatile in use and allow quick manoeuvring even in busy urban environments.

In modern automotive engineering, when designing multi-articulated vehicle combinations, the choice of trailer units remains a relevant issue and is actively discussed in the literature. Two main layouts of multi-articulated vehicle combinations are distinguished: trailer and semitrailer configurations. In the trailer configuration, each unit rests on its own axles, whereas in the semitrailer configuration, each unit rests both on its own axles and on the axle(s) of the preceding unit. Among three-unit vehicle combinations, 35-m-long road trains with two trailers (Figure 1) or with three semitrailers capable of transporting three containers can be considered promising [2]. The use of multi-articulated vehicle combinations must be supported not only by progressive transport legislation, but also by the solution of specific technical problems aimed at ensuring a high level of safety of both the vehicle combination itself and the overall traffic flow, among which manoeuvrability and directional stability are undoubtedly key.

https://doi.org/10.14669/AM/225308

**Fig. 1. Three-link road train**

The basis of multi-articulated vehicle combinations, along with tractor units, is formed by trailers and semitrailers. At the same time, despite the considerable diversity of layout schemes of multi-articulated vehicles, their design can be composed of a relatively small number of structurally complete functional elements – modules. Thus, the unified system "road train" can be represented as consisting of two or more subsystems articulated with each other: the "tractor unit" and the "trailer units (trailer or semitrailer)", depending on the layout scheme of the combination. At present, two main layout schemes are typically used in multi-articulated vehicle combinations: trailer-based (vehicle + n-trailers) and semitrailer-based (tractor unit + n-semitrailers). Semitrailer schemes, in particular the "B-triple" type, require further investigation.

Articulated heavy vehicles or road trains consisting of a truck tractor and a semitrailer (or trailer) are among the most widely used means of road transport [3–4]. On the other hand, in contrast to single-unit vehicles, road trains create significant safety challenges on highways due to their large mass and considerable overall dimensions, the high position of the centres of mass of individual units, and the internal forces acting at their articulation points [5].

The presence of articulated joints helps to reduce the effective wheelbase of the vehicle, thereby improving low-speed turning capability, whereas the large road train mass and high centre of mass reduce stability at high speeds in three potentially unstable motion modes: folding, rollover and trailer sway [6–7]. Trailer sway and rollover are directly related to the increase in lateral acceleration acting at the centre of mass of the semitrailer (trailer). This occurs due to the generation of lateral forces at the semitrailer tyres with a time delay relative to the tractor unit, which results in larger yaw angles and higher lateral accelerations of the semitrailer [8]. Among the measures aimed at reducing lateral forces at the wheels of the road train, semitrailer steering systems appear to be the most effective. Various steering

https://doi.org/10.14669/AM/225308

concepts for articulated and multi-steered vehicles have been investigated since the early stages of road train research [9]. The most common passive steering systems include direct steering systems, self-steering axles and steering bogies. Studies of passive semitrailer steering systems have shown that, while these systems improve low-speed manoeuvrability and reduce tyre scrubbing, they degrade high-speed stability of the road train. Active steering systems allow designers to eliminate the complex mechanical mechanisms required for passive systems and, if properly tuned, can also provide appropriate steering performance for different objectives, regardless of vehicle speed. However, low-speed and high-speed requirements are inherently conflicting due to the different lateral slip behaviour of vehicles at different speeds [10–11]. In view of this, two critical safety problems of the road trains require improvement: insufficient static stability of the semitrailer (trailer) and a high tendency to rollover at high speeds, which can easily lead to accidents. This can be avoided by lateral control system based on braking and semitrailer (trailer) steering [12]. In this work, the amplification of lateral acceleration feedback is used to evaluate the lateral performance of the road train. The problem of cornering and lateral dynamics control is considered for the road train with a trailer on a "Dolly" trolley. Integrated chassis control approaches combining braking, steering and stability control functions for multi-axle vehicles are comprehensively reviewed in [13]. In [14], a control system for a tractor and semitrailer with a dolly combination with active feedback and feedforward steering control is investigated. In [15], the influence of steering-related parameters on articulated vehicle lateral stability is investigated using a single-track dynamic model.

Unlike passenger cars, for which static or divergent instability associated with oversteer is the primary concern, road trains may also exhibit dynamic or oscillatory instability, mainly due to reduced yaw damping at high speeds [16]. The road train is characterised by a unique dynamic phenomenon known as the rearward amplification (RWA) factor, defined as the ratio between the maximum peak lateral acceleration at the centre of mass of the semitrailer (trailer) and that of the tractor unit during a manoeuvre [17]. From the RWA perspective, road train is more prone to unstable motion modes, for example during a "slalom" manoeuvre. For most passenger cars, the rollover threshold exceeds 1.0 g; for light trucks, vans, and SUVs, it ranges from 0.8 g to 1.2 g; whereas for loaded road trains, it often lies well below 0.5 g [18].

The dominant factors affecting the lateral stability of the road train include the moment of inertia of the semitrailer (trailer), the longitudinal position of the centre of mass, the location of the coupling device, and the layout configuration (axle positions relative to the centre of mass) [19]. Trailer oscillations can be mitigated using appropriate controllers. Thus, in [20], the measured folding angle and its time derivative are used to generate asymmetric braking pressure at the semitrailer (trailer) wheels to limit trailer sway. In [21], an adaptive sliding-mode controller was proposed to improve the lateral stability of articulated vehicles based on yaw rate and articulation angle control. In [22], an active trailer differential braking controller based on a linear quadratic regulator (LQR) is investigated, whose integration with active differential braking of the tractor further improves the lateral stability of both the

https://doi.org/10.14669/AM/225308

vehicle and the trailer. In [23] a comprehensive review and classification of control schemes for improving the lateral stability of car-trailer combinations is presented. The study analyses various active steering and yaw stability control approaches for articulated vehicle systems, including coordinated steering strategies for both tractor and trailer units. The review also compares the effectiveness of different control methods under various driving conditions using standard manoeuvres and lateral stability performance indicators.

The safe operation of any vehicle is largely determined by its dynamic properties, particularly its stability and handling performance. At present, the problem of determining the stability conditions of two-link articulated vehicle combinations has been extensively studied. In particular, studies [24–25] investigate the dynamic stability and motion behaviour of articulated transport systems using multi-body and linearised dynamic models. The equations of motion are derived considering the influence of braking, acceleration and lateral dynamic effects. However, for three-link road trains, such mathematical models still require further development, particularly for manoeuvrability and stability analysis. Therefore, the aim of this study is to determine the lateral manoeuvrability and stability indices of "B-triple" type road trains and to identify ways to improve them.

## 2. Materials and Methods

We use a well-known system of equations that describes the motion of a three-link road train type "B-double". This system of equations was taken as the basis for formulating the equations of motion for the road train type "B-triple" (Figure 2 and 3).

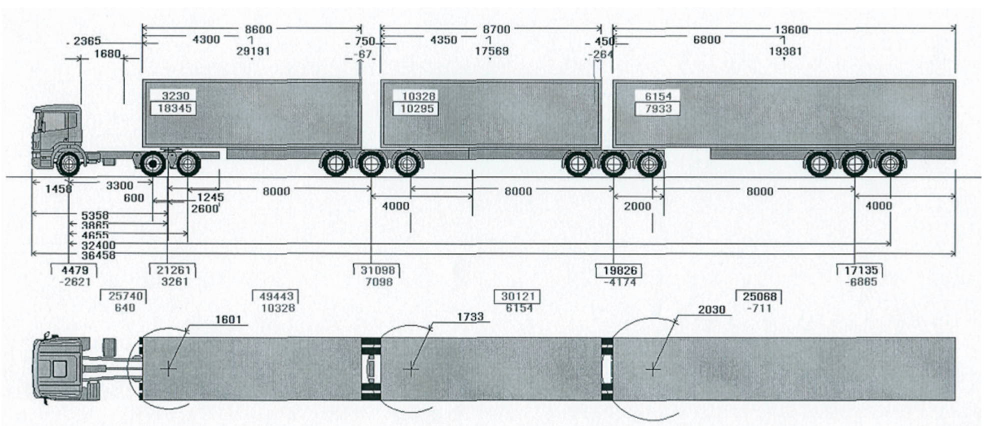



**Fig. 2. Three-link road train type "B-triple"**

https://doi.org/10.14669/AM/225308

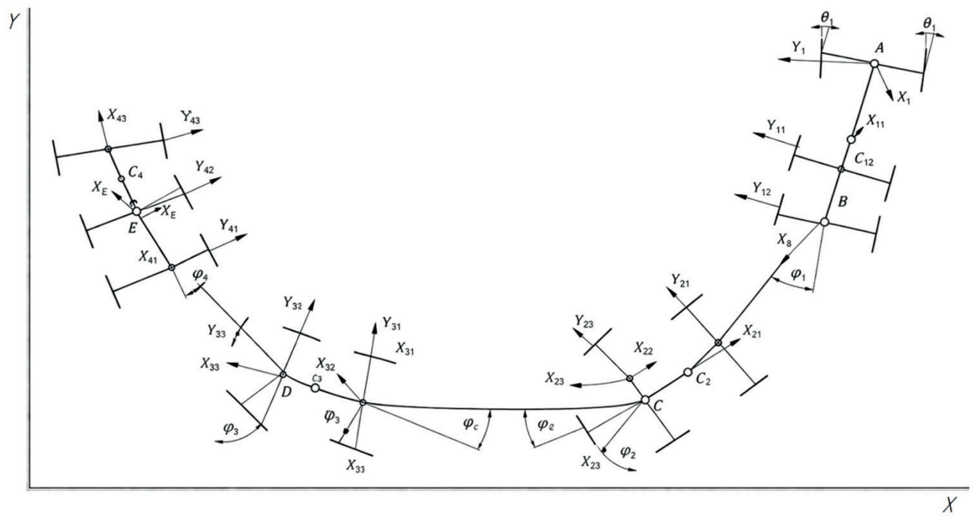



**Fig. 3. Computational scheme of a three-link road train type “B-triple”**

When formulating the equations of motion, the following notation is introduced:

$m$, $I$; $m_1$, $I_1$; $m_2$, $I_2$; $m_3$, $I_3$; $m_4$, $I_4$; $m_5$, $I_5$; $m_6$, $I_6$ – mass and central moment of inertia of the tractor unit’s, the bogie of the first semitrailer, the frame of the first semitrailer, the bogie of the second semitrailer, the frame of the second semitrailer, the bogie of the third semitrailer, and the frame of the third semitrailer, respectively, kg and kg·m²;

$\gamma_1$, $\gamma_2$, $\gamma_3$ – articulation angles between the road train links, rad;

$\dot{\gamma}_i$, $\ddot{\gamma}_i$ – derivative of angular velocity and derivative of angular acceleration, rad/s$^2$ and rad/s$^3$;

$\varphi_1$, $\varphi_2$, $\varphi_3$, $\varphi_4$, $\varphi_5$, $\varphi_6$ – absolute angles between the tractor frame and the frame of the first semitrailer; between the frame of the first semitrailer and its bogie; between the bogie of the first semitrailer and the frame of the second semitrailer; between the frame of the second semitrailer and its bogie; between the bogie of the second semitrailer and the frame of the third semitrailer; and between the frame of the third semitrailer and its bogie, respectively, rad;

$V$, $U$ – longitudinal andlateral, velocities of the tractor unit’s centre of mass, m/s;

$\omega$ – yaw velocity of the tractor unit’s centre of mass, rad/s;

$\dot{\omega}$ – yaw acceleration of the tractor unit’s centre of mass, rad/s$^2$;

$M$ – mass of the tractor (equivalent mass of the system reduced to its centre of mass), kg;

$M_{11}$ - inertial coefficient (equivalent mass / additional mass term), kg;

$M_i = f(\varphi_i)$ – turning resistance moments of the road train links, N·m;

$X_{ij}$, $Y_{ij}$, $Z_{ij}$ – longitudinal, lateral, and vertical reactions of the supporting surface acting on the road train wheels, N;

$i$ – index denoting the vehicle segment (tractor unit or semitrailer, $i$ = 1 to 4);

$j$ – index denoting the axle number of the corresponding segment (j = 1 to 3);

https://doi.org/10.14669/AM/225308

a – distance from the front axle to the centre of mass of the tractor unit's, m;
c – distance from the centre of mass of the tractor unit's to the coupling point with the first semitrailer, m;
$d_1, d_2, d_3, d_4, d_5, d_6$ – distances from the centre of mass of the first semitrailer to the coupling point with the tractor; from the centre of mass of the bogie of the first semitrailer to its front axle; from the centre of mass of the second semitrailer to the coupling point with the first semitrailer; from the centre of mass of the bogie of the second semitrailer to its front axle; from the centre of mass of the third semitrailer to the coupling point with the second semitrailer; and from the centre of mass of the bogie of the third semitrailer to its front axle, respectively, m;
$l_1, l_3, l_5$ – distances from the bogie centre of mass to the coupling point with the first, second, and third semitrailers, respectively, m;
$l_2, l_4, l_6$ – distances from the bogie centre of mass to the point located midway between the first and second axles of the first, second, and third semitrailers, respectively, m;
θ – steering angle of the tractor wheels, rad;
$\theta_1, \theta_2, \theta_3$ – steering angles of the bogie axles of the first, second, and third semitrailers, respectively, rad.

This is an example of an equation:

$$M\big(\dot{U} - V\omega\big) + M_{11}c\omega^2 - (m_1d_1 + m_2l_1 + (m_3 + m_4)l_2) \cdot (\ddot{\gamma}_1 cos\varphi_1 - \dot{\gamma}_1^2 sin\varphi_1) - m_2d_2 \cdot (\dot{\gamma}_1 cos\gamma_1 - \dot{\gamma}_1^2 sin\gamma_1) - (m_3d_3 + m_4l_3) \cdot (\dot{\gamma}_2 cos\gamma_2 - \dot{\gamma}_2^2 sin\gamma_2) - m_4d_4 \cdot (\dot{\gamma}_2 cos\gamma_2 - \dot{\gamma}_2^2 sin\gamma_2) - (m_5d_5 + m_6l_5) \cdot (\ddot{\gamma}_3 cos\varphi_5 + \dot{\gamma}_3^2 sin\varphi_5) = -\sum_{i=1}^{n}(X_{1i} + Y_{1i} sin\theta_{1i}) - \sum_{j=1}^{n}\big(X_{2j} sin\gamma_1 + Y_{2j} cos\gamma_1\big) - \sum_{k=1}^{n}\big(X_{3k} sin(\theta_2 + \gamma_2) + Y_{3k} cos(\theta_2 + \gamma_2)\big) - \sum_{p=1}^{n}\Big(X_{4p} sin(\theta_3 + \gamma_3) + Y_{4p} cos(\theta_3 + \gamma_3)\Big) \tag{1}$$

$$I\dot{\omega} + cM_1 \cdot \left(\dot{\omega} - \frac{U + V\omega}{c}\right) + c(m_1d_1 + m_2l_1 + (m_3 + m_4)l_2) \cdot (\ddot{\gamma}_1 cos\varphi_1 + \dot{\gamma}_1^2 sin\varphi_1) + cm_2d_2(\dot{\gamma}_1 cos\gamma_1 + \dot{\gamma}_1^2 sin\gamma_1) + c(m_3d_3 + m_4l_3) \cdot (\dot{\gamma}_2 cos\gamma_2 + \dot{\gamma}_2^2 sin\gamma_2) = a\sum_{i=1}^{n}(Y_{1i} cos\theta_{1i} - X_{1i} sin\theta_{1i}) - c\sum_{j,k=1}^{n}\big(X_{jk} sin\gamma_j + Y_{jk} cos\gamma_j\big) \tag{2}$$

– for the frame of the first semitrailer:

$$(I_1 + m_1d_1^2 + m_2l_1^2 + (m_3 + m_4)l_2^2)\ddot{\gamma}_1 + (m_1d_1 + m_2l_1 + (m_3 + m_4)l_2) \cdot \big[\big(\dot{V} - U\omega\big) sin\varphi_1 + \big(V\omega - \dot{U}\big) cos\varphi_1\big] - m_2d_2l_1(\dot{\gamma}_1 cos\varphi_2 + \dot{\gamma}_1^2 sin\varphi_2) = d_1\sum_{j=1}^{n} X_{2j} sin\gamma_1 + l_1\sum_{j=1}^{n} X_{2j} sin\varphi_2 + d_2\sum_{j=1}^{n} Y_{2j} cos\gamma_1 + l_2\sum_{j=1}^{n} Y_{2j} cos\varphi_2 \tag{3}$$

– for the bogie of the first semitrailer:

$$(I_2 + m_2d_2^2)\ddot{\gamma}_1 + (m_3d_3 + m_4l_3) \cdot \big[\big(\dot{V} - U\omega\big) \cdot sin(\varphi_1 + \varphi_2) + \big(V\omega - \dot{U}\big) \cdot cos(\varphi_1 + \varphi_2)\big] = d_2\sum_{j=1}^{n} Y_{2j} + l_2\sum_{j=1}^{n} Y_{2j} + M_1 \tag{4}$$

https://doi.org/10.14669/AM/225308

– for the frame of the second semitrailer:

$$(I_3+m_3d_3^2+m_4l_3^2)\ddot{\gamma}_2 + (m_3d_3+m_4l_3)\cdot\left[\left(\dot{U}-V\omega\right)sin\gamma_2 + \left(V\omega-\dot{U}\right)cos\gamma_2\right] = d_3\sum_{k=1}^{n}X_{3k}sin\gamma_2 + l_3\sum_{k=1}^{n}X_{3k}sin\varphi_3 + d_3\sum_{k=1}^{n}Y_{3k}cos\gamma_2 + l_3\sum_{k=1}^{n}Y_{3k}cos\varphi_3 \tag{5}$$

– for the bogie of the second semitrailer:

$$(I_4+m_4d_4^2)\ddot{\gamma}_2 + m_4d_4\left[\left(\dot{U}-V\omega\right)sin\gamma_2 + \left(V\omega-\dot{U}\right)cos\gamma_2\right] = d_4\sum_{k=1}^{n}Y_{3k} + l_4\sum_{k=1}^{n}Y_{3k} + M_2 \tag{6}$$

– for the frame of the third semitrailer:

$$\left(I_5+m_5d_5^2+m_6l_5^2\right)\ddot{\gamma}_3 - l_5(m_5d_5+m_6l_5)\dot{\gamma}_3^2 = d_5\sum_{p=1}^{n}X_{4p}sin\gamma_3 + l_5\sum_{p=1}^{n}X_{4p}sin\varphi_5 + d_5\sum_{p=1}^{n}Y_{4p}cos\gamma_3 + l_5\sum_{p=1}^{n}Y_{4p}cos\varphi_5 \tag{7}$$

– for the bogie of the third semitrailer:

$$(I_6+m_6d_6^2)\ddot{\gamma}_3 + m_6d_6\left[\left(\dot{U}-V\omega\right)sin\gamma_3 + \left(V\omega-\dot{U}\right)cos\gamma_3\right] = d_6\sum_{p=1}^{n}Y_{4p} + l_6\sum_{p=1}^{n}Y_{4p} + M_3 \tag{8}$$

The system of equations (1–8) includes the longitudinal and lateral forces acting on the road train wheels, as well as the road train layout and mass parameters.

The longitudinal forces $X_i$ are defined as:

$$X_i = f\cdot Z_i \tag{9}$$

f – rolling resistance coefficient of the road train wheels, assumed to be identical for all wheels;
$Z_i$ – normal reaction of the supporting surface acting on the wheels of the $i$–th axle of the road train, N.

The lateral forces acting on the wheels of the road train axles are determined using the D.A. Antonov formula:

$$Y = qk_{y0e}\delta \tag{10}$$

$$Y_{ij} = q_{ij}k_{y0e,ij}\delta_{ij} \tag{11}$$

$q_{ij}$ – coefficient accounting for the influence on tyre slip of the normal support reactions redistribution, traction and braking forces, the inclination angle of the wheel plane relative to the supporting surface, tyre inflation pressure, slip of the rear non–steered wheels, transient slip, road surface quality, wheel oscillations when motion over an uneven surface and wheel motion on unpaved roads;
$k_{y0e,ij}$ – extreme (maximum) lateral stiffness coefficient;
$\delta_{ij}$ – tyre (axle) slip angle, rad.

https://doi.org/10.14669/AM/225308

The viscous friction moments in the steering systems of the tractor unit and semitrailer are proportional to the angular velocity of the equivalent wheels rotation:

$$M_{hi} = h_i \cdot \dot{\theta}_i \tag{12}$$

$h_i$ – viscous friction coefficients in the steering system components, $h_i$= 15 Nms/rad;
$\dot{\theta}_i$ – angular velocity of the steered wheels rotation, rad/s.

The elastic moments in the steering systems of the tractor unit and semitrailer are proportional to the steering angles of the equivalent wheels and to the stiffness of the steering mechanism:

$$M_{pi} = \chi_i \cdot \theta_i \tag{13}$$

$\chi_i$ – steering mechanism stiffness coefficients, $\chi_i$ = 110 to 170 Nm/degree.

The steering angle of the semitrailer bogie axle is determined as:

$$\theta_{1,2,3} = u_2 \varphi_{2,4,6} - u_1 \varphi_{1,3,5} \tag{14}$$

$u_1$, $u_2$ – transmission ratios of the direct and feedback drives determine the dual steering drive transmission ratio of the semitrailer axle.

## 3. Results

Integration the equations of motion (1–8) was performed using the following initial data: $m$ = 2400 kg; $m_1$ = $m_3$ = $m_5$ = 1700 kg; $m_2$ = $m_4$ = $m_6$ = 2500 kg (with the semitrailer loaded with a 40-ft container) and $m_2$ = $m_4$ = $m_6$ = 1830 kg (with the semitrailer loaded with a 20-ft container); $k_{y1}$ = 160000 Nm; $k_{y12}$ = $k_{y13}$ = 220000 Nm; $k_{y2}$ = $k_{y21}$ = $k_{y22}$ = $k_{y3}$ = $k_{y31}$ = $k_{y32}$ = $k_{y4}$ = $k_{y41}$ = $k_{y42}$ = 240000 Nm; $I_y$ = 36288 Ns/m; $I_{y1}$ = $I_{y3}$ = $I_{y5}$ = 275.4 Ns/m (with the semitrailer loaded with a 20-ft container); $I_{y2}$ = $I_{y4}$ = $I_{y6}$ = 15550 Ns/m (with the semitrailer loaded with a 20-ft container); $I_{y2}$ = $I_{y4}$ = $I_{y6}$ = 29350 Ns/m (with the semitrailer loaded with a 40-ft container); $h$ = 15 Nms/rad; $\chi$ = 145 Nm/degree; $a$ = 2.593 m; $b$ = 1.987 m; $c$ = 4.505 m; $d_1$ = $d_3$ = $d_5$ = 8.5 m (with the semitrailer loaded with a 40-ft container); $d_1$ = $d_3$ = $d_5$ = 5.6 m (with the semitrailer loaded with a 20-ft container); $d_2$ = $d_4$ = $d_6$ = 1.6 m; $l_1$ = l2 = l3 = 1.05 m; l4 = l5 = l6 = 1.65 m; $u_1$ = 0.95; $u_2$ = 1.15.

For these initial conditions, as well as for the road train turning mode defined by the steering mode coefficient ($k_t = \dot{\theta}/V_a$, $\dot{\theta}$ – is the equivalent angular steering speed of the tractor wheels), the road train manoeuvrability and motion stability indices were determined.

https://doi.org/10.14669/AM/225308

Figures 4–6 present the calculated yaw velocity results of the road train links, lateral acceleration, wheel slip angles of the axles and articulation angles of the units (links) during the execution of the "entry into a turn" manoeuvre.

The stable nature of motion of the road train links during this manoeuvre is indicated by the decaying oscillations of the links yaw velocity (Figure 4). However, it should be noted that the absolute values of the yaw velocity of the third semitrailer are significantly higher than those of the first and second semitrailers, which may lead to a loss of stability as the motion speed of the road train increases.

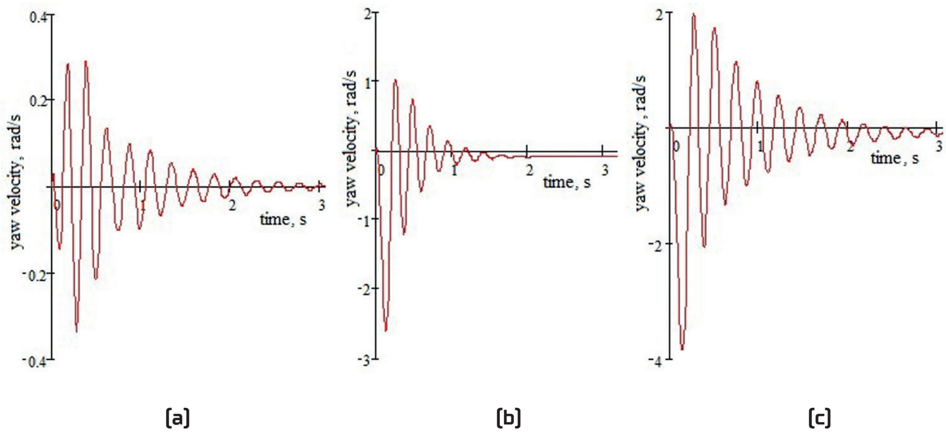
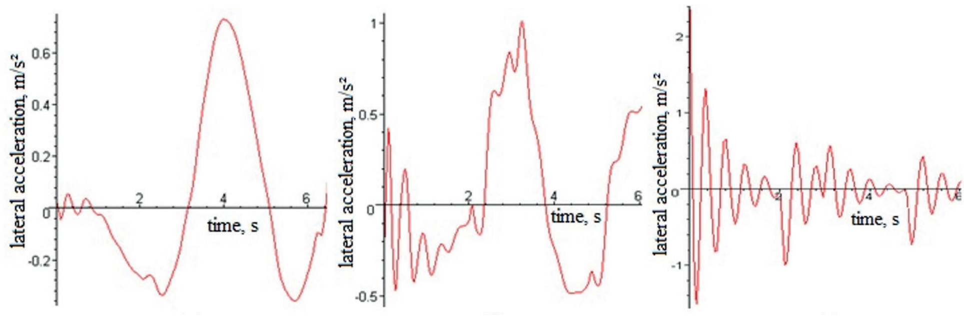



**Fig. 4. Variation of the yaw velocity during the road train entry into a turn; (a) the first semitrailer, (b) the second semitrailer, (c) the third semitrailer**

The motion stability can be assessed to a greater extent by the magnitude of the lateral accelerations acting at the centres of mass of the individual links. According to the literature, motion stability can be considered satisfactory if the lateral accelerations at the centre of mass of a link do not exceed 0.45 g [2]. This condition is satisfied by the road train under consideration (Figure 5). In addition, the decaying nature of the oscillations also indicates stable motion of the road train. However, the accelerations acting at the centre of mass of the third semitrailer exhibit a cyclic character, which, with an increase in motion speed, may lead to a loss of motion stability of the road train.

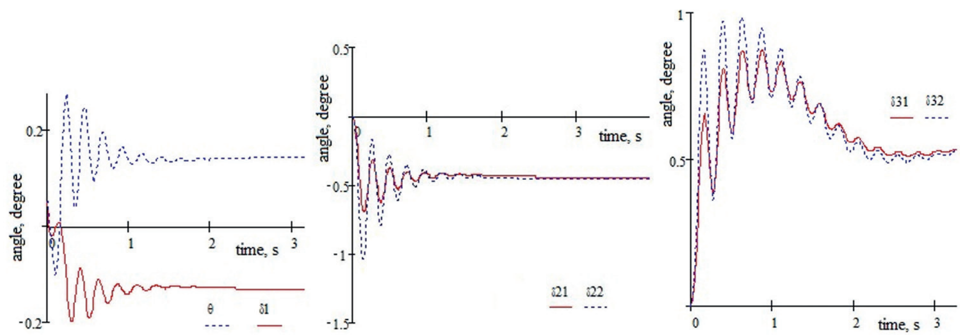
https://doi.org/10.14669/AM/225308




(a) (b) (c)

**Fig. 5. Variation of the lateral acceleration during the road train entry into a turn over the transient process; (a) the first semitrailer, (b) the second semitrailer, (c) the third semitrailer**

Based on the values of lateral accelerations, the lateral forces acting at the centres of mass of the individual links were determined and subsequently the lateral forces and slip angles of the links were obtained (Figure 6).



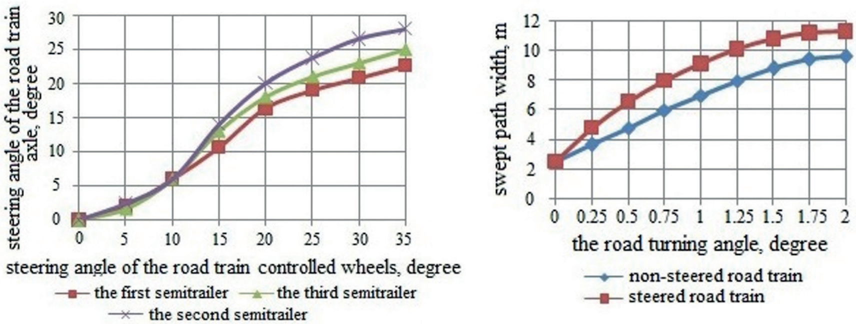

(a) (b) (c)

**Fig. 6. Variation of the steering angle and slip angles of the steered axle of the first semitrailer (a) and the non-steered axles of the second (b) and third (c) semitrailers during the road train entry into a turn**

It should be noted that the maximum slip angles of the second and third semitrailers are approximately 2–5 times greater than those of the steered wheels of the tractor unit. The variable nature of lateral acceleration and lateral forces leads to oscillations of the lateral slip angles of the links, which may result in excessive road train oversteer and a loss of stability as the motion speed increases.

https://doi.org/10.14669/AM/225308

The slip angles, together with the steering angle of the tractor controlled wheels, determine the folding angles of the road train links, which in turn define the steering angles of the semi-trailer axles, the displacement of the trajectories of characteristic points of the semitrailers relative to the trajectory of the tractor unit, and the road train swept path width (Figure 7).




(a) (b)

**Fig. 7. Variation of the steering angles of the controlled axles of the semi-trailers (a) and the road train swept path width (b) as a function of the trajectory turning angle**

Thus, the displacement of the trajectories of the semitrailers' characteristic points increases progressively as the steering angle of the tractor steered wheels changes from 0 (at turn entry) to $2\pi$ (during circular motion), as shown in Figure 6a. Based on the magnitude of the semitrailers trajectory displacement, the road train swept path width was determined. The baseline configuration was taken as a three-link "B-triple" road train with non-steered semi-trailers. For this road train, the swept path width during circular motion amounted to 11.3 m, which significantly exceeds the permissible value $Bg$ = 7.2 m. Equipping the semitrailers with dual steering systems on the rear bogie axles significantly improved the road train manoeuvrability indices. In particular, the swept path width of the three-link "B-triple" road train with a dual steering system on the steered rear axles of the semitrailers was reduced to 9.13 m, which still does not meet the requirements of regulatory documents (Figure 6b). At the same time, both the steered and non-steered road trains exhibit insufficient turning ability, which may serve as a prerequisite for motion stability.

The analysis of the calculated yaw velocity and lateral accelerations acting at the centres of mass of the semitrailers at a vehicle speed of 5 m/s indicates a steady-state motion of the road train.

An increase in the road train speed up to 15 m/s during circular motion also confirms satis-factory motion stability of the road train (Figure 8) and the possibility of its operation on the

https://doi.org/10.14669/AM/225308

general road network. In this case, increasing the outer turning radius to 15 m ensures acceptable values of handling and stability indicators, which allows the operating conditions of the road train to be considered satisfactory.

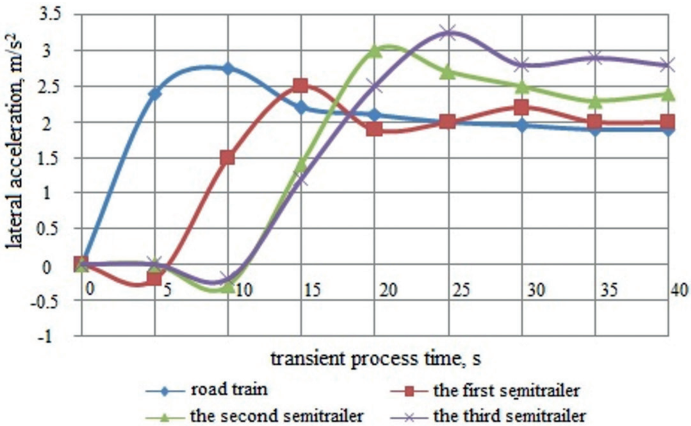



**Fig. 8. Lateral acceleration of the road train links at a speed of 15 m/s and circular motion**

## 4. Discussion

The integration of the equations of motion was performed for a road train consisting of a Scania tractor and three Krone semitrailers loaded with two 20-foot containers and one 40-foot container. The yaw velocity of the road train links, lateral accelerations, wheel slip angles, folding angles and swept path width during the manoeuvre "entry into a turn" were determined. The stable nature of the motion of the road train links during this manoeuvre is confirmed by the decaying oscillations of the yaw velocity and lateral acceleration, whose magnitudes do not exceed the permissible limits.

The slip angles, together with the steering angle of the tractor's controlled wheels, determine the folding angles of the road train links. In turn, these angles define the steering angles of the controlled axles of the semitrailers, the lateral displacement of the characteristic points of the semitrailers relative to the tractor trajectory, and the swept path width of the road train. For a three-unit road train, type "B-triple" with non-steered semitrailers, the swept path width reached 11.3 m, which significantly exceeds the permissible value of 7.2 m. Equipping the semitrailers with dual steering systems on the rear bogie axles significantly improved the manoeuvrability of the road train (the swept path width decreased to 9.13 m); however, even in this case, the road train with steered semitrailers does not meet the regulatory requirements. At the same time, both steered and non-steered road trains exhibit insufficient turning ability, which may serve as a prerequisite for stable motion.

## 5. Conclusions

As a result of the conducted research, the following conclusions can be drawn:

- the system of equations describing the planar motion of a three-unit "B-triple" road train with semitrailers equipped with dual steering systems on the rear bogie axles has been improved,
- it was established that, according to the yaw velocity and the magnitude of lateral accelerations acting at the centres of mass of the semitrailers at a speed of 5 m/s, the motion of the road train is stable. Increasing the road train speed to 15 m/s during circular motion also indicates satisfactory motion stability and the possibility of its operation on the general road network,
- it was shown that for a three-unit "B-triple" road train with non-steered semitrailers, the swept path width is 11.3 m, which significantly exceeds the permissible value of 7.2 m. Equipping the semitrailers with dual steering systems on the rear bogie axles significantly improves the manoeuvrability of the road train (the swept path width decreases to 9.13 m); however, the road train with steered semitrailers still does not meet regulatory requirements. At the same time, both steered and non-steered road trains demonstrate insufficient turning ability, which may contribute to motion stability.

Based on the results, further research on improving manoeuvrability indicators for three-link road trains type "B-triple" can be achieved in two ways: by revising transport legislation regarding their manoeuvrability or by developing more advanced control systems.

## 6. References

[1] Song D. A literature review, container shipping supply chain: planning problems and research opportunities. Logistics. 2021;5(2):41. https://doi.org/10.3390/logistics5020041.

[2] Sakhno VP, Sharai SM, Murovanyi IS, Chovcha IV. Do vyznachennia stiikosti rukhu trylankovykh avtopoizdiv [To determination of stability movement three-link road trains]. Modern Technologies in Mechanical Engineering and Transport. 2022;1(18):155–166. https://doi.org/10.36910/automash.v1i18.772.

[3] Sharma T, He Y. On trade-off relationship between static and dynamic lateral stabilities of articulated heavy vehicles. Designs. 2024;8(5):103. https://doi.org/10.3390/designs8050103.

[4] Sharma T, He Y. Design of a tracking controller for autonomous articulated heavy vehicles using a nonlinear model predictive control technique. Proceedings of the Institution of Mechanical Engineers, Part K: Journal of Multi-body Dynamics. 2024;238(2):334–362. https://doi.org/10.1177/14644193241232353.

[5] Hou Y, Xu X. High-speed lateral stability and trajectory tracking performance for a tractor-semitrailer with active trailer steering. PLoS One. 2022;17(11):e0277358. https://doi.org/10.1371/journal.pone.0277358.

[6] Milani S, Ünlüsoy YS, Marzbani H, Jazar RN. Semitrailer steering control for improved articulated vehicle manoeuvrability and stability. Nonlinear Engineering. 2019;8(1):568–581. https://doi.org/10.1515/nleng-2018-0124.

[7] Tian J, Zeng Q, Wang P, Wang X. Active steering control based on preview theory for articulated heavy vehicles. PLoS One. 2021;16(5):e0252098. https://doi.org/10.1371/journal.pone.0252098.

[8] Deng Z, Jin Y, Gao W, Wang B. A closed-loop directional dynamics control with LQR active trailer steering for articulated heavy vehicle. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering. 2023;237(12):2741–2758. https://doi.org/10.1177/09544070221121859.

[9] Ljungqvist O, Axehill D. A predictive path-following controller for multi-steered articulated vehicles. IFAC-PapersOnLine. 2020;53(2):15725–15732. https://doi.org/10.1016/j.ifacol.2020.12.053.

[10] Vempaty S, He Y. A review of car-trailer lateral stability control approaches. SAE Technical Paper. 2017. https://doi.org/10.4271/2017-01-1580.

[11] Emheisen MA, Emirler MT, Ozkan B. Lateral stability control of articulated heavy vehicles based on active steering system. International Journal of Mechanical Engineering and Robotics Research. 2022;11(8):575–582. https://doi.org/10.18178/ijmerr.11.8.575-582.

[12] Cai H, Xu X. Lateral stability control of a tractor-semitrailer at high speed. Machines. 2022;10(8):716. https://doi.org/10.3390/machines10080716.

[13] Zakaria MSM, Singh ASP, Aras MSM. Integrated chassis control for multi-axle vehicles: a comprehensive review. International Journal of Automotive and Mechanical Engineering. 2024;21(3):11663–11681. https://doi.org/10.15282/ijame.21.3.2024.17.0900.

[14] Sadeghi Kati M, Fredriksson J, Jacobson B, Laine L. Gain-scheduled H∞ controller synthesis for actively steered longer and heavier commercial vehicles. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering. 2020;234(7):1740–1758. https://doi.org/10.1177/0954407019870352.

[15] Peng D, Fang K, Kuang J, Hassan MA, Tan G. Conflict and sensitivity analysis of articulated vehicle lateral stability based on single-track model. Shock and Vibration. 2021;2021:5893993. https://doi.org/10.1155/2021/5893993.

[16] Lei T, Wang J, Yao Z. Modelling and stability analysis of articulated vehicles. Applied Sciences. 2021;11(8):3663. https://doi.org/10.3390/app11083663.

[17] He Y, Islam MM, Zhu S, Hu T. A design synthesis framework for directional performance optimization of multi-trailer articulated heavy vehicles with trailer lateral dynamic control systems. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering. 2017;231(8):1096–1125. https://doi.org/10.1177/0954407016671284.

[18] Tan VV, Sename O, Gáspár P. Improving roll stability of tractor semi-trailer vehicles by using H∞ active anti-roll bar control system. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering. 2021;235(14):3509–3520. https://doi.org/10.1177/09544070211013949.

[19] Marumo Y, Yokota T, Aoki A. Improving stability and lane-keeping performance for multi-articulated vehicles using vector follower control. Vehicle System Dynamics. 2020;58(12):1859–1872. https://doi.org/10.1080/00423114.2019.1651877.

[20] Catterick J, Botha T, Els S. Articulated vehicle stability control using brake-based torque vectoring on trailer using nonlinear model predictive control. SAE International Journal of Commercial Vehicles. 2023;16(1):111–126. https://doi.org/10.4271/02-16-01-0007.

[21] Esmaeili N, Kazemi R, Tabatabaei Oreh SH. An adaptive sliding mode controller for the lateral control of articulated long vehicles. Proceedings of the Institution of Mechanical Engineers, Part K: Journal of Multi-body Dynamics. 2019;233(3):487–515. https://doi.org/10.1177/1464419318806801.

[22] Abroshan M, Hajiloo R, Hashemi E, Khajepour A. Model predictive-based tractor-trailer stabilisation using differential braking with experimental verification. Vehicle System Dynamics. 2021;59(8):1190–1213. https://doi.org/10.1080/00423114.2020.1744024.

https://doi.org/10.14669/AM/225308

[23] Vempaty S, He Y, Zhao L. An overview of control schemes for improving the lateral stability of car-trailer combinations. International Journal of Vehicle Performance. 2020;6(2):151–199. https://doi.org/10.1504/IJVP.2020.106985.

[24] Sorge F. A nonlinear dynamical approach to the path correction of multi-steering articulated vehicles. Vehicle System Dynamics. 2021;59(11):1759–1780. https://doi.org/10.1080/00423114.2020.1791350.

[25] De Felice A, Mercantini M, Sorrentino S. Stability analysis of articulated bus in straight-ahead running manoeuvre. Journal of Applied and Computational Mechanics. 2021;7(3):1649–1662. https://doi.org/10.22055/JACM.2021.36566.2869.