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ASSESSMENT OF THE INFLUENCE OF A LOAD POSITION ON A TWO-AXLE TRAILER ON DRIVING SAFETY BY MEANS OF COMPUTER SIMULATIONS

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ABSTRACT

Car-trailer combinations are relatively popular in many countries. A trailer is a means of transport that increases the load-carrying capacity of a car. However, coupling a trailer to a car significantly affects the dynamic behaviour of the resulting car–trailer combination. The longitudinal position of the load on the trailer superstructure represents one of the key factors influencing its driving characteristics. The goal of this research is to assess the driving properties of a car-trailer combination from the perspective of driving safety. The analysis was carried out using computer simulations in a commercial multibody system software environment. Driving safety is assessed for a scenario in which the car–trailer combination passes over a road obstacle at various vehicle speeds. This approach allows the identification of load positions at which the combination becomes unstable, potentially leading to an accident. Two load levels were considered, corresponding to 20% and 80% of the trailer's maximum load capacity. For each load case, three longitudinal positions of the centre of gravity were analysed: in *front of the first axle, between the axles, and behind the second axle of the trailer. The results show that the lowest level of driving safety occurs when the car–trailer combination passes over the obstacle at high speed with the load positioned behind the second axle of the trailer. Based on the obtained results, a suitable load position ensuring improved stability and driving safety is proposed.*

Keywords: *two-axle trailer, simulation computations, driving safety, multibody model*

1. INTRODUCTION

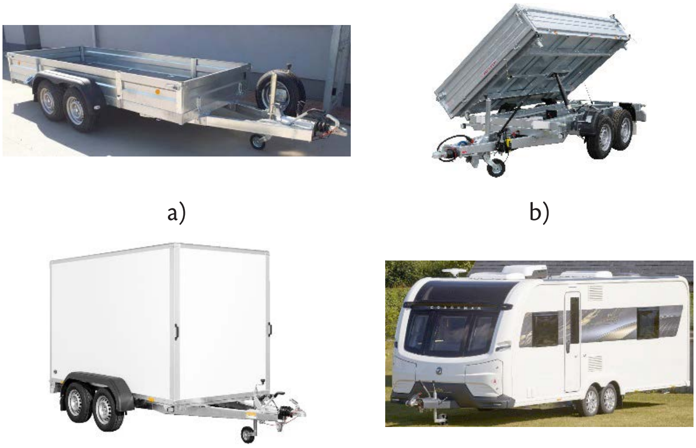
Individual passenger transport remains highly popular in many countries. Passenger cars provide a degree of personal freedom, allowing users to travel from one place to another at a time of their choosing (Thomas and Cabrera Serrenho, 2026). Modern passenger cars are also sufficiently powerful to tow trailers, which significantly increases their transport capacity.

A wide range of trailer types is currently available, differing in both design and load capacity. From a design perspective, trailers range from simple single-axle configurations to larger units equipped with two or three axles. In the context of passenger car–trailer combinations, two principal weight categories are defined: O1 and O2. Category O1 includes trailers with a maximum permissible mass of up to 750 kg. These lighter trailers typically require only a standard passenger car driving licence and are usually equipped with a single axle. Category O2 comprises trailers with a maximum permissible mass exceeding 750 kg and up to 3,500 kg. These trailers are generally larger and enable the transport of significantly greater loads. Their design often includes two axles, particularly when the total weight exceeds 1,800 kg.

Trailers in the O2 category, especially those with two axles, are manufactured in a wide variety of configurations. Their superstructures are designed for diverse applications, and their weight and dimensions are non-negligible, even when towed by SUVs or vans (Synák and Jakubovičová, 2024). The present research focuses specifically on this category of trailers. Two-axle trailers may have wheels positioned either alongside the loading platform or beneath it. Their superstructures can take various forms, including flatbed platforms for general transport, tilting platforms, car transporters, mini-excavator carriers, box trailers, caravans, and others. Selected examples of these trailer types are shown in Fig. 1.

Transport and traffic safety remain critical considerations for all road users. The driver is responsible not only for the proper distribution of the load on the trailer but also for the safe manoeuvring of the car–trailer combination during operation. It is well established that improper load distribution, in combination with certain driving speeds, can destabilise the entire vehicle combination. If such motion is not promptly controlled, it may lead to a serious accident. This phenomenon is referred to as sway motion and is illustrated in Fig. 2.

Figure 1. Examples of types of two-axle trailers: a) a universal platform trailer with wheels along the platform, b) a tilting trailer with wheels under the platform, c) a box trailer, d) a caravan



a) b)

c) d)

Source: a) Unikol (2026); b), c) Pongratz (2026); d) Coachman (2026)

Figure 2. An illustration of a road accident due to the sway motion of a trailer

Source: Caravan crash! Caravan accident! Stabiliser no safety! (2026)

Experimental investigation of the driving behaviour of a car–trailer combination with respect to load distribution is associated with significant safety risks and is therefore practically infeasible under real operating conditions. For this reason, some researchers employ scaled models to evaluate the influence of load distribution on driving properties. However, such models do not fully capture the complexities of real vehicles or actual operating conditions.

Therefore, computational modelling and simulation represent effective and accessible approaches for analysing the behaviour of car–trailer combinations under defined load cases. The present study employs a multibody simulation model of a car–trailer combination to investigate its dynamic response for various load configurations. Specifically, a two-axle trailer is analysed at load levels of 20% and 80% of its maximum load capacity. For each load level, three longitudinal positions of the load centre of gravity are considered: in front of the first axle, between the axles, and behind the second axle.

The main goal of the study is to determine the vehicle speed at which stable motion of the car–trailer combination is lost when passing over a road obstacle. The limiting condition is defined as the maximum driving speed corresponding to the most critical load position, i.e., when the load is located behind the rear axle. This case is subsequently compared with configurations where the load is positioned between the axles and in front of the first axle. Loss of stability is evaluated based on the wheel–road contact forces acting on the trailer. The results are presented in the form of graphical outputs.

2. A LITERATURE REVIEW OF CURRENT RESEARCH

A literature review provides an appropriate foundation for identifying current trends in the analysis and investigation of the driving behaviour of car–trailer combinations under real operating conditions.

Robat *et al.* (2024) developed a multibody model of a vehicle–trailer system configured as a differentially driven wheeled robot, aiming to enhance both payload capacity and manoeuvrability. The simulation results demonstrate the superior performance of the proposed approach compared to existing methods. The influence of high driving speeds on the stability of car–trailer combinations was further investigated by Zhang *et al.* (2025a), who showed that lateral stability can be significantly compromised, potentially leading to severe sway motion. To address this issue, they proposed a phase-portrait-based sliding mode control strategy using coordinated braking of both the towing vehicle and the trailer, enabling direct yaw moment control and improving dynamic stability.

Viganico *et al.* (2025) assessed the stability and safety of vehicle–trailer systems through handling and stability manoeuvres; however, their work focuses on heavier tractor–semi-trailer combinations. Their study employed multibody simulations using Adams/Car software. Charles *et al.* (2021) conducted a numerical investigation of turbulent airflow around a tractor–trailer combination, analysing three configurations with different drag-reduction devices.

Harlecki *et al.* (2022) developed a mathematical model of a car–trailer combination to analyse its dynamic properties under road traffic conditions. Their simulations examined system behaviour during typical driving manoeuvres, considering variations in design parameters and trailer load. The resulting model can be regarded as a virtual prototype of the system. Zhang *et al.* (2025b) proposed an innovative path-planning method for intelligent semi-trailers operating on straight road segments. Their approach dynamically adapts the planned trajectory to changing driving conditions, improving trackability by reducing steering angle, lateral acceleration, and yaw rate.

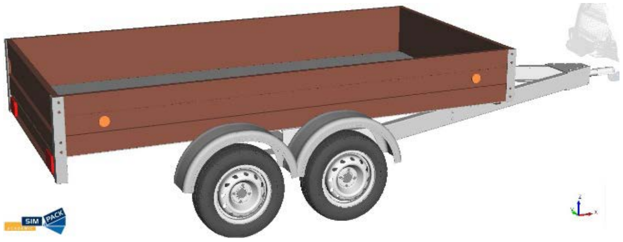
Ta *et al.* (2022) investigated rollover stability indicators of a semi-trailer vehicle during turning manoeuvres using multibody dynamics and Newton–Euler equations. Their work resulted in a comprehensive dynamic model incorporating adaptive sub-models, including suspension, nonlinear tyre behaviour, and fifth-wheel coupling. Wang *et al.* (2023) highlighted the increased complexity of car–trailer dynamics compared to non-articulated vehicles due to their articulated structure. They developed a five-degree-of-freedom model that includes damping characteristics of trailer shock absorbers. Their findings indicate that lower damping coefficients of the trailer shock absorbers reduce rearward amplification, thereby enhancing overall driving safety.

3. MATERIALS AND METHODS

The research is based on numerical modelling. A multibody model of a two-axle trailer was developed using the commercial simulation software Simpack. From a mathematical perspective, a complete analytical description of the system using equations of motion becomes highly complex for a full two-axle trailer model (Frej *et al.*, 2023; Sakhno *et al.*, 2025). Therefore, a simplified representation of the trailer dynamics is considered in this study. The developed multibody model of the two-axle trailer is shown in Fig. 3, without the applied load.

The total mass of the trailer is assumed to be 2,500 kg, with a payload capacity of 2,050 kg, resulting in a curb weight of 450 kg. In the simulations, the trailer is towed by a passenger car of the SUV category. As the main goal of this study is to investigate the driving behaviour of the trailer, detailed parameters of the towing vehicle are not explicitly considered. For reference, the SUV has a total mass of 3,500 kg, which provides relatively favourable stabilising conditions during the analysed manoeuvres. It should be noted that a lighter towing vehicle would likely lead to different simulation results, reflecting the changed dynamics of the car–trailer system.

Figure 3. A multibody model of the two-axle trailer in the Simpack software



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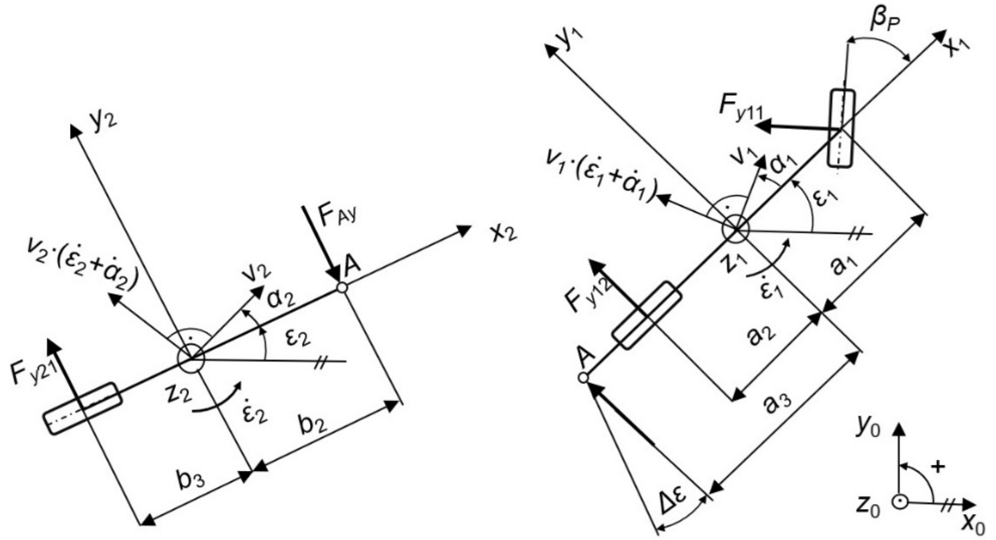
The simplified analytical model is based on a single-track representation of the car–trailer system. Although this approach is less conventional for describing the dynamics of car–trailer motion (Manrique *et al.*, 2022; Molnár and Kun, 2024; Sokil *et al.*, 2022), it provides a tractable framework for analysis. The corresponding calculation scheme is shown in Fig. 4. As illustrated, the model consists of three wheels, and the relevant forces acting during motion are indicated.

Both the towing vehicle and the trailer are modelled as rigid bodies. It is assumed that they move at the same longitudinal velocity v. Under this assumption, only the lateral force F_{Ay} acts at the tow hitch, while longitudinal interaction forces are neglected.

The simplified dynamic model has three degrees of freedom (DOF): the yaw rate of the car (denoted as $\dot{\varepsilon}_{1\,car}$), the sideslip (deviation) angle at the vehicle's centre of gravity (denoted as α_1), and the articulation (hitch) angle of the car–trailer combination (denoted as $\Delta\varepsilon$). The equations of motion, which form a system of second-order differential equations, are derived using either a free-body diagram approach or Lagrange's equations of the second kind.

Figure 4. A dynamical scheme of a single-track model of a car-trailer combination



Source: Vlk (2003)

Their form is as follows:

$$m \cdot \dot{\varepsilon}_1 \cdot v + m \cdot \dot{\alpha}_1 \cdot v + m_t \cdot \dot{\varepsilon}_2 \cdot v + m_t \cdot \dot{\alpha}_2 \cdot v = F_{y11} + F_{y12} + F_{y21} \tag{1}$$

$$m \cdot v \cdot \dot{\varepsilon}_1 \cdot a_3 + m \cdot v \cdot \dot{\alpha}_1 \cdot a_3 + I_{z1} \cdot \dot{\varepsilon}_1 = F_{y11} \cdot a_1 - F_{y12} \cdot a_2 + F_{y11} \cdot a_3 - F_{y12} \cdot a_3 \tag{2}$$

$$m_t \cdot v \cdot \dot{\varepsilon}_1 \cdot b_2 + m_t \cdot v \cdot \dot{\alpha}_1 \cdot b_2 + I_{z2} \cdot \dot{\varepsilon}_2 = F_{y21} \cdot b_3 + F_{y11} \cdot b_2 + F_{y12} \cdot b_2 \tag{3}$$

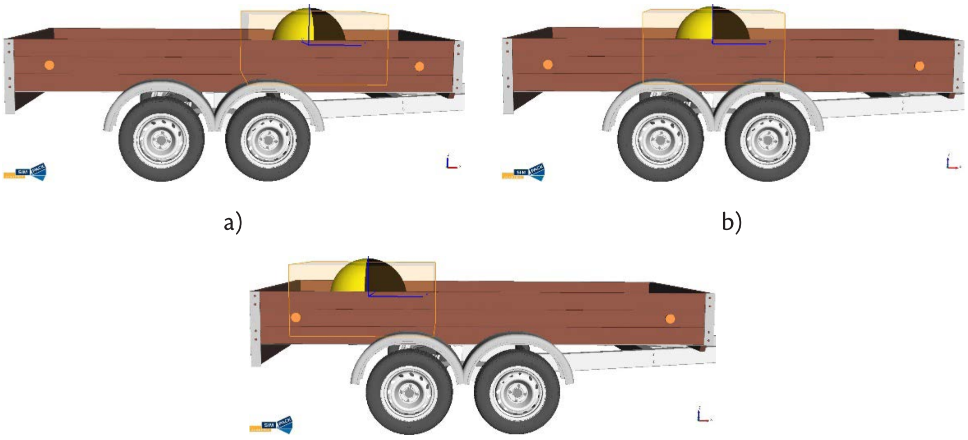
where m [kg] – mass of the car, m_t – mass of the trailer, v [m/s] – driving speed, I_{z1} [kg·m^2] – moment of inertia of the car, I_{z2} [kg·m^2] – moment of inertia of the trailer, $\dot{\varepsilon}_1$ [rad/s] – yaw angular speed of the car, $\dot{\varepsilon}_2$ – yaw angular speed of the trailer, $\dot{\alpha}_1$ [rad/s] – yaw angular speed deviation of the car, $\dot{\alpha}_2$[rad/s] – yaw angular speed deviation of the trailer, F_{y11}, F_{y12} [N] – the lateral forces in the tyre/road contact of the car for the front and rear wheel, respectively, F_{y21} [N] – the lateral force in the tyre/road contact of the trailer, a_1, a_2, a_3 [m] – dimensions of the car obvious from Fig. 4, b_1, b_2, b_3 [m] – dimensions of the trailer obvious from Fig. 4.

In the derived mathematical model, Eq. (1) represents the equilibrium of lateral forces acting on the car–trailer combination, Eq. (2) describes the moment equilibrium of the towing vehicle, and Eq. (3) expresses the moment equilibrium of the trailer.

Figure 4 also defines the coordinate systems used in the analysis: the longitudinal (x_1), lateral (y_1), and vertical (z_1) axes of the car, and the longitudinal (x_2), lateral (y_2), and vertical (z_2) axes of the trailer. The angle β_P [rad] denotes the steering angle of the front wheels of the car, while the parameter $\Delta\varepsilon$ [rad] represents the articulation angle, i.e., the deviation of the trailer drawbar from the longitudinal axis.

In the simulation study, the two-axle trailer was analysed under two loading conditions: 20% and 80% of its maximum payload capacity. In addition, three longitudinal positions of the load were considered, as described previously. These load configurations are illustrated in Fig. 5, and a detailed overview of the load cases and their respective positions on the trailer platform is provided in Table 1.

Figure 5. Positions of the load on the two-axle trailer: a) Position A, b) Position B, c) Position C



a) b)

c)

Source: Authors

Table 1. The load cases and the load positions of the two-axle trailer

Designation	Position A		Position B		Position C	
	The load in front of the 1st axle		**Between axles**		**Behind the 2nd axle**	
Load-80	Percentage of the payload	kg	Percentage of the payload	kg	Percentage of the payload	kg
	80%	1,640	80%	1,640		1,640
Load-20	Percentage of the payload	kg	Percentage of the payload	kg	Percentage of the payload	kg
	20%	410	20%	410	20%	410

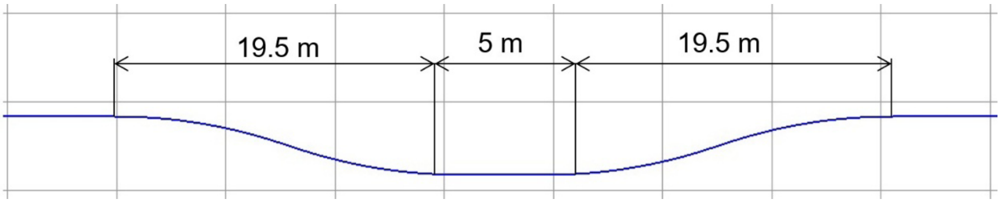
Source: Authors

The results of the simulation computations are presented in the following section in graphs. The individual results are labelled in accordance with Table 1.

4. RESULTS AND DISCUSSION

This section presents the results of the simulation analyses performed using the multibody model of the two-axle trailer. A series of simulations was conducted for different driving speeds and defined load cases (Table 1). The analysed driving manoeuvre corresponds to a vehicle passing over a road obstacle, in accordance with relevant standards (ISO 3888-2, 2011; ISO 4138, 2021). The road profile implemented in the simulation software was defined based on the geometry shown in Fig. 6.

Figure 6. A multibody model of the two-axle trailer in the Simpack software



Source: Authors

The driving behaviour of the two-axle trailer (and, in principle, the car–trailer combination) is evaluated according to the following methodology:

- The Load-80 case was defined for Position C in the multibody system (MBS) model
- The maximum driving speed v_{max} for this loading condition was determined. This critical speed was identified as the point at which the vertical wheel–road contact force reached zero.
- The Load-80 case was subsequently applied to Position A and Position B, and the corresponding maximum driving speed v_{max} was determined. The vertical wheel–road contact forces were evaluated and compared with those obtained for Position C.
- The Load-20 case was then implemented in the MBS model, and the same simulation procedure was repeated as for Load-80, with a primary focus on Position C. The maximum driving speed was identified, followed by a comparison of the vertical wheel–road contact forces for Position A and Position B.
- Finally, graphical outputs were generated, which are presented and discussed in the following section.

The first load case was analysed for the Load-80 configuration. A series of simulations was performed to determine the maximum driving speed at which the trailer can safely pass over the road obstacle. The safety criterion was defined as the condition in which the vertical wheel–road contact force becomes zero. A zero vertical wheel–road force indicates the loss of contact between the wheel and the road surface, meaning that no contact forces can be transmitted. This behaviour is represented by the red curve in Fig. 7 and is observed for Position C under the Load-80 condition. As the trailer traverses the obstacle in this configuration, the system exhibits dynamic instability, leading to unsafe and potentially uncontrollable motion of the car–trailer combination. For Load-80 applied at Position B (i.e., between the axles, Fig. 5b), the vertical wheel–road contact force remains non-zero (orange curve in Fig. 7), indicating that the trailer maintains contact with the road and passes the obstacle safely. An even higher level of stability is observed for Position A, where the vertical wheel–road contact force further increases, resulting in safe traversal of the manoeuvre. The simulation results show that the maximum safe driving speed of the car–trailer combination under the defined conditions is 59.73 km/h.

Figure 8 presents the results of the simulation for the Load-20 condition. The same procedure was applied as in the case of Load-80. The maximum driving speed was identified as 66.85 km/h. At this speed, the stability of the trailer during the obstacle-avoidance manoeuvre begins to be compromised for Load-20 in Position C. The vertical wheel–road contact force reaches zero (red curve), indicating loss of contact with the road surface. As a result, no contact forces can be transmitted, and the trailer enters an unstable and uncontrolled motion state. An interesting observation is that, in contrast to the Load-80 case, shifting the load to Position A or Position B does not lead to such a significant improvement in driving safety (orange and green curves in Fig. 8). The difference lies in the timing of instability onset: for Load-20, Positions A and B become critical at the entry phase of the obstacle manoeuvre rather than at its end. This indicates that, for a lighter trailer load, the longitudinal position of the payload has a less pronounced influence on overall stability during the analysed manoeuvre compared to the higher load condition.

Figure 7. The results of the simulation computations for the Load-80

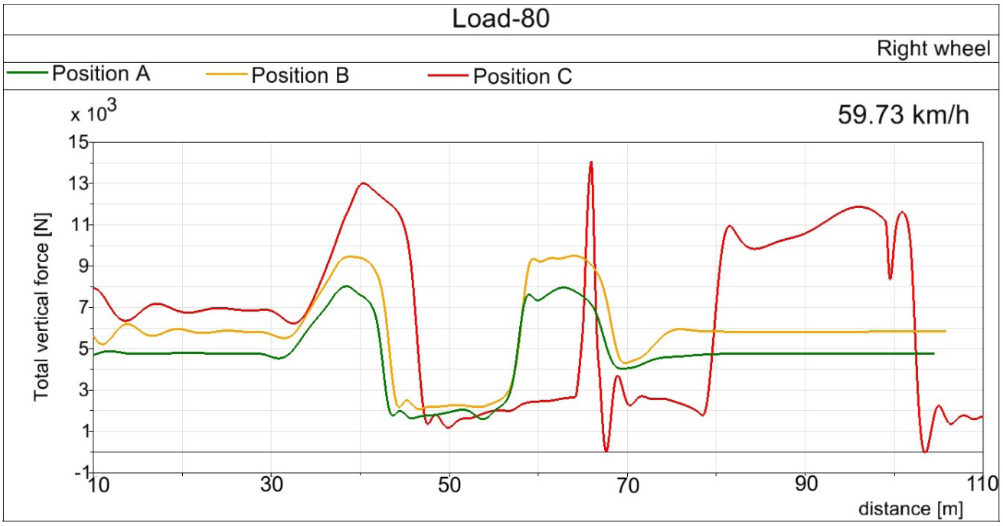


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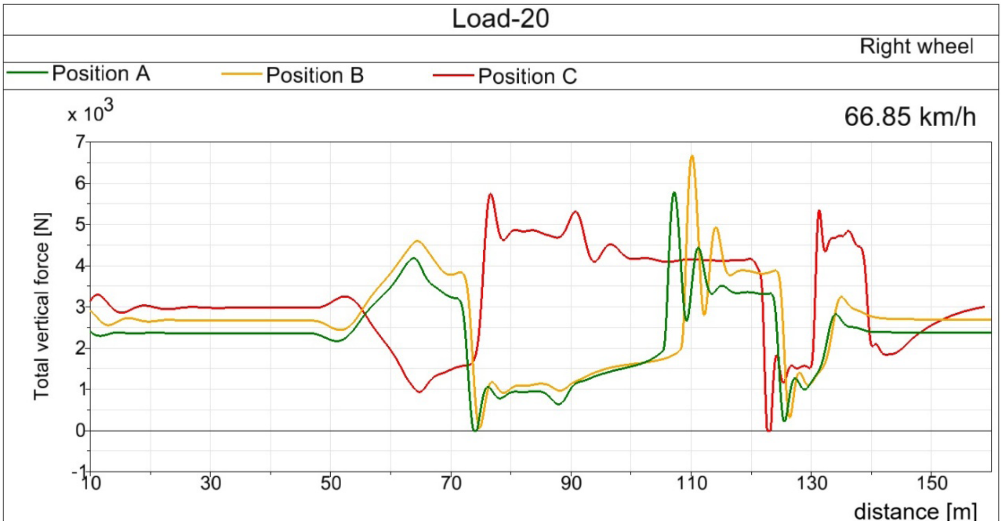
Figure 8. The results of the simulation computations for the Load-20



Source: Authors

The present research confirms the findings of previous studies (Jagelčák and Kubanová, 2024; Zerbato *et al*., 2022; Zhang *et al*., 2020; Zhang *et al*., 2016), which focus on mathematical modelling of vehicle–trailer combinations, their numerical solution, and the resulting dynamic behaviour. These results show good agreement with the outcomes of the present numerical simulations. These studies similarly indicate that an appropriate longitudinal position of the load on the trailer platform plays a significant role in improving system stability. In particular, they conclude that a loaded towbar contributes positively to the longitudinal stability of the car–trailer combination. Conversely, positioning the load behind the rear axle of the trailer increases the risk of unstable motion of the trailer, and in extreme cases, of the entire vehicle–trailer combination.

Future research in this field will focus on a more comprehensive assessment of the driving behaviour of car–trailer combinations under a wider range of driving scenarios. As previously introduced, three specific load positions were considered in this study; however, in practice, the load may be placed at positions located both closer to and further from the front and rear axles of the trailer. Such configurations may lead to more unfavourable dynamic behaviour. It is expected that positioning the load closer to the towing hitch may result in overloading of the rear axle of the towing vehicle (Oh and Choi, 2024). Conversely, placing the load further behind the rear axle of the trailer may significantly accelerate the onset of unstable behaviour of the car–trailer combination. In addition, the vertical position of the load centre of gravity also plays an important role in driving stability. A higher centre of gravity may increase the risk of trailer rollover, particularly during cornering or obstacle-avoidance manoeuvres, potentially leading to serious road accidents. Future studies should also examine the influence of the mass ratio between the towing vehicle and the trailer on system stability, as it is expected to significantly influence the dynamic response of the car–trailer combination.

5. CONCLUSIONS

Passenger cars are widely used for individual transport worldwide. The use of trailers significantly increases their load-carrying capacity; however, a car–trailer combination exhibits substantially different dynamic behaviour compared to a single passenger vehicle. For this reason, it is necessary to investigate its driving properties in order to identify potentially hazardous situations and to define safe operating conditions.

As shown in this study, one of the most influential factors affecting the dynamic behaviour of a car–trailer combination is the longitudinal position of the load on the trailer platform. The research focused on driving safety during a specific manoeuvre, namely the passage over a road obstacle. Two loading conditions were considered, corresponding to 80% and 20% of the trailer's payload capacity.

The maximum safe driving speed was determined for the most critical configuration, i.e., when the load was positioned behind the rear axle of the two-axle trailer. This case was subsequently compared with configurations where the load was positioned between the axles and in front of the first axle.

The results indicate that the driving safety of the car–trailer combination is more sensitive to load position when the trailer is highly loaded. In contrast, for lower load levels or an unloaded trailer, the maximum safe driving speed is only slightly affected by the longitudinal position of the load on the trailer platform.

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NOTE

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PROCJENA UTJECAJA POLOŽAJA TERETA NA DVOOSOVINSKOJ PRIKOLICI NA SIGURNOST VOŽNJE PRIMJENOM RAČUNALNIH SIMULACIJA

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SAŽETAK

Kombinacije osobnog i priključnog vozila relativno su popularne u mnogim zemljama. Prikolica je transportno sredstvo koje povećava nosivost automobila. Povezivanje prikolice s vučnim vozilom utječe na vozna svojstva kombinacije automobil-prikolica. Položaj tereta na nadogradnji prikolice jedan je od najvažnijih čimbenika koji utječu na vozna svojstva takve kombinacije. Cilj je ovog istraživanja procijeniti vozna svojstva kombinacije automobila i prikolice s gledišta sigurnosti vožnje. Istraživanje je provedeno pomoću računalnih simulacija u komercijalnom softveru za simulaciju sustava s više tijela (multibody simulation software)*. Sigurnost vožnje ispitana je kroz scenarij u kojem kombinacija automobil-prikolica prelazi preko prepreke na kolniku pri različitim brzinama vožnje. Time se utvrđuje položaj tereta pri kojem kombinacija vozila i prikolice postaje nestabilna, što može dovesti do prometne nesreće. Opterećenje prikolice odabrano je u vrijednostima od 20 % i 80 % ukupne nosivosti, uz tri položaja težišta tereta u uzdužnom smjeru: ispred prve osovine, između osovina te iza druge osovine prikolice. Utvrđeno je da se najniža razina sigurnosti vožnje postiže kada kombinacija automobil-prikolica prelazi preko prepreke na cesti pri velikoj brzini uz teret postavljen iza druge osovine prikolice. U konačnici, na temelju rezultata istraživanja predložen je optimalan položaj tereta.*

Ključne riječi: *dvoosovinska prikolica, simulacijski proračuni, sigurnost vožnje,* multibody model