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STRUCTURAL ANALYSIS OF TRUCK TRAILER FRAMES WITH TWO CENTRAL AXLES DESIGNED FOR TRANSPORT OF SWAP BODIES

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Abstract

This article presents a method for strength calculations of truck trailer frames with two central axles designed for the transport of swap bodies. The finite element method formalism was used to carry out the calculations. In the design of truck trailer frames, the commonly applied static finite element method does not reflect the actual operating conditions of the structure, as it does not account for dynamic loads occurring during road manoeuvres. Therefore, the authors decided to employ a calculation approach based on an analysis of the dynamics of the truck–trailer vehicle combination using the MSC Adams and NX Nastran/Femap interface. The Craig–Bampton method was used to perform the computations. For the purposes of the analysis, a diagram of the pneumatic suspension system of the trailer was also developed. The dynamic characteristics of the air springs were determined on the basis of a polytropic process resulting from changes in their volume during operation. Sample results of reduced stress calculations in the trailer frame during three selected road tests are presented, namely braking, a U-turn and a series of two S-turns. In the authors' opinion, the method could be utilised in the design process of the trailers under analysis.

Keywords: swap body trailer; trailer frame strength; finite element method; MSC Adams – NX Nastran interface; Craig–Bampton method

1. Introduction

Swap bodies, which revolutionised the transport of general cargo, were introduced in Germany at the turn of the 1960s and 1970s. The German transport association Bundesverband des Deutschen Güterfernverkehrs (BDF) standardised the technical solutions for this

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type of body emerging in transport companies – hence the term "BDF swap bodies" is some-times used.

A vehicle combination consisting of a truck and a trailer designed for transporting swap bodies (Figure 1), unlike a truck and a trailer with a fixed body, allows the body to be easily detached from the chassis on which it is transported.

These bodies can also be used as storage space, and the truck–trailer vehicle combination – which is to transport them – can carry out other transport tasks whilst goods are being loaded into them.

Swap bodies are equipped with so-called support legs, which allow them to be positioned at a specific height, enabling the chassis of the truck and trailer to drive underneath them. These chassis are then lifted together with the swap bodies using their air suspension, allowing the support legs to be retracted. In the final stage of loading, each body must be secured ("locked") at four locations to the chassis of the truck or trailer. The main dimensions of swap bodies are specified in standard EN 284:2006 [1].

Fig. 1. View of swap bodies during loading onto truck and trailer

Curtain-sided bodies are a common type of swap body. Such bodies (Figure 2) usually consist of a floor frame with the floor, a front wall, rear doors, a roof (which can be slid forwards or backwards along roof-mounted rails and can also be lifted during loading) and side walls in the form of a tarpaulin, which can be slid forwards or backwards along rails forming part of the roof. Due to their design, these bodies are typically characterised by low flexural and torsional stiffness.

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Fig. 2. View of curtain-sided swap body

Truck trailers used to transport swap bodies are, in many cases, trailers with two central axles positioned halfway along their length (Figures 1 and 2).

This article will focus on trailers of this type used to transport curtain-sided swap bodies.

The aim of this article is to present a method for strength calculations of frames for this type of trailer using the Finite Element Method (FEM) formalism.

2. Essence of MSC Adams and NX Nastran/Femap interface

In the design of truck trailers, static finite element method analysis is still generally used today. However, this does not accurately reflect the "behaviour" of these vehicles' components – particularly their frames – under dynamic road traffic conditions. The use of static FEM analysis does not allow for the identification of locations within the frame where elevated stress levels may occur as a result of vibrations of the frame itself or in components attached to it. Therefore, in the modern design process for truck trailers, and in particular their frames, it is essential to take dynamic phenomena into account. However, this is not possible without appropriate computer tools that allow for advanced dynamic analysis. Calculations limited to statics, including the frequently used calculations employing so-called coefficients of mass forces (elementary inertial forces distributed throughout the volume of the modelled body), mean that the actual "behaviour" of the frame under dynamic conditions will not be accurately reflected. In such cases, the designer's experience is crucial, as it enables the identification of critical areas where dynamic phenomena may have a significant impact on the strength of the designed structure. However, relying solely on this experience may lead

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to the structure being "over-engineered" or its strength being "overestimated", and consequently necessitate ongoing repairs to the frame in vehicles already in service. The difficulties in determining the impact of dynamic phenomena on the strength of the trailer frame mean that a structure designed with insufficient precision is subjected only to minor modifications for many years, which consequently limits product development.

Given the limitations of static FEM analysis outlined above, the present study employs an approach based on analysing the dynamics of the vehicle combination under consideration using the interface between the advanced software packages MSC Adams (for multibody dynamics analysis) and NX Nastran/Femap (for FEM based strength analysis). This approach may be referred to as a dynamic finite element method.

As stated in monograph [2], the motion of a flexible member can be considered as the motion of that member treated as rigid, onto which its deformations resulting from natural vibrations are superimposed. This approach, referred to in the literature as the Floating Frame of Reference System, is used in the modelling of flexible members undergoing large reference frame motion using FEM. The frame of the trailer under analysis, moving under road traffic conditions, is precisely such a member. The MSC Adams and NX Nastran/Femap interface is an advanced tool that allows large reference frame motion of modelled members to be taken into account during the design process. Typical FEM programmes offering so-called "transient" analysis do indeed allow for dynamic loads on components, but their large reference-frame motion is not taken into account here. It should be borne in mind that the number of degrees of freedom in FEM models for large systems can reach as many as several million. To make the computational process feasible, a special procedure must be applied, described in detail also in the PhD thesis [3] and in the article [4], which allows for a significant reduction in the number of degrees of freedom of the FEM model under consideration. This is referred to in the literature as the Modal Reduction Technique. As part of this procedure, for the interface under consideration, the Craig-Bampton method [5] is used, which belongs to a group of methods referred to in the literature as Component Mode Synthesis – CMS. Before proceeding to create the FEM model of a given component, its CAD model must be prepared (for this analysis, the SolidWorks and Onshape environments were used for this purpose).

Figure 3 illustrates the transfer of information between individual computational environments, carried out via the interface between the MSC Adams and NX Nastran/Femap packages. Geometric models of the components of the vehicle combination under analysis, developed using the commercial SolidWorks (although another CAD programme could also be used), are exported to a universal STEP file, after which they can be imported into the MSC Adams or Femap environment. In this environment (in the pre-processor), an FEM model of the trailer frame is created and its modal analysis is performed using the NX Nastran solver. During this analysis, a *.mnf file is created, containing, amongst other things, information on the frequencies and modes of natural vibration of the FEM model. This information

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is transferred to the MSC Adams environment, and in particular to its MSC Adams/View module, where the model of the analysed vehicle combination is built. The next stage involves simulating the motion of the vehicle combination using the MSC Adams/Solver module, taking into account a specific method of applying external forces. The results obtained, in the form of time-varying deformations of the modelled member (which may be the trailer frame), are exported to the NX Nastran/Femap environment, where calculations are performed using the NX Nastran processor, concerning, for example, the determination of reduced stress contours or deformations. The results of this analysis are presented using the Femap post-processor, whilst the results of the dynamic analysis carried out in the MSC Adams environment are presented using the MSC Adams/Postprocessor module.

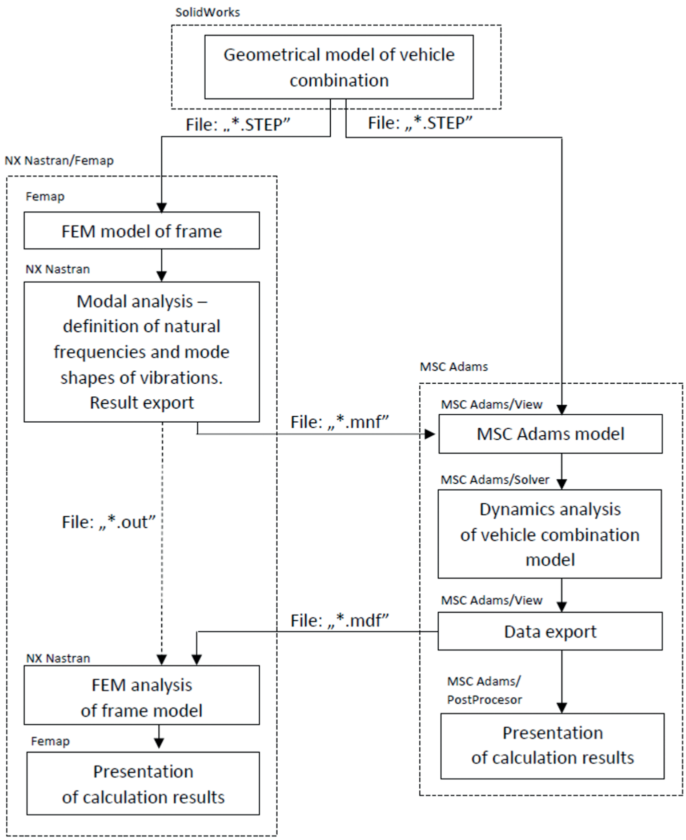

Fig. 3. Transfer of information between individual computational environments

The ability to utilise the illustrated information transfer in the described structural analysis method constitutes the essence of this method.

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3. CAD model of truck trailer designed for transport of swap bodies

Figure 4 shows a view of the analysed trailer with two central axles designed for the transport of swap bodies. The permissible total mass of the trailer is 18000 kg. The applied pneumatic suspension ensures compliance with the requirements for equal load distribution between the trailer axles.

Fig. 4. View of trailer under analysis

Figure 5 shows the developed CAD model of the trailer under analysis. In addition to the frame, axles and drawbar, this model also included the pneumatic suspension and braking systems. A model prepared in this way can be expanded to include further components or simplified – depending on the specific calculation scenario under consideration.

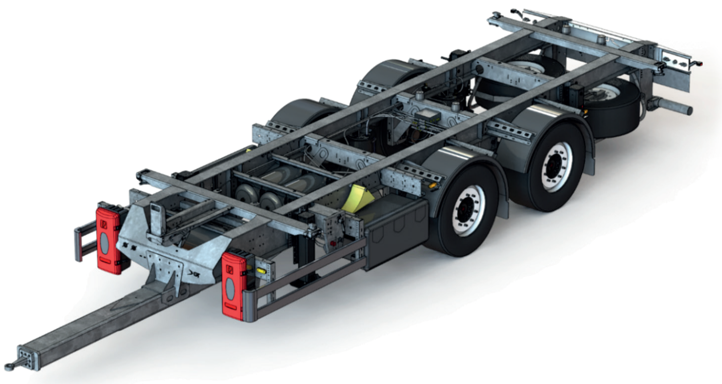

Fig. 5. CAD model of trailer

Figure 6 shows the CAD model of the trailer frame, which consists of two parallel side members made of welded steel plates in the form of an I-beam. The side members are connected by cross members in the form of a channel section and locking beams, which are used to secure the swap body. At the front of the frame there is a front plate connecting the side members.

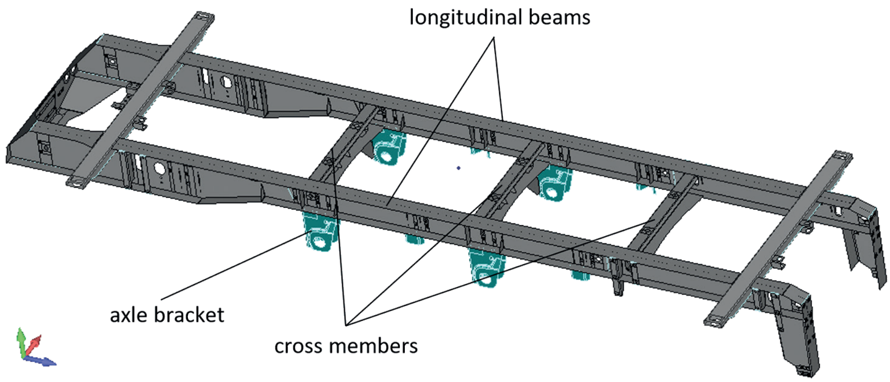

Fig. 6. CAD model of trailer frame

Figure 7 shows a CAD model of the trailer axle, which is constructed from a circular-section tube, at the ends of which are wheel hubs, brake discs and callipers. Two longitudinal control arms, which form part of the trailer's suspension, are permanently attached to the axle; air springs, acting as spring elements, are located on their extensions. The control arms are connected to brackets welded to the trailer frame. Shock absorbers, which act as vibration dampers, are located between the axle and its brackets.

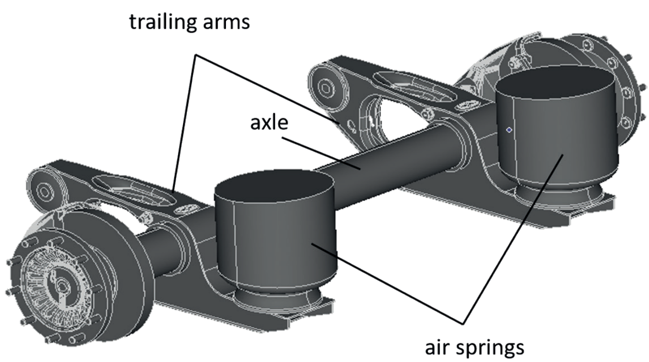

Fig. 7. CAD model of trailer axle

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The main component of the trailer drawbar is a rectangular-section tube with appropriate reinforcements, drawbar eye mounting elements and elements mounting the drawbar to the trailer frame. The drawbar is one of the most important components of the trailer – its stiffness and method of attachment can influence the stress values in the frame. The drawbar, included in the trailer's CAD model, has been type-approved in accordance with the procedure described in Regulation [6] – consequently, the determination of the stresses occurring within it was not the subject of the present analysis.

An important consideration when creating the trailer's CAD model was the correct modelling of its load. To this end, a CAD model of the curtain-sided swap body frame, including the plywood floor (Figure 8), was prepared. This served as the basis for attaching models of the remaining parts of the body, such as the corner posts.

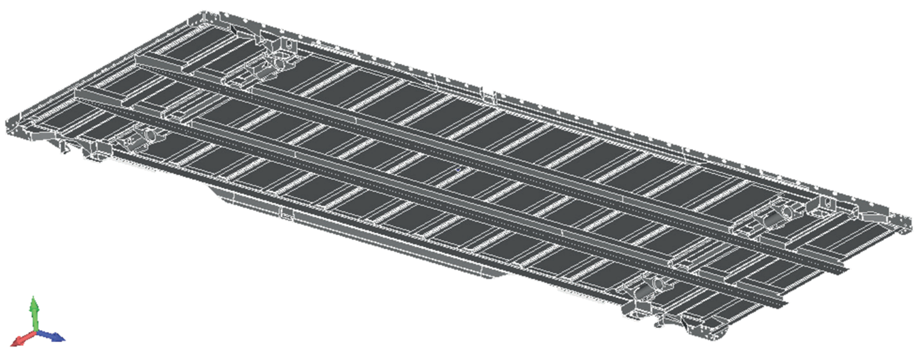

Fig. 8. CAD model of curtain-sided swap body frame together with floor

In addition to the CAD model of the frame, a CAD model of this frame loaded with cargo in the form of a set of cuboid blocks placed on pallets was also developed.

4. FEM model of truck trailer frame designed for transport of swap bodies

Publications on the strength analysis of heavy vehicle frames, i.e., trucks and trailers, are primarily focused on static FEM analysis. An exception is the study [7]. Its authors – as they already described in their earlier paper [8] – initially developed a simplified mathematical model of a truck with six degrees of freedom, which could be treated as a type of multi-body system. Subsequently, according to this description, using Lagrange's equations, they analysed the vehicle's dynamics, determining the values of reaction forces acting from the wheels on the vehicle frame. In the subsequent stage – described in [7] – based on a procedure known as the "Inertial Relief Technique", they carried out a FEM analysis of the vehicle frame loaded with forces with the values thus determined. This analysis provided more accurate results than the classical static FEM analysis.

Among the publications mentioned later in this section that use static FEM analysis, one may cite the study [9], in which four loading cases of a truck frame were considered – namely, cases of its bending or torsion, as well as cases of its loading during steering or braking manoeuvres. The authors of the study performed a modal analysis of the frame model and subsequently optimized its design, which allowed for a reduction in its mass. Publication [10] presents a comparison of the strength of truck frames made of mild steel or titanium alloy, demonstrating that the frame made of titanium alloy exhibits higher strength properties, whereas the frame made of mild steel shows less tendency to bend. The authors of study [11] analysed the so-called stress concentration factor (SCF) in the welded joints of truck frame elements, demonstrating that an increase in the fillet radius of the weld from 0 mm to 10 mm reduces this factor by 30%, which is significant for the fatigue life of the structure. In study [12], the authors presented a FEM analysis of two types of modular frames for a high-power specialized firefighting vehicle equipped with an all-wheel drive system, used for intervention and firefighting at civil and military airports, belonging to the Aircraft Rescue and Fire Fighting (ARFF) vehicle group. The analysis was conducted strictly to meet the design process requirements of the manufacturer of such vehicles.

For this study, quadrilateral shell elements, mainly of the "CQUAD8" type, were used to construct the FEM model of the frame of the trailer under analysis (Figure 9).

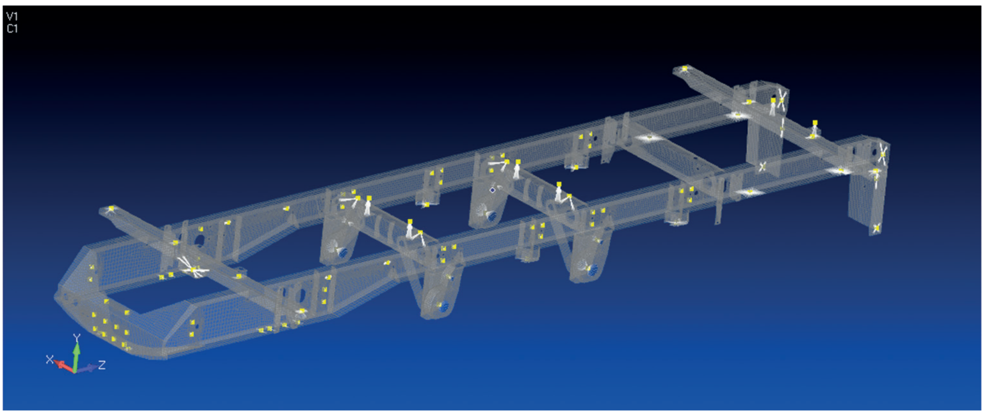

Fig. 9. FEM model of trailer frame designed for transport of swap bodies

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The FEM model of the trailer axle was created using shell elements of both quadrilateral types “CQUAD8” and “CQUAD4”, as well as triangular types “CTRIA6” and “CTRIA3”. The FEM model of the drawbar was created using “CQUAD8” and “CTRIA6” elements, as well as “CTETRA” solid elements. The FEM model of the curtain-sided swap body (Figure 10), created on the basis of the CAD model of the body frame together with the floor (Figure 8), was constructed predominantly from “CQUAD8” elements, with “CQUAD4” and “CTRIA3” elements used in selected regions. The plywood forming the body floor was modelled using elements referred to in Femap as “laminate plate” and in NX Nastran as “CQUAD8” with the assigned material type designated as “MAT8”. In addition, “CBEAM” beam elements were used in the model. Based on the CAD model of the load, an FEM model was created consisting of “CHEXA” solid elements.

Rigid body elements (RBEs) were introduced into the FEM models, connecting selected FEM mesh nodes with interface nodes, at which, within the MSC Adams environment, these models are coupled with one another or with other components.

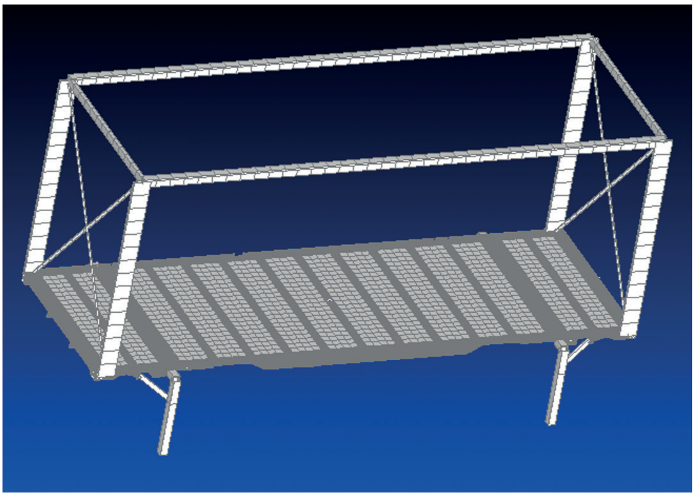

Fig. 10. FEM model of curtain-sided swap body

During the calculations, the discretisation density was gradually increased in search of its optimal value. The correctness of the FEM model discretisation was also assessed by verifying the values of the so-called aspect ratio of individual finite elements and the values of their deviation angle from the optimal shape.

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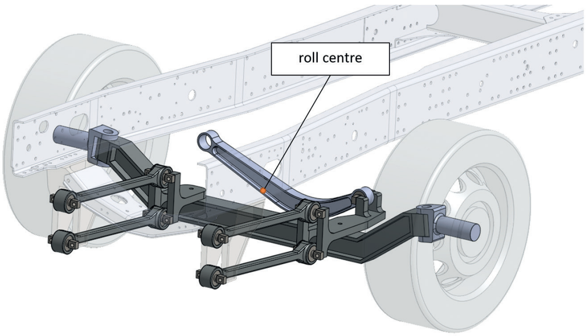
5. MSC Adams models

5.1. MSC Adams model of truck towing trailer designed for transport of swap bodies

In order to create, within the MSC Adams environment, a model of a truck towing a trailer designed for transporting swap bodies (hereinafter referred to as the MSC Adams truck model), the CAD model of this vehicle was imported into the environment. The truck's suspension was modelled in a simplified manner, allowing for only two motions of the truck's axles relative to the truck frame, namely translational motion and rotational motion about the longitudinal axis passing through the so-called centre of lateral roll of the given truck axle [13]. The positions of these centres for the individual axles of the truck are shown in Figure 11.

a



b

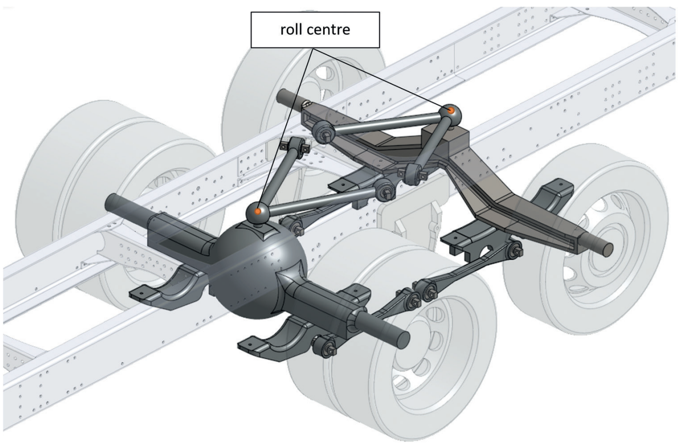


Fig. 11. Position of centres of lateral roll in case of: (a) the first (front) axle of truck, (b) second and third axles of truck

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5.2. MSC Adams model of trailer designed for transport of swap bodies

The MSC Adams model of a trailer designed for the transport of swap bodies, with a curtain-sided body loaded with cargo, is shown in Figure 12.

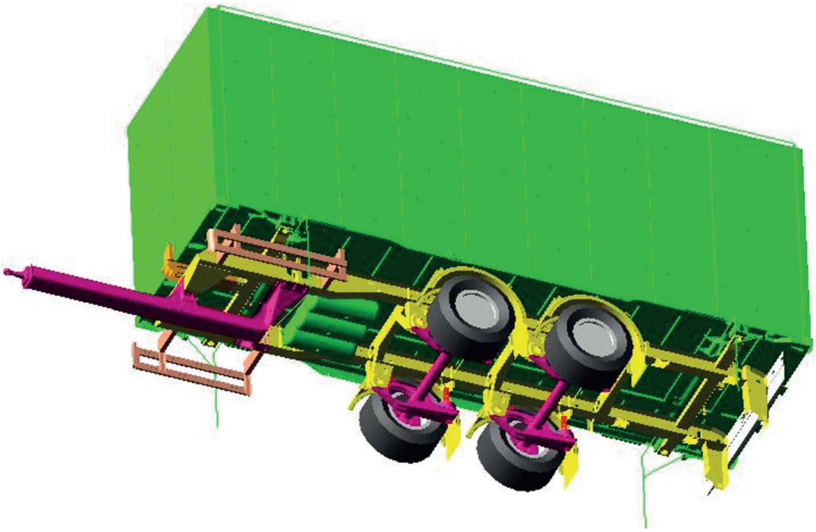

Fig. 12. MSC Adams model of trailer under analysis

After modifying the characteristics of the suspension springs, it was also possible to simulate the motion of the trailer without the swap body (Figure 13).

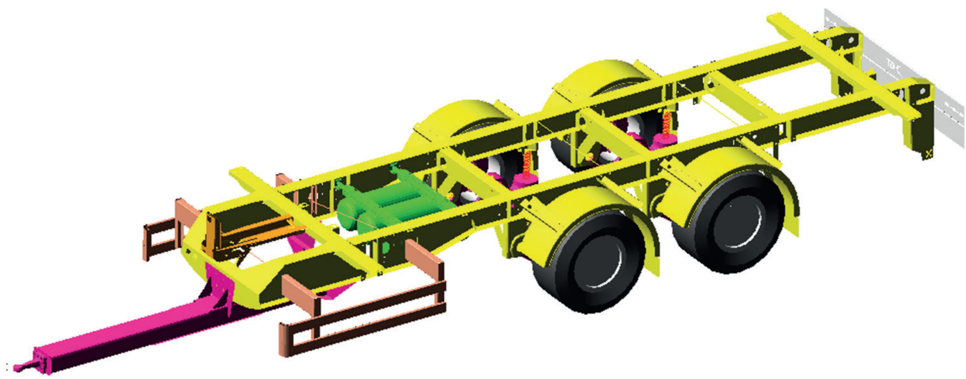

Fig. 13. MSC Adams model of analysed trailer without swap body

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The MSC Adams model of the trailer under analysis took into account the characteristic of the shock absorbers (Figure 14) forming part of its suspension. This was determined as part of the PhD thesis [3], based on data contained in the catalogue of the manufacturer of truck trailer suspensions – SAF-Holland [14].

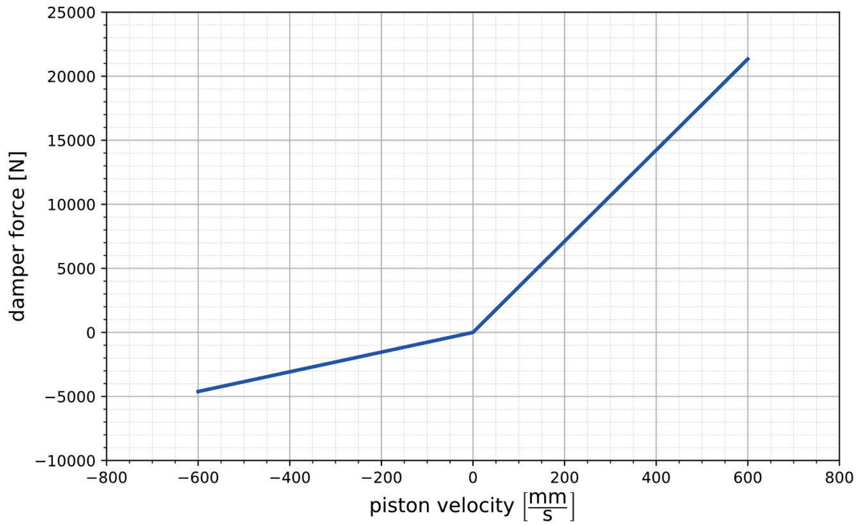

Fig. 14. Shock absorber characteristics

It is worth noting that the damping coefficient of shock absorbers has different values in their extension and compression phases.

6. Determination of dynamic characteristics of trailer air suspension air springs

6.1. Developed diagram of pneumatic system

Figure 15 shows the developed diagram of the pneumatic suspension system of the analysed trailer.

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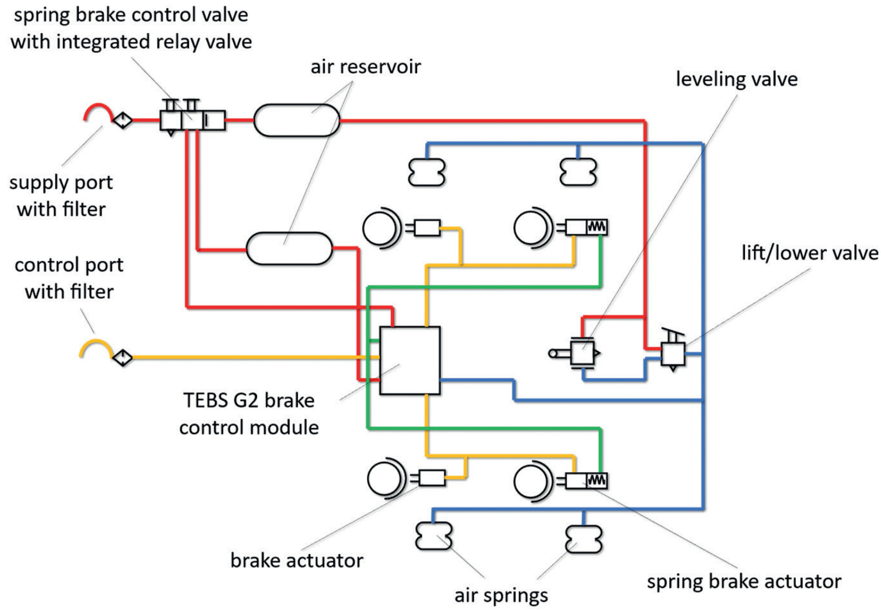

Fig. 15. Diagram of pneumatic suspension system of trailer under analysis

When the trailer is in motion – that is, when the lift/lower valve is set to the "driving" position – air is supplied to the air springs via the levelling valve, which maintains a constant trailer height regardless of its load. It is important to note that there is only one levelling valve in the entire trailer. Air from this valve is directed simultaneously to all the air springs. Their interconnection ensures that, in a static state (i.e. when the trailer is stationary), the pressure in all the air springs is equal, regardless of whether a particular air spring is compressed or extended. Furthermore, air can flow between the individual air springs, and its flow rate is limited by internal resistance in the pneumatic system components, particularly in the levelling valve and the pipes and connectors linking the air springs. The restriction of air flow within the levelling valve means that during rapid, short-term changes in the position of the trailer's axle relative to its frame (i.e. whilst the trailer is in motion), the volume of air filling the air springs does not change significantly. The described method of interconnecting the pneumatic components of the suspension system has certain significant implications for the trailer model: 1. the dynamic characteristics of the suspension's spring elements depend on the amount of air in the air spring – therefore, separate characteristics must be determined for each load case under analysis; 2. given that the static pressure values in all air springs are equal, it is necessary to check, prior to commencing the actual simulation, whether there is a difference in the values of the forces acting on individual spring elements in the model – for example, due to the trailer tilting.

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6.2 Determination of dynamic characteristics

At the outset of the procedure, the relationship between the trailer axle load and the pressure in the air spring was determined. For this purpose, the formula provided by the manufacturers of truck suspension systems and trailer axles – SAF-Holand [14] and BPW [15] – was used to determine the static pressure in the air spring:

$$p_s(Q') = \frac{(Q' - A)\ i\ p^*}{2}, \tag{1}$$

where:
Q' – the "load" on a given axle, expressed as the portion of the trailer's mass carried by that axle,
A – unsprung mass, i.e. the mass of the trailer components located below the suspension's spring elements (in this case, air springs) which do not exert a force on them, comprising mainly wheels, hubs, brake discs, callipers with brake cylinders and axles; in BPW's materials [15], it is suggested that this value should be equal to 0.08 of the permissible axle "load",
$p*$ – the pressure conversion factor in the air spring per unit of mass; according to data from the manufacturer of the suspension used in the trailer under analysis, namely SAF-Holland [14], for the air spring used, this value is $p^* = 0.00187\,\frac{\mathrm{Pa}}{\mathrm{kg}}$,
i – the so-called suspension "ratio", which, based on Figure 16, can be determined as:

$$i = \frac{L_1}{L_1 + L_2}. \tag{2}$$

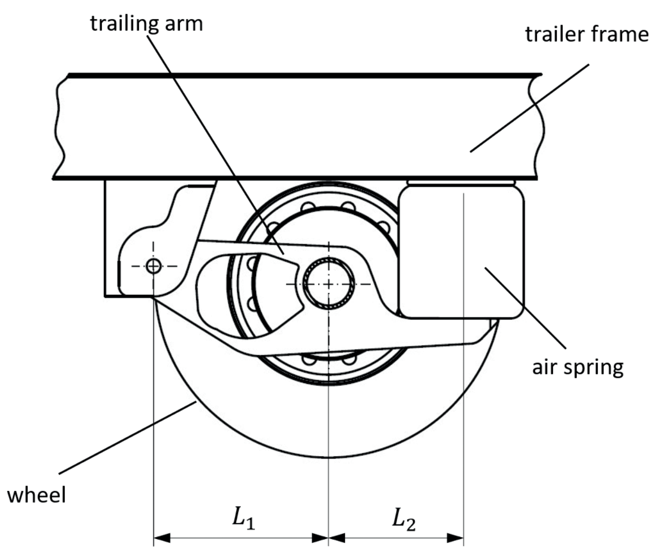

Fig. 16. Main suspension dimensions

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It is assumed that whilst the trailer is in motion, as a result of changes in the height of the air spring Δh (Figure 17), the amount of air filling it remains constant, and its volume changes in accordance with the following formula:

$$V(\Delta h) = \pi \frac{d_z^2}{4}(h_0 + \Delta h) + \left(\pi \frac{d_z^2}{4} - \pi \frac{d_w^2}{4}\right)\left(\frac{h_{max} - h_0 - \Delta h}{2}\right), \tag{3}$$

where:
d_z – outer diameter of the air spring,
d_w – piston diameter,
h_0 – the height of the air spring in a static state,
h_{max} – maximum air spring height,
Δh – change (increase) in air spring height.

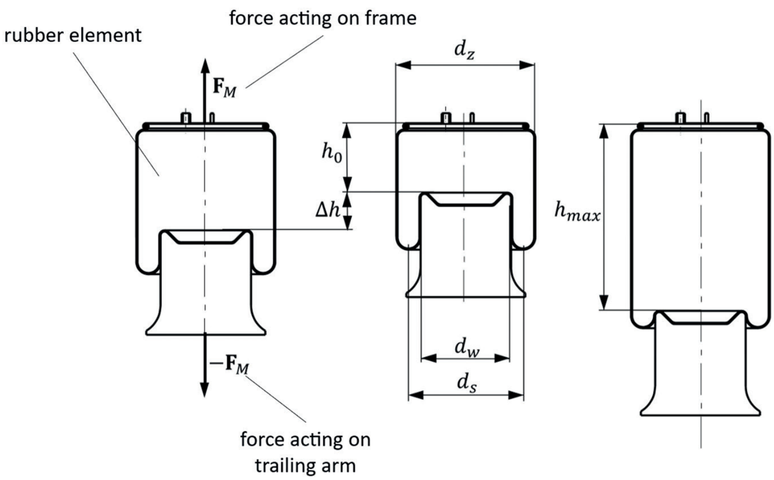

Fig. 17. Diagram of air spring under analysis

The change in pressure inside the air spring, resulting from the change in its volume, was determined on the basis of the ideal gas law. In accordance with the recommendations of Firestone Industrial Products Company [16], in the case of rapid suspension motions, assuming an adiabatic process may be acceptable; however, in typical cases, more accurate results are obtained by taking into account a polytropic process with an exponent of 1.38. The authors of this article have proposed a way to express the varying pressure inside the air spring (dynamic pressure) – dependent on the static pressure p_s (Q') and the change in air spring volume $V(\Delta h)$ – using the following formula:

$$p(Q', \Delta h) = p_s(Q')\left[\frac{V(h_0)}{V(\Delta h)}\right]^{1.38}. \tag{4}$$

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Therefore, the value of the forces exerted by the air spring on the frame and the swing arm can be expressed by the formula:

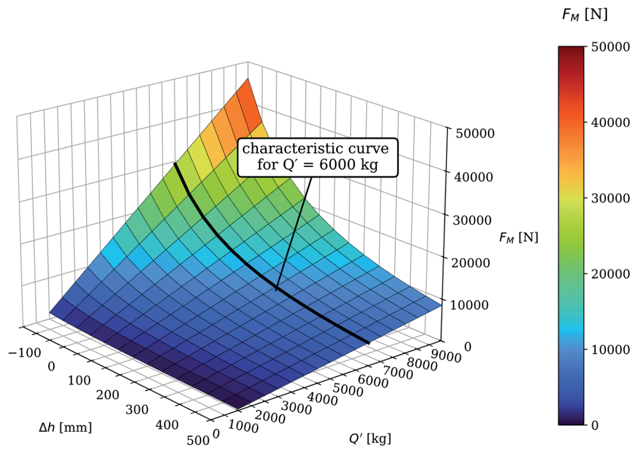
$$F_M(Q', \Delta h) = p(Q', \Delta h)\pi \frac{{d_s}^2}{4}, \tag{5}$$

where the diameter d_s [Figure 17] is defined as:

$$d_s = \frac{d_z + d_w}{2}. \tag{6}$$

Figure 18a shows the three-dimensional dynamic characteristic of the air spring derived from equation [5]. As can be seen, the force value F_M, illustrated in colour, depends on the parameters Q' and Δh. Figure 18b shows the characteristic curve for a selected example value of the parameter Q'.

a.



b.

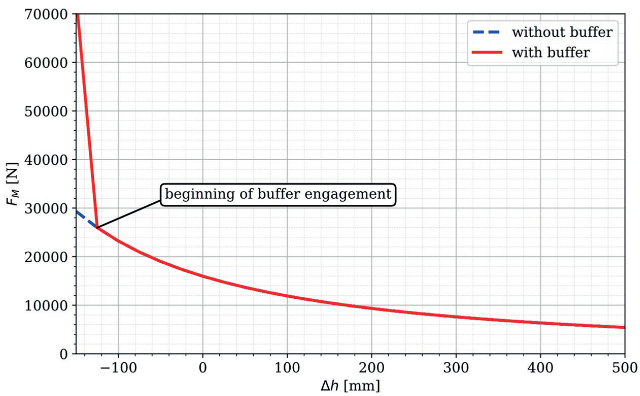


Fig. 18. [a] Three-dimensional dynamic characteristic of air spring, [b] Characteristic curve for Q' = 6000 kg

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In a real air spring, a rubber buffer is built in, which operates when the air spring is fully compressed – therefore, the characteristic shown in Figure 18b has been modified to account for its effect by adding a sharply rising curve within the range of the air springs' stroke where it operates.

7. MSC Adams model of "truck–trailer" vehicle combination

In the prepared MSC Adams model of vehicles (Figure 19), the connection between the truck and the trailer designed for transporting swap bodies – which in the actual system is achieved via a so-called coupling device (on the truck side) and a drawbar eye (on the trailer side) – has been simplified by introducing a ball joint, disregarding the friction and play that actually occur within it.

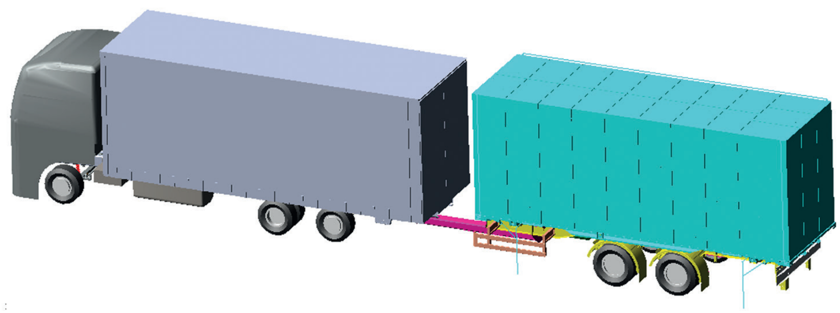

Fig. 19. MSC Adams model of "truck–trailer" vehicle combination

The developed MSC Adams model of the vehicle combination was used to simulate selected road manoeuvres. To this end, it was necessary to take into account both the time histories of the drive torques acting on the wheels of the truck's second axle (as in the actual system) and the steering angles of the wheels on its first axle. In the adopted truck model, the steering geometry was simplified and it was assumed that both steering knuckles, and thus the front wheels, had the same steering angles. The remaining elements of the steering system's geometry, such as the camber angle and the caster angle of the steering knuckle, were taken into account during the preparation of the truck's CAD model. The parameters of the tyres of the truck and the trailer were selected based on the PAC2002 tyre model offered in the MSC Adams environment, which is based on Pacejka's "magic formula" [17].

All simulations began with the vehicle combination at a standstill. During this phase, the correctness of the MSC Adams model was verified by assessing the static forces generated by the spring elements of the suspension of the individual trailer axles and the deflection of these elements.

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8. Results of selected numerical simulations performed in MSC Adams environment

As part of the numerical experiments, simulations were carried out for three typical driving manoeuvres, described in Annex B to standard PN-EN 12642:2016 [18], which resulted in high dynamic loads on vehicle components and, consequently, significant stresses.

8.1. Braking

The braking manoeuvre was carried out in accordance with the description given in section B.5.2 of Appendix B of the aforementioned standard. The braking effect was achieved by applying appropriate braking torques to all wheels of the truck and trailer. Figure 20 shows a section of the velocity curve of the centre of mass of the MSC Adams truck model during braking.

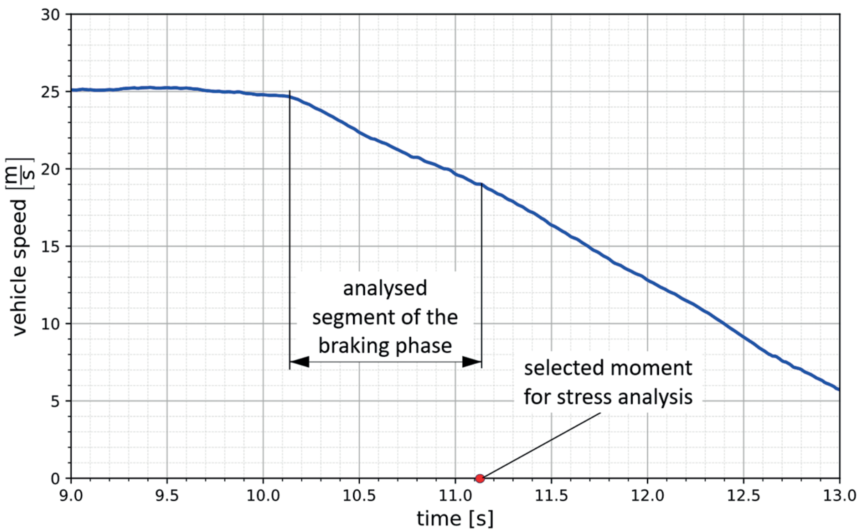

Fig. 20. Velocity curve of centre of mass of MSC Adams truck model during braking

To assess the reduced stresses in the trailer frame, the initial 1-second segment of the braking phase was selected. The observed stresses do not cause permanent deformation of the frame, but their apparent variability is significant, which is particularly relevant from the standpoint of fatigue strength. Figure 21 illustrates the contour lines of reduced stresses in the frame at the selected moment of analysis (Figure 20).

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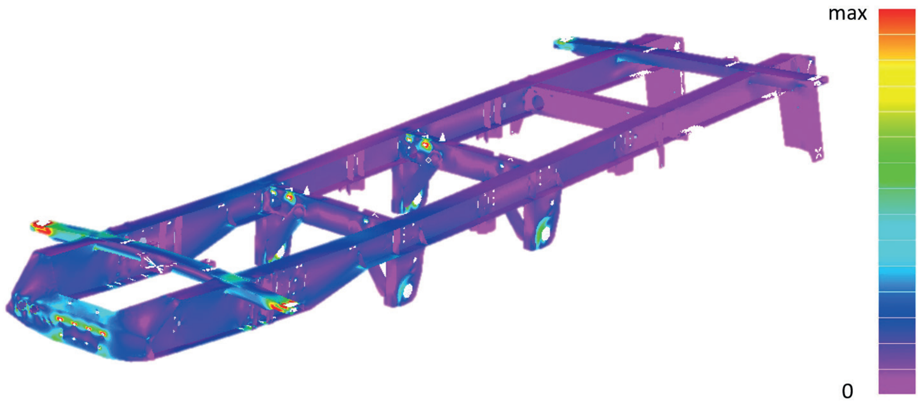

Fig. 21. Contour lines of reduced stresses in frame at selected moment of analysis

The relatively high reduced stresses in the area where the swap body is attached to the trailer frame also result from the method of modelling the connection using RBEs, which causes a local, unrealistic increase in the stiffness of this area, as well as from the simplifications adopted.

8.2. U-turn

This manoeuvre, described in the cited standard, involved performing a turn during which the turning radius of the centre of symmetry of the truck's front axle was within the range $R = 25 \pm 2$ m (Figure 22).

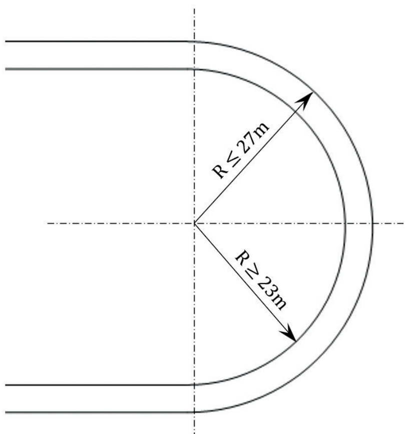

Fig. 22. Track geometry for U-turn

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The manoeuvre – performed at a vehicle combination speed with an initial value of approximately 30 km/h, i.e. approximately 8.3 m/s – causes a centripetal acceleration (normal) of the centre of mass of the trailer's load with a value of approximately 5 m/s^2. During the manoeuvre, drive torques were applied to the wheels of the truck's second axle with time histories varying in such a way as to increase the truck's speed to a set value and then "maintain" it. The truck's steering system incorporated kinematic forcing of the steering knuckles to replicate the assumed steering angle of the wheels during the manoeuvre. Whilst cornering, the acceleration of the centre of mass of the trailer's load gradually increased until the inner wheels of the trailer lost contact with the road surface and the trailer overturned (Figure 23).

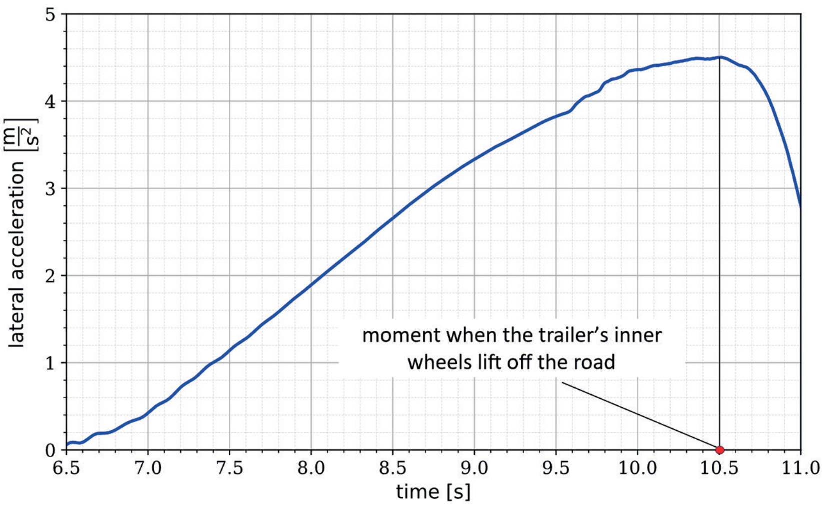

Fig. 23. Centripetal acceleration of centre of mass of trailer load during manoeuvre

Figure 24 illustrates the contours of reduced stresses in the frame of the analysed trailer, which arose at the moment when the trailer's inner wheels began to lift off the road surface. The analysis showed that significant reduced stresses appeared at the suspension mounting locations and in the cross member near the holes. The elevated reduced stresses may have resulted from the manner in which the load was applied via the RBEs. In reality, additional elements are bolted in these locations, which will cause the load to be "distributed" and the reduced stresses to decrease.

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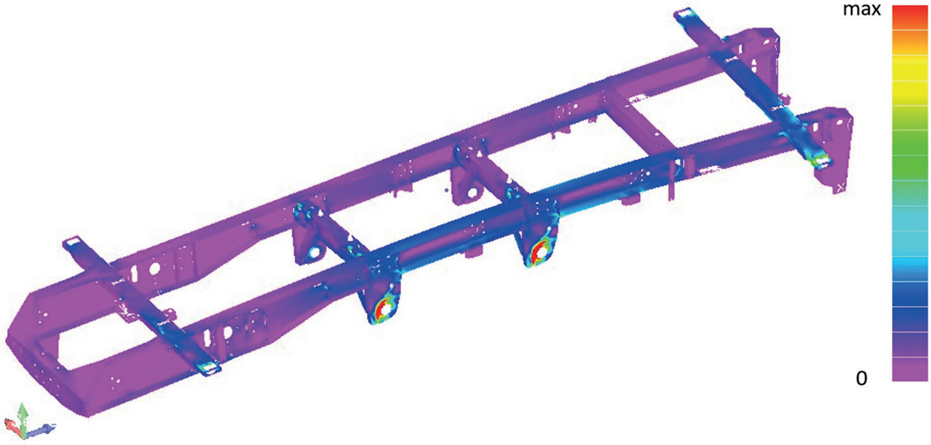

Fig. 24. Contour lines of reduced stresses at selected moment of analysis

Due to the strength of the entire trailer structure, the level of reduced stresses in the frame side members is of particular importance, both near the mounting holes for other components and near the manufacturing holes (Figure 25).

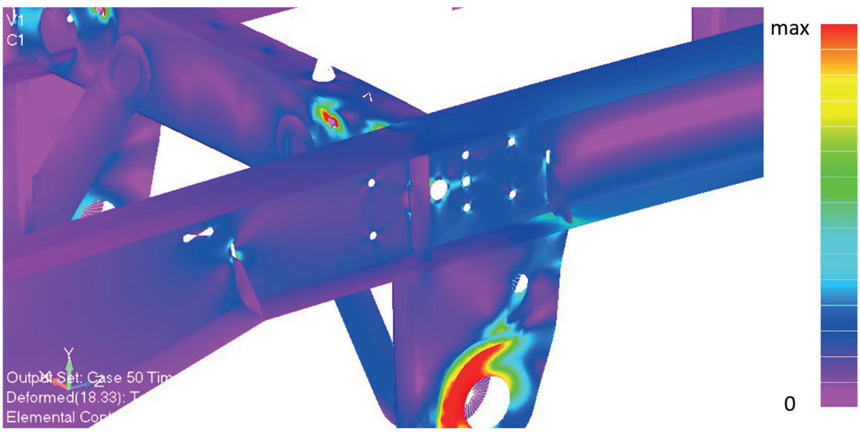

Fig. 25. Contours of reduced stresses in a critical section of frame at selected moment in the analysis

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The level of reduced stresses in the lower web of the side member in the area of the rear part of the axle bracket is also significant. The use of a so-called partial rib has allowed the loads to be "distributed" over a larger area, thereby reducing the maximum stress values. Calculations have shown that the partial rib is sufficient and there is no need to use a full rib connecting the lower flange of the girder to its upper flange.

8.3 Series of two S-turns

This manoeuvre, described in the cited standard, involved performing a series of two consecutive turns in one direction and then the other, during which the track radius of the centre of symmetry of the truck's front axle, as in the previous manoeuvre, was within the range of R = 25 ± 2 m (Figure 26). The manoeuvre was performed at an initial speed of approximately 30 km/h, i.e. approximately 8.3 m/s. During the simulation, the speed was limited to the value at which the inner wheels of the trailer lifted off the road surface and the trailer overturned.

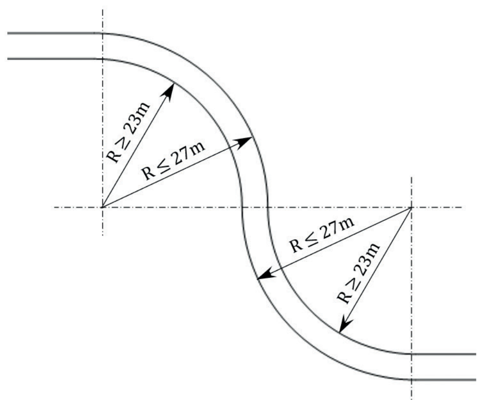

Fig. 26. Track geometry for series of two S-turns

Figure 27 illustrates the reduced stress contours in the analysed trailer frame at the moment when the inner wheels of the trailer began to lift off the road surface.

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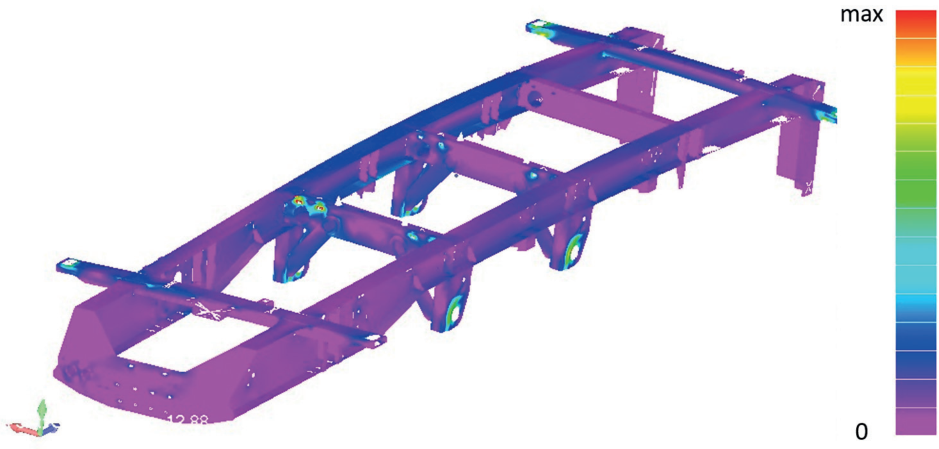

Fig. 27. Contours of reduced stresses at selected moment in analysis

In the analysed S-turns manoeuvre, the distribution of reduced stresses in the frame was similar to that observed in the case of a U-turn. The highest values again appeared at the suspension mounting locations. Despite local increases in stress, their level remained below the material's yield strength.

9. Summary

The dynamic FEM analysis described in this article, based on the interface between the MSC Adams and NX Nastran/Femap software packages, is significantly more advanced than the static FEM analysis and provides more accurate calculation results. The presented method can significantly accelerate the design process of truck trailers and may therefore have significant practical applications.

The results of the performed calculations may serve as a basis for fatigue analyses of the frame, enabling an assessment of its predicted service life.

The article presents the results of reduced stress calculations in the trailer frame obtained during selected road tests performed on a flat road surface. In future work, the passage of the vehicle combination over an uneven road surface could be considered, modelling its bumps and depressions.

In the authors' opinion, an important advantage of the method is the incorporation into the MSC Adams model of the findings from the analysis of the pneumatic suspension system of the trailer, as well as the development of a three dimensional dynamic characteristic of the air springs. This made the resulting model more realistic.

The method presented utilises the commercial software MSC Adams and NX Nastran/Femap – however, the authors of the article do not rule out the possibility of using other programmes, including those from the so-called "free software". The use of such software would require identifying how to practically implement the interface between the relevant programmes comprising it. It should also be borne in mind, however, that commercial programmes guarantee correct operation. Another advantage is that they have a database of "standard" components, such as a ready-made wheel model complete with data suitable for various tyre sizes, which is a significant advantage over "free software".

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