Application of Finite Element Method and MSC Adams Software in Design Process of Truck Trailers

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Abstract. A method of a dynamic analysis of a special kind of trailers with the central axle (axles) towed by truck is presented in the paper. Within the method, vibration and also strength calculations of frames and other components of the selected trailer can be carried out. Interface of MSC Adams software, for the dynamic analysis of mechanical systems, and NX Nastran/Femap software, for the finite element method (FEM) analysis, was used. The information transfer between both types of the software was realised within this interface. In the approach proposed, a significant reduction of degrees of freedom of the FEM model of the trailer frame was performed based on the Craig-Bampton method. Some numerical results were verified experimentally by using a trailer towed by a truck driving on the test track. The main goal of the proposed method is to ensure the reliability of the designed trailers in traffic conditions.

Keywords: Truck Trailer · Dynamics · FEM · MSC Adams · NX Nastran/Femap · Craig-Bampton Method

1 Introduction

In the case of heavy goods vehicles, a part of routes could be travelled with partial load or without load, also it is possible to travel with fully filled loading space by light goods with relative low density. Such cases of transport do not cause high stresses in the trailer frame and in other components, but they increase a relationship of the so-called unsprung mass to sprung one, what is an adverse phenomenon in vehicle dynamics. It could cause high-amplitude vibrations of the trailer frame and the components attached directly to it, e.g. mudguard brackets. These vibrations could be also a problem while transporting fragile goods. Therefore, during the design process, it is desirable to analyse possible vibrations. The vibrations of an empty trailer with two central axles (see Fig. 1) were analysed in this paper and a series of numerical simulations was performed in order to compare different variants of the trailer construction. For this purpose, the interface of programs MSC Adams and NX Nastran/Femap was used. This interface is an example of applying a new methodology [2, 3, 6, 7, 9] that combines aspects of both tools – finite element analysis and multibody system simulation.

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2 Materials and Methods

The CAD model of the vehicle combination in question was worked out by using software SolidWorks and Onshape. The truck model includes the sub-models of the frame, the cabin, the load space, the axles, the engine with the clutch and the gearbox, the rear differential gear, the fuel tanks and the battery box. The trailer model includes the sub-models of the frame, the drawbar, the load space, the axles with the brake disks and the callipers and the suspension system with the trailing arms.

2.1 FEM Model

The FEM model of the trailer frame was made by using pre-processor FEMAP on the basis of its CAD model. Shell elements, which represented the real components of the frame made of steel sheet, were used mainly in the model. The trailer drawbar and the trailer corner pillars (see Fig. 1) were modelled as beam elements.

2.2 MSC Adams Model

The MSC Adams model of vehicle combination (see Fig. 1) includes the model of the truck considered as a system of rigid bodies (a multi-body system) connected by spring and damping elements and the model of the trailer including the flexible frame and other rigid bodies.

In the model considered, the trailer was connected with the truck by a spherical joint. The suspension system of the trailer (see Fig. 2) was modelled in detail taking into account all its important parts. For example, the rubber bushings were modelled by the spring and damping elements located in the place of connection of trailing arms with suspension brackets, which are components of the frame. Torsional flexibility of axles was taken into account by introducing flexible connections in the place of the contact of axles with trailing arms. The air bags were modelled as spring elements having nonlinear characteristics based on the equation of the polytropic process for

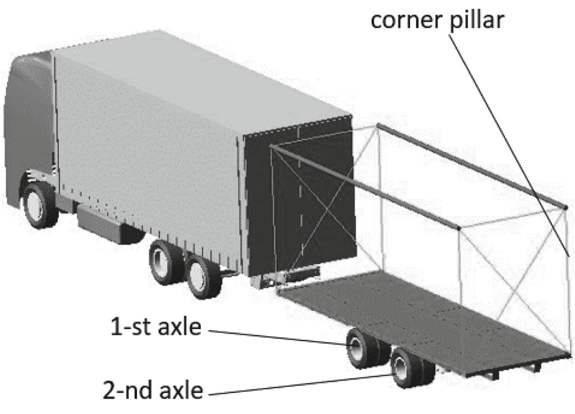

Fig. 1. Model of vehicle combination in MSC Adams environment.

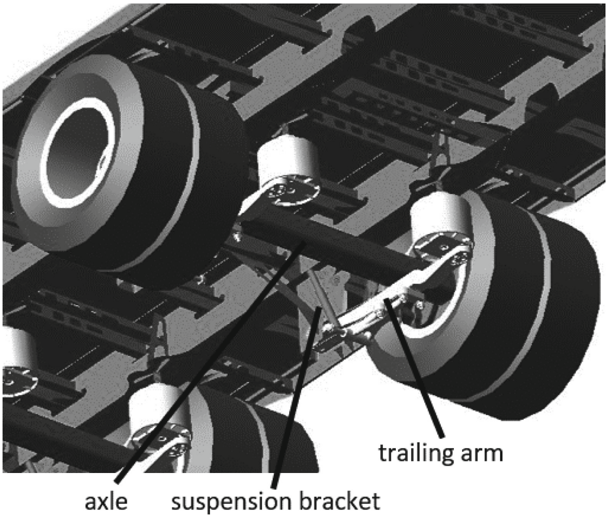

Fig. 2. Model of trailer suspension in MSC Adams environment.

ideal gas – according to Firestone Industrial Products Company recommendations [11]. Characteristics of damping elements were assumed on the basis of parameters of shock absorbers produced by BPW Bergische Achsen KG [12]. Parameters of tires of the truck and the trailer were taken on the basis of a tire model known as "Magic Formula", developed by Pacejka [8], which is offered in the MSC Adams environment.

2.3 MSC Adams – NX Nastran/Femap Interface

The motion of a deformable body, which is subjected to an action of constraints from the adjacent bodies, can be considered as motion of this body treated as non-deformable one, on which its deformations, being a result of its free vibrations, are superimposed [10]. This approach is called as "Floating Frame of Reference Method". It is used also in the case of interface of MSC Adams software and software using FEM based on the implementation of Craig-Bampton Method [1]. This method, which is based on the works of Guyan [4] and Hurty [5], belongs to the group of methods known as "Modal Reduction Technique" and it is used in order to reduce significantly the size of FEM model, in other words to diminish the number of its degrees of freedom. In the approach assumed, the nodes of the FEM model are divided into two groups: the boundary nodes (also called as exterior nodes) and interior nodes. In the boundary nodes constraints may be introduced or external forces/torques may act from adjacent bodies.

The equation of motion of the constrained FEM model (when damping is omitted) can be expressed as:

$$\boldsymbol{M}\ddot{\boldsymbol{u}} + \boldsymbol{K}\boldsymbol{u} = \boldsymbol{f} \quad (1)$$

where $\boldsymbol{M}$ is mass matrix, $\boldsymbol{K}$ is stiffness matrix, $\boldsymbol{u}$ is a vector of displacements (generalized coordinates) of all nodes, and $\boldsymbol{f}$ is vector of forces/torques acting at nodes.

This equation can be presented in the form of two equations referring to boundary or interior nodes:

$$\begin{bmatrix} \boldsymbol{M_{BxB}} & \boldsymbol{M_{BxI}} \\ \boldsymbol{M_{IxB}} & \boldsymbol{M_{IxI}} \end{bmatrix} \begin{bmatrix} \ddot{\boldsymbol{u}}_{\boldsymbol{B}} \\ \ddot{\boldsymbol{u}}_{\boldsymbol{I}} \end{bmatrix} + \begin{bmatrix} \boldsymbol{K_{BxB}} & \boldsymbol{K_{BxI}} \\ \boldsymbol{K_{IxB}} & \boldsymbol{K_{IxI}} \end{bmatrix} \begin{bmatrix} \boldsymbol{u_B} \\ \boldsymbol{u_I} \end{bmatrix} = \begin{bmatrix} \boldsymbol{f_B} \\ \boldsymbol{f_I} \end{bmatrix} \quad (2)$$

where $\boldsymbol{B}$ and $\boldsymbol{I}$ are respectively the numbers of degrees of freedom of boundary and interior nodes.

Taking into account fact that in interior nodes there is no external load (in the form of force or torque), so $\boldsymbol{f_I} = 0_{\boldsymbol{I}}$, hence this equation can be written as:

$$\begin{bmatrix} \boldsymbol{M_{BxB}} & \boldsymbol{M_{BxI}} \\ \boldsymbol{M_{IxB}} & \boldsymbol{M_{IxI}} \end{bmatrix} \begin{bmatrix} \ddot{\boldsymbol{u}}_{\boldsymbol{B}} \\ \ddot{\boldsymbol{u}}_{\boldsymbol{I}} \end{bmatrix} + \begin{bmatrix} \boldsymbol{K_{BxB}} & \boldsymbol{K_{BxI}} \\ \boldsymbol{K_{IxB}} & \boldsymbol{K_{IxI}} \end{bmatrix} \begin{bmatrix} \boldsymbol{u_B} \\ \boldsymbol{u_I} \end{bmatrix} = \begin{bmatrix} \boldsymbol{f_B} \\ 0_{\boldsymbol{I}} \end{bmatrix} \quad (3)$$

Vector of displacements of the interior nodes can be presented as the sum of vectors:

$$\boldsymbol{K_{IxB}u_B} + \boldsymbol{K_{IxI}u_I'} = 0 \quad (4)$$

After transformation, the following formula can be obtained:

$$\boldsymbol{u_I'} = -\boldsymbol{K_{IxI}^{-1}K_{IxB}u_B} = \boldsymbol{G_{IxB}u_B} \quad (5)$$

$$\boldsymbol{u_I'} = \boldsymbol{G_{IxB}u_B} \quad (6)$$

where $\boldsymbol{G_{IxB}}$ is matrix of static correction modes (matrix of static deformation shapes). The elements of this matrix are obtained by fixing all the boundary nodes and applying a unit linear or angular displacement to one degree of freedom of only one of these nodes. This procedure results in obtaining as many static deformation shapes of the FEM model as there are degrees of freedom of its boundary nodes. An image of the static deformations of the FEM model in question is obtained as the result of the procedure assumed.

Vector $\boldsymbol{u_I''}$ can be determined as:

$$\boldsymbol{u_I''} = \boldsymbol{\Phi_{Ixj}q_j} \quad (7)$$

where:

$\boldsymbol{q_j}$ – vector of modal displacements of FEM model,

$\boldsymbol{j}$ – number of modal displacements (where $\boldsymbol{j} \leq \boldsymbol{I}$),

$\boldsymbol{\Phi_{Ixj}}$ – matrix of normal modes of vibrations (matrix of mode shapes of vibrations).

Consequently, the image of the dynamic deformations of the FEM model in question is obtained, being a result of its free vibrations. Natural frequencies and normal modes of these vibrations are determined during the modal analysis. In computational practice, a number $\boldsymbol{j}$ of the modal displacements taken into account in the analysis can be assumed as much lower than a number $\boldsymbol{I}$ of the degrees of freedom of the interior nodes of the FEM model. Obviously, it should be ensured that it will not affect accuracy of the calculations performed. As a consequence of the procedure presented, the motion of the FEM model is described by the reduced system of generalized coordinates, being a sum of generalized coordinates of the boundary nodes and the modal displacements. The Craig-Bampton transformation can be applied by using relationships (7) and (8):

$$\begin{bmatrix} \boldsymbol{u_B} \\ \boldsymbol{u_I} \end{bmatrix} = \begin{bmatrix} \boldsymbol{I_{BxB}} & 0_{\boldsymbol{Bxj}} \\ \boldsymbol{G_{IxB}} & \boldsymbol{\Phi_{Ixj}} \end{bmatrix} \begin{bmatrix} \boldsymbol{u_B} \\ \boldsymbol{q_j} \end{bmatrix} \tag{8}$$

where:

$\boldsymbol{I_{BxB}}$ – unit matrix.

Including this transformation in Eq. (2) and making further transformations the equation of motion can be written in a form of a new system of which mass matrix $\boldsymbol{M}$ and stiffness matrix $\boldsymbol{K}$ have reduced dimensions (additionally, the stiffness matrix contains lots of zero elements). Consequently, a number of generalized coordinates describing motion of the FEM model in question decreases significantly what enables to speed up radically the computation process:

$$\boldsymbol{K_{CB}} = \begin{bmatrix} \boldsymbol{I_{BxB}} & 0_{\boldsymbol{Bxj}} \\ \boldsymbol{G_{IxB}} & \boldsymbol{\Phi_{Ixj}} \end{bmatrix}^{\boldsymbol{T}} \boldsymbol{K} \begin{bmatrix} \boldsymbol{I_{BxB}} & 0_{\boldsymbol{Bxj}} \\ \boldsymbol{G_{IxB}} & \boldsymbol{\Phi_{Ixj}} \end{bmatrix} \tag{9}$$

$$\boldsymbol{M_{CB}} = \begin{bmatrix} \boldsymbol{I_{BxB}} & 0_{\boldsymbol{Bxj}} \\ \boldsymbol{G_{IxB}} & \boldsymbol{\Phi_{Ixj}} \end{bmatrix}^{\boldsymbol{T}} \boldsymbol{M} \begin{bmatrix} \boldsymbol{I_{BxB}} & 0_{\boldsymbol{Bxj}} \\ \boldsymbol{G_{IxB}} & \boldsymbol{\Phi_{Ixj}} \end{bmatrix} \tag{10}$$

A transfer of information between the programs realised within the interface MSC Adams and NX Nastran/Femap is presented in Fig. 3.

2.4 Experimental Tests

Driving of the trailer towed by a truck, at different speeds, over a single obstacle in the form of a speed bump of the dimensions as shown in Fig. 4 was performed, among others, as a part of the experiments carried out.

While driving, time courses of accelerations were determined by sensors located in the selected places (points) of the trailer and they were compared with the time courses of these accelerations determined by computing. In each case considered, satisfactory compliance of both types of the courses was found. According to the authors, it confirms correctness of FEM and MSC Adams models assumed. In the next figures, there are some examples of the results of the experiments and the calculations. They deal with one driving of the vehicle combination over the obstacle at 3, 4 m/s speed (ca. 12 km/h). In Fig. 5, location of the sensors is shown: first one is placed in symmetry centre of 2-nd axle of the trailer, and second near the gravity centre of the frame.

As it can be seen, vibrations of the symmetry centre of 2-nd axle are intensified between the eighth and ninth second of motion when the wheels of this axle overcome the obstacle (see Fig. 6). A smaller increase of the vibrations also occurred right before the eight second of motion when the obstacle was overcome by the wheels of 1-st axle what indicates an impact of the vibrations of 1-st axle on the vibrations of 2-nd axle. The vibrations of the gravity centre of the trailer frame (see Fig. 7) occur already in the fifth second of motion when the rear wheels of the truck overcome the obstacle, then they disappear and increase again when the trailer wheels drive over the obstacle.

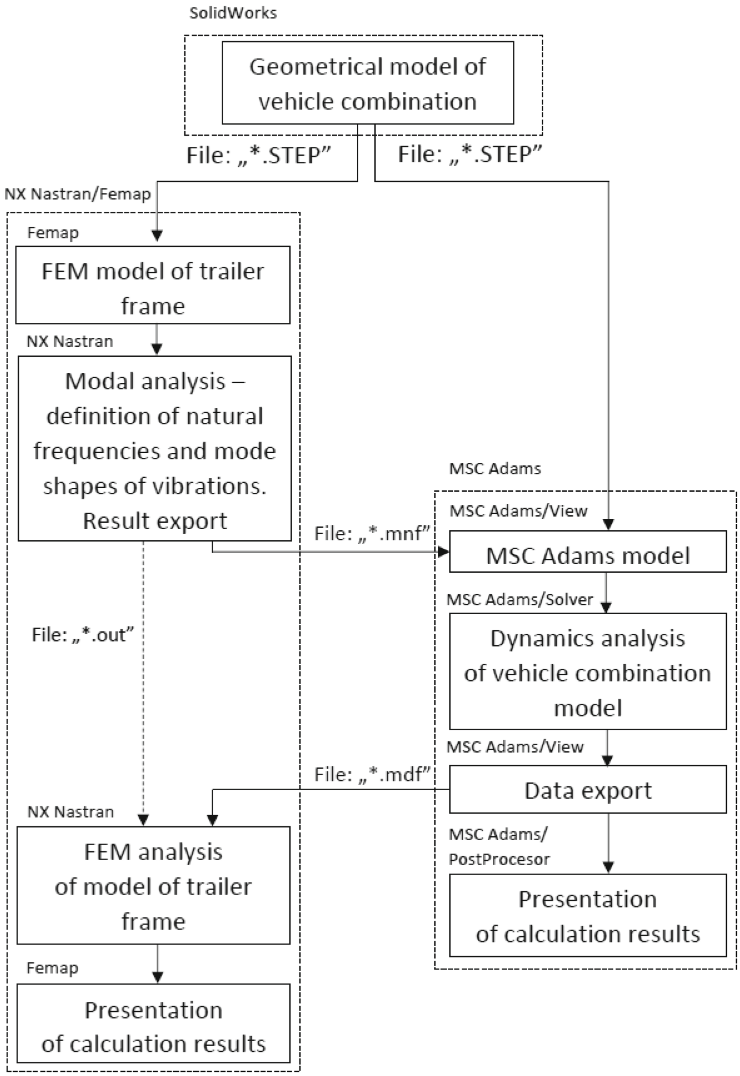

Fig. 3. Interface of programs MSC Adams and NX Nastran/Femap.

3 Results

Within the numerical simulations, two cases of motion of vehicle combination were considered, that is driving over the obstacle shown in Fig. 4 (case A) and driving over uneven road surface, characterized by International Roughness Index equal 5 m/km (case B). Two versions of the trailer were considered: a standard version and a version with lifted 1-st axle. In case B, the vehicle combination accelerates from the initial speed equal zero to speed equal 14 m/s (ca. 50 km/h) during 10 s, and then it continues to drive at constant velocity.

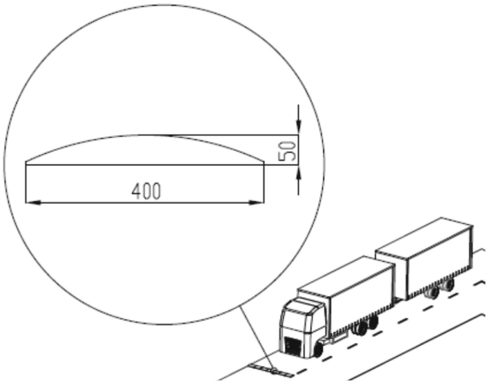

Fig. 4. Dimensions of speed bump in mm.

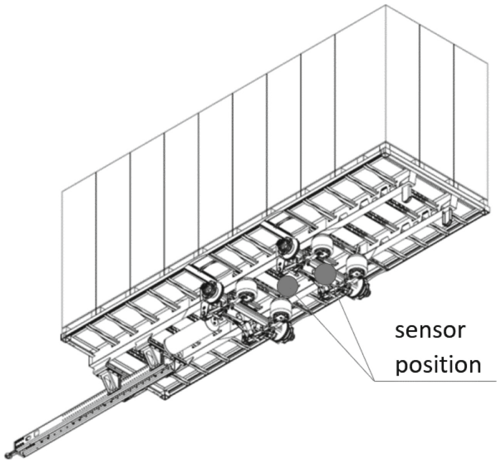

Fig. 5. Sensor position.

The results of case A of motion are illustrated in Fig. 8. The time courses of accelerations of frame gravity centre of trailer in the standard version and with lifted 1-st axle are presented in this figure. The accelerations reach the similar maximum values, although slower disappearance of vibrations is clearly noticeable in the case of the trailer with lifted 1-st axle.

The results of case B of motion are illustrated in Fig. 9. The time courses of accelerations of 2-nd axle symmetry centre and the gravity centre of trailer frame are presented in Fig. 9 and 10, respectively.

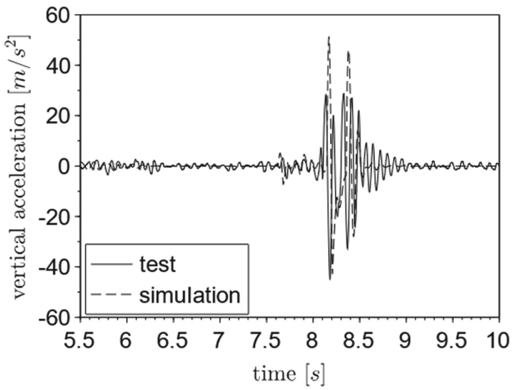

Fig. 6. Vertical accelerations of 2-nd axle symmetry centre.

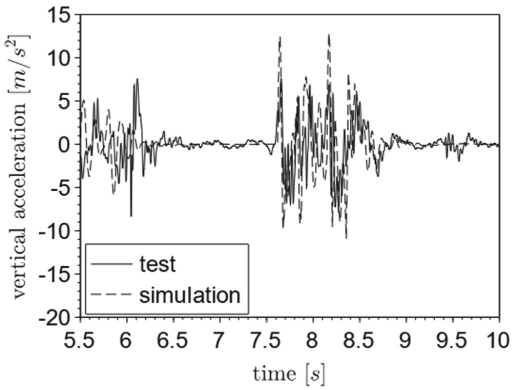

Fig. 7. Vertical accelerations of gravity centre of trailer frame.

Much lower values of the accelerations and lower frequencies of vibrations are clearly visible in the case of motion of the trailer with lifted 1-st axle. Therefore, it seems to be well-founded to use lifting mechanism of 1-st axle in the construction of trailer. It could significantly limit vibrations of the frame and components connected to it, and thus lifetime of these elements could increase.

In case B of motion, forces and torques acting in the place of connection of the mudguard bracket of the selected wheel (see Fig. 11) with the trailer frame were also determined. The time courses of these forces and torques, including three directions of trailer motion, are presented in Fig. 12 and 13, respectively.

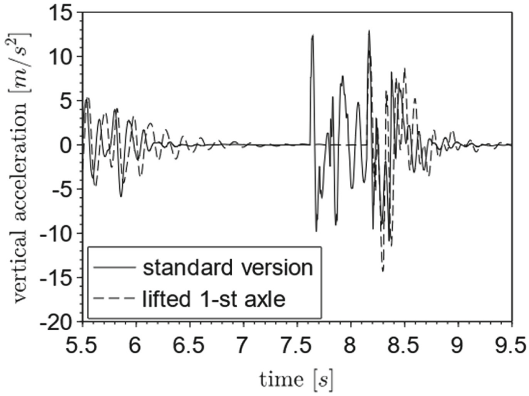

Fig. 8. Vertical accelerations of gravity centre of trailer frame – case A of motion.

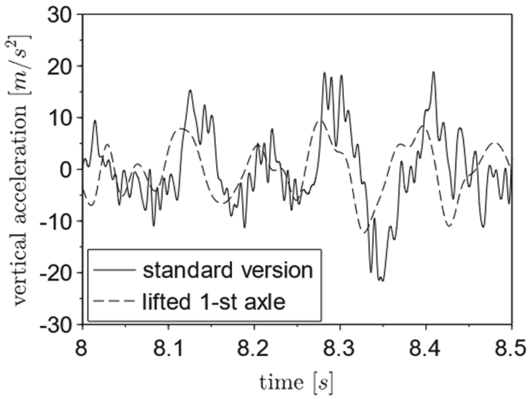

Fig. 9. Vertical accelerations of 2-nd axle symmetry centre – case B of motion.

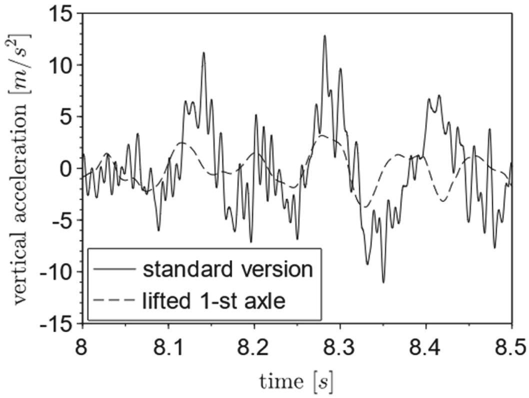

Fig. 10. Vertical accelerations of gravity centre of trailer frame – case B of motion.

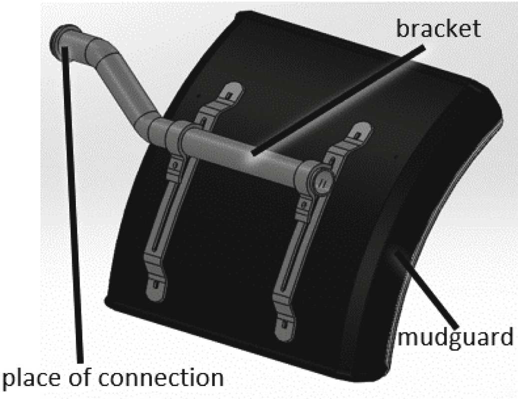

Fig. 11. Mudguard with bracket.

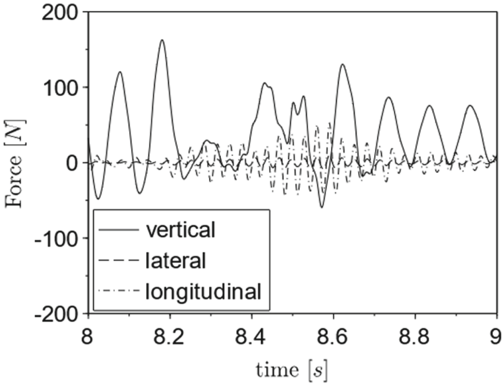

Fig. 12. Forces in connection of mudguard bracket with trailer frame.

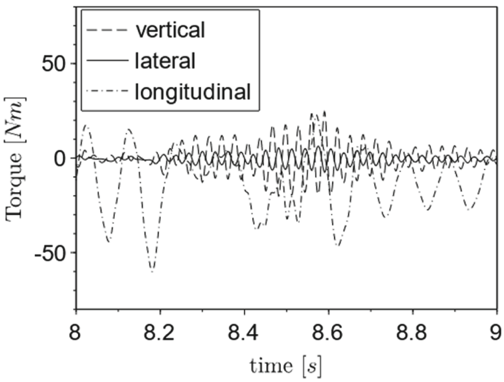

Fig. 13. Torques in connection of mudguard bracket with trailer frame.

Values of forces and torques are not significant, although their oscillations can cause loosening of the bolt connecting the bracket with the trailer frame.

4 Conclusions

According to the authors, the presented method can be of interest for engineers dealing with design process of truck trailers. The model of the trailer developed by use of interface of programs MSC Adams and NX Nastran/Femap, understood as a virtual prototype, enables to perform the required numerical simulations in design process. Virtual simulations allow for quick analysis of many alternative solutions. An analysis of dynamics of the virtual prototypes of trailers may become the basis for preparing its real prototypes. The proposed approach should shorten significantly design process duration and by that it can lower its costs.

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