MULTI-BODY SIMULATIONS OF RAILWAY WAGON DYNAMICS

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Abstract

The main aim of the presented paper is to verify the dynamics of chosen railway wagon using multi-body methods. Thus, in tests a wagon prototype with rotational rail-car body was simulated, which is going to be used for TIR vehicles trailer bodies transporting. Implemented prototype is authored by the team from Department of Applied Mechanics and Computer Science from Faculty of Mechanical Engineering in Military University of Technology.

For this purpose CAD model, and subsequently numerical model, of mentioned wagon was developed. Using MSC Adams software initial boundary conditions, material parameters and contact characteristics were defined. In the previous authors' paper [1], preliminary analyses of the loads acting on the railway platform were carried out, which verified chosen parameters and coefficients. The main aim of presently proposed 3D multi-body simulations of railway wagon is to investigate its motion with full load and to determine the operation limit safety. Thus, in the performed analyses the railway wagon was accelerated to the desired velocity and drove through the curved tracks with four different velocities. From the carried out simulations the wagon dynamic behaviour was investigated and obtained results including contact forces characteristics were compared from all analyses cases. Moreover, performed study shows that the MSC. Adams software is capable for analysing and simultaneously validating various complex engineering problems, such as the one investigated by the authors

Keywords: *multi-body, MSC Adams, railway wagon, dynamics*

1. Introduction

The main aim of the carried out investigations is to verify the dynamics of chosen railway wagon using multi-body methods. Thus, for testing a wagon prototype with rotational rail-car body was simulated, which is going to be used for TIR vehicles trailer bodies transporting. Implemented prototype is authored by the team from Department of Applied Mechanics and Computer Science from Faculty of Mechanical Engineering in Military University of Technology.

Within proposed investigations the CAD model of mentioned railway wagon was developed, which consequently was the basis for numerical model implementation. Subsequently, multi-body model was validated in preliminary dynamic simulations, which proofed that adopted mechanical parameters and initial boundary conditions were properly used. Results of these simulations of railway wagon crossing straight and curved tracks can be found in the previous authors' paper [1]. Both previous and current numerical tests were carried out using MSC Adams software, which effectiveness was proven by many authors [2,3].

The idea for developing such vehicle came from the increasing demand for alternative, more economical and ecological forms of inland transport of goods. Also, what can be found in [4], one heavy goods vehicle within 3 seconds causes road surface damage comparable to 163 840 passengers' cars. This gives approximately 400 deaths per year. Hence, one of the solutions to reduce these negative consequences can be the intermodal transport, which could optimize the relationship between various means of transport in Poland and other countries. It should be pointed out that intermodal transport market in Poland is very small: it was estimated to be 2%, while in

Europe is about 25% [4]. The issue of the rail transport in Poland was also provoked by the actions of various institutions and organizations. It has also turned out that intermodal transport is increasingly being used in the European Union with success, especially in France (Modalohr) and Sweden (Megaswing) (Fig. 1).

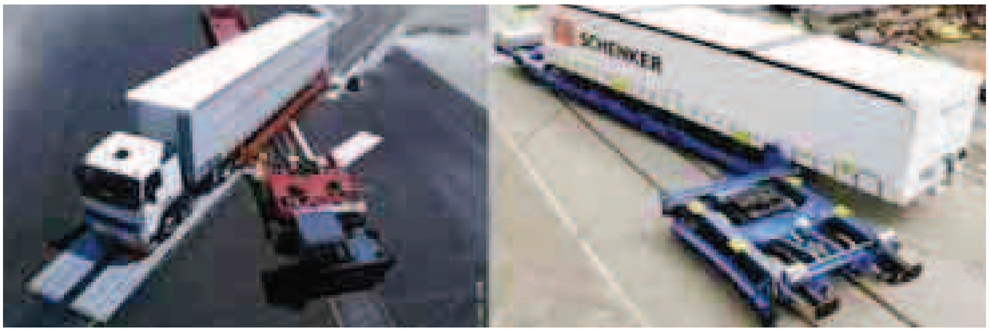

Fig. 1. Intermodal systems used in France: Modalohr (left) and in Sweden: Megaswing (right) [5, 6]

2. Object of investigation

In the presented paper one of the first prototypes of railway wagon is modelled and tested, which is presented in Fig. 2. Wagon structure allows for easy heavy goods trucks loading on the suitable railway platforms. Rotation of the moving part of the wagon is achieved thanks to hydraulic actuators (between fixed frame and moving part). Moreover, in the central part of the wagon a rotary mechanism is placed that helps to rotate the platform with the suitable angle around the axis of bearing. Also, at both ends of the platform two rolls are mounted which are an additional support and allow it to move freely.

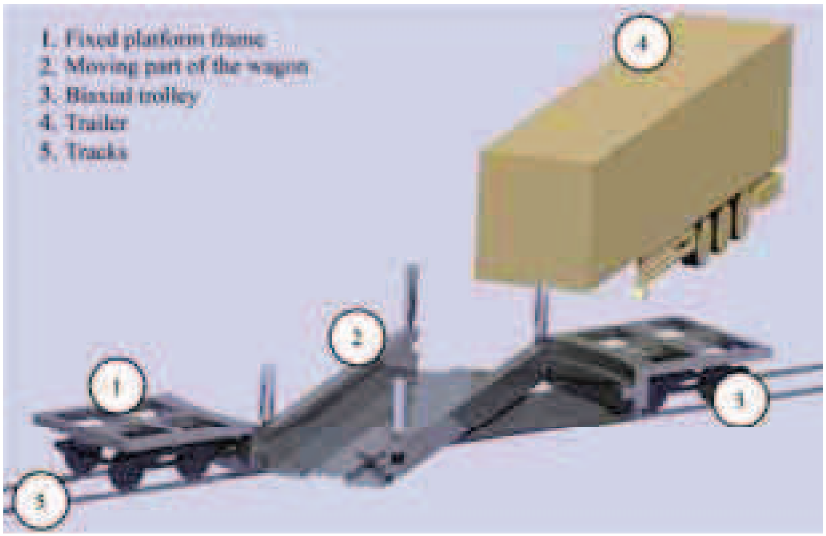

Fig. 2. Schematic model of investigated railway wagon [1, 7, 8]

3. Numerical model of railway wagon

Numerical model of railway wagon was developed based on the CAD model presented previously. For the purpose of carrying out dynamic simulation, there was a need to adopt certain simplification. Thus, braking system elements, wagon bumpers, elements responsible for stiffness increase of platform frame and devices required to operate and control gear assembly were omitted at the stage of numerical modelling. These particular elements were chosen due to their low weight, which (comparing them to the overall mass of wagon) did not influence the dynamics of vehicle and its behaviour during tests. Also, in order to simplify the model and shorten computational time the side slides were not included (Fig. 3).

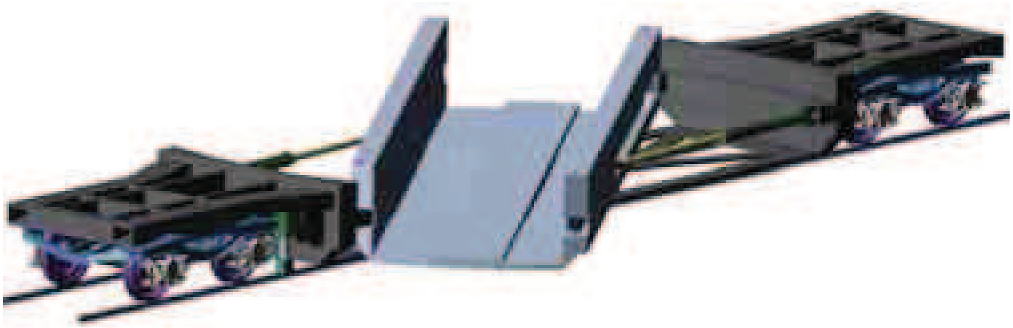

Fig. 3. Numerical model of railway wagon adopted for analyses [1]

The collaboration between trucks and mainstay was implemented by using an articulated spherical joint with applied rotational constraint about Y global axis, which reflected the cooperation with omitted side slides. Biaxial trolleys (Fig. 4), as well as their frameworks, were considered as a solids combined into the single component. This resulted in reduction of needed relations and kinematic constraints. Moreover, trolleys suspension was modelled with spring dumpers and translational joints allowing for one possible direction of movement perpendicular the frame.

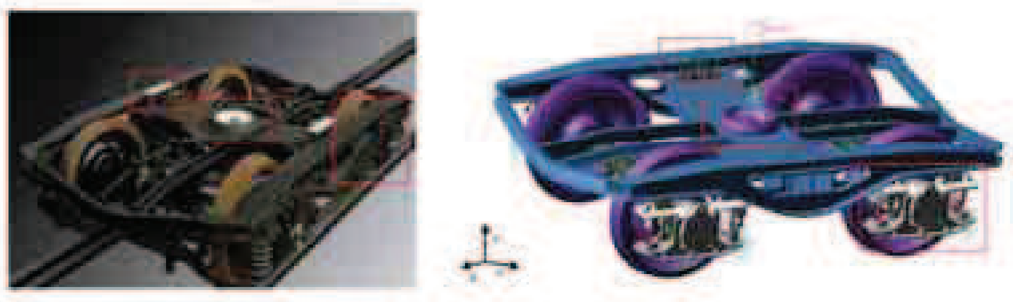

Fig. 4. Actual trolley Y25 [9] and its corresponding numerical model [1]

The chassis and simplified suspension system of semi-trailer were modelled as a uniform mass with possibility to rotate in ZY plane (about trailer's wheels axis). Similar to trolleys, deflection of the trailer was limited by four spring dampers connecting the chassis with wheels. Discussed numerical model of the semi-trailer is presented in Fig. 5. Adopted mass of the trailer was chosen based on the technical specification of the original KÖGEL curtain trailer [10].

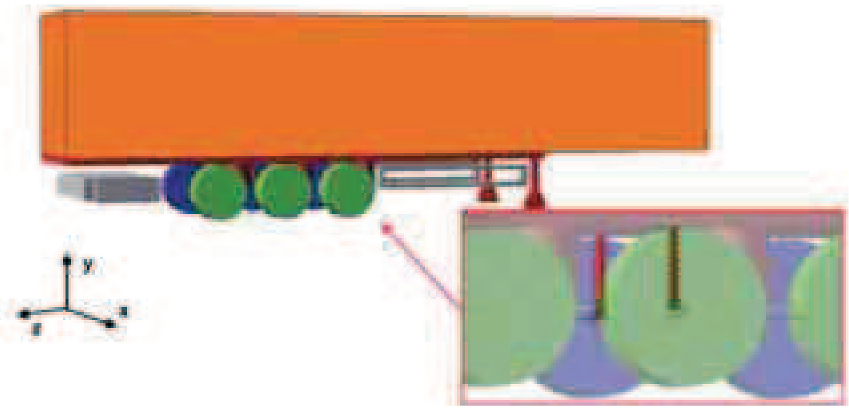

Fig. 5. Actual trolley Y25 [9] and its corresponding numerical model [1]

4. Numerical simulations description

4.1. Theoretical fundamentals of carried out simulations

An analysis of multi-body kinematics can be performed in natural or absolute coordinates that is implemented in MSC Adams software as well [11]. In this method, equations of motion of a single component are described by twelve first-order differential and three algebraic equations. Also, for all zero value velocities and accelerations an algebraic equation is obtained which can be solved using Newton-Raphson scheme in terms of unknowns (Fig. 6):
– r: displacement vector of local coordinate system relative to the global one,
– $\varphi = [\psi, \theta, \phi]$: three Euler angles system,
– λ: Lagrange multiplayer vector.
Subsequently, the static equilibrium and reaction forces values are calculated.

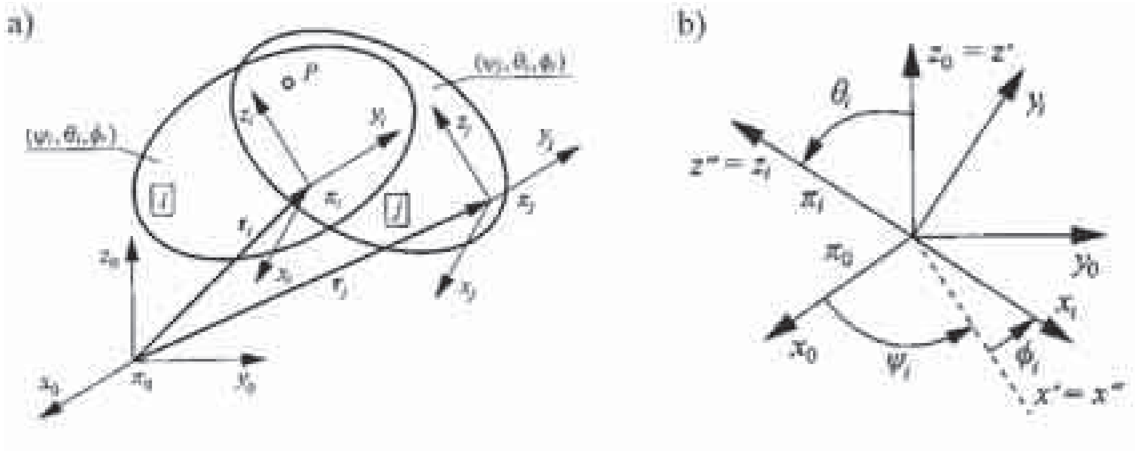

Fig. 6. a) Absolute coordinates and b) Euler angles [11]

If there is a need to analyse multi-body dynamics, some general assumptions have to be adopted including that the global reference coordinate is inertial and origins of local coordinate systems are in the components centres of mass to which they belong. Within simulated problem equation of motions are formulated in absolute coordinates using mentioned first-order Lagrange equations as follows [11]:

$$\frac{d}{dt}\left(L_{\dot{q}}^{T}\right) - L_{q}^{T} + \phi_{q}^{T}\lambda = Q\,, \tag{1}$$

where:
L – Lagrange function: difference between kinetic and potential energy $L = (T - V)$,
λ – Lagrange multiplier vector,
Q – vector of generalized forces acting on the multi-body system ,
ϕ – left side constraint equation vector in the following form $\phi(q,t) = \begin{bmatrix} \phi^{K}(q) \\ \phi^{D}(q,t) \end{bmatrix}$.

4.2. Initial-boundary condition for analyses

In the previous authors paper [1] the results of preliminary simulation of the railway wagon were presented. By carrying out the analysis of loads acting on the platform the model was validated. Also, checking the equilibrium condition between the Contact forces and inertia forces confirmed the correctness of used coefficients and parameters. This model is used in presented investigations also.

In order to properly carry out numerical analysis it was necessary to define correct initial-conditions including railway wagon velocity, loading and friction characteristic. Thus, for simulation the railway wheels and tracks commonly used in Poland were implemented (according to the PK-92/K-91056 standard). Currently, in the major of operating railway vehicles the standard biaxial trolleys are used, thus narrowed edges in UIC60 outlines of wheels were not included in computations.

The rails were modelled in a simplified way but with upper curvature taking into account. Relation between trolleys wheels and railway was simulated using contact procedure available in MSC Adams/View software in which contact forces are calculated using an “impact” method [11]. Moreover, all joint required to define mechanical parameters presented in Tab. 1. Their values were chosen based on authors experience in this field and from literature [2, 3]. Additionally, static and kinetic friction coefficients were taken from literature [12]. Moreover, frictional effects occurring in spherical joints connecting trolleys with the platform were considered and modelled by applying a coefficient of friction for greased steel, i.e. 0.08 for static friction and 0.05 for kinetic friction [2].

Tab. 1. Statistic data of discrete suspension system model

No.	Parameter	Value	Unit
1	Stiffness	13	-
2	Force coefficient	1.5	-
3	Damping	50	[N*mm/s]
4	Penetration depth	0.1	[mm]
5	Max. velocity for static friction	100	[mm/s]
6	Min. velocity for dynamic friction	1000	[mm/s]
7	Static friction coefficient	0.7	-
8	Dynamic friction coefficient	0.57	-

In order to obtain a certificate of entry into service of any freight wagon it is necessary to thoroughly test a vehicle including the passage through the arc with minimum radius. Thus, the main aim of proposed 3D multi-body simulations of railway wagon is to investigate its motion with full load and to determine the operation limit safety according to [13]. In performed analyses the railway wagon was accelerated to the desired velocity. Subsequently, it drove through first left turn with the radius of 250m and second right turn with the same radius (Fig. 7). Due to the scientific character of the work tracks widening, super elevations and transient radiuses were included in simulations.

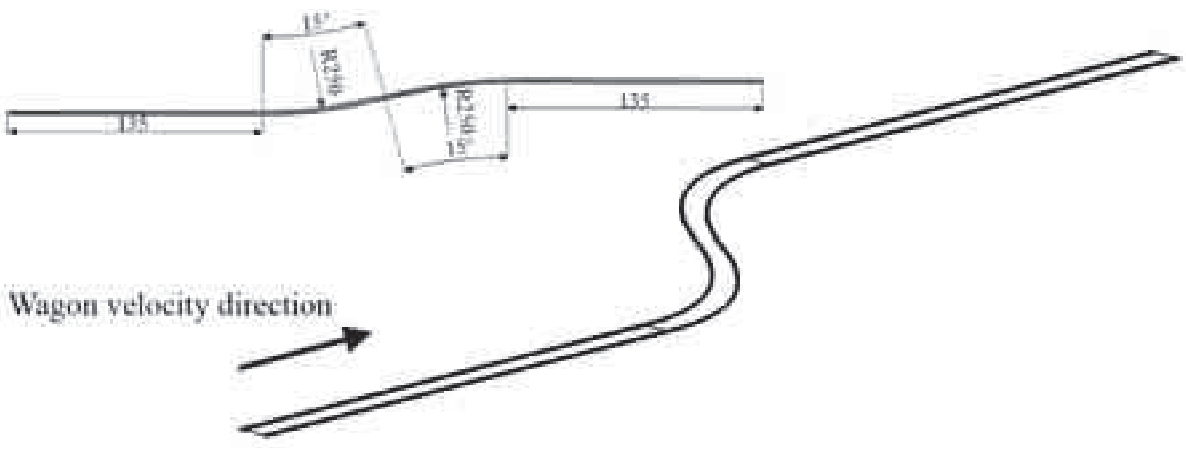

Fig. 7. Railway tracks used in simulations

For analyses authors choose four wagon velocities: 90 km/h, 95 km/h, 100 km/h and 110 km/h, which were implemented by angular rotational speeds applied on each platform axle. Also, there was a need to avoid wheels slips in the initial phase of movement, therefore the velocity had increasing characteristic.

5. Numerical simulations results analysis

From the performed simulations the wagon dynamic behaviour was obtained. In order to present the results more precisely and clearly, the contact force values during the passage for all cases were gathered and analysed (Fig. 8-11). In the presented graphs the force values were taken from the rear axle wheels of the first (front) trolley. In all cases, it can be seen that Contact force values are different for left and right wheel. This can be explained as the effect of inertia forces acting on the platform when riding on the chicane.

By taking closer look at below figures it can be noticed, that for cases with velocities of 90 km/h and 95 km/h there was no continuity brake of the contact between wheels and rails (Fig. 8 and 9).

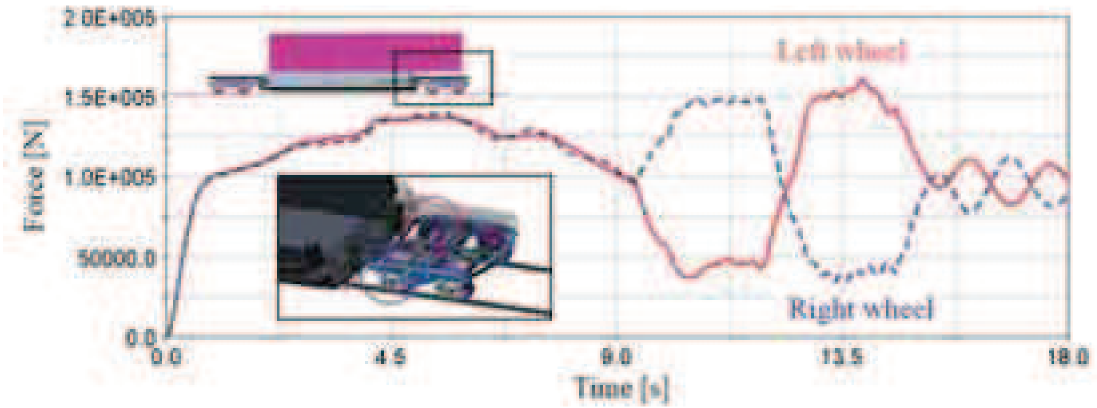

Fig. 8. Contact force versus time for 90 km/h case

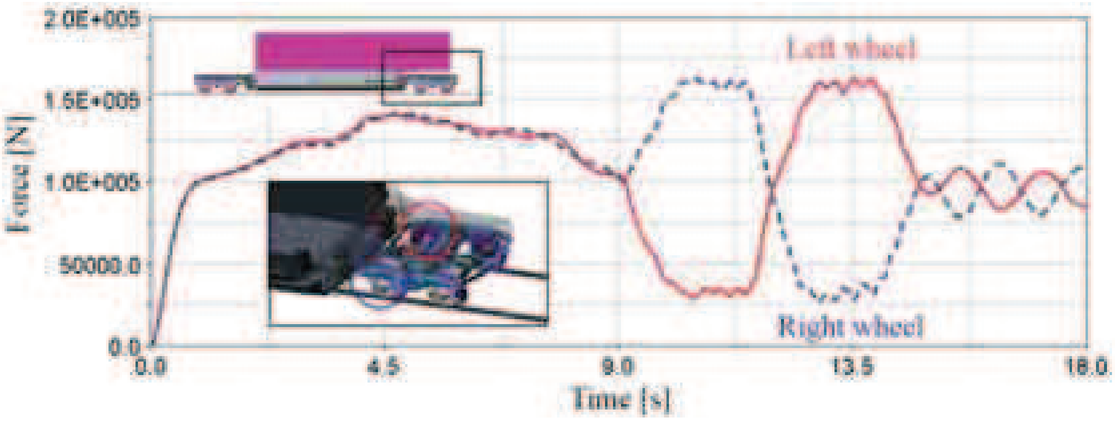

Fig. 9. Contact force versus time for 95 km/h case

In the subsequent analyses the railway wagon had the velocity 100 km/h and 110 km/h. In both cases the discontinuity of the wheel and rail contact was noticed. In case with 100 km/h the right wheel broke away after reaching second turn (at approximately 12.25 s). Also, in 110 km/h limit safety was exceeded, which resulted in platform overturning due to the high centrifugal and inertial forces: it was noticed after 9 s of simulation. The contact force versus time characteristics for both cases are presented in Fig. 10 and Fig. 11. The wagon overturning after reaching the first chicane for 110 km/h is presented in Fig. 12.

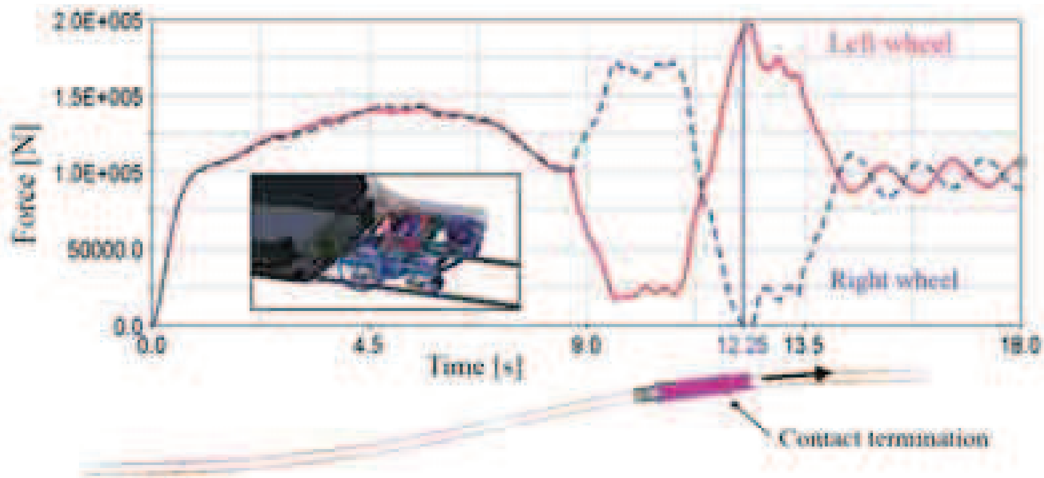

Fig. 10. Contact force versus time for 100 km/h case with the moment of wheel connectivity brake

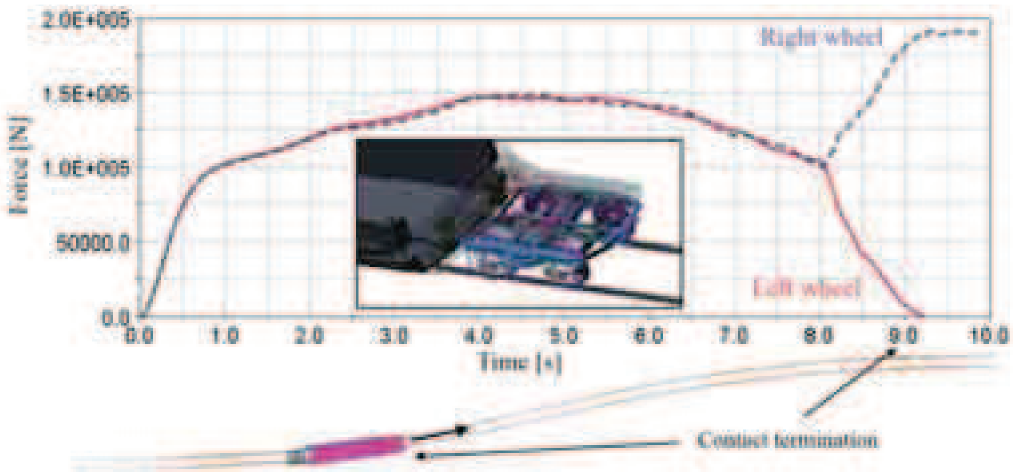

Fig. 11. Contact force versus time for 110 km/h case with the moment of wheel connectivity brake

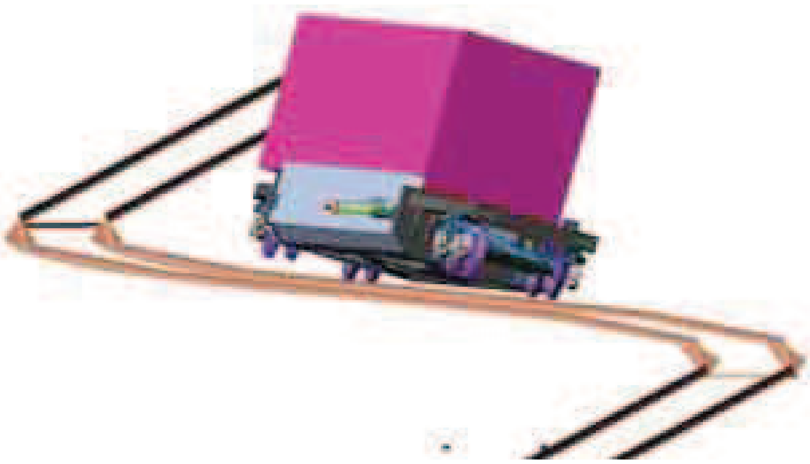

Fig. 12. Railway wagon overturning after reaching the first chicane

6. Conclusions

The presented paper shows the subsequent stages of multi-body modelling and simulations. General conclusion from the performed study is that that the MSC. Adams software is capable for analysing and simultaneously validating various complex engineering problems, such as the one investigated by the authors.

Finally, as a result of the dynamic tests at different wagon velocities the safety limit for the platform was obtained. It was concluded that (within investigated speeds) the 95 km/h is the value of velocity when the connectivity brake was not observed. Although, it should be pointed out that presented simulations had a lot of simplifications, i.e. tracks widening, super elevations and transient radiuses were included in simulations. Also, extreme load conditions applied in the model, minimum track radius and unreal cargo load disposition do not occur in reality. Other factors also influenced obtained safety limit, i.e.: deficiencies of the trolleys masses or rigid side slides modelling. Therefore, those rigorous analysis conditions leave users a wide safety margin. However, the presented study is the basis for further, more complex numerical and strength tests of investigated railway wagon prototype.

Acknowledgements

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