Simulation of Freight Car Dynamics: Mathematical Models, Safety, Wear

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Abstract

Mathematical models of freight cars, numerical methods, simulation results and some related topics such as safety, durability, wear are considered in this paper.

Keywords: freight car, three-piece bogie, railway vehicle dynamics, safety, wear

Introduction

Railway industry needs in accurate models of freight cars. It is a very important problem because this type of rail vehicles substantially defines operating costs of railway transport. It is also well known that freight cars with three-piece bogies, especially empty ones, sometimes show not really satisfactory safety factor. A significant part of actual researches is devoted to the dynamic analysis of new designed bogies and modifications of existing bogies.

All techniques, approaches, mathematical models and results that are considered in this paper are referred to Universal Mechanism software, (UM, www.umlab.ru) that has been developed in Bryansk State Technical University since the end of the eighties. One of the main applications of UM is the simulation of 3D dynamics of rail vehicles.

A CAE-based approach for stress load and durability analysis to predict the fatigue damage of parts of mechanical systems is described. The method predicts the fatigue strength of structural components of machines and mechanisms based on results of simulation of their dynamics taking into account real working conditions. Estimation of stress load and fatigue strength of long-wheelbase container flat-cars of a new type is discussed.

Wheel and rail profile wear is a serious and costly problem for a railway industry, especially for freight car traffic. The main purpose is the reduction of wheel and rail profile wear and the improvement of operational safety. The technique for solving such problems as well as profile selection policies and real engineering practices by using the multibody approach is described in this paper. An example of practical approach to profile evaluation is given. A scientific approach to profile evolution and optimization is described and some results which were obtained for a freight car with three-piece bogies are presented.

Model of three-piece bogie

Recent publications concerning to the simulation of dynamics of three-piece bogies are generally focused on models of a wedge frictional system [1-3]. The considered models usually treat appreciable simplifications. Inertia properties of wedges are ignored as well as the linear model of tangential contact forces is used in [3]. Mathematical models in [1, 2] consider mass and moments of inertia for frictional wedges with 4 d.o.f.

A freight car with three-piece bogies is one of the most difficult models of rail vehicles developed with the help of UM, Figure 1. Its difficulty is caused by the fact that almost all interactions between bodies in the model are contacts with friction.

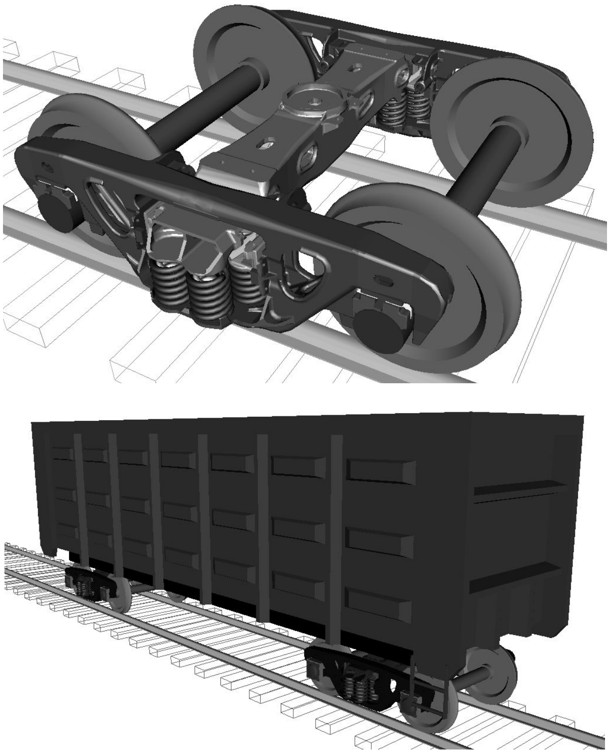


Figure 1. UM model of three piece bogie and freight car

To make the model more universal and accurate, frictional wedges are presented by rigid bodies with six degrees of freedom each. 16 points describe contacts of each wedge with the bolster and the side frame. Parameterized coordinates of contact points allow usage of the model for wedges with different shapes, Figure 2.

All rest bodies such as the bolster, side frames and wheelsets have six degrees of freedom. Interactions between frictional wedges and the bolster, wedges and side frames, bolsters and the car body, side frames and axle boxes are modeled as contact forces with taking into account real clearances between parts.

Thus, the model of a typical freight car contains 19 rigid bodies and has 114 degrees of freedom. Considering a mechanism of auto-coupling for creation of trains and tractive connections adds 3 bodies, 8 degrees of freedom and about a dozen of contact points.

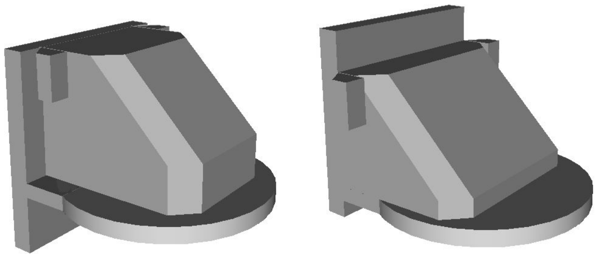


Figure 2. Contact wedges

Compliant contact model of point-plane type is used for mathematical models of the three-piece bogie. Mathematical model of contact forces is based on ideas from [4] and considered in details in [5].

Numerous contacts make the equations of motion of the model stiff. For example, the contacts of a wedge introduce frequencies about 1000 Hz in the model. Implicit Park method accompanied with computation of the approximate Jacobian matrices for contact forces implemented in UM make simulation of the freight car model fast enough for practical goals, [6-8]. Evaluation of Jacobian matrices by finite differences is a very CPU time-consuming operation, even if it can be computed not on every simulation step. An alternative method is proposed in [7, 8]. The method consists in evaluation of approximate Jacobian matrix based on simplified analytic expressions for local Jacobian matrices of stiff forces. Using Jacobian matrices is a key element of a computation procedure that make possible to obtain relatively big step-size and as a result leads to extremely fast simulation of freight car dynamics.

Rail-wheel contact forces

To obtain adequate and reliable results with the help of the considered above model of the three-piece bogie and to get all of advantages of such highly detailed model it needs that all other components of mathematical model of the freight car as a railway vehicle should be accurate enough.

A rail is considered as a massless force element. This means, both stiffness and damping of the rail is taken into account, but not the inertia properties. Generalized coordinates are not introduced for the rail and its lateral and vertical deformations must be computed from the equilibrium equations, Figure 3.

The following assumptions take place:

- rail deformations for different wheelsets are independent and can be computed separately;
- deformations of the left and right rails are independent;
- rail deformations include independent lateral and vertical deformations, Figure 2, which are parallel to the corresponding system of coordinates of the track;
- the rail as a linear force element both in the lateral and vertical directions.

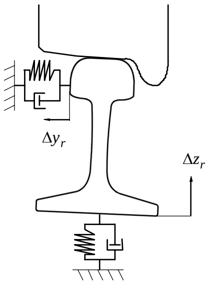


Figure 3. Rail as massless force element

Hertzian solution and FASTSIM algorithm by Kalker from [9] as well as modified non-elliptic multipoint contact model by Kik and Piotrowski from [10, 11] are used, Figures 4, 5.

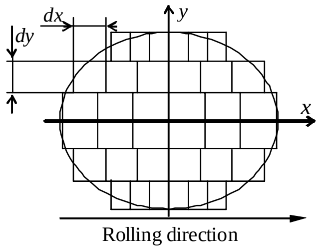


Figure 4. Elliptical contact patch for FASTSIM model

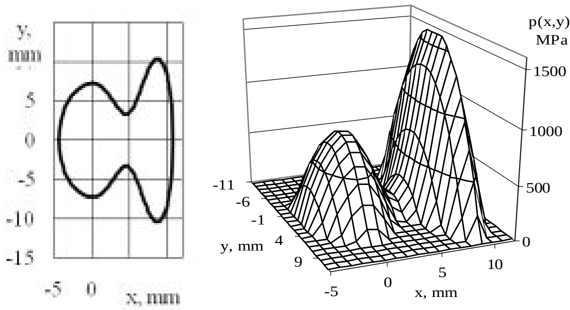


Figure 5. Non-elliptical contact patch and normal pressure distribution

Prediction of wheel and rail profile wear

Increasing wheel and rail profile wear now is a very serious problem especially for freight railway traffic. To compute profile wear in UM, four wear intensity models together with the models of freight car and wheel-rail contact forces described above are used. These wear intensity models are:

1. Archard model [12]. The model is based on the hypothesis of the linear dependence between the volume wear I and the work of creep forces A:

$$I = k_V \, A, \qquad (1)$$

where I is the volume wear, m^3;

k_V is the factor of the volume wear, m^3/J;
A is the work of friction, J.

2. Specht model [13]. This model also uses the linear dependence between the volume wear and the work, but it is assumed that there exist two areas within the contact patch: the areas of moderate and intense wear with different values of wear coefficients:

$$\begin{cases} I = k_V\, A, & w < w_{cr}, \\ I = k_V\, \alpha\, A, & w \ge w_{cr}, \end{cases} \tag{2}$$

where w is the friction power, W/m^2;
w_{cr} is the critical power, W/m^2,
α is the jump coefficient.

3. VNIIZhT-1 model [14] proposed by specialists of All-Russian Railway Research Institute (VNIIZhT) is

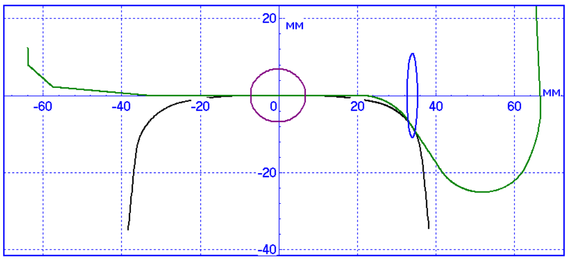
$$I = k\xi^2 p\,, \tag{3}$$

where p is the pressure in the contact patch,
ξ is the full creepage.

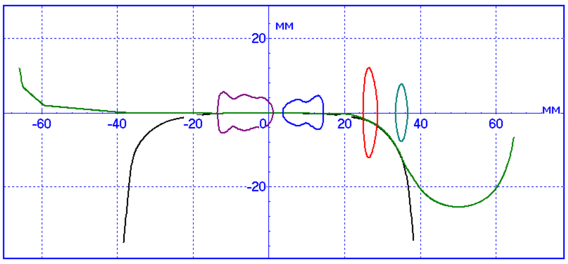
4. Plasticity model [14] takes into account the plasticity:

$$I = k \cdot \xi^2 \cdot p^* \cdot \min\left(tg(p / p^*), 1.5\right), \tag{4}$$

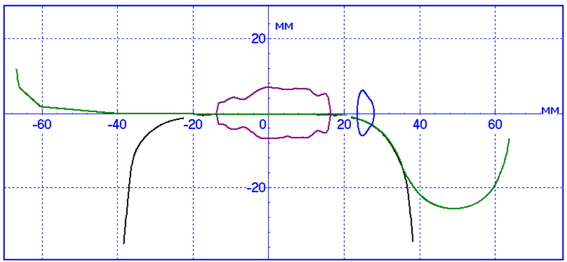
where p^* is the critical pressure.



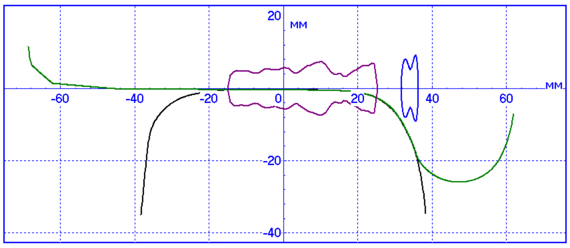
Initial profiles



Profiles after 20th iteration



Profiles after 40th iteration



Profiles after 60th iteration

Figure 6. Wheel and rail contact patches

The prediction of wheel and rail profile wear in UM is organized on the base of the scanning tool supplemented by the *evolution* concept. The evolution is series of multivariant calculations of identical structure (iterations) which differ in external conditions. In case of prediction of wheel profile wear the external conditions are wheel profile curves which are changed at the end of every multivariant calculation (iteration) in compliance with the tribological wear model. One of the wear problems solved by using UM was the simulation of mutual wheel and rail profiles for freight car in a curve R = 300 m. The total number of iterations is 80. The maximal profile material removal for every iteration is 0.1 mm. In Figure 6, the typical contact patches on 1, 20, 40, 60 iterations calculated with the help of the non-elliptical contact model are shown.

One can see that for the initial profiles the contact patches are very close to elliptical. In the sequel contact patches differ from elliptical ones more and more and contact becomes more mutual.

Another example presented in Figure 7, is the simulation of rail profile wear in a curve R = 300 m.

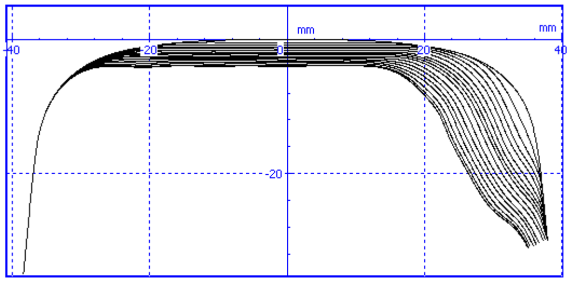


Figure 7. Worn rail profiles

Three-piece bogie model: experimental verification

In order to verify developed computer model of the three-piece bogie, a series of full scale experiments were done by the biggest Russian producer of freight cars JSC Uralvagonzavod (Nizhniy Tagil, Russia). The detailed report with comparison of experimental and simulation results is published in [15]. Taking into account small number of printed copies and Russian language of original publication let us repeat here some key results. All materials are cited with kind permission of the authors of [15].

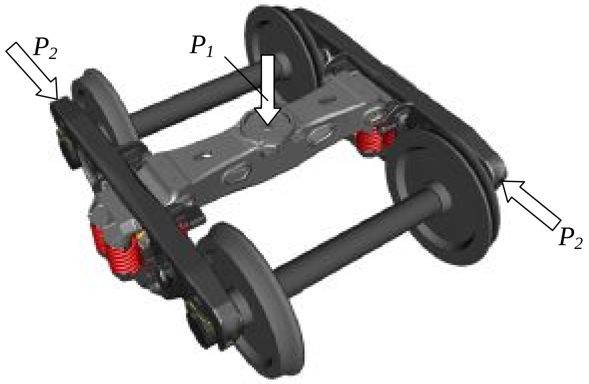
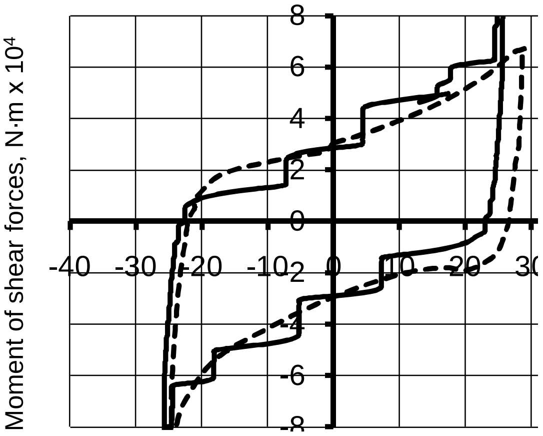


Figure 8. Experiment outline

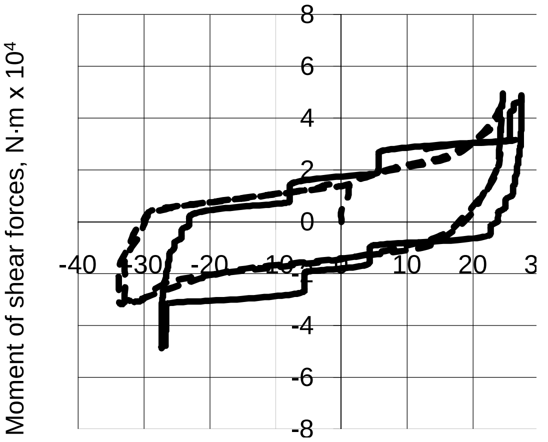
The description of experiments is given below. A three-piece bogie is loaded with the vertical force P_1

that is applied to the central plate and two longitudinal shear forces P_2, see Figure 8. Vertical force was applied through the intermediate bearing that allows to avoid frictional torques while yaw of the bolster. Firstly the vertical force P_1 was applied and after that the longitudinal forces were applied. As a result the yaw angle of the bolster vs. moment of longitudinal forces was registered.

Experimental measurements and results of computer simulation varying vertical load are presented in Figure 9. Comparison of obtained results shows good qualitative and quantitative agreement between experimental and simulation results. Maximal relative error between experimental and simulation results in terms of area of a hysteresis loop is 12%.



a) Vertical load P_1 = 45 kN



b) Vertical load P_1 = 20 kN

Figure 9. Yaw angle of bolster vs. moment of shear forces; experimental (dashed lines) and simulation (solid lines) results

One more series of numerical experiments for checking the dependence between normal forces between frictional wedges and side frames and vertical deflection of the central suspension were performed. Longitudinal shear forces P_2 were set to zero so the only slow harmonic vertical force P_1 was applied to the bogie, see Figure 8.

Comparison of experimental and numerical results shows again a good agreement between them, Figure 10. Maximal relative error between experimental and numerical results is about 16%.

So the obtained experimental results for two different experiments are in quite good agreement with results of computer simulation of the developed model of three-piece bogie.

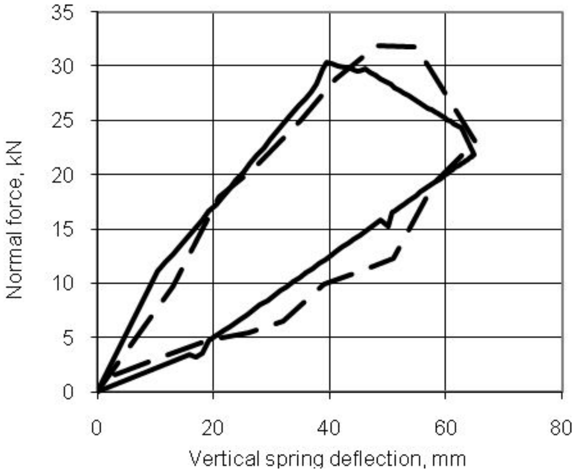
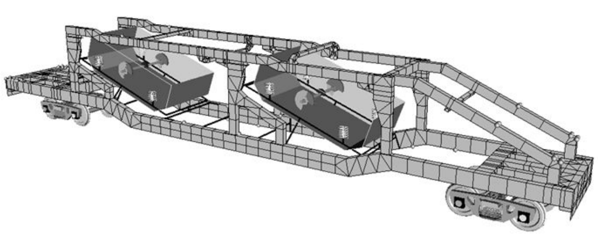


Figure 10. Vertical spring deflection vs. normal force on a wedge; experimental (dashed lines) and simulation (solid lines) results

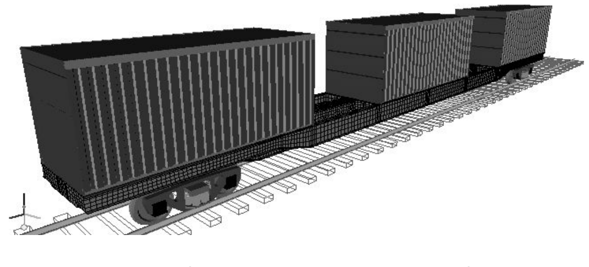
Freight car model with flexible bodies

UM software gives the user a possibility to introduce into a mechanical system flexible bodies, which are undergone large reference displacement but small deformation [16].

This approach enables to consider dynamics of flexible bodies. Such an approach can be useful for stress load and durability analysis as well as for taking into account dynamics of flexible bodies itself. It is quite important issue for example for such systems that are depicted in Figure 11.



a) Model of ballast cleaning vehicle with flexible frame



b) Model of long-wheelbase wagon-platform with flexible frame

Figure 11. Models of wagons with flexible bodies

Long frames of a ballast cleaner and a long-wheelbase flat car have relatively low first eigen frequencies that makes considering frames as flexible bodies to be required. For example it can be used for the analysis of vibration of frames during the motion of a railway vehicle with respect to the influence of railway track irregularities and power generating set, see Figure 11.

Flexible bodies are introduced with the help of finite-element models, see Figure 12. Modal formulation is used for obtaining flexible deformation, stresses and strains of flexible bodies. UM software supports import of finite-element models from ANSYS and NASTRAN software. The approach is used successfully for simulation of freight car dynamics as well as stress load and durability analysis.

Figure 12. Flexible side frame of three-piece boogie

Strain/stress state and durability analysis

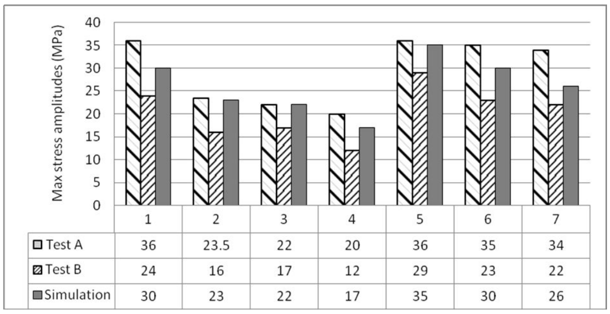
Simulation of dynamics of hybrid systems that include both rigid and flexible bodies gives a researcher a possibility to obtain stress or strain time histories in any node of finite-element mesh of flexible bodies. Such stress time histories are the input data for posterior stress load and durability analysis. This feature is useful for development of durable constructions from the point of view of real working conditions on early design stages.

Maximum stress amplitudes for several nodes on the frame of the long-wheelbase flat car (see Figures 11b) for two experimental and one numerical runs are given in Figure 13. The comparison of experimental and numerical results shows very good agreement between them. It is clearly seen that results of numerical simulation lie within the range between results of two experimental runs.

Two basic inputs are necessary for fatigue strength estimation: stress loading data and fatigue resistance properties described in terms of applicable method. UM software includes tools for stress loading analysis based on simulation results and further fatigue analysis.

Let us consider durability analysis for a frame of long-wheelbase flat car, Figure 14. Series of numerical experiments were used for the investigation of various loading cases describing working conditions of the vehicle. Simulation results are used for stress loading and S-N durability analysis resulted in qualitative and quantitative estimations of loading parameters and vehicle strength.

Comparison of experimental and numerical results of durability analysis, which was fulfilled for the frame of the long-wheelbase flat car, showed a good qualitative agreement between them [17].



	1	2	3	4	5	6	7
☐ Test A	36	23.5	22	20	36	35	34
▨ Test B	24	16	17	12	29	23	22
■ Simulation	30	23	22	17	35	30	26

Figure 13. Comparison of experimental and simulation results

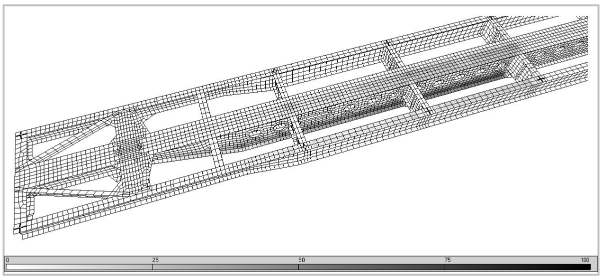


Figure 14. Computed maximal stress cycle amplitude distribution for frame of long-wheelbase flat car

Results and Discussion

The developed model includes frictional wedges as rigid bodies with six degrees of freedom. Contact interactions between bodies of the bogie are modeled with the help of point-plane contact model that calculates viscous-elastic normal force and Coulomb friction force in both sticking and sliding modes. Lots of contact forces lead to stiff equations of motion but used Park solver along with Jacobian matrices for contact forces made simulation of the model fast enough.

The developed model is fully parameterized and can be easily adjusted for any specific three-piece bogie modification.

Such detailed model of the three-piece bogie along with such UM tools as import of flexible bodies, wear, stress load and durability prediction gives a researcher a fast, powerful and proved tool for solving various problems of railway industry.

Conclusions

The detailed model of the three-piece bogie is developed. Two different full scale experiments with the bogie were performed. Comparison of experimental and simulation results showed quite good agreement between them and proved adequacy and accuracy of suggested model.

Acknowledgment

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