Experimentální Analýza Napětí 2007

COMPUTER SIMULATIONS OF THE COAL WAGON LABORATORY KINEMATIC EXCITATION

Pavel Polach,[1] Michal Hajžman,[2] Jaroslav Václavík,[3] Jan Chvojan[4]

Abstract: *Using SIMPACK simulation tool multibody models of the MGR Coal Hopper HAA two-axle open coal wagon were created. They are intended for the laboratory tests simulations. Multibody models of an empty wagon and three variants of a partly loaded one, which correspond to the wagon loading by concrete panels during testing in the Accredited Dynamic Testing Laboratory ŠKODA VÝZKUM s.r.o test stand, were created. With multibody models it is possible to simulate ten variants of the laboratory kinematic excitation of wagon wheels by a wideband sweep signal in vertical direction, which correspond to real loading states performed on the test stand. Time histories or frequency domain responses of kinematic and dynamic quantities reflecting the wagon examined properties are the output of the computer simulations and the experimental measurements. The aim of the laboratory tests and computer simulations is especially the evaluation of the impact of using two-leaf composite springs of the wagon suspension instead of five-leaf parabolic steel ones on the monitored kinematic and dynamic quantities.*

Keywords: *coal wagon, laboratory tests, computer simulations, experiment*

1. Introduction

Computer simulations of mechanical systems should be performed hand in hand with experimental measurements on real subjects. This paper is intended to connect the numerical and experimental investigations in the field of rail vehicle dynamics. It is focused on the kinematic excitation of a coal wagon on a test stand. In comparison with [7] another simulation tool and another approach to the wagon modelling were used.

The rail vehicle will be considered a multibody system consisted of rigid bodies coupled by kinematic joints. This is a common approach in multibody dynamics when the dynamic behaviour is influenced mainly by suspension elements. Such rail vehicle models can be used for studying dynamics of complex vehicles in different driving situations or for various laboratory excitations.

The aim of the presented work was the investigation of the dynamic properties of the two types of leaf springs mounted on the MGR Coal Hopper HAA wagon (Fig. 1). The standardly utilized type of leaf springs is a parabolic steel one (see Fig. 6). These springs have some undesirable properties such as corrosion of leaves and silting of an inter-leaf space. Therefore it can be efficient to substitute them by composite leaf springs (see Fig. 6) of better properties. The aim of the numerical simulations and experimental works is the comparison of the steel

[1] Dr. Ing. Pavel Polach, ŠKODA VÝZKUM s.r.o., Tylova 1/57, 316 00 Plzeň, tel.: +420 378182246, e-mail: pavel.polach@skodavyzkum.cz

[2] Ing. Michal Hajžman, Ph.D., ŠKODA VÝZKUM s.r.o., Tylova 1/57, 316 00 Plzeň, tel.: +420 378182268, e-mail: michal.hajzman@skodavyzkum.cz

[3] Ing. Jaroslav Václavík, ŠKODA VÝZKUM s.r.o., Tylova 1/57, 316 00 Plzeň, tel.: +420 378182193, e-mail: jaroslav.vaclavik@skodavyzkum.cz

[4] Ing. Jan Chvojan, ŠKODA VÝZKUM s.r.o., Tylova 1/57, 316 00 Plzeň, tel.: +420 378182162, e-mail: jan.chvojan@skodavyzkum.cz

parabolic leaf spring and the composite GRP leaf spring. However the comparison based on the computer simulations is influenced by the proper computer model that has to be verified in the course of its development. This paper shows the verification of the coal wagon multibody models created in SIMPACK simulation tool on the basis of the experimental laboratory measurements on the test stand.

Fig. 1. The MGR Coal Hopper HAA wagon on the test stand in the Dynamic Testing Laboratory ŠKODA VÝZKUM s.r.o.

2. SIMPACK software

SIMPACK simulation tool [9] is being developed in INTEC GmbH, Weßling, Germany. Similarly as other MBS software it is intended for investigating kinematic and dynamic properties of a nonlinear three-dimensional coupled mechanical system consisted of many bodies. The approach to solving the tasks in the field of mechanics using computer models, which is based on the systems of bodies, enables to solve substantially more general problems than the approach based on the finite element method because it is not dependent on the continual model of the investigated system. As a consequence of a greater generality of this approach and of the character of the studied mechanical systems demands for the computing time of the solution of the nonlinear equations system are growing. When creating a multibody model it is necessary to pay attention to choosing the number of bodies, the number of kinematic pairs and especially the total number of degrees of freedom in kinematic pairs of a mechanical system, i.e. to optimally interpret the physical substance of the solved problem. The total number of degrees of freedom in kinematic pairs determines the number of constructed nonlinear equations of motion, solution of which should be within a real period of time.

Multibody models are created by a finite number of bodies connected by kinematic pairs and massless force elements, which enable to model spring-damper structural parts. With respect to the multibody models creating methodology and automatic generating of the differential equations in SIMPACK simulation tool kinematic pairs are classified into two types (two separate groups within the framework of modelling in SIMPACK simulation tool) – *joints* and *constraints*. Exactly one joint with a given number of degrees of freedom belongs to each body, which enables a body motion considering the previous body in a kinematic chain. Constraints are utilized for the closing of kinematic chains, i.e. for creating kinematic loops, and constraining the relevant degree of freedom. Bodies can move in space in the framework of joints, constraints, force elements, the way of coupling to the reference frame and boundary conditions. Each body is defined by inertial properties (mass, centre of mass coordinates and moments of inertia). It is possible to bind different *markers* to the bodies. A marker is a point, in which a local coordinate system is defined. Markers can be used to locate reference frames, to define the centre of mass. Through the markers it is possible to couple bodies by joints, constraints and force elements, it is possible to act on bodies by applied forces and torques, etc. After creating a multibody model it is possible to simulate the modelled system motion. In simulating motion with multibody models in the MBS software non-linear equations of motion are generated. The equations are solved by means of numerical time integration. Generally, displacements, velocities and accelerations of the individual bodies, forces and torques acting in kinematic pairs and force elements are the monitored quantities. It is possible to obtain results in the form of time series, in the form of graphs or in the form of multibody model visualization (static or with animation). In outputs in the form of graphs it is possible to compare e.g. influences of changes of various parameters of the multibody model on the simulations results, it means operatively evaluate the influences of permitted design adjustment to the desired kinematic and dynamic properties of the real structure.

Besides the basic SIMPACK Kinematics & Dynamics module it is possible to buy additional SIMPACK simulation tool modules and data interfaces with other software. In ŠKODA VÝZKUM s.r.o. the SIMPACK Automotive+ module (support of road vehicles modelling including tire models), the SIMPACK Wheel/Rail module (support of rail vehicles modelling including wheel-rail contact models) and the SIMPACK Contact module (support of contacts between bodies modelling) are at disposal.

3. Multibody model description

The multibody models of the MGR Coal Hopper HAA goods wagon were created mainly on the basis of the report [6] and publications [8], [3]. The x-axis of the reference coordinate system is defined parallel to the direction of the wagon motion (driving direction), the z-axis is downwards and y-axis steers to the right (see Fig. 2).

The model consists of four bodies (including frame) connected by three kinematic joints (see Fig. 3). Considering the aim of the modelling, the wagon body can be represented by one rigid body that has six degrees of freedom with respect to the frame (BUNC, unconstrained joint). The laboratory stand is considered to be the rigid reference frame. The front and rear wheelsets are connected with the frame using special user defined joint (USR) that allows the rotation about the x-axis and translation along the z-axis. The wagon body and the wheelsets are mutually connected by four leaf springs (two leaf springs between the wagon body and the front wheelset and two leaf springs between the wagon body and the rear wheelset), which are modelled by component force elements. The leaf springs are complex spatial suspension elements and they can transmit forces and torques in all three directions.

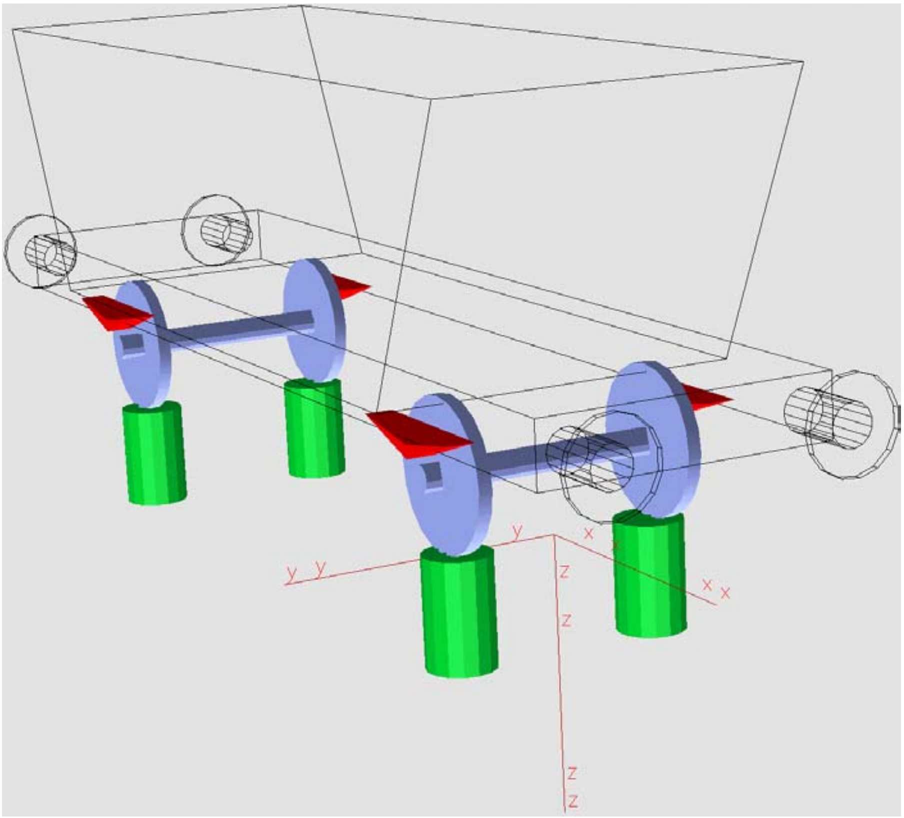

Fig. 2. Visualization of the MGR Coal Hopper HAA wagon multibody model in SIMPACK simulation tool including the reference coordinate system

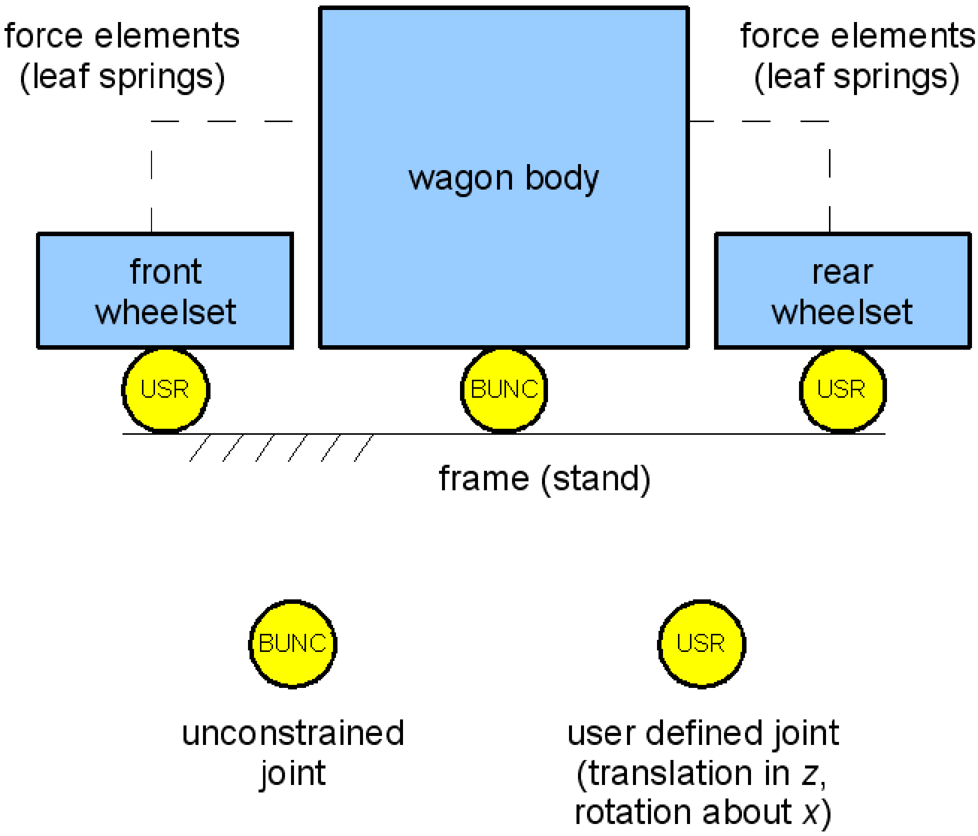

Fig. 3. Kinematic scheme of the MGR Coal Hopper HAA wagon multibody model

Multibody models of an empty wagon and three variants of a partly loaded one, which correspond to the wagon loading by concrete panels during testing in the Accredited Dynamic Testing Laboratory ŠKODA VÝZKUM s.r.o test stand (see chapter 4), were created. With multibody models it is possible to simulate ten variants of the laboratory kinematic excitation of the wagon wheels by a wideband sweep signal in vertical direction, which correspond to the real loading states performed on the test stand (see Tab. 1).

The behaviour of the leaf springs is very complex. It is influenced by the flexibility of leaves, by contact and friction forces between leaves and by other operational conditions. Various types of the leaf spring models in the framework of multibody models are described in [4]. The main characteristic of the leaf springs is the force-deformation characteristic in vertical direction (z-axis). This static characteristic was measured and nonlinear curves characterized by hysteresis were obtained (see Fig. 4). These curves were processed in order to identify the simple nonlinear curves describing leaf springs vertical behaviour. The horizontal (in lateral and longitudinal directions) characteristics of the leaf springs were estimated on the basis of numerical tests and computational experience and linear stiffnesses were used.

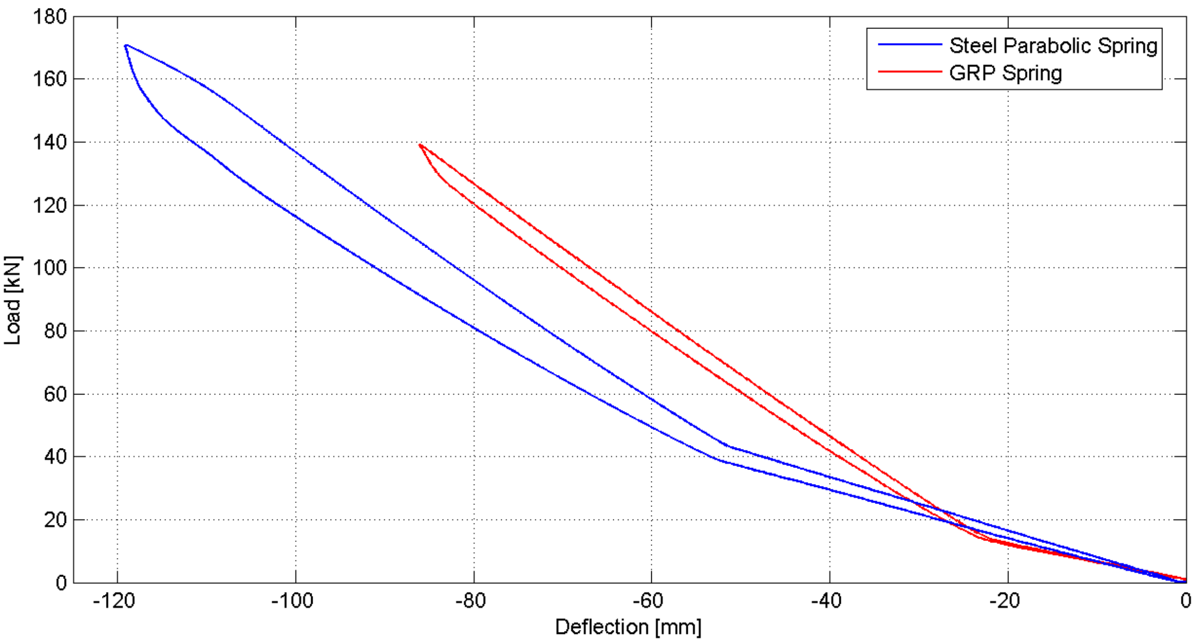

Fig. 4. Vertical static characteristics of both parabolic steel five-leaf and two-leaf composite springs

The wheels of the wagon were mounted on the hydraulic servo valves of the test stand that can experimentally simulate vertical kinematic excitation of the wagon. The contact between the wheels and the loading servo valves was modelled by a contact force with defined stiffness and damping.

4. Experimental laboratory tests

The possibility of variable changing in some vehicle parameters, which can influence their dynamic properties, and the possibility of choosing various loading modes are the advantage of vehicles verification in laboratories. In case of the wagon the parameters are e.g. wagon load, suspension elements change etc., the loading modes can simulate a different quality of railway line, driving speed etc.

The MGR Coal Hopper HAA goods wagon was tested empty and partly loaded three times in three variants in the laboratory. Loading was realized by concrete panels. The empty wagon total mass was 13 967 kg, 1st load variant wagon mass was 22 846 kg, 2nd load variant wagon mass was 31 562 kg and 3rd load variant wagon mass was 39 839 kg. Wagon dimensional drawing (3rd load variant) is in Fig. 5.

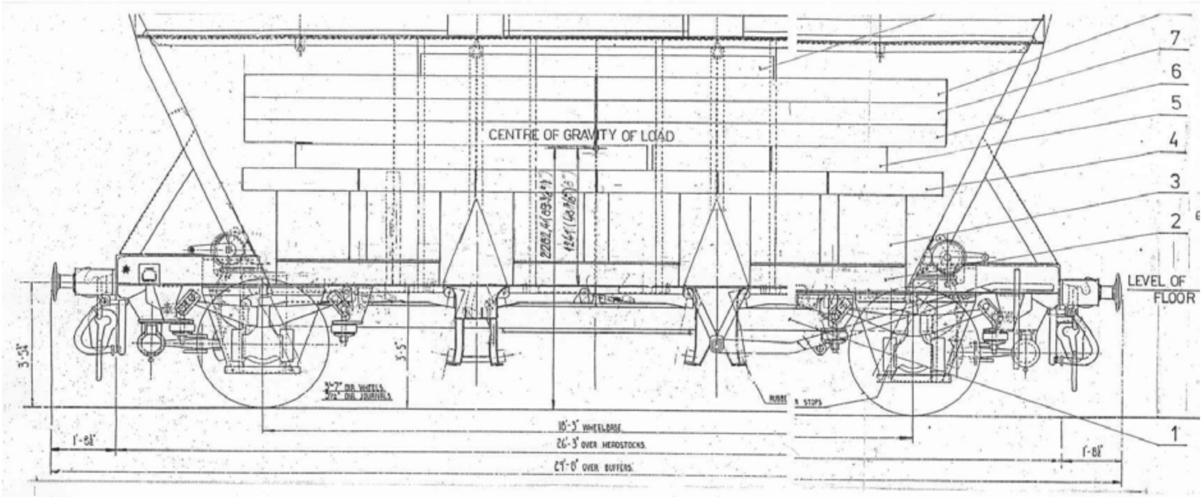

Fig. 5. Dimensional drawing of the loaded wagon (taken from [8])

The five-leaf parabolic steel springs and the two-leaf composite springs (see Fig. 6) were successively used in the wagon suspension.

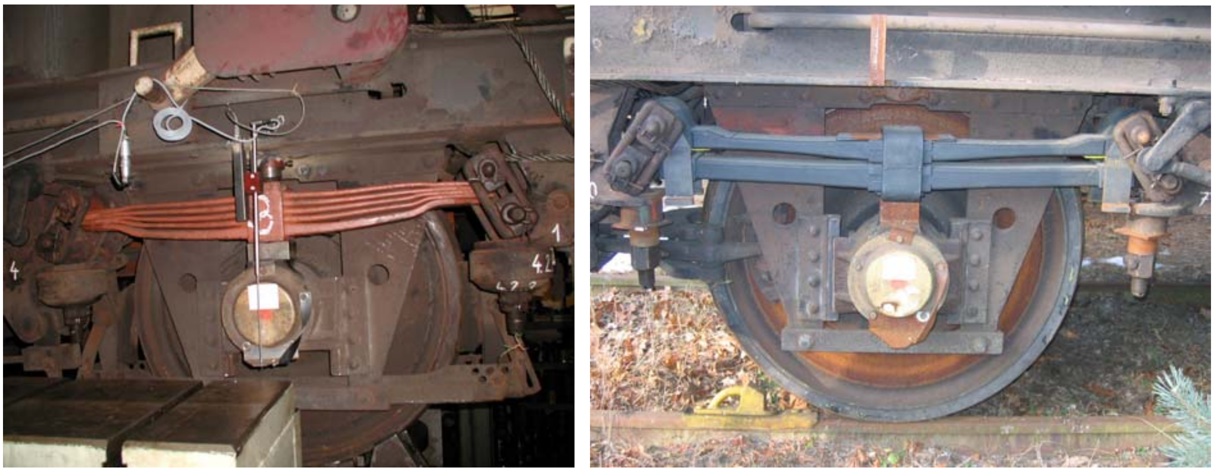

Fig. 6. Five-leaf parabolic steel spring and two-leaf composite spring

Kinematic and dynamic properties of the MGR Coal Hopper HAA goods wagon were investigated on the computer controlled Instron Schenck electrohydraulic loading system. The test stand base was the multi-purpose modular clamping and support Schenck 4000 system. The wagon wheels were placed on the units, which prevented the wagon from both lateral and longitudinal movement; the wheels brakes were taken off during the testing. The wagon front wheels were excited kinematically by the Schenck PL 630 kN loading servo valves with three-stage valves, the rear wheels by the Schenck PL 400 kN loading servo valves with two-stage valves. Movement of the loading servo valves was controlled by the Schenck S59 four-channel digital system.

The wagon was subjected to several loading modes on the test stand. The wagon natural frequencies and natural mode shapes were identified during the kinematic excitation of the wheels by a wideband sweep signal in vertical direction (“sweep test”). The wheels were excited by loading servo valves in phase (“bump test”) or out of phase (“roll test”). The wagon stability was examined at front wheels excitation by „sweep“ (in phase) in vertical direction (“cyclic top test”); the excitation signal simulated the wagon running over the unevenness railway line, or rail joining. The list of the chosen loading modes and their parameters are given in Tab. 1.

Time history of vertical displacements $y(t)$ of the wagon wheels kinematically excited by a wideband sweep signal (wideband sweep signal is especially appropriate to the nonlinearity study) on the test stand can be described using relation

$$y(t) = A \cdot \sin(\omega(t) \cdot t) \quad , \tag{1}$$

where A is an amplitude of the vertical displacements, $\omega(t)$ is time depending linearly variable angular frequency, for which is valid (e.g. [1])

$$\omega(t) = 2 \cdot \pi \cdot f(t) \quad , \tag{2}$$

where $f(t)$ is time depending linearly variable frequency, which can be formularized by relation $f(t) = k \cdot t$ (k is a constant). Then the time history of vertical displacements of kinematically excited wheels can be determined using relation

$$y(t) = A \cdot \sin(2 \cdot \pi \cdot k \cdot t^2) \quad . \tag{3}$$

Kinematic excitation of wagon wheels		
Loading mode	**Parameters of wideband sweep signal**	
“bump test”	A = 1 mm, range of excitation frequencies f from 0 Hz to 30 Hz, excitation of the front wheelset wheels in phase	$k = 0.2$ Hz·s^{-1}
		$k = 0.03$ Hz·s^{-1}
“bump test”	A = 0.5 mm, range of excitation frequencies f from 0 Hz to 30 Hz, excitation of the front wheelset wheels in phase	$k = 0.2$ Hz·s^{-1}
		$k = 0.03$ Hz·s^{-1}
“roll test”	A = 0.5 mm, range of excitation frequencies f from 0 Hz to 30 Hz, excitation of the front wheelset wheels out of phase	$k = 0.2$ Hz·s^{-1}
		$k = 0.03$ Hz·s^{-1}
“cyclic top test”	A = 6 mm, range of excitation frequencies f from 0 Hz to 7 Hz, excitation of the front wheelset wheels in phase	$k = 0.2$ Hz·s^{-1}
		$k = 0.03$ Hz·s^{-1}
“cyclic top test”	A = 2 mm, range of excitation frequencies f from 0 Hz to 7 Hz, excitation of the front wheelset wheels in phase	$k = 0.2$ Hz·s^{-1}
		$k = 0.03$ Hz·s^{-1}

Tab. 1. List of the selected loading modes

Measured (and documented) kinematic and dynamic quantities were relative displacements (DW) and accelerations (VAP) of loading servo valves, forces acting between the servo valves and the wagon wheels (LW), accelerations of the wheels in contact points with the servo valves (VAW), relative displacements between the wheels and the wagon body (DS), acceleration of the center of the front wheelset axle (VAA) and accelerations on six points of the wagon body (VACH, BACH) – see Fig. 7 and Fig. 8. In addition, two acoustic microphones monitored the noise (AN) – see Fig. 8.

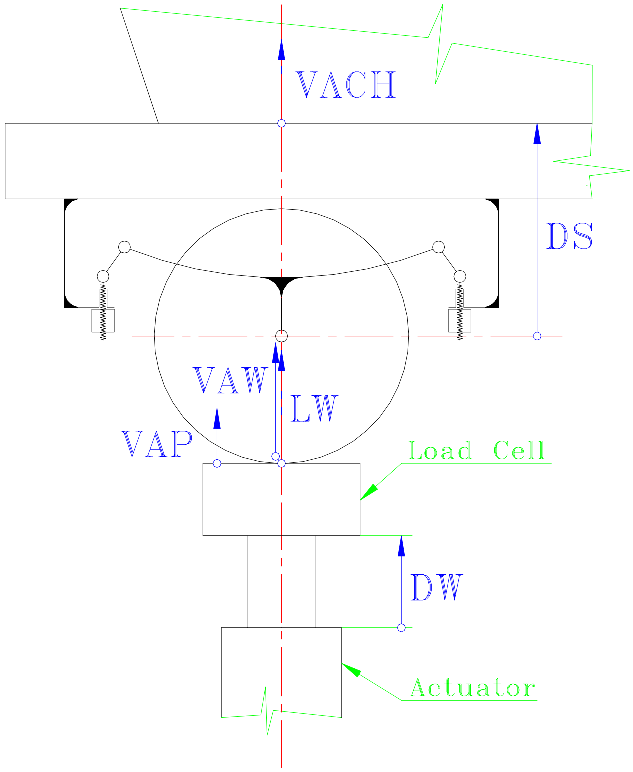

Fig. 7. Distribution of measuring sensors in one wheel scheme (taken from [8])

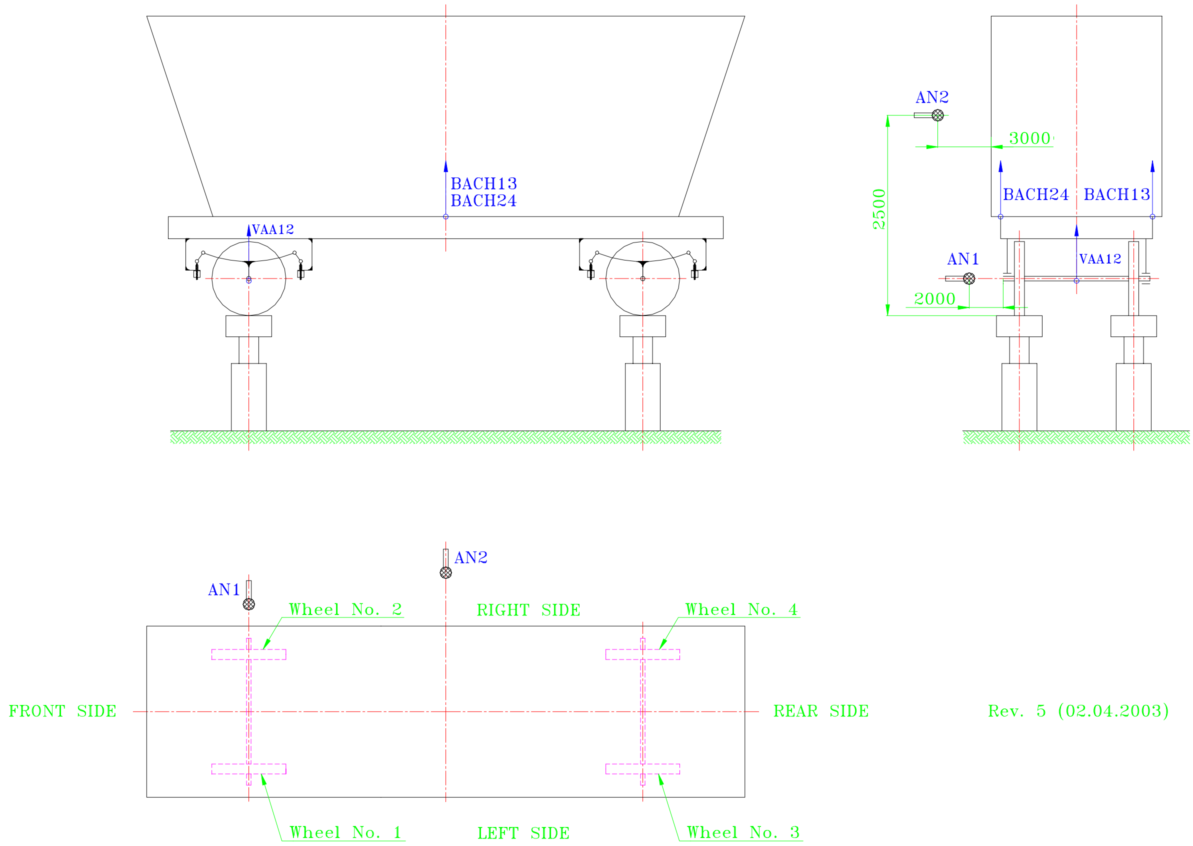

Fig. 8. Distribution of measuring sensors and wheels designation schemes (taken from [8])

5. Results

Simulations of all the loading modes on the test stand (see Tab. 1) and research computations were performed with the multibody models of the MGR Coal Hopper HAA goods wagon.

All the kinematic and dynamic quantities measured in the laboratory were compared with the corresponding quantities calculated at computer simulations. In this paper only the results of the "bump test" with the empty wagon at vertical displacement amplitudes on front wheels of $A = 0.5$ mm are given. Pieces of knowledge following from that laboratory test and its computer simulation can be generalized for all the loading modes:

1. The resonant frequencies identified from the laboratory tests records (in case of the empty wagon with five-leaf parabolic steel springs of the front suspension and two-leaf composite springs of the rear suspension approx. 4 Hz – see Fig. 9) are higher than identified using computer simulations (in case of the empty wagon with five-leaf parabolic steel springs of the front suspension and two-leaf composite springs of the rear suspension approx. 3 Hz – see Fig. 10). But the coincidence of resonant frequencies is better than in [7].
2. The magnitudes of the monitored quantities amplitudes at resonant states measured at laboratory tests (in case of displacements between the front wheels and the wagon body of the empty wagon with five-leaf parabolic steel springs of the front suspension and two-leaf composite springs of the rear suspension approx. ±7 mm – see Fig. 9) are lower than determined at computer simulations (in case of the empty wagon with five-leaf parabolic steel springs of the front suspension and two-leaf composite springs of the rear suspension approx. ±8 mm – see Fig. 10). But coincidence of the magnitudes of the monitored quantities amplitudes at resonant states is better than in [7].
3. The character of the monitored quantities curves measured at the laboratory tests and determined at the computer simulations is similar.

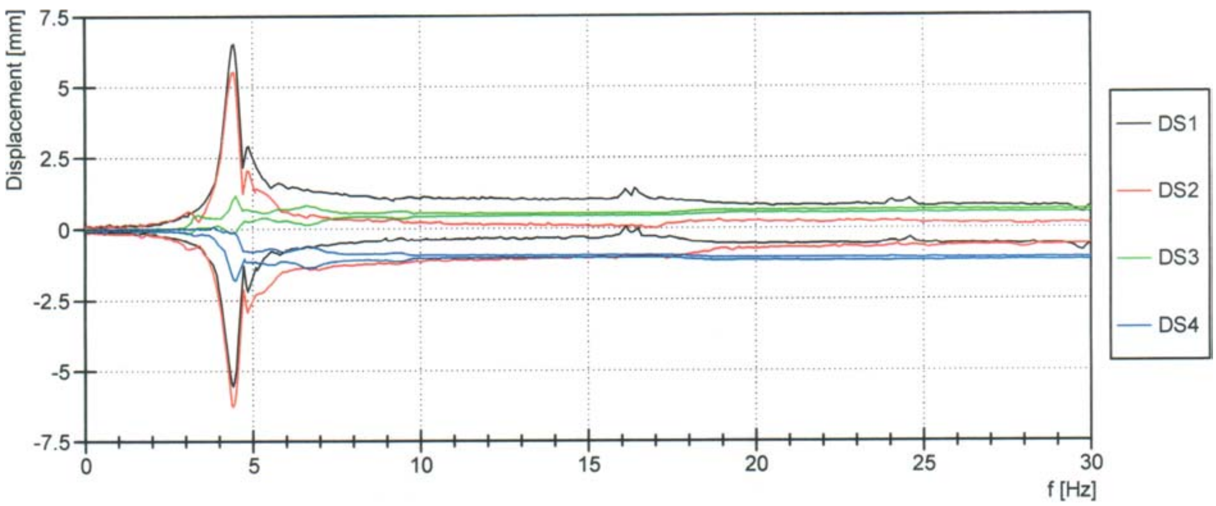

Fig. 9. Envelopes of the experimentally measured relative displacements between the wheels and the wagon body of the empty wagon (with five-leaf parabolic steel springs of the front suspension and two-leaf composite springs of the rear suspension) at "bump test" and vertical displacements amplitudes on front wheels of $A = 0.5$ mm (DS1 – front left wheel, DS2 – front right wheel, DS3 – rear left wheel, DS4 – rear right wheel)

On the basis of the research computations the leaf springs model (according to presumption in [7]):

1. The characteristics of leaf springs vertical stiffness used in the wagon multibody models were measured in the laboratory conditions at their static loading state, not at dynamic loadings states [2].
2. It is evident from the structural design of the five-leaf parabolic steel spring that friction forces act between the single spring leaves (at their deformation). Magnitudes of these forces were not experimentally measured.
3. Laboratory determined vertical damping coefficients of leaf springs evidently do not correspond to reality (damping influences magnitudes of the monitored quantities amplitudes at resonant states). In available literature there are shown the values of vertical damping of leaf springs of goods wagons higher than in [8] – e.g. ten times higher in [5].

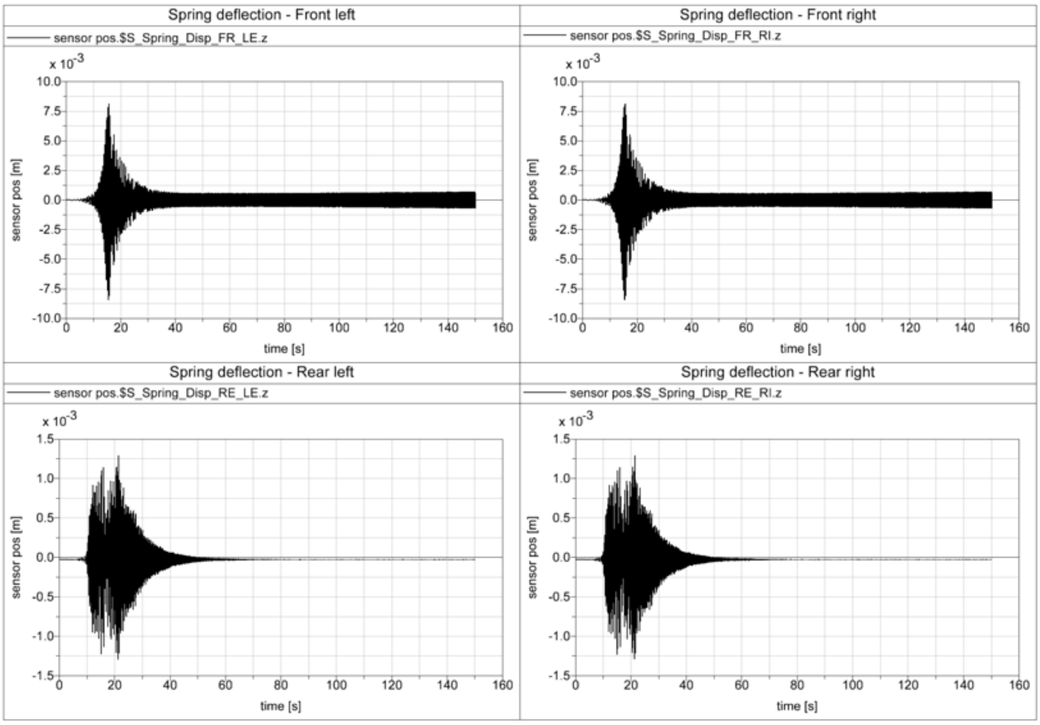

Fig. 10. Computed relative displacements between the wheels and the empty wagon body (with five-leaf parabolic steel springs of the front suspension and two-leaf composite springs of the rear suspension) at “bump test” and vertical displacements amplitudes on the front wheels of $A = 0.5$ mm

In order to improve the coincidence of the laboratory tests and the computer simulations results the values of vertical damping coefficients of leaf springs in the multibody models were increased. In Fig. 11 there are given computation results of relative displacements between the wheels and the empty wagon body (with five-leaf parabolic steel springs of the front suspension and two-leaf composite springs of the rear suspension) considering three times higher vertical damping coefficient of springs.

In order to investigate the sensitivity of horizontal (in lateral and longitudinal directions) stiffnesses and damping coefficients of leaf springs these stiffnesses were varied. The influence of these elements values was very small. For example, in Fig. 12 there are given research computation results of relative displacements between the wheels and the empty wagon body (with five-leaf parabolic steel springs of the front suspension and two-leaf composite springs of the rear suspension) considering proportionally lower horizontal stiffnesses and damping coefficients of leaf springs.

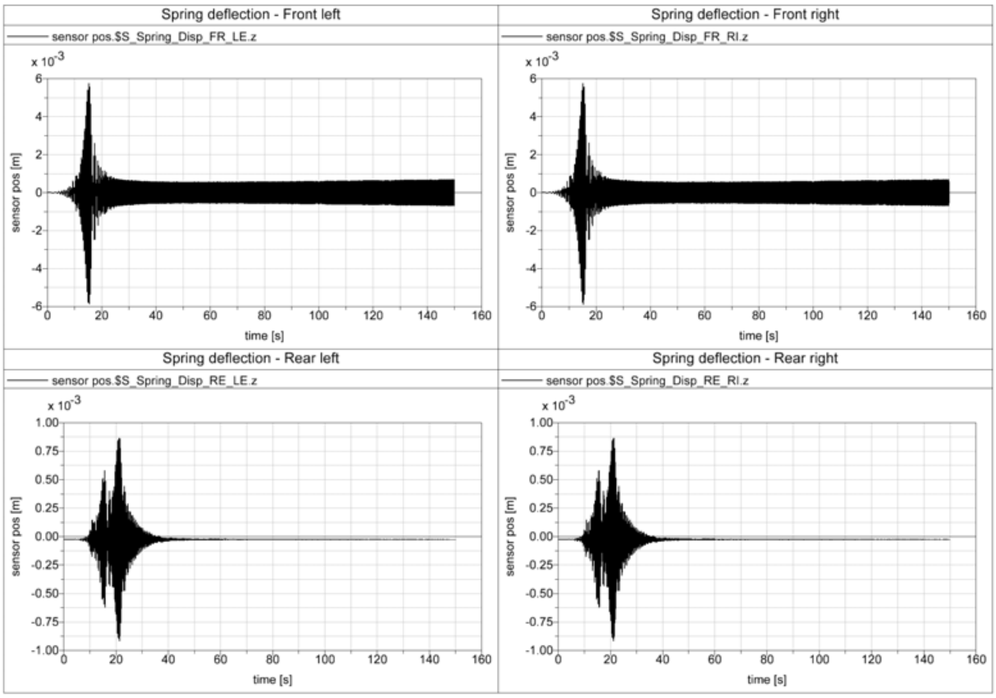

Fig. 11. Computed relative displacements between the wheels and the empty wagon body (with five-leaf parabolic steel springs of the front suspension and two-leaf composite springs of the rear suspension) at "bump test" and vertical displacements amplitudes on the front wheels of $A = 0.5$ mm – three times higher vertical damping coefficient of leaf springs

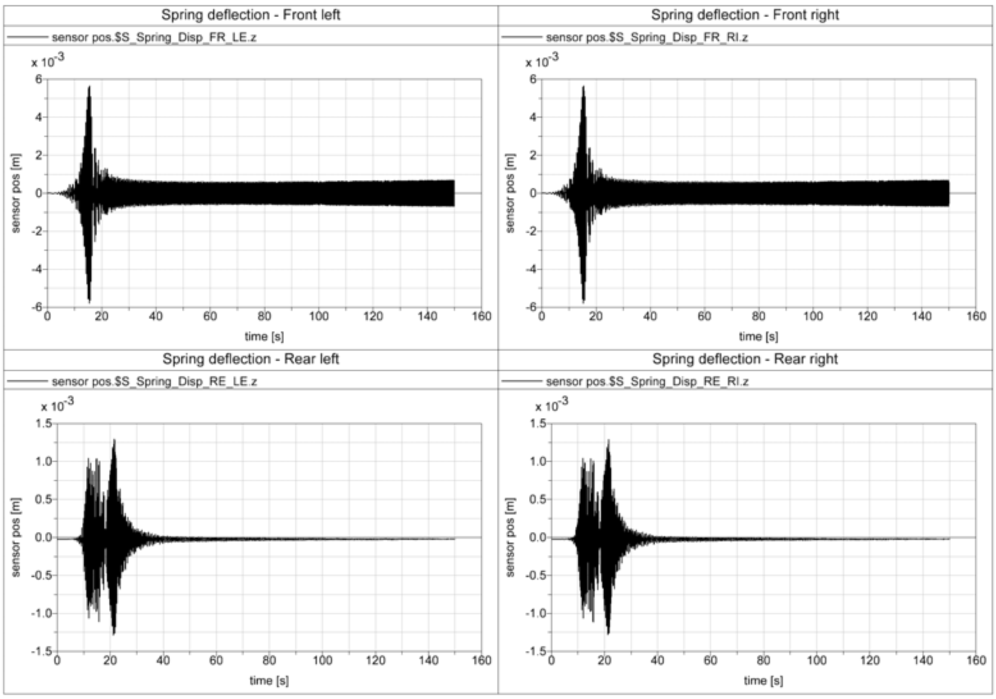

Fig. 12. Computed relative displacements between the wheels and the empty wagon body (with five-leaf parabolic steel springs of the front suspension and two-leaf composite springs of the rear suspension) at "bump test" and vertical displacements amplitudes on the front wheels of $A = 0.5$ mm – proportionally lower horizontal (in lateral and longitudinal directions) stiffnesses and damping coefficients of leaf springs

6. Conclusions

The influence of using the composite leaf springs instead of the original parabolic steel leaf springs on the kinematic and dynamic qualities of the MGR Coal Hopper HAA two-axle open goods wagon was examined during the laboratory tests on the test stand in the Accredited Dynamic Testing Laboratory ŠKODA VÝZKUM s.r.o. The multibody models of the empty wagon and three variants of partly loaded wagon, which correspond to the wagon loading by concrete panels at the laboratory tests, were created in SIMPACK simulation tool. It is possible to simulate ten variants of laboratory kinematic wheels excitation, which correspond to the loading modes realized on the test stand, with the existing multibody models.

The presented multibody models are the second approximation (the first approximation was presented in [7]) of the computational wagon models, intended for the kinematic and dynamic quantities investigation, to the real rail vehicle. Before the multibody models extension by the wheel-rail contact model, which enables to perform the simulations of the wagon driving on the railway, and by the improvement of the wheelset model, which enables to consider its elastic properties, in will be necessary to improve the leaf spring model first (in the way discussed in [4]).

Acknowledgements: The paper has originated in the framework of solving the project of the Ministry of Education, Youth and Sports of the Czech Republic 1M0519 "Research Centre of Rail Vehicles".

References

[1] Brousil, J., Slavík, J., Zeman, V.: *Dynamics*. SNTL, Prague, 1989. (in Czech)

[2] Černý, I.: *Static and fatigue tests of GRP spring leaves for rail wagons*. Technical Report SVÚM a.s., No. 40E148/1, Prague, 2005. (in Czech)

[3] Chvojan, J., Cherruault, J.Y., Kotas, M., Mayer, R., Gil Marcos, N.M., Václavík, J.: *Dynamic investigation on freight wagon suspensions*. Proceedings, 12th International Conference on Experimental Mechanics, 2004.

[4] Hajžman, M., Polach, P.: *Modelling of leaf springs in the framework of multibody models*. Proceedings, 9th International Scientific Conference Applied Mechanics 2007, 2007.

[5] Petersen, D.E., Hoffmann, M.: *Dry friction and impact dynamics in railway vehicles*. M.Sc.Eng. Thesis, Informatics and Mathematical Modelling, Technical University of Denmark, Lyngby, 2003.

[6] Polach, P.: *Multibody models of MGR Coal Hopper HAA goods wagon in* ***alaska*** *software intended for laboratory tests simulations*. Research Report ŠKODA VÝZKUM s.r.o., VYZ 0773/2005, Plzeň, 2005. (in Czech)

[7] Polach, P., Hajžman, M., Václavík, J., Chvojan, J.: *Computer simulations of coal wagon excitations on a test stand and comparison with experimental results*. Acta Mechanica Slovaca, EAN 2006, 1/2006, 10, 405-412. (in Czech)

[8] Václavík, J., Chvojan, J., Kotas, M.: *Interaction between coal wagon and rail at shaker rig at different performance conditions using UIC parabolic five leave steel suspensions*. Technical Report ŠKODA VÝZKUM s.r.o., VYZ 54/02/2003, Plzeň, 2003.

[9] *SIMPACK 8.800, User Manual*. INTEC GmbH, Weßling, 2006.