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TROLLEYBUS DYNAMIC RESPONSE AND IDENTIFICATION OF THE TIRE RADIAL PROPERTIES

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Summary: *This contribution deals with a methodology of tire radial properties identification. An experimental measurement was realized on a designed equipment. Multibody models of the ŠKODA 21 Tr trolleybus created in MSA (Multibody System Analysis) software and in MATLAB are presented. The alternative MATLAB model is implemented on a basis of analytical derivation. Experimental results of real trolleybus dynamic behaviour during driving over a set of normalized obstacles are described. These results are compared with numerical simulations while identified tire radial properties are used.*

1. Introduction

Investigations of road vehicles vertical dynamics, i.e. numerical simulations and experimental measurements of driving along an uneven road surface, are common and important stages in road vehicles design. The results of multibody simulations are significant especially for suitable suspension elements design and also for the vehicle bodyworks, stress analysis and fatigue life assessment. On the other hand, the experimental measurements of the road vehicles dynamic response in the course of driving along a real uneven road surface provide data sets comparable with data obtained by numerical simulations. It allows to verify and to improve the numerical models.

The specific experiments are intended to obtain parameters of design elements such as force-velocity characteristics of shock absorbers, stiffness properties of air springs or mechanical characteristics of tires. Various tire models need different input data depending on their complexity. The review of the tire models used in the field of vehicle multibody dynamics can be found in the monography (Pacejka, 2002). Supposed usage of the multibody model is an important factor that determinates the type of necessary characteristics. For the problems of vehicle dynamics mainly radial properties of tires are used in appropriate models.

The first part of this paper presents a specially proposed experimental measurement for determining these radial properties (stiffness and damping) of a real trolleybus tire. The next parts briefly describe two types of multibody models of the ŠKODA 21 Tr trolleybus and deal

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with numerical simulations and an experimental measurement performed with this trolleybus running along an artificial track composed of normalized obstacles.

This work is one of the introductory papers focused on the driving along the artificial track that should contribute to development and improvement of the ŠKODA 21 Tr trolleybus mathematical models, applicable in numerical simulations of trolleybus driving along a virtual uneven track. The virtual uneven tracks will be generated on the basis of the statistical analysis.

2. Identification of the tire radial properties

The most important tire characteristics needed for solving the vehicle vertical dynamics tasks are their radial properties (Vlk, 2000). The lateral and longitudinal properties are used mainly in horizontal vehicle dynamics (Pacejka, 2002) for studying different driving manoeuvres. Lateral, slip and longitudinal forces can be mostly neglected for the direct driving along an uneven dry road surface. The simplest tire model for vertical dynamics is the contact point model (Kovanda et al., 1997) based on the tire substitution by a single serial spring and damper. This tire model, also called the normal force tire model, is used for the simple trolleybus model in MATLAB presented in the next paragraph. More sophisticated numerical models need longer computational time and better present real behaviour of the tires for relatively sharp road unevennesses. The authors (Bartoš & Kopenec, 2003) proposed an improved tire model usable for the vertical and horizontal vehicle dynamics problems. Their tire model is implemented in MSA software (Kopenec, 2004). Another publication (Harth, Fayet & Maiffredy, 2004) introduces the tire model based on the air volume optimization.

The straightest way to determine the radial properties is experimental measurement. The measurement was performed in the Dynamic Accredited Testing Laboratory ŠKODA VÝZKUM s.r.o. (Bártík & Jozefy, 2004). The standard ŠKODA 21 Tr trolleybus tire MATADOR with specified tire inflating was measured. The scheme of the measurement is in Fig. 1. The experimental set was composed of the tire (P) mounted to a rigid frame and of a movable weight. The weight was placed over the tire in a given height and then released. The vertical position of the weight over the tire was measured by the displacement transducer on the axis o. The force between the tire and the weight was measured using the force transducer (S). The weight acceleration, that was measured by accelerometer (A), wasn't used in identification after all. The illustrative photos of the experimental arrangement are in Fig. 2.

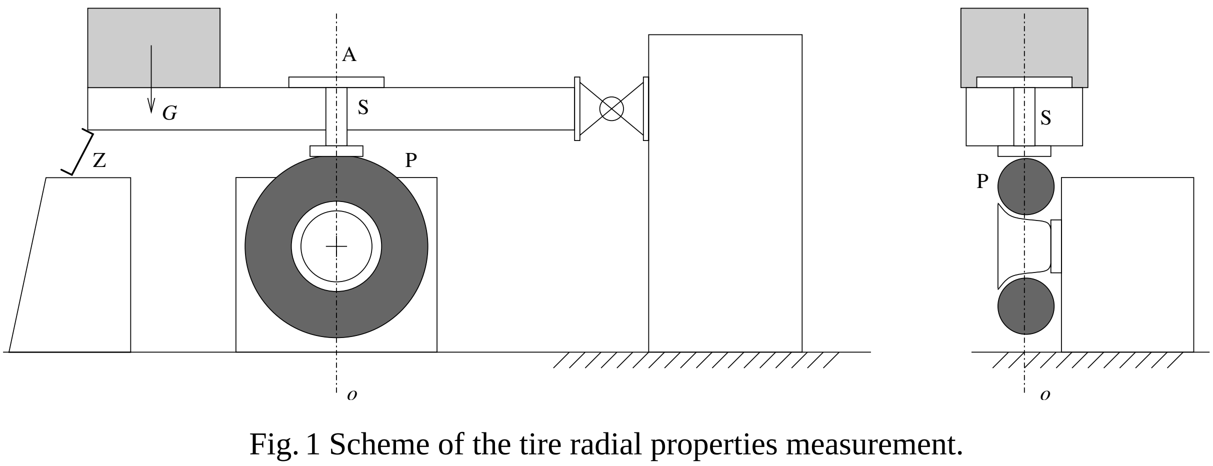


Fig. 1 Scheme of the tire radial properties measurement.

Fig. 2 Illustrative photos of the assembly for the tire radial properties measurement.

For the purpose of tire radial damping identification, the weight and the tire were simply substituted by the single degree of freedom system

$$m\ddot{x} + b\dot{x} + kx = 0, \quad \text{rewritten as} \quad \ddot{x} + 2D\Omega\dot{x} + \Omega^2 x = 0, \tag{1}$$

with serial spring k and damper b characterizing the tire radial properties and vibrating mass m over the tire. The particular value of mass m was calculated using the measured static force F_{st} acting on the tire by relation

$$m = \frac{F_{st}}{g} = \frac{18.8 \cdot 10^3}{9.81} = 1916.4 \text{ kg}. \tag{2}$$

The damping coefficient b was evaluated on the basis of logarithmic decrement δ (Zeman & Hlaváč, 1999), that can be used for damping ratio D calculation

$$D = \frac{\delta}{\sqrt{4\pi^2 + \delta^2}}. \tag{3}$$

Fifteen measurements were performed for five given values of the weight initial position over the tire. Each measurement was processed by means of own software in MATLAB, that allows to read the extreme values of signal and their sampling times and to calculate damping ratio D and corresponding frequency f. In each case, the measured time history of force (see Fig. 3 for illustration) was used to determine the moment, from which any mass bounce doesn't occur, and the time history of displacement (see Fig. 3) was used to evaluate frequency f and damping ratio D. Then the radial damping coefficient was

$$b = 2D\Omega m = 4\pi D f m. \tag{4}$$

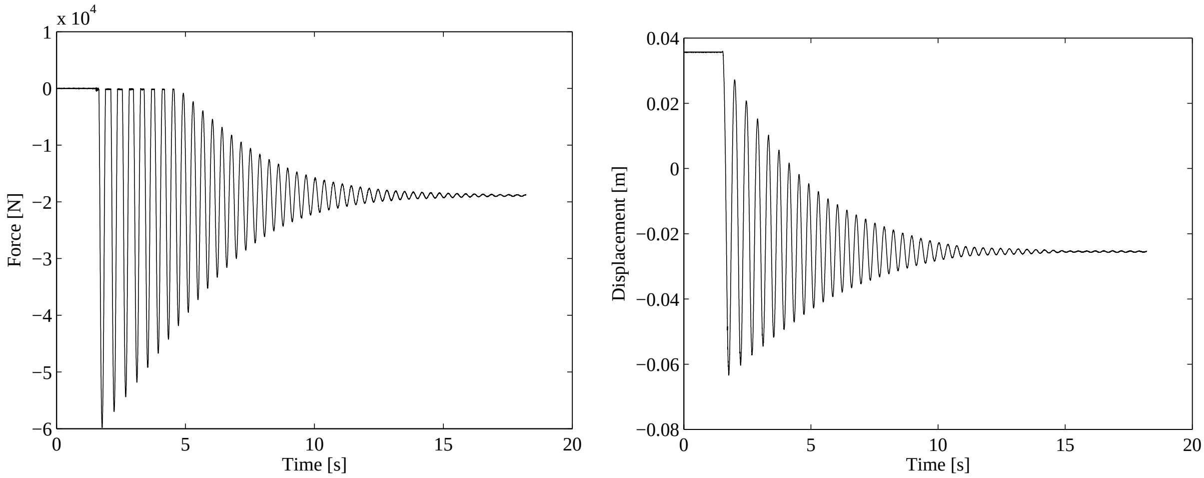


Fig. 3 Illustrative plots of the force and displacement time history for the mass initial height approximately 0.035 m.

The average radial damping obtained from all processed signals is $b_R = 1568.13$ Ns/m.

The radial stiffness characteristic was supposed to be nonlinear. Conveniently this characteristic is approximated by a quadratic function. The measured force can be plotted with respect to the measured displacement and this points (blue color in Fig. 4) can be fitted by the chosen curve. In order to find the coefficients of the quadratic fitting function

$$F(x) = \begin{cases} c_2 x^2 + c_1 x & \text{for} \quad x \geq 0 \\ 0 & \text{for} \quad x < 0 \end{cases},$$

the procedure `lsqcurvefit` from MATLAB Optimization Toolbox (The Mathworks, 2003) was used. Better approximation was achieved by the polynomial fitting function

$$F(x) = \begin{cases} p_5 x^5 + p_4 x^4 + p_3 x^3 + p_2 x^2 + p_1 x & \text{for} \quad x \geq 0 \\ 0 & \text{for} \quad x < 0 \end{cases}.$$

The identified radial characteristics are shown in Fig. 4 and the coefficients are in Tab. 1.

The identified radial characteristics have to be used carefully, because the measurement was proposed for the specific shape of the contact surface interacting with the tire. This surface corresponds rather to smoother surfaces such as the roadway with the normalized obstacle discussed in next paragraphs. However different deformation characteristics could be obtained for sharper shapes of the contact surface.

Tab. 1 Coefficients of the quadratic and polynomial fitting functions.

Quadratic characteristic		Polynomial characteristic	
c_1	$0.74782600729576 \cdot 10^6$	p_1	$0.00004886763560 \cdot 10^{10}$
c_2	$4.05562818386256 \cdot 10^6$	p_2	$0.00186079018689 \cdot 10^{10}$
		p_3	$-0.01357303481591 \cdot 10^{10}$
		p_4	$-0.13976604737495 \cdot 10^{10}$
		p_5	$1.30776870046186 \cdot 10^{10}$

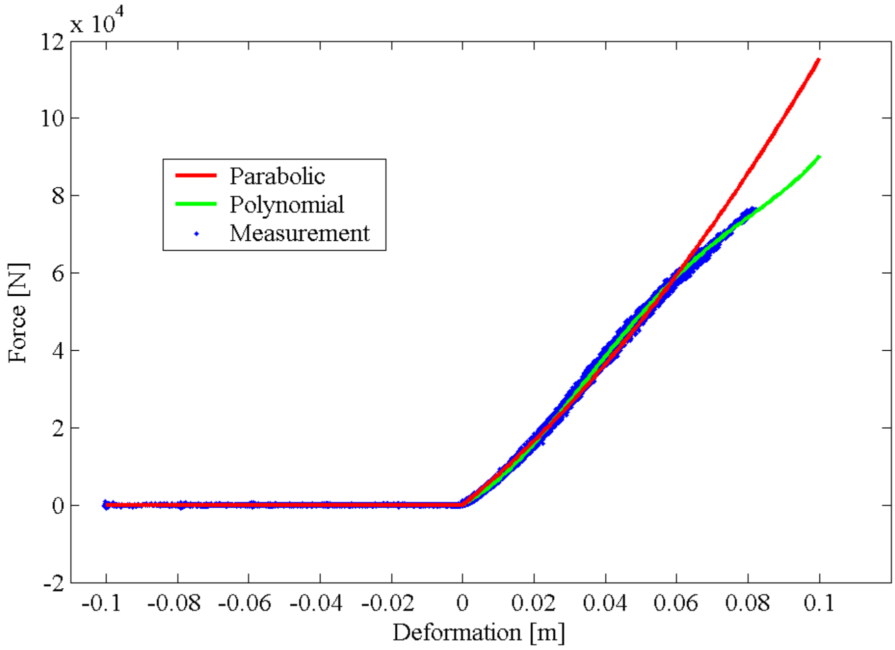


Fig. 4 Radial deformation characteristics obtained from identification.

3. The ŠKODA 21 Tr trolleybus multibody models

Two types of the ŠKODA 21 Tr (Fig. 5) trolleybus multibody models will be discussed in this contribution. Both types were created according to the previous models (Polach, 2003) in **alaska** simulation toolbox (Maisser et al., 1998).

Fig. 5 ŠKODA 21 Tr trolleybus in Hradec Králové depot.

The first one was built up by means of MSA software (Kopenec, 2003), that is specialized in the general modelling and analysis of multibody systems. Significant advantage is ability to use its efficient tire model for the simulations of driving along an uneven road surface. Tires have to be represented by the parabolic deformation curve and linear damping. The ŠKODA 21 Tr multibody model has sixteen degrees of freedom with simplified suspensions kinematics. The

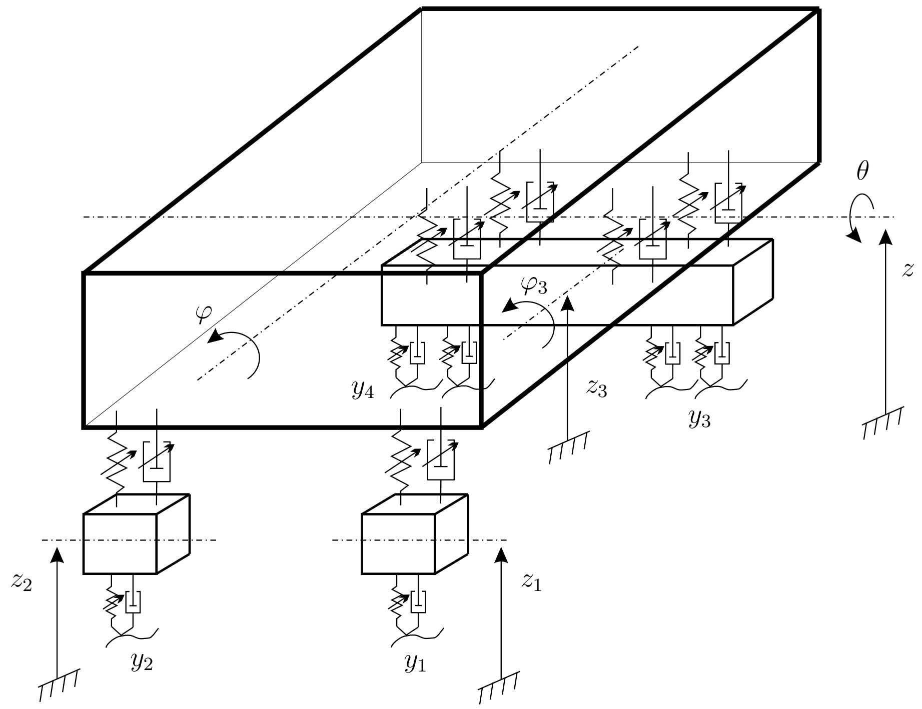


Fig. 6 Scheme of the simple nonlinear model implemented in MATLAB code.

shock absorbers are represented by their nonlinear velocity characteristics (force in dependence on velocity) newly measured for more than ± 0.8 m/s (Polach & Hajžman, 2005). Similarly the air springs are described by their nonlinear deformation characteristics (Polach, 2003). MSA software enables to consider the drive system of a vehicle with given drive torque characteristic. This property brings better behaviour closer to reality.

The second trolleybus model was implemented in MATLAB on the basis of analytical derivation (Hajžman & Polach, 2004). The simple full vehicle model (see Fig. 6) with seven degrees of freedom is composed of four bodies (one divided front axle, portal rear axle and sprung mass). The mathematical model was created by means of the free body principle and can be written in the form

$$\boldsymbol{M}\ddot{\boldsymbol{q}} = \boldsymbol{f}(\boldsymbol{q}, \dot{\boldsymbol{q}}, t)\,, \tag{5}$$

where $\boldsymbol{M}$ is a diagonal mass matrix, $\boldsymbol{q}$ is a vector of generalized coordinates of the bodies and $\boldsymbol{f}(\boldsymbol{q}, \dot{\boldsymbol{q}}, t)$ is a vector of nonlinear forces representing air springs, shock absorbers, tires and gravity forces. The tires are modelled by their identified radial properties considering the possibilities of the tire bounce from the road surface. The quadratic deformation characteristic is used because of easy comparison with the MSA model. This approach to multibody trolleybus modelling is advantageous because of its clear mathematical expression, easy parametrization and relatively short computational time. It can be used for optimization problems from these reasons.

4. Comparison of the experimental measurement and numerical simulations

All virtual mathematical models and numerical simulations need experimental verification. Eventually there must exist experimental results to improve the numerical models on their basis.

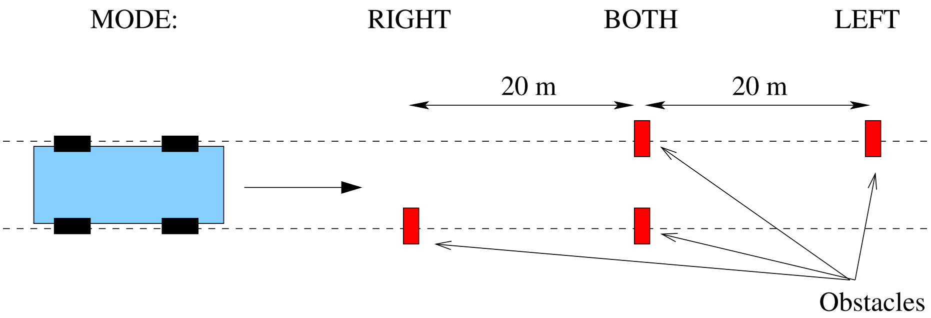


Fig. 7 Scheme of the artificial track.

The experimental measurements of the empty ŠKODA 21 Tr low-floor trolleybus was performed in depot of Hradec Králove Public City Transit Co. Inc. in October 2004. An artificial test track was composed of three normalized obstacles placed on the smooth road surface (the distance between the obstacles was 20 metres). The first obstacle was run over only with right wheels, the second one with both and the third one only with left wheels (see Fig. 7). The normalized obstacle is of the cylindrical segment shape (see Fig. 8) with height $h = 60$ mm and length $d = 500$ mm. Vertical coordinates of the obstacle are given by

$$y(x) = \sqrt{R^2 - \left(x - \frac{d}{2}\right)^2} - (R - h), \tag{6}$$

where R is the obstacle radius ($R = 551$ mm). The velocity of trolleybus was $v = 43$ km/h.

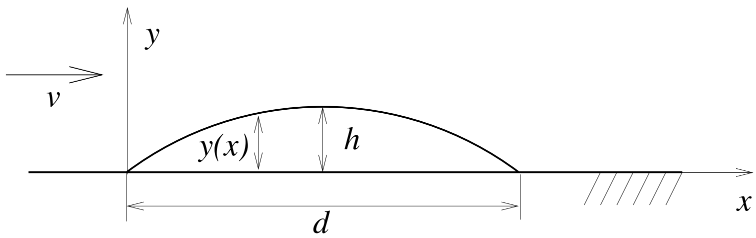
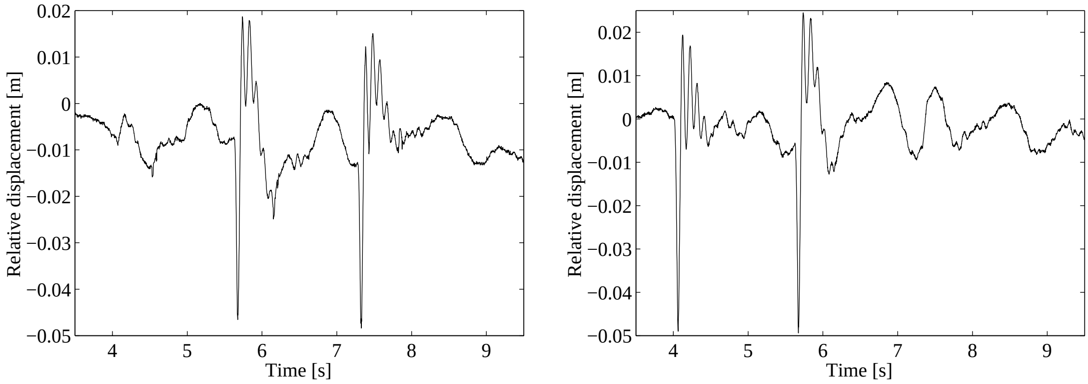


Fig. 8 Normalized obstacle according to ČSN 30 0560.

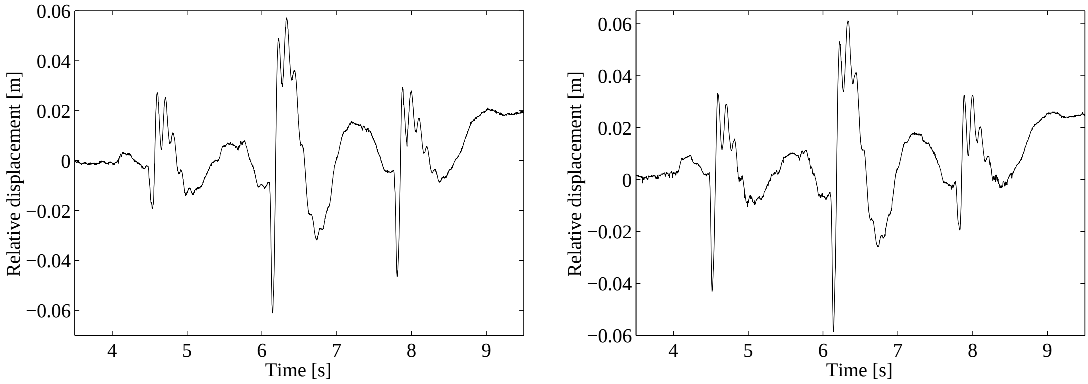
In the course of test driving the time histories of relative displacements between the axles and the chassis frame were measured (Fig. 9). Altogether four displacement transducers were placed in the vertical direction approximately on the level of air springs (on the left front half-axle, on the right front half-axle, on the rear axle to the left and on the rear axle to the right). These quantities are compared with the results of numerical simulations. Time histories of acceleration and of stresses in specific places were recorded.

The numerical simulations of the running along the same artificial test track as in the case of the measurement were performed with both types of the presented ŠKODA 21 Tr multibody models. The procedure `ode12s` in MATLAB was used for numerical integration. MSA software uses the Wilson-θ integration procedure. The time histories of relative displacements between the trolleybus axles and chassis frame calculated in MSA software, respectively in MATLAB, are shown in Fig. 10, respectively in Fig. 11. Extreme values are compared in Tab. 2.

The quadratic tire deformation characteristic was used in both models. Comparison of the simulations results obtained in MATLAB with quadratic and polynomial tire characteristics shows, that it has a very small effect.

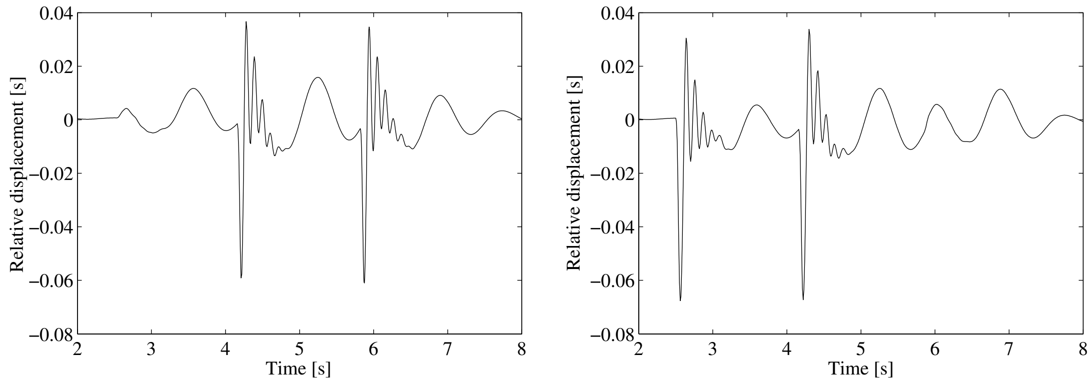


(a) Left front side (b) Right front side

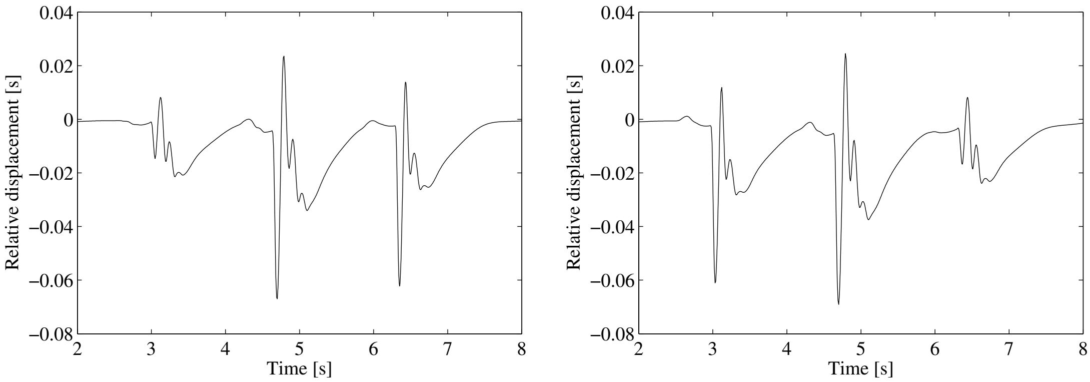


(c) Left rear side (d) Right rear side

Fig. 9 Measured time histories of relative displacements between the axles and the chassis.

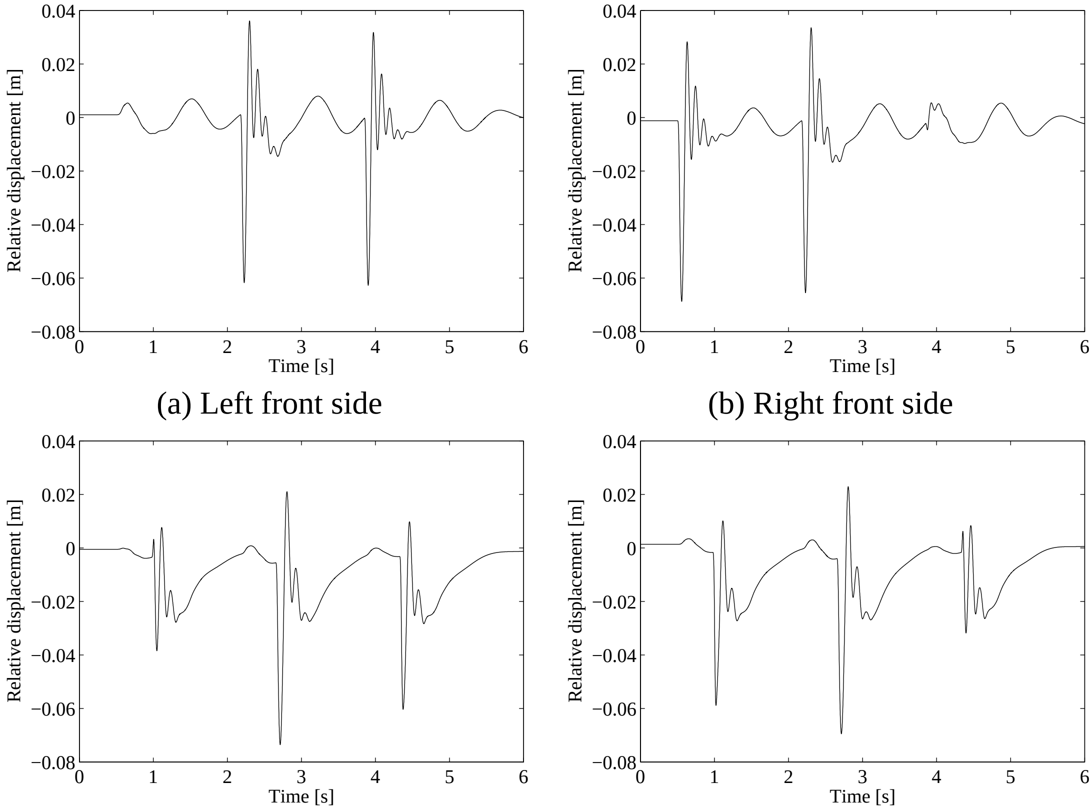


(a) Left front side (b) Right front side



(c) Left rear side (d) Right rear side

Fig. 10 Calculated time histories of relative displacements between the trolleybus axles and the chassis frame in MSA software.



(a) Left front side (b) Right front side

(c) Left rear side (d) Right rear side

Fig. 11 Calculated time histories of relative displacements between the trolleybus axles and the chassis frame in MATLAB.

5. Conclusion

The methodology of the tire radial properties identification using experimental data was presented. The radial damping and the nonlinear deformation characteristic of tire were determined. These parameters were used in the multibody models of the ŠKODA 21 Tr trolleybus. The trolleybus multibody models were implemented in MATLAB on the basis of analytical derivation and in specialized MSA software.

The dynamic response of the ŠKODA 21 Tr trolleybus in the course of running along the artificial test track was measured. The results of the experiment were compared with the numerical simulations performed in MATLAB and MSA systems.

From the unpublished numerical simulations it can be concluded, that usage of quadratic or polynomial tire radial characteristic has a negligible effect for the agreement of simulations and experiments. With respect to published results in (Polach & Hažman, 2005) it can be further concluded that models with complicated suspension kinematics aren't significantly more accurate.

The greatest differences are in suspension elements rebound stage of the rear axle. It is evident from time histories of the monitored deflections (Figs 9 to 11) that in the field of rebound of air springs of the rear axle more significant damping of calculated relative deflections occurs. It is possible to conclude on the basis of the simulations results, that the shock absorbers

Tab. 2 Measured and computed extreme values of the air springs relative displacements.

EXPERIMENT		Value	Relative displacement between the axles the chassis frame [mm]			
			On right front side	On left front side	On right rear side	On left rear side
1st obstacle		minimum	-49	-16	-43	-18
		maximum	19	-2	32	27
2nd obstacle		minimum	-49	-47	-59	-61
		maximum	24	18	61	57
3rd obstacle		minimum	-9	-48	-19	-46
		maximum	8	14	32	29
SIMULATIONS		Value	Relative deflection of air springs [mm]			
Type of model	Obstacle		Right front	Left front	Right rear	Left rear
MSA	1st	minimum	-68	-5	-61	-21
		maximum	30	4	12	8
	2nd	minimum	-67	-60	-69	-67
		maximum	34	37	35	34
	3rd	minimum	-8	-61	-24	-62
		maximum	6	35	9	14
MATLAB	1st	minimum	-69	-6	-60	-38
		maximum	28	5	10	8
	2nd	minimum	-66	-62	-70	-74
		maximum	34	37	23	21
	3rd	minimum	-10	-63	-61	-32
		maximum	5	32	9	10

characteristics measured on the laboratory testing device under specified conditions (i.e. under harmonic exciting and displacement 100 mm) do not correspond in rebound field with loading conditions of shock absorbers in the real vehicle in the course of running over the significant road unevennesses.

The following stage of the ŠKODA 21 Tr trolleybus multibody models verification will be aimed at the determination of the improved shock absorber characteristics. Especially the multibody model created in MATLAB code is suitable for solving this task.

6. Acknowledgement

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