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USE OF MULTIBODY SYSTEM DYNAMICS AS A TOOL FOR RAIL VEHICLE BEHAVIOUR DIAGNOSTICS

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Abstract

Computer simulations are currently often utilized tools in vehicle design, diagnostics of vehicle behaviour and evaluation of their characteristics. The vehicle multibody system (MBS) represents a complicated mechanical system especially when flexible bodies are considered, modelled and implemented in the calculation. Multibody system dynamics is based on classical mechanics and its engineering applications range from mechanisms, through means of transport, to biomechanics. Results of multibody system dynamics are most important for an assessment and diagnostics of transport means behaviour even in the development phase. This paper deals with the multibody system dynamic approach and its application in vehicle multibody systems. The first part of this paper focuses on a general approach to the multibody system of vehicles, especially of rail vehicles, and it includes fundamentals of multibody system dynamics. The next part deals with practical use of multibody system software. A multibody system of a passenger car have been modelled using commercial software, we have carried out simulations of a passenger car running on the real track, and subsequently we assessed its ride properties and evaluated its passenger comfort level.

Key words: multibody system, computer tools, vehicle dynamics, dynamic analyses.

INTRODUCTION

Performing dynamic analysis of a vehicle multibody system is permanently in focus. Analysis of multibody system dynamic of a vehicle uses special software tools.

In terms of mechanics, a vehicle is a multibody mechanical system. The multibody system consists of rigid bodies connected by joints that limit relative motion of pairs of bodies. Forces and moments are caused by elements such as springs, dampers, tires, shock absorbers, actuators or rods and other elements that give rise to reaction forces and moments. The above-mentioned coupling elements themselves are generally considered to be massless. In the vehicle multibody system approach, the properties of inertia, elasticity, viscosity and force are associated with certain discrete elements. Analysis of multibody systems requires appropriate modelling techniques, reliable numerical solvers for the differential equations obtained and sophisticated graphical representation of the results which, in turn, are subject to optimization [16, 17].

1. COMPUTER TOOLS FOR MULTIBODY SYSTEM DYNAMICS

The biggest advantages of analysis of multibody system dynamics is the production costs reduction. Computer simulations enable predicting and diagnosing ride properties of vehicles in general, as well as of other transport means, e.g. road vehicle [14], motorcycle, rail vehicle [20, 21 and 22], aircraft or their subsystems (engine, gearbox, etc.) [2], and allow performing adjustments and optimization of the vehicle design without the need to have the real product. Further, experiments are useful for the final validation of the results obtained.

The development of multibody system software has always been driven by the problem arising from the connection of vehicles to road and rail. Nowadays, the most common multibody system programs are SIMPACK, ADAMS, VI-GRADE, DADS and WorkingModel [8, 12, 15 and 16]. These software products work with equations of motion.

The system of a vehicle and its development processes as they exist today, pose certain requirements on an MBS system, which are partially of general nature and partially specific to MBS. As seen from the users´ point of view, one could group these requirements as follows: ease of use, scalability, re-usability, reliability.

In general, vehicles are highly moving mechanical systems. Vehicles move over surfaces that are characterized by the rolling contact between wheels and the guideway, modelled as a continuum mechanics problem considering the elasticity in the contact patch. Moreover, surfaces are randomly excited by the lateral and vertical irregularities of the guideway resulting from stochastic processes [13, 17 and 19]. The response of the wheeled vehicle multibody system strongly depends, inter alia, on the tire pressure [3]. Road vehicles are controlled by drivers representing an operator-in-the-loop problem. Driver assistance devices make a vehicle a mechatronic system, too. Analyses of rail vehicle multibody systems introduce into the computation the phenomena of wheel/rail contact [7, 9]. Interaction between wheel and rail is a very complex nonlinear element in the rail vehicle multibody system [10, 11].

2. FUNDAMENTALS OF MULTIBODY DYNAMICS

With the advent of personal computers with faster processors and growing performance, the use of analytical modelling programs has become less complicated, and far more practical. This is true in the area of vehicle modelling as well. Lower costs of computer modelling compared to real-world testing are a significant benefit of the use of numerical modelling of road as well as rail vehicles [23, 24].

The first step in every vehicle computer simulation is to set up a mechanical model appropriate to fulfil the desired simulation task [4, 5]. This model constitutes the basis for mathematical description using the equations of motion, obtained through physical principles and laws (Newton's laws, etc.).

2.1 Equations of motion of a vehicle multibody system

Software for multibody system dynamics works with equations of motion.

Numerical equations of motion have to be generated for each timestep of the integration code and for each parameter variation. The symbolical equations were generated only once, they are especially helpful for real time applications and parameter optimization. The first stage in setting up a computer model is to prepare a set of mathematical equations that represent the vehicle multibody system. Application of constraint equations results in a set of equations of motion that are ordinary differential equations (ODE), or linear algebraic equations (LAE) and ODEs, depending on how the constraint equations are used. Combination of algebraic equations and ordinary differential equations leads to differential-algebraic equations (DAE) [17, 18].

A rail vehicle multibody system with proportional or proportional-differential forces results in an ordinary multibody system. The equations of motion follow from the Newton-Euler equations, applying d'Alambert's principle [17].

The equations of motions of the vehicle multibody system are found according to d'Alambert's principle as:

$$\boldsymbol{M}(\boldsymbol{y},t)\ddot{\boldsymbol{y}} + \boldsymbol{k}(\boldsymbol{y},\dot{\boldsymbol{y}},t) = \boldsymbol{q}(\boldsymbol{y},\dot{\boldsymbol{y}},t)\,. \tag{1}$$

Here the number of equations is reduced from *6p* to *f*, the f x f -inertia matrix $\boldsymbol{M}(\boldsymbol{y},t)$ is completely symmetrized, $\boldsymbol{M}(\boldsymbol{y},t) = \bar{\boldsymbol{J}}^T \bar{\bar{\boldsymbol{M}}}\bar{\boldsymbol{J}} > 0$, and the constraint forces and moments are eliminated.

Equation (1) is also true for unconstraint systems. Then, it yields $\boldsymbol{y} = \boldsymbol{x}$ and the global Jacobian matrix $\bar{\boldsymbol{J}}$ is a quadratic *6p* x *6p* – matrix. Adding the implicit constraint $\boldsymbol{x} = \boldsymbol{x}(\boldsymbol{y},t)$ or $\boldsymbol{\Phi}(\boldsymbol{x},t) = 0$, Lagrange's equation of the first kind are obtained as

$$\boldsymbol{M}(\boldsymbol{x},t)\ddot{\boldsymbol{x}} + \boldsymbol{k}(\boldsymbol{x},\dot{\boldsymbol{x}},t) = \boldsymbol{q}(\boldsymbol{x},\dot{\boldsymbol{x}},t) - \boldsymbol{\Phi}_X^T \boldsymbol{\lambda} \tag{2}$$

where λ is the *q x 1* – vector of Lagrangian multipliers. The *6p* scalar equations (2) cannot be solved due to the *6p+q* unknowns in the vectors *x*, *λ*. The implicit constraint equations $\boldsymbol{x} = \boldsymbol{x}(\boldsymbol{y},t)$ or $\boldsymbol{\Phi}(\boldsymbol{x},t) = 0$ have to be called again

$$\boldsymbol{\Phi}(\boldsymbol{x},t) = 0 \tag{3}$$

This maintains a set of *6p+q* differential algebraic equations. The resulting system of differential-algebraic equations for the vehicle multibody system is

$$\begin{bmatrix} \boldsymbol{M} & \boldsymbol{\Phi}_X^T \\ \boldsymbol{\Phi}_X & 0 \end{bmatrix} \begin{bmatrix} \ddot{\mathbf{x}} \\ \boldsymbol{\lambda} \end{bmatrix} = \begin{bmatrix} \boldsymbol{q} - \boldsymbol{k} \\ -\boldsymbol{\Phi}_t - \dot{\boldsymbol{\Phi}}_X \dot{\mathbf{x}} \end{bmatrix} \tag{4}$$

Solving the equations presents numerical difficulties resulting from the need to ensure that the kinematic constraints are not violated during the integration process.

After grouping the force matrixes according to coordinate vectors, the equations of motion for the rail vehicle will have the following form:

$$\boldsymbol{M}\ddot{\boldsymbol{x}}(t) + (\boldsymbol{D} + \Omega_0 \boldsymbol{G})\dot{\boldsymbol{x}}(t) + (\boldsymbol{K} + \Omega_0^2 \boldsymbol{Z})\boldsymbol{x}(t) = \boldsymbol{f}(t) \tag{5}$$

where the vector ***x(t)*** contains the generalized coordinates, matrix ***M*** is the mass matrix, matrix ***D*** is the damping matrix resulting from viscous coupling elements between the wheel (wheelset), bogie frame and car body and matrix ***K*** is the stiffness matrix resulting from the elastic coupling elements and centrifugal forces, respectively. The vector ***f(t)*** represents generalized external forces resulting from wheel/track (or wheel/road) interface, from gravitation and from nonlinear yaw damping. Because of the symmetric structure of the vehicle, the equations of motion can be split up into two separate systems for symmetric and asymmetric motions. The vehicle multibody dynamics problem is reduced to solving a set of differential-algebraic equations. All time-domain analyses are performed with the second order Runge-Kutta integration method [17, 18].

Once equations of motion can be compiled and integrated, they can provide us with the information about variation in response for different vehicle designs.

3. DYNAMIC ANALYSIS OF A PASSENGER CAR

This part of the paper focuses on the diagnostics and assessment of ride properties and ride comfort evaluation of a passenger car running on a real track. Analyses and assessment of ride properties of a rail vehicle and, for passenger cars, also ride comfort evaluation are important aspects in the process of designing new or renovating the existing rail vehicles [1]. We used the MBS Simpack/Rail software product to create the passenger car model and track model. This software allows modelling and analysing rail vehicles and similar devices of every kind, from trams to high-speed trains. Most parts of a rail vehicle model in Simpack/Rail are made using standard Modelling Elements, but calculation of wheel/rail contact states requires to use specialized Modelling Elements, such as Rail-Wheel Contact, Rail-Wheel Pairs, Rails, Wheelsets, and Result Elements.

3.1 Ride properties assessment of the passenger car

The analysed referenced rail vehicle was a passenger car with four wheelset. The computer model of the passenger car consists of three main subsystems – two bogies and the passenger car body. Wheelset are guided by swinging arms in bogies. Between wheelsets and the bogie frame there is a primary suspension is mounted which is composed of spring coils and hydraulic dampers with non-linear characteristics. A secondary suspension is situated between the bogies and car body, and it also consists of spring coils and hydraulic dampers with non-linear characteristics. In addition, hydraulic dampers are also components of secondary suspension for damping of yawing oscillations and lateral movements of bogies. Figure 1 shows the dynamic schematic of the passenger car multibody system. In the computational model of a passenger car, rigid bodies (body of wagon, wheelsets, axleboxes, frames of bogies) are connected by force elements, joints, constraints, etc. [4]. Figure 2 shows a dynamic schematic of a passenger car multibody system.

Important parameters for creating the passenger car model are shown in the following Tables.

Tab. 1. Parameters of the passenger car body

Body of passenger car:		
Mass	$m_1 = 32e3$	[kg]
Moments of	$J_{x1} = 5.68e6$	[kg.m^2]
	$J_{y1} = 1.970e6$	
	$J_{z1} = 1.970e6$	

Tab. 2. Parameters of the passenger car bogie

Bogie frame:		
Mass	$m_2 = 2.6e3$	[kg]
Moments of inertia	$J_{x1} = 1.722e3$	[kg.m^2]
	$J_{y1} = 1.476e3$	
	$J_{z1} = 3.067e3$	
Wheelset:		
Mass	$m_3 = 1.5e3$	[kg]
Moments of inertia	$J_{x1} = 1.722e3$	[kg.m^2]
	$J_{y1} = 1.476e3$	
	$J_{z1} = 3.067e3$	
Axlebox:		
Mass	$m_4 = 0.150e3$	[kg]
Moments of inertia	$J_{x1} = 0.002e3$	[kg.m^2]
	$J_{y1} = 0.006e3$	
	$J_{z1} = 0.006e3$	

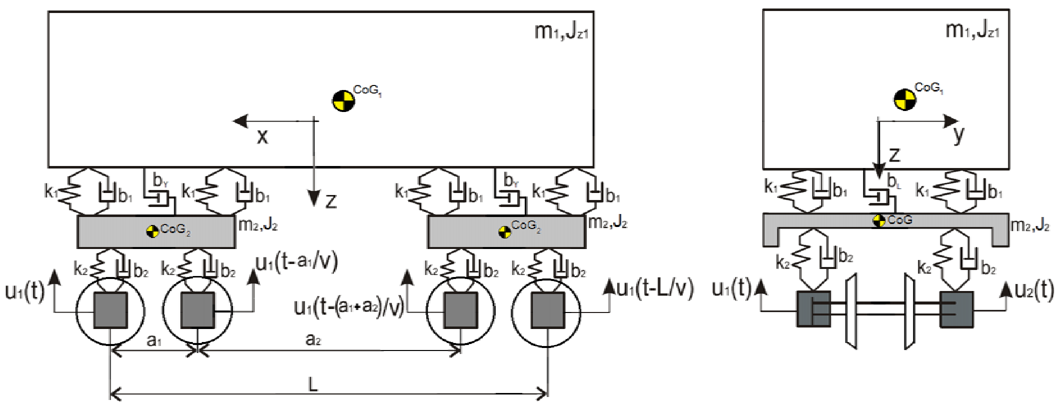

Fig. 1. Passenger car multibody system – dynamic schematic

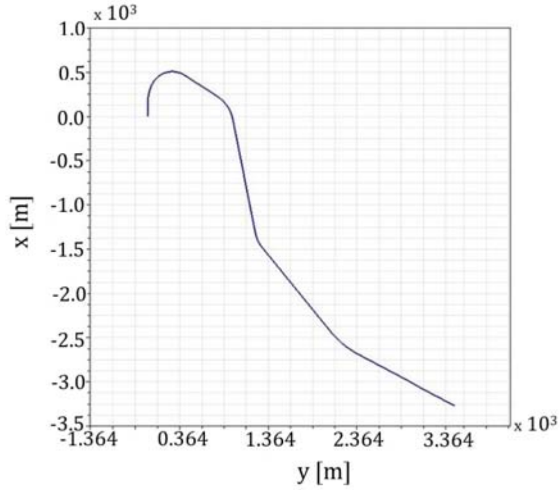

Fig. 2. Horizontal profile of the track

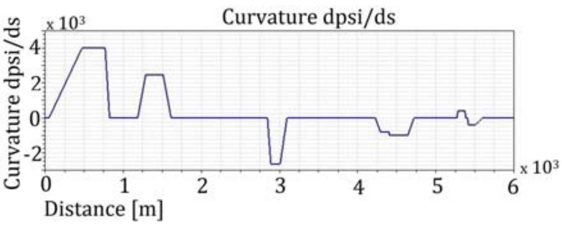

Fig. 3. Curvature of the track

The Simpack software has been used to create the model and analyse its performance.

The passenger car was run on the track model created from real geometric parameters of the Šurany – Úľany nad Žitavou track section. This track is suitable for simulations performing because it consists of several sections with different curve radii and the resulting sections of transmission curves and superelevation ramps. The total length of this track section is 5850 m, and it has the normal rail gauge of 1435 mm with UIC60 rail profile. All wheels of the passenger car are equipped with the S1002 profile. The track model contains also measured track irregularities prescribed by discrete form with step of 0.5 m, which excited the passenger car during run.

The horizontal profile of the track is shown in Figure 3, and the curvature of the track is shown in Figure 4.

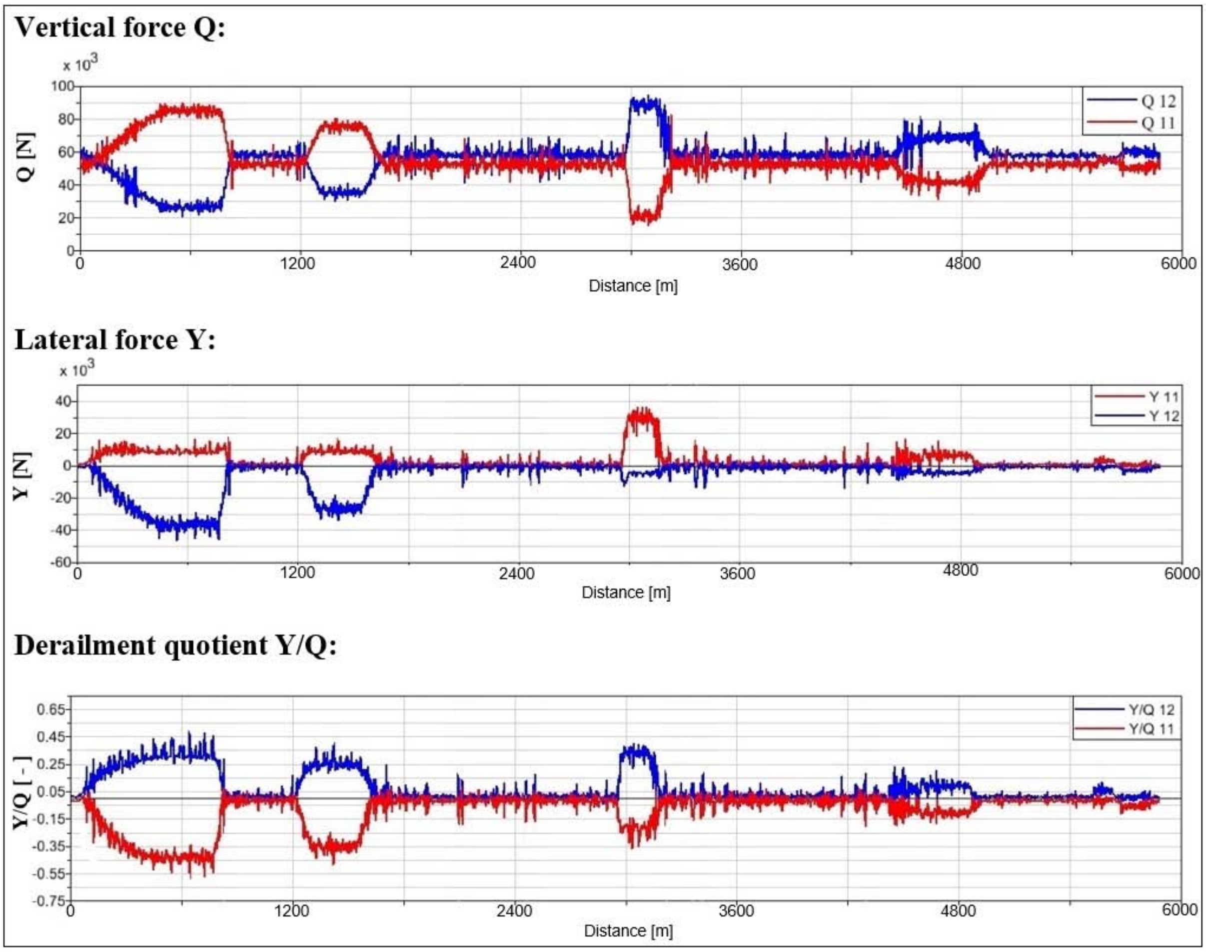

Fig. 4. Results of passenger car dynamic analysis

Figure 5 shows the cumulative graph of values for the right wheel of the 1st wheelset (Q11, Y11, Y/Q11, in red) and for the left wheel (Q12, Y12, Y/Q12, in blue). These graphs include three parts. The vertical forces are in the 1st part, lateral forces in the 2nd part, and derailment quotient in the 3rd part. The derailment quotient (Y/Q) is given as ratio of the lateral force (Y) and the vertical force (Q) and we take into account its absolute value. Vertical and lateral forces but mainly the derailment quotient must be known for diagnostics and evaluation of the car running safety.

We can see from the graphs in Figure 5 that when the passenger car enters curves, values of vertical wheel forces increase (consistent with the track curvature, Figure 4). In straight track sections, vertical forces correspond to the gravitational load of wheels of the 1st wheelset.

The second part of Figure 5 shows that lateral forces increase also when the passenger car enters curves. In straight track sections lateral forces achieve very small values compared to the values in curves. Finally, the third part of Figure 5 shows the derailment quotient values. As written above, derailment quotient is the ratio of the lateral and vertical forces, and it expresses safety of the rail vehicle running. As we can see, the maximum values are reached when the passenger car is running in the first curve in the running direction. This curve is the smallest one in the created track section, i.e. its curvature (1/R [1/m]) has the highest value (Figure 4). The maximum value approaches 0.55, therefore the passenger car runs safely [25]. We can also see that track irregularities cause excitation of the passenger car mechanical system, and thus the deterioration of running behaviour and safety.

3.2 Ride comfort evaluation of the passenger car

The passenger car ride comfort can be evaluated by the so-called indirect method. For this method it is necessary to know values of accelerations of the analysed passenger car at specificed points. Acceleration values are filtered and weighted by functions that take into account the human sensitivity to vibration in reference directions During the vehicle running, dynamic movements of the body are generated. These movements are manifested as vibration. Passengers are subjceted to these negative effects during operation. The ride comfort is the total sensation, which the rail vehicle body movements generate in a passenger body.

Rail vehicle body movements are transmitted to the whole passenger body at passenger–vehicle contact points:

- **standing positon** – floor – feet,
- **seated position** – headrest – neck,
  – arm rest – arms,
  – seat – hip,
  – backrest – back,
  – floor – feet.

Ride comfort level is expressed by the value of the ride comfort indices. To assess the ride comfort indices we have to know the values of acceleration in the x, y and z directions. Based on the sampling freguency f_n we determin the number of samples that are recorded in the time interval of 5 sec. At the sampling frequency of 200 Hz during 5 seconds we obtain 1000 samples. Based on the condition of samples occurance in 5 sec. intervals, the sensing time is divided into 5-sec. time periods. Each section has clearly established its beginning T_1 and end T_2 in time. Fast Fourier transform (FFT) is performed for the dataset in each time interval. CAW calculation is performed for the range of frequences 0.4 Hz – 100 Hz.

The weighting filter w, which takes into accout the human body sensitivity to different frequencies, is applied depending on the type of evaluation – floor, standing position, seated position.

Weighting function modified acceleration values are statistically evaluated and summation functions in histograms are determined.

The procedure of the ride comfort calculation procedure is shown in Figure 5.

Resulting values of passenger ride comfort indices for the average comfort are calculated according to the EN 12299:2009 (2009): *Railway Applications – Ride Comfort for Passengers – Measurement and Evaluation.* European Committee for Standardization, Brussels. standard. Analyses in this paper are focused on the calculation of the ride comfort index on floor N_{MV}, which is expressed as follows [6]:

$$N_{MV} = 6 \cdot \sqrt{\left(a_{XP95}^{W_d}\right)^2 + \left(a_{YP95}^{W_d}\right)^2 + \left(a_{ZP95}^{W_b}\right)^2} \qquad (6)$$

As seen in equation (6), calculation of the N_{MV} comfort index requires to input the accelerations in longitudinal (*x*), lateral (*y*) and vertical (*z*) directions for each point of interest.

Sometimes, depending on the application, it can be useful to calculate partial indices of the ride comfort [6]:

$$N_{MV_x} = 6 \cdot a_{XP95}^{W_d} \qquad (7)$$

$$N_{MV_y} = 6 \cdot a_{YP95}^{W_d} \qquad (8)$$

$$N_{MV_z} = 6 \cdot a_{ZP95}^{W_d} \qquad (9)$$

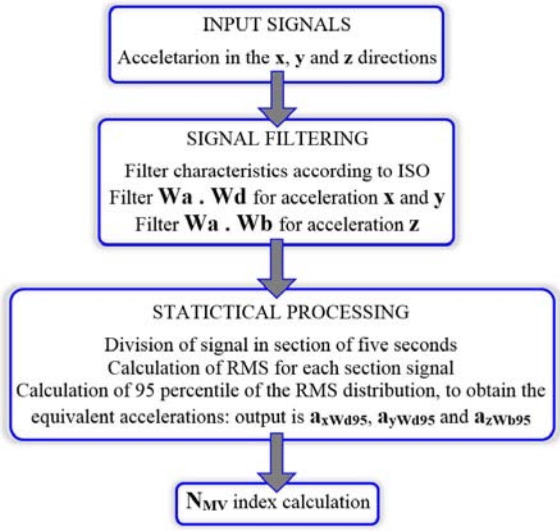

Fig. 5. Calculation process for N_{MV} comfort index

For ride comfort evaluation of the analyzed passenger car we defined a total of fifteen points on the floor (Figure 7). Therefore, specialized Control Elements (Accelerometer) were applied into these point.

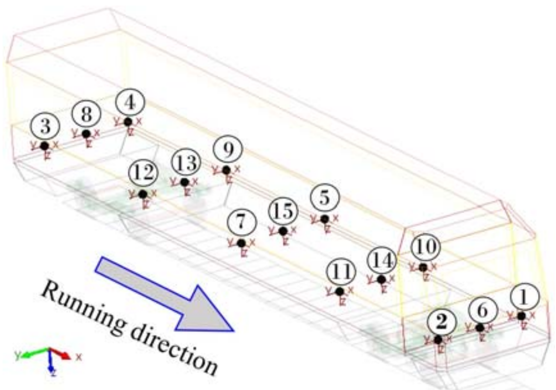

Fig. 6. Locations of points for the ride comfort evaluation of the passenger car

Figures 7, 8 and 9 show computation procedures of the passenger car ride comfort index on wagon floor level for Point 1 in Simpack PostProcessor, in x, y and z directions respectively. Outputs of procedures are frequency weighted RMS values of acclereation $a_{XP95}^{W_d}$, $a_{YP95}^{W_d}$ and $a_{ZP95}^{W_d}$ respectively, where [6]:

a – RMS-values of acceleration,

X, Y, Z – directions of acceleration,

P – measurement position (floor),

95 – 95 percentil,

W_d – weighting curve in *x* and *y* direction,

W_b – weighting curve in *z* direction.

As described in Subchapter 3.1, the passeger car runs on the track at the constant speed of 85 km/h.

The five-second RMS-values of the frequency weighted accelerations are calculated as [6]:

$$a_{xj}^{W_d}(t)=\sqrt{\left[\frac{1}{\tau}\cdot\int_{t-\tau}^{t}\left(\ddot{x}_{W_d}^{*}(\tau)\right)^2 d\tau\right]} \tag{10}$$

$$a_{zj}^{W_b}(t)=\sqrt{\left[\frac{1}{\tau}\cdot\int_{t-\tau}^{t}\left(\ddot{z}_{W_b}^{*}(\tau)\right)^2 d\tau\right]} \tag{11}$$

$$a_{zj}^{W_b}(t)=\sqrt{\left[\frac{1}{\tau}\cdot\int_{t-\tau}^{t}\left(\ddot{z}_{W_b}^{*}(\tau)\right)^2 d\tau\right]} \tag{12}$$

The N_{MV} passanger comfort index was calculated according to equation (1) from the values of frequency weighted acceleration, which are shown in the second step in Figures 7, 8 and 9. The calculated indices of the passenger ride comfort allow objectification of the expected subjective discomfort during travelling.

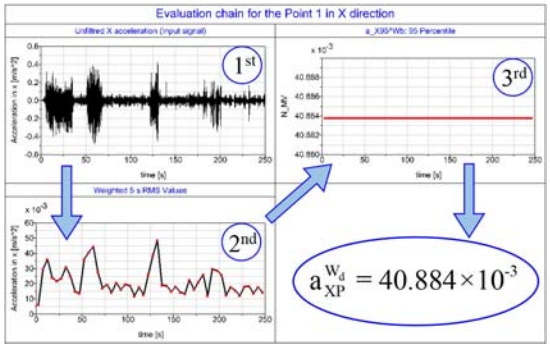

Fig. 7. Computation of the frequency weighted acceleration for Point 1 in x direction

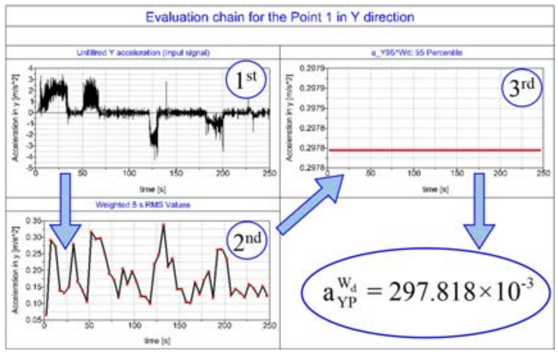

Fig. 8. Computation of the frequency weighted acceleration for Point 1 in y direction

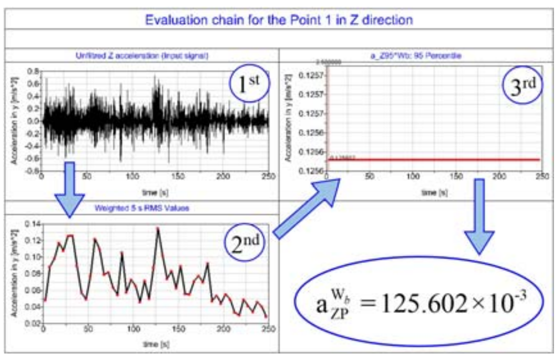

Fig. 9. Computation of the frequency weighted acceleration for Point 1 in z direction

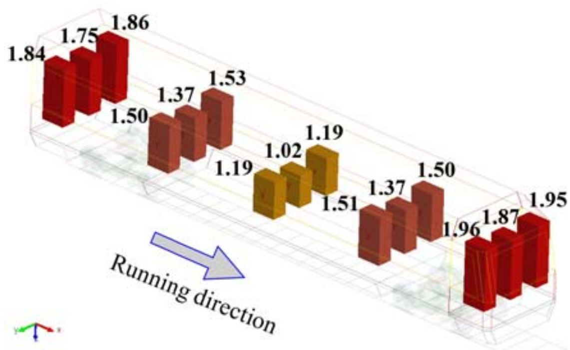

Fig. 10. N_{MV} indices for the speed of 85 km/h

Results of ride comfort indices calculation are shown in Figure 10. As we can see, the position in which the passenger expirences the least influence of acceleration on the body is near the centre of the wagon body. In contrast, the highest values of acceleration are generated in the front and rear parts of the passenger car.

Tab. 3. Scale for N_{MV} comfort index [6]

$N_{MV} < 1.5$	Very comfortable
$1.5 \leq N_{MV} < 2.5$	Comfortable
$2.5 \leq N_{MV} < 3.5$	Medium
$3.5 \leq N_{MV} < 4.5$	Uncomfortable
$N_{MV} > 4.5$	Very comfortable

When we compare the calculated ride comfort index N_{MV} with the scale according to Table 3 (EN 12299:2009), the passenger car has been clasified as "comfortable" ($1.5 \leq N_{MV} < 2.5$). Five points near the centre of gravity were assessed as "very comfortable" ($N_{MV} < 1.5$) [6].

4. SUMMARY

The advent of personal computers and fast processors made the use of analytical modelling software less complicated and far more practical. The relatively low price of computer modelling in dynamic analysis compared to real-life vehicle testing is a significant benefit in vehicle modelling. Computer modelling of vehicle multibody mechanical systems and of their behaviour in terms of dynamics allows a customer to test and diagnose a new car design without having to build a prototype and testing track, which increases the productivity through savings in time and manpower. Modelling of vehicle multibody systems provides equipment to test vehicle parameters, properties and behaviour, and also allows predicting the forces and accelerations acting on various construction assemblies of vehicle.

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