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SOFTWARE TESTING OF THE RAIL VEHICLE DYNAMIC CHARACTERISTICS

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Sanjin Troha[1], Miloš Milovančević[2], Alireza Kuchak[3]

[1]Faculty for Technical Engineering, University in Rijeka, Croatia
[2]Faculty for Mechanical Engineering, University in Niš, Serbia
[3]Department of Structural Analysis, TU Berlin, Germany

Abstract. *The modern construction concept and the determination of the machine system characteristics anticipate CAD design. Creating model that will be tested using FEM and other methods for determining stress-strain is a very important part of rail vehicle construction. Applicative software package consists of linear and non–linear methods for the prediction of railway vehicle behavior and various methods of analysis have been assembled into a single coherent package in order to allow real problems in railway vehicle dynamics to be solved.*

Key Words: *Software, Virtual Testing, Rail Vehicle*

1. INTRODUCTION

The dynamic behavior of railway vehicles has been a research topic for more than a century. The dynamic fluctuations that occur as a result of the wheel rails contacts were described for the first time by George Stephenson in 1821. Since then a lot has been done concerning the control system of the railway vehicle vibration, and, in particular, the dynamic behavior of vehicle in a curve. Thus, at the sixties when computers became more accessible and mathematical formulas more accurate, quality results in determining the railway vehicle dynamics were obtained. Further advance in computer technology has led to the rapid development of numerical techniques for problem solving of vehicle dynamics [1, 2].

As part of the British Association for Research railways (now the AEA Technology Rail) a large number of programs were developed in last twenty five years in order to solve the movement dynamics. This development includes the creation of linear models used in the

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Corresponding author: Sanjin Troha
Faculty of Engineering, University of Rijeka, Vukovarska 58, 51000 Rijeka, Croatia
E-mail: sanjin.troha@riteh.hr

linear analysis and programs based on the linear theory of movement. The linear program for vehicle dynamics in curve track is time progressed and has basic understanding of nonlinear force on the wheel-rail contact. Further development led to the non-linear program analysis of transitional states that are researching the characteristics of the vehicle using discrete inputs and features rail road on the basis of known vertical and lateral displacement. It has been concluded that the linear program has some limitations related to the theoretical assumptions about wheel-rail contact, but, if recognized, these linear programs are extremely useful [3, 4].

2. Development of Linear Analysis of Rail Vehicle

2.1. Basic software characteristics

The equations of the rolling process are generally nonlinear. The first studies that examined the dynamics of vehicles used the linearization techniques to obtain a set of linear equations of motion that were then solved leading to understanding the vehicle dynamic behavior. These early works of the British Institute for the development of railways resulted in the CEI Award 1975. They still represent a large portion of the linear analysis techniques used in the software. This involved the use of linearized models of contact-point rail (which is generated by the static procedure Booton) and linearization forces that occur during a slow rolling process (creep). Also back then there were some simplifications related to the symmetry of the vehicle. It was noted that the coordinates that describe the movement of vehicles can be divided into groups of vertical and lateral. Vertical coordinates relate to the movement of vehicles together with those that cause displacement in the vertical and longitudinal plane. The group gives the lateral movement of the coordinate administration compared to the previously stated levels. To simplify small movements of vertical and lateral groups the coordinate-equation can be divided; a large number of the first models were actually only vertical or lateral models. Today, exclusively three-dimensional models are used and the only disadvantage of the linear software is that it is limited to small displacements. Linear methods allow for an analysis of the dynamic behavior of the vehicle on the right tracks provided that linearization of the suspension elements is done. In essence, the basic problem related to the linear model is that the mathematical model is fully adequate but it is very difficult to assign adequate parameters of the model. Giving the correct parameters is the main factor affecting the accuracy of predicting the dynamic behavior of the vehicle. The linear techniques limitations exist only in the data selected for analysis. It also automatically performs suspension elements linearization. Linear solving of the wheel-rail contact has its limitations but if recognized the program is still very useful because it allows for the variation of the taper point parameters and understanding of the dynamics of movement [5, 6, 7].

3. Development of Nonlinear Analysis of Rail Vehicle

Experiences with the linear theory of the dynamics of movement have led to the studies on the behavior of the vehicle in a curve and the vehicle's behavior when there are discrete inputs; it has led to the use of nonlinear equations of motion. The quasi-static theory of movement has been developed in 1978. This is essentially a nonlinear theory that deals with the problems of the geometry of the wheel-rail contact and the forces that occur in slow moving vehicle (creep, creep forces), the Kalker's method. Two different methods are used for calculating and solving equations and both give similar results. The first method uses a partial linearization of the contact wheel-rail information so that it can carry out their interpolation; quasi-static force equilibrium solution is obtained by rapid convergence. The second method uses step-by-step integration which includes creating time range with the vehicle in a curve; such convergence is also a quasi-static solution. The integration technique is slower but it involves attenuation of the integration that arises as a result of lateral and longitudinal accelerations. The advantage of this second method is that it can be extended to predict the behavior of the vehicle passing through the switch, that is, generally speaking, for any kind of prediction of nonlinear behavior. The developed theoretical method that allows for an analysis of the friction braking torque, has, in addition to the creep forces arising from Kelkerovog work, variations of the ellipse contact surface size in order to predict the behavior of the axle assembly and derailing the rails. Nonlinear programming method is tested by the movement simulation of a two-axle rail vehicle-trolley at a certain line. The simulation results are lined with experimental results on a given line. The simulation results comparing to the results of the previous measurements are located in the range of up to 20% deviation. It is experimentally confirmed that the movement simulation can be performed in order to predict the vehicle dynamic behavior on a straight track [8, 9, 10, 11].

4. Program Analysis of Transient States of Rail Vehicle

Program analysis of transient states is used to calculate the vehicle’s dynamic response to external effect of railroad geometric imperfections and external forces in general. The calculation covers full nonlinear model vehicles and oscillations that occur as a result of movement or wheel-rail contact. The program analysis of transient state in the software package is presented through a complex and highly demanding user interface. A dynamic vehicle’s response to outside influences is conditioned by a number of different parameters of the vehicle, wheel-rail or rails contact and geometry, geometry point. The transient states analysis represents the most complex form of analysis in addition to the contents of the calculation inputs and double wheel-rail contact. The process of analyzing the rail vehicle movement begins with defining vehicles (geometry and physical properties) in terms of defining and determining the file from which the software that reads the data on the vehicle is taken, as well as the appointment of the analysis. The file is used to define geometry and physical characteristics of vehicles (trolleys). Since this is a dynamic test, the trolleys in a file bearing the code record on trolleys enter the following data:

- Trolley is a vehicle with two axles (2 axles)
- The weight of the axle assembly 1313.63 kg
- The total maximum weight of loaded trolley 15373 kg
- Stiffness spring primary suspension is 1.1 MN/m
- Modulus 80 kN/mm^2
- The distance between the axles 3.9 m

Furthermore, it is necessary to define the conditions of dynamic testing of vehicles, in terms of defining precisely the desired speed of the vehicle (in our case, 50 km/h and 60 km/h). It is also possible to define the range of speeds at which we test. Since the standard UIC 518 defines the length of test track sections, it must also be defined in the software on which the dynamic response of the vehicle is determined. Furthermore, it is necessary to define the time steps for the calculation and the recommended values are 1-5 milliseconds.

Quality of rail road greatly affects the quality of the movement of rail vehicles. Software package that is used to simulate the movement of the trolley calculates the impact of rail road defined through the file that carries information about the track. Since this is a rail road which is in the third category and whose tracks are observed, according to the type of deformation (stability, warping and camber). Some data on the state of rail are captured using measuring equipment given in Table 1. The data presented in Table 1 shows data discrepancies rail geometry of constructive listed values (found rates). We can see that the camber on the part of the railroad from 4716 m – 4718 m railway line is -15 mm compared to constructively inventoried value, while in the act of 4979 m – 4982 m railway line is -17 mm.

Table 1 Rail way track geometry parameters

Type of deformation	Stationing		length	Measures
	From	To	(m)	
Overshoot	4.716	4.718	2	-15 mm
Stability-D	4.808	4.811	3	-23 mm
Stability-D	4.815	4.817	2	25 mm
Stability-L	4.815	4.818	3	25 mm
Twisting 3.5m	4.979	4.981	2	-19 mm
Overshoot	4.979	4.982	3	-17 mm

The complexity of nonlinear dynamic model demonstrates the need to define the parameters required for the calculation of the relative speed of the axle assembly. In this sense, it is necessary to define the profile of wheel- rail contact, geometry axle assembly, coefficients of friction at the contact point between the wheel surface and rails as well as the coefficient of friction between the side rails and wreaths bandage given that there is a possibility of contact in two points.

Profile point trolley is defined by UIC regulations as well as for all other rail vehicles so that the profile data point contained in the FAIL S1002-SW is determined on the basis of standard UIC 510-2. Profile rail is also regulated by the UIC regulations so that the data file on the profile rail is fixed.

Defining the geometry of the axle assembly is based on adding the value of half the distance between the wheels and the gap between the wheel and rail. The wheel-rail contact is fully determined; it is necessary to define the coefficients of friction at the contact point between the tread and the rail for the wheels, as well as the coefficients of friction between the wheel and the side rails. In the case of tested trolley, all these coefficients are 0.3. Since the contact nonlinearity is based on the complex conditions determined by geometry of vehicle and track, it is necessary to define the parameters of the tracks.

In the context of defining the track characteristics it is possible to enter values for the lateral and vertical damping and rigidity. After complete definition of all the previously mentioned parameters it is necessary to define the parameters that should be output from the analysis. Considering that there are experimental data on the vertical acceleration at the axle bearing housing assembly in the center of mass, the defined output channels carry supple on the vertical accelerations on the bearing housing and the center of gravity of mass trolleys.

5. RESULTS OF DYNAMIC ANALYSES

5.1. Testing of the vertical acceleration

Vertical acceleration significantly affects the stability and tranquility of rail vehicle. The standard UIC 518 fully defined the conditions of measurement, analysis and limit values of vertical acceleration. In addition to the general conditions, there are those related to the testing of vehicles which limit the speed or when the measurement is done at speeds below 120 km/h as well as axle loads must not exceed the limit value of $2Q_0 <$ 225 kN, provided that the test speed: $V=1.1V_{DOZ}$ with a minimum V_{DOZ}+10[km/h] of tolerance: ± 5km/h.

The conditions related to the state and the selection of tracks for testing are also listed as follows: the test is performed on the track that is used for regular excavation, on the condition that rail must be dry. The measurement is carried out at speeds of less than 120 km/h. The limit values of vertical acceleration on the body -for vehicles towing vehicles (with single suspension $(Z_S)_{lim} = 4$ m/s^2.

The virtual measuring of vertical acceleration in a box above the front axle assembly trolley and above the point on the bearing housing shaft has been done. The measured values are analyzed in the time and frequency domain. Maximum value of lateral acceleration is $(Y_S)_{lim} = 4$ m/s^2. Software package has the ability to identify and graphically display lateral acceleration, as well as the relationship between lateral and vertical forces Y/Q and other dynamic parameters that determine the safety and tranquility of vehicles.

The values of vertical acceleration under different conditions are given in the time and frequency domain. The vertical acceleration, at the bearing of axle, is presented in Fig. 1.

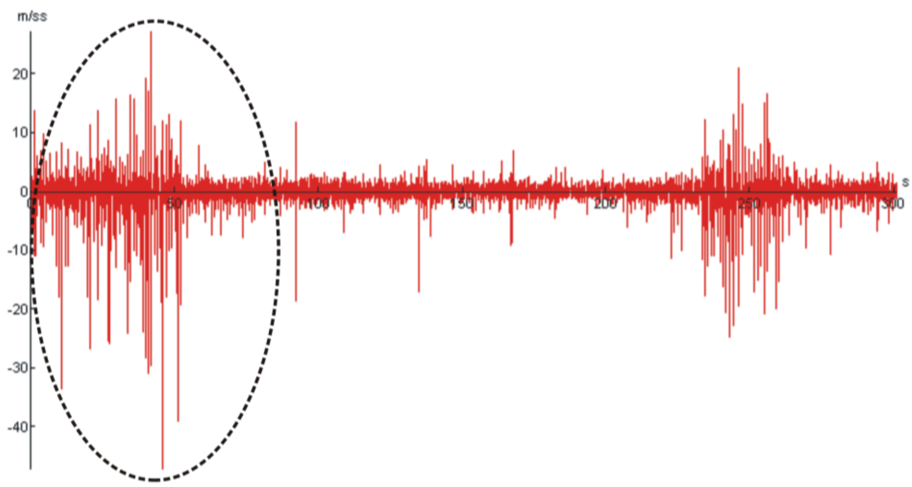

Fig. 1 Vertical acceleration in time domain

The values of the maximum amplitude of acceleration are in the range of 0-60 s. The time range of 0-60 s corresponds with the scope of the distance from 0-1000 m presented in Fig. 2. In the period of 80-230 s the movement stabilization occurs and the maximum amplitude of acceleration is in the range of from -10 to 10 m/s^2. Then, in the period of 230-270 s amplitude appears in the range of from -20 to 20 m/s^2, and then the vehicle is stable.

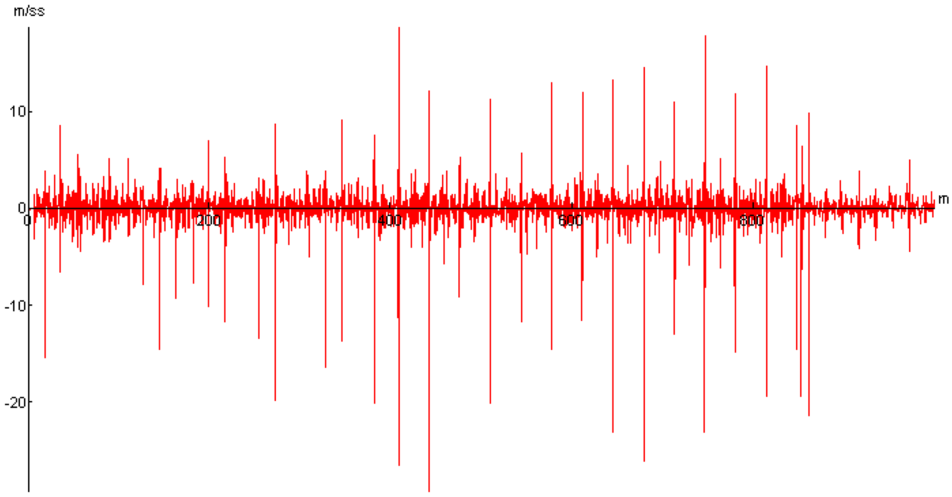

Fig. 2 Vertical acceleration with respect to distance traveled 0-1000m

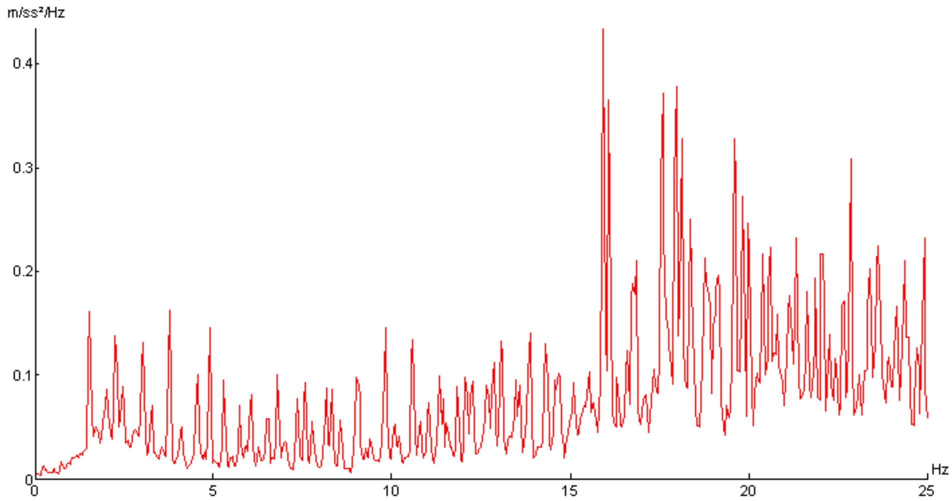

Fig. 3 Vertical acceleration in frequency domain

Displaying data in frequency domain $S(f)$ depending on frequency f is presented in Fig.3. Frequency functions of the signal have the unit of acceleration ((mm/s^2)^2)/Hz. If amplitude a mm/s^2 of a signal is G(f) =S(f)Δf the acceleration can be calculated

$a = \sqrt{2G(f)}$ as stated in report ERRI B 153/RP 21 [12]. Length of FFT analysis in post processing is 1024 points, which gives 512 sampling points in the frequency domain $\Delta f=1/(1024\Delta t)$, time step is set by software while the maximum frequency can be $f_{max}=1/(2\Delta t)$. The result of measuring the vertical acceleration over the trolleys bearing housing axle assembly at a speed indicating extreme values of vertical acceleration -29.21 and 18.64 m/s^2, Fig. 4. These values greatly exceed the limit, according to UIC 518, of 4 m/s^2 or are not relevant to the comparison given that it is not a suspended part.

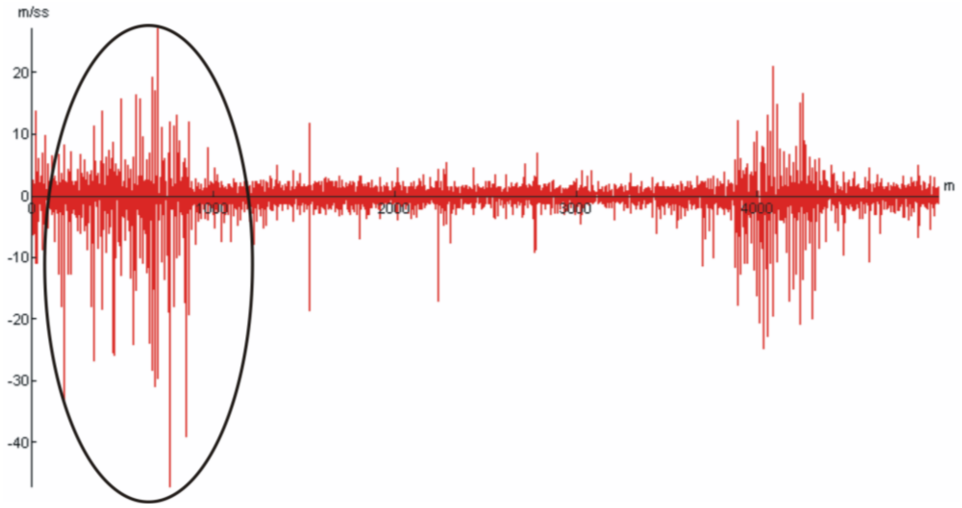

Fig. 4 Vertical accelerations at speed of 60 km/h

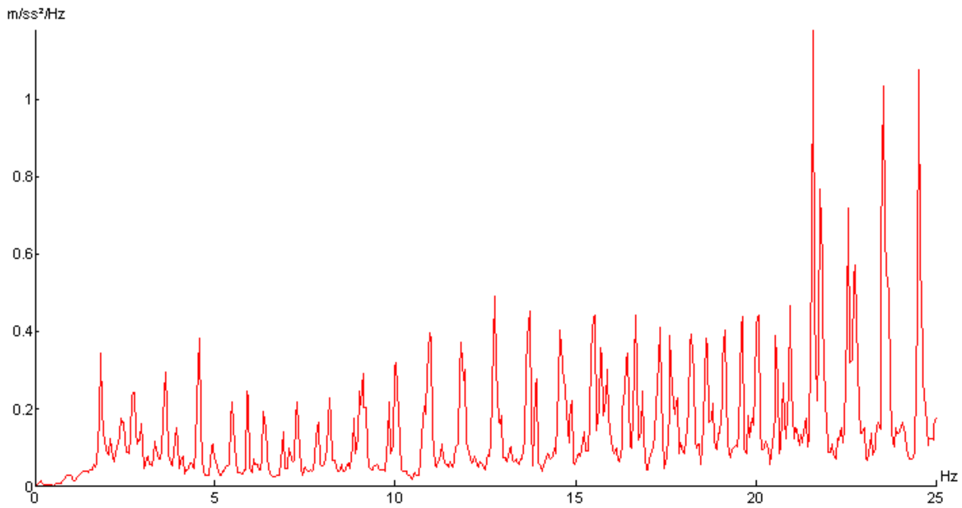

Fig. 5 Vertical acceleration in frequency domain at speed of 60 km/h

The result of measuring the vertical acceleration of trolleys at the measuring point shaft bearing housing assembly at a speed indicating extreme values of vertical acceleration -47.25 m/s^2 and 27.28 m/s^2.

Value of vertical acceleration at the axle bearing housing assembly reaching a value of 47.25 m/s^2 indicates that there are very large dynamic changes - "dynamic impact" - on the bearing. Such data are expected considering that it is a simulation of movement of not suspended wheel-rail contact. Such high values, peaks of maximum amplitude of vertical acceleration are a consequence of information entered on the state of the rails. It is a track that, according to UIC regulations falls into category 3. The vertical acceleration presented in the frequency domain (Fig. 5) is the pillar of the development of software diagnostics of railway vehicles. Substantial research in the field of application of FFT analysis in the diagnosis of the working state condition of rail vehicles are made, and are intended to determine the significant parameters that define the image of the FFT of response. Based on previously presented FFT diagram it can be concluded that the maximum amplitude of the acceleration is in the range of 15-25 Hz. Also, with the increase of speed, the amplitude range is shifted towards higher frequencies.

5.2. The results of measurements of vertical acceleration on the floor of the trolley

Acceleration values under different conditions are given in the time and frequency domain. The vertical acceleration at the measuring point on the trolleys floor is presented in Fig.6. Thus, in the range from 0-100 s, there are peaks of acceleration that exceed 1 m/s^2. Then, in the range of 100-250 s stable movement of the vehicle occurs, and dynamic instability again. In Fig. 7 range 0-80 s is not present in the time domain, but in the domain of the transit distance 0-1000 m.

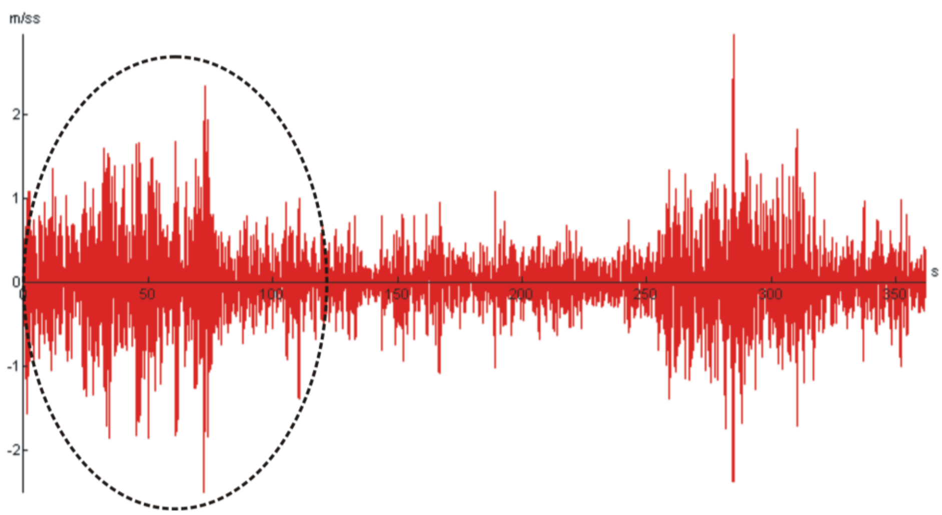

Fig. 6 Vertical acceleration in time domain (50 km/h)

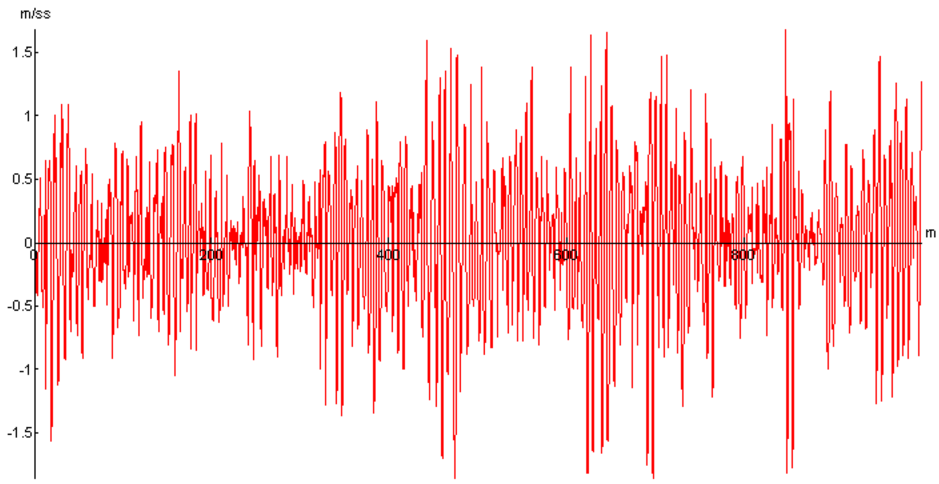

Fig. 7 Vertical acceleration with respect to distance traveled 0-1000 m

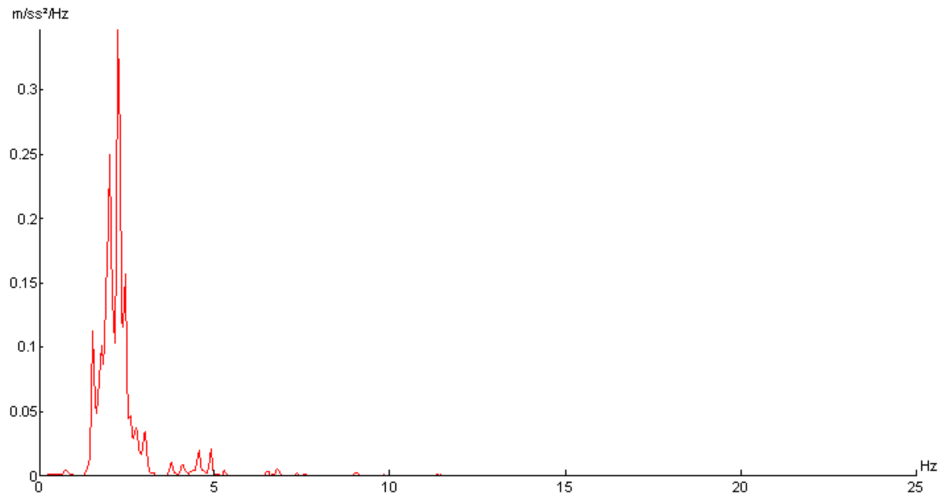

Fig. 8 Vertical acceleration in the frequency domain (50 km/h)

The result of measuring the vertical acceleration of trolleys at the floor measuring point at the speed of 50km/h indicates the extreme values of vertical acceleration of -2.54m/s^2 and 2.946 m/s^2, these values are less than the maximum allowed vertical acceleration defined by the UIC 518, Fig. 8.

The result of measuring the vertical acceleration of trolleys at the floor measuring at the speed of 60km/h indicates the maximum value of the vertical acceleration -2814 m/s^2 and 3122 m/s^2, Fig. 9. The maximum values of vertical acceleration are less than the maximum permissible value according to UIC 518. This indicates the following: provided that a software machine that simulates the dynamics is adequate, trolleys can safely run on the railway track class 3 at speed of 60 km/h, without the risk from the aspect of allowed vertical acceleration. The fact that there is a difference in the diagrams of vertical acceleration at speeds of 50 and 60 km/h indicates that there is no linear correlation of speed and dynamic response of the vehicle. The diagrams of the frequency spectrum of the vertical acceleration indicate that the maximal acceleration amplitude of suspended part of the trolleys, are in the range of from 0-5Hz.

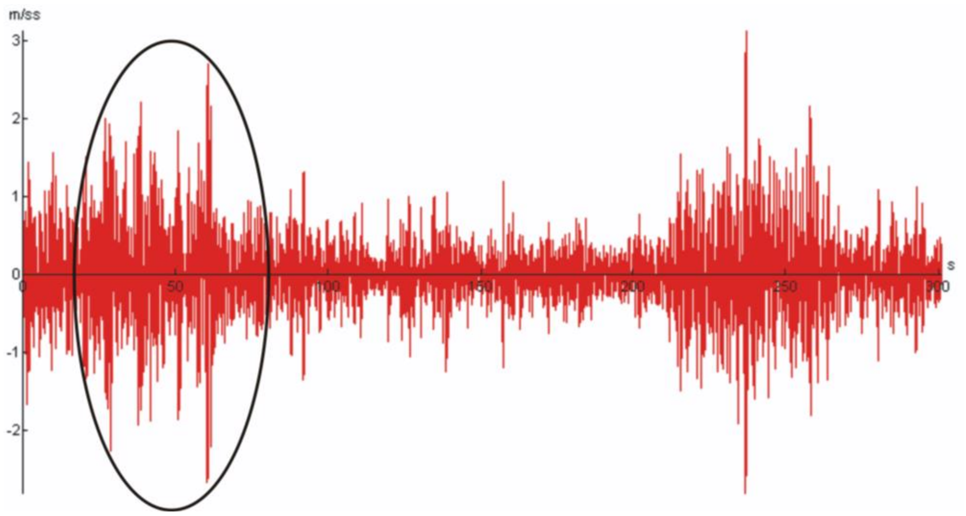

Fig. 9 Vertical acceleration in time domain (60 km/h)

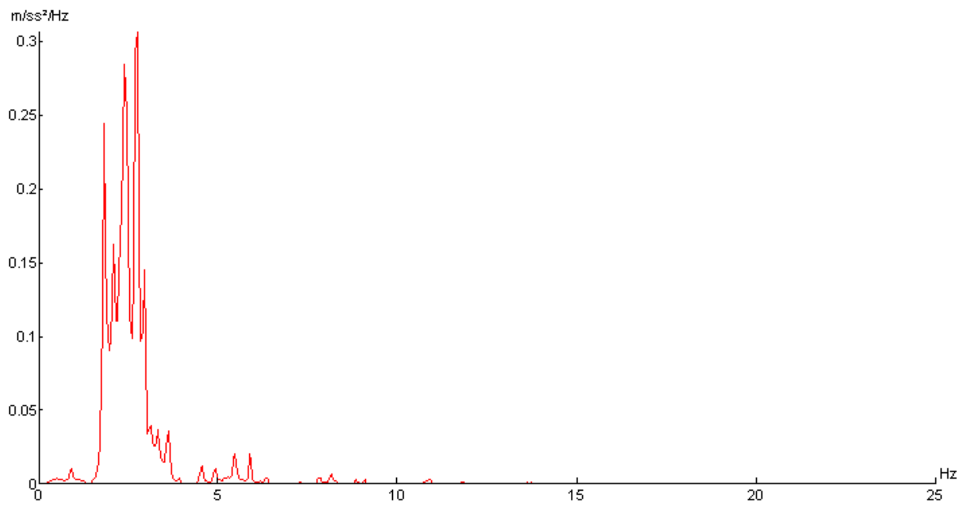

Fig. 10 Vertical acceleration in frequency domain (60 km/h)

6. CONCLUSIONS

Results of the simulation of trolleys speeds 50 to 60 km/h clearly indicate the following: The vertical acceleration at the axle bearing housing assembly is significantly greater than that on the trolleys floor. This is due to the fact that the bearing housing has no damper for axle assembly oscillations, while on the floor the oscillations are damped; damping is in proportion to the suspension (springs and friction dampers).

The vertical accelerations on the trolleys floor and on the bearing housings are greater with increasing speed; this is a consequence of the fact that increase in speed increases the kinetic and potential energy of the vehicle and thus amplitude of oscillation.

In comparative display in the frequency domain differences in frequencies are observed that are accompanied by maximum amplitude of vertical acceleration. Fig. 11 presents a comparative view of vertical acceleration on the bearing housing in the frequency domain at 50 and 60 km/h. From the diagram, it is evident that there is certain amplitude schedule dependence in the frequency range of 0-25Hz. It is also easily noticeable that the maximum amplitude of vertical acceleration at a speed of 60km/h occurs in the range of oscillation frequency of 20-25Hz, and the maximum amplitude of vertical acceleration at a speed of 50km/h occur in the range of oscillation frequency of 15-20Hz.

Fig. 12 shows a comparative view of vertical acceleration on the trolleys floor in the frequency domain of 50 and 60 km/h. From the diagram, it is evident that there is certain amplitude dependence schedule in the frequency range of 0-15Hz. It is also easy noticeable that the amplitude of vertical acceleration at speeds of 50 km/h and 60 km/h occurring in the range of the oscillation frequency of 0-5Hz.

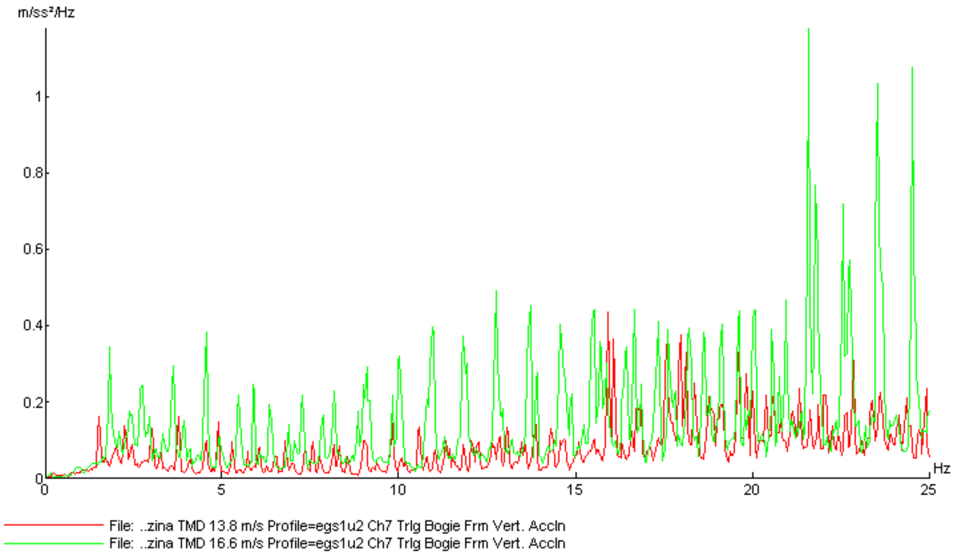

Fig. 11 Frequency spectrum of vertical acceleration of bogie frame

Elements of suspension (springs and dampers) have a role to absorb fluctuations that occur as a result of geometric wheel-rail imperfections. The importance of the suspension elements is presented on comparative diagram in Fig. 13 with the vertical amplitude of vibration in the frequency domain 0-25Hz at the trolleys floor measuring points and bearing housing at a speed of 60km/h.

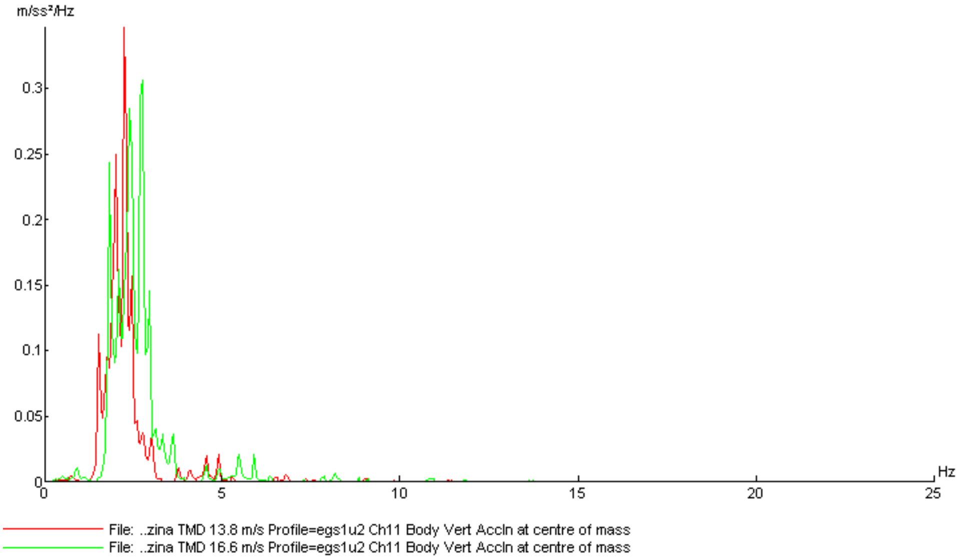

Fig. 12 Frequency spectrum of vertical acceleration at center of mass

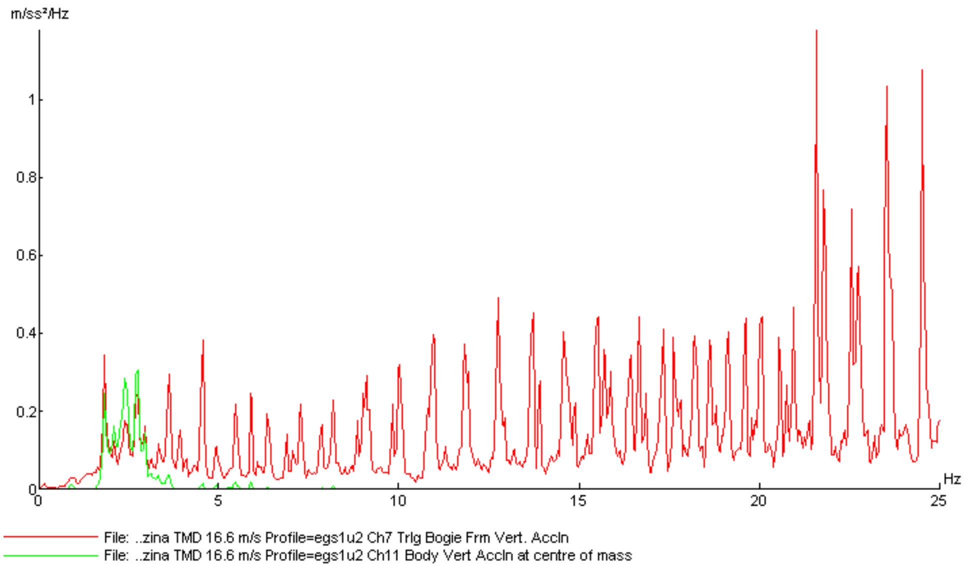

Fig. 13 Compiled frequency spectrum of bogie frame and center of the mass of vertical acceleration

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