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Analysis of the Rail Roughness Influence on Vehicle Dynamic Behavior by Means of Multibody Simulation

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Abstract

The paper presents the results of multibody simulations performed for the model of rail vehicle in order to study the influence of introduced rail roughness on dynamic characteristics of vehicle. Dynamic behavior has been assessed with the accelerations of vibrations measured for chosen points in the model of five-piece tram. Moreover there have been shown calculated normal modes and related natural frequencies of the multibody model. The influence of rail roughness has been analyzed with the tram runs performed for given speeds. Yielded results have enabled to conclude about sources and propagation of vibrations considering kinematic excitation resulting from geometric imperfections of rail.

Keywords: rail vehicle, rail roughness, multibody analysis, vibrations, measurements

1. Introduction

The process of designing a new rail vehicle requires a number of analyses to be performed for the assessment of its static and dynamic properties [3,4]. All analyses can be generally divided into three groups respectively related to safety

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requirements, performance and comfort of traveling. First group deals with material strength and fatigue analyses performed mostly for critical components of a vehicle as well as the analysis of stability of system wheel-rail [12,15]. Second group of carried out analyses stands for the search for possibly most efficient performance of rail vehicle considering less power consumption of drives due to mass reduction and streamlined shape as well as energy harvesting while breaking. Moreover, the comfort of traveling can be taken into account and reduction of acceleration of vibrations and emitted level of acoustic noise [14]. Many of mentioned above analyses can be supported with numerical simulations available within procedures of virtual prototyping. There is a possibility to assess static and dynamic properties of vehicles by the application of both finite element and multibody analyses also including tasks of design optimization [5,8]. There are also known numerical analyses carried out in order to find possibly most robust design configuration under given design constraints [7]. The problem however may arise when quite complicated models are studied and a long computational time is needed. This may disturb sensitivity, an uncertainty analyses as well as design optimization in which significant number of model realizations have to be checked to determine final results. Response surface method can stand for the solution of mentioned problem [2,9,10]. Elaborated metamodels can effectively support performed analyses also including introduced uncertainties [6].

Amongst all rail vehicles trams stand for an interesting group since their specific operation conditions and scope of applications. As they differ from common rail vehicles they cannot be treated as belonging to that group. They characterize smaller dimensions and they are designed for lower operational speeds, usually not exceeding 80km/h [1]. Assumed loads per axis usually do not exceed 10t. Moreover recently designed trams characterize low floor and independent wheels of small turning radius enabling for turn even with radius of down to 15m [11]. It also turns out that the problems unexpected elsewhere can appear in trams. Very small turning radius may cause plastic deformations of both rails and wheels even for quite small speeds [1]. Appearing failures stand for source of rail roughness which in turn manifests itself in progressive force of interaction between rail and wheel. Hence additional kinematic excitations cause continuous degradation of rail. Apart from pure technical aspects of this phenomenon the comfort of traveling gets worse as well.

In the paper there have been presented the results of dynamic analyses performed for the multibody model of five-piece tram. In the analysis the kinematic excitation resulting from rail roughness has been taken into account. Elaborated normal modes and measured accelerations of vibrations have enabled for the study of mostly excited modes of vibrations and helped to find most critical components of the vehicle. The results of performed analysis will be used in further optimization task especially to set up the domain of input design parameters. Applied scheme of undertaken workflow is presented in Fig. 1.

Parallelly determined results of modal and transient analyses can be applied to study normal modes, interactions in the system rail-wheel, acceleration of vibrations and finally may help to find decision parameters, objective function as well as criteria for planned optimization task.

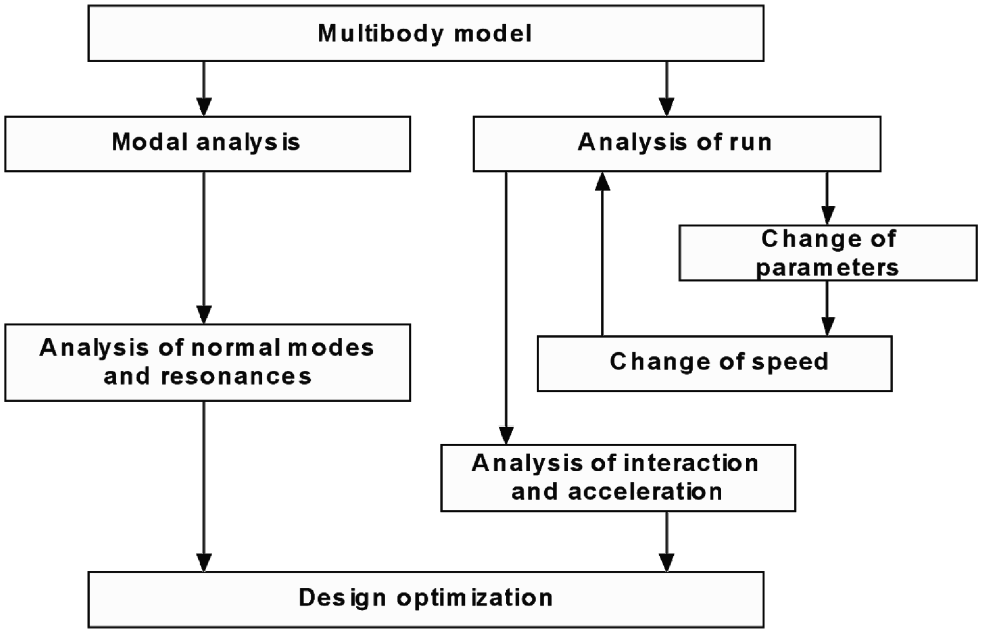

Fig. 1. Exemplary workflow for rail vehicle design

2. Multibody Model

Numerical simulations and data processing have been performed with MSC/Adams and Mathoworks/MATLAB. The object of analysis has stood for the multibody model of five-piece tram and rail with introduced roughness. It is equipped with three bogies with bumpstops, dampers as well as springs for primary and secondary systems. The model is presented in Fig. 2. The total mass of vehicle with passengers equals 48,8t.

The rail and related structural elements have been modeled as flexible and mounted on ideally rigid foundation. The vehicle consists of the system of rigid bodies with determined masses and moments of inertia connected by the weightless joints and components of primary and secondary suspension systems including springs and dampers. For elaborated model there has been performed the assessment of dynamic properties of tram including both modal and transient analyses. As the elements characterizing infinite stiffness the following have been modeled:

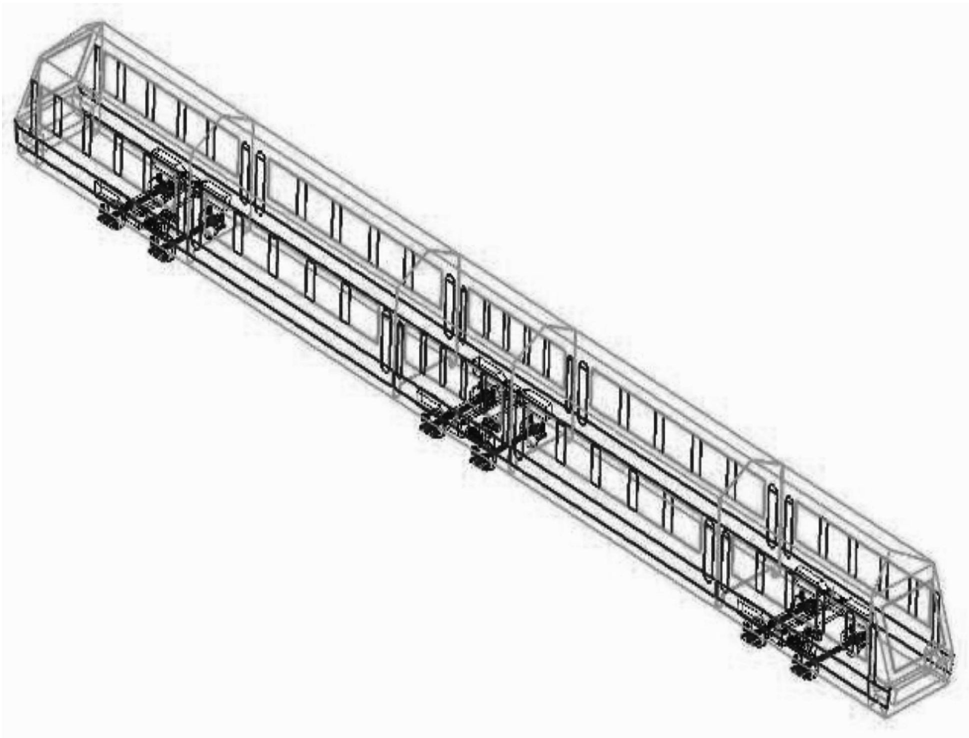

Fig. 2. Multibody model of tram

- bogie frames,
- axles,
- wheels,
- grease-boxes,
- tram frame bodies with passengers.

The suspension systems have included the following substitute elements representing behavior of real construction:

- springs with nonlinear characteristics with introduced damping,
- components with nonlinear stiffness characteristics with defined backlash,
- damping elements with defined damping coefficients for given axes,
- reduced revolution pivot of bogies with defined cylindrical joint to enable for the connection between tram body and bogie.

The first level of suspension system consists of metal-rubber springs. The connection between the tram body and bogie is made with screw springs of the second level of suspension. Damping of vertical vibrations in the first level of suspension is performed with metal-rubber springs. In case of the second level of suspension the vertical and horizontal vibrations are damped with hydraulic dampers.

In the model there has been introduced statistically determined roughness of rails accordingly to the standard ORE B176 [13]. The vertical roughness S_{ZZ}, horizontal roughness S_{YY} and inclination $S_{\phi\phi}$ have been defined as follows:

$$S_{ZZ} = \frac{A_V \Omega_c^2}{(\Omega^2 + \Omega_r^2)(\Omega^2 + \Omega_c^2)} \left[\frac{m^2}{rad/m}\right] \tag{1}$$

$$S_{YY} = \frac{A_A \Omega_c^2}{(\Omega^2 + \Omega_r^2)(\Omega^2 + \Omega_c^2)} \left[\frac{m^2}{rad/m}\right] \tag{2}$$

$$S_{\phi\phi} = \frac{A_V \Omega_c^2 \Omega^2}{(\Omega^2 + \Omega_r^2)(\Omega^2 + \Omega_c^2)(\Omega^2 + \Omega_s^2)} \frac{1}{b^2} \left[\frac{rad^2}{rad/m}\right] \tag{3}$$

The values of auxiliary parameters are: $\Omega_c = 0.8246 rad/m$, $\Omega_r = 0.0206 rad/m$, $\Omega_s = 0.4380 rad/m$, $b = 0.75m$, $A_V = 1.08 \cdot 10^{-6} m \cdot rad$, $A_A = 6.125 \cdot 10^{-7} m \cdot rad$. Introduced roughness causes the wave lengths to be within the range from 2m to 100m. Equations (1), (2) and (3) are elaborated accordingly to performed studies and measurements for main railway tracks in the countries belonging to European Union. Although the conditions considered for tramway tracks may be different from those ones assumed for main railway tracks, mainly on permissible levels, there has been still treated as justified the application of available data anyway for tram rails as there is no authoritative reference guidelines like in case of main railway tracks. The consideration of vertical and horizontal roughness as well as their mutual relationships enables for the simulation of all possible cases that occur while common run on rail.

3. Modal Analysis

The modal analysis has been performed in order to find normal modes of vibrations of the model and related resonance frequencies and damping coefficients. This analysis seems to be important in the context of correct concluding about expected dynamic behavior of the tram under given kinematic excitations. Modal analysis makes possible the prediction of contributions of found normal modes in overall vibrations which appear for assumed rail roughness and speed. Moreover the analysis also allows for the assessment of the properties of suspension system including stiffness and damping coefficients of springs and dampers respectively. Usually modal analysis is carried out already at an early stage of design process to prevent from unexpected dynamic behavior of tram that could result in failure mainly due to instability of wheel motion. Crude model of tram consisting of rigid bodies, joints and parameterized suspension systems allows for the search of most critical components and then change their characteristics before any further and more detailed analysis is performed.

Selected resonance frequencies and damping coefficients which have been determined for low-frequency normal modes are listed in Table 1.

Table 1

Resonance frequencies and damping coefficients for selected normal modes

No. of mode	Frequency [Hz]	Damping coefficient [-]	No. of mode	Frequency [Hz]	Damping coefficient [-]
1	0.59	0.03	20	11.10	0.30
2	0.72	0.01	21	11.66	0.30
3	0.97	0.03	22	12.10	0.02
4	1.17	0.01	23	12.11	0.02
5	1.17	0.02	24	12.15	0.02
6	1.25	0.08	25	12.15	0.02
7	1.47	0.06	26	12.15	0.02
8	1.51	0.05	27	12.15	0.02
9	1.53	0.10	28	13.50	0.06
10	1.74	0.02	29	13.53	0.07
11	2.05	0.03	30	14.83	0.33
12	5.71	0.02	31	14.85	0.33
13	6.49	0.02	32	14.22	0.10
14	6.50	0.02	33	15.78	0.26
15	9.63	0.23	34	24.00	0.04
16	9.56	0.02	35	24.22	0.04
17	9.63	0.02	36	27.54	0.03
18	9.63	0.02	37	28.79	0.03
19	10.32	0.06	38	28.82	0.03

The most important modes of vibrations are those ones which characterize the lowest values of resonance frequencies. It is so because their presence can mostly disturb normal operation and stability of the construction of tram as well as they play important role in terms of the comfort of travelling subjectively assessed by passengers. For example the vibrations present for frequency 0.5 Hz with large enough amplitude of displacement may cause a travel sickness. Vibrations for higher frequencies in turn do not result in significant decrease of the traveling comfort however they may result in instable structural behavior, especially in case of coupling observed between vibrations of bogies and bodies. Fig. 3 up to Fig. 9 present exemplary determined normal modes.

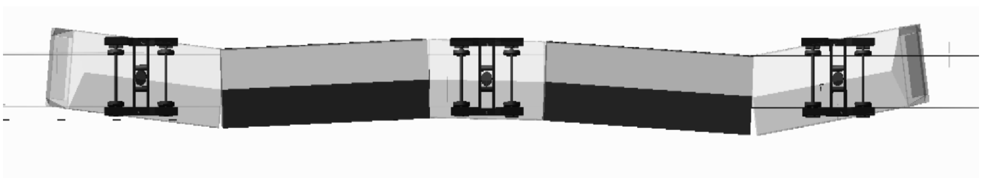

Fig. 3. Normal mode no. 1 – transversal vibrations for 0.59Hz

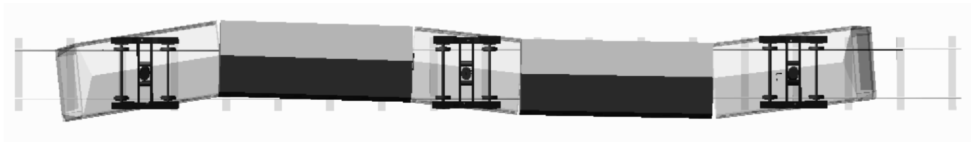

Fig. 4. Normal mode no. 2 – transversal vibrations for 0.72Hz

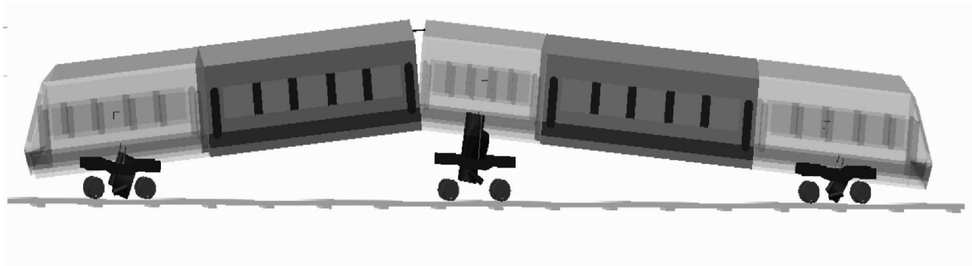

Fig. 5. Normal mode no. 6 – vertical vibrations for 1.25Hz

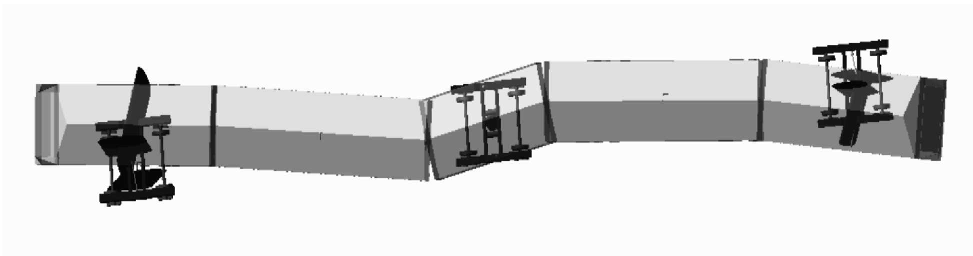

Fig. 6. Normal mode no. 10 – vertical vibrations for 1.74Hz

Fig. 7. Normal mode no. 19 – torsion of bogie for10.32Hz

For safety of tram run it is very important to prevent from coupling between vibrations of components localized in the areas of bogies. Normal modes which are observed for mentioned areas (examples are presented in Fig.s 7, 8 and 9) when excited by introduced geometric imperfections of rail may finally lead to failure because of loosing stability in the system wheel-rail. Moreover there may be exceeded allowed material strength due to fatigue phenomenon in case of long operation of critical components at resonance frequencies. Failures which occur

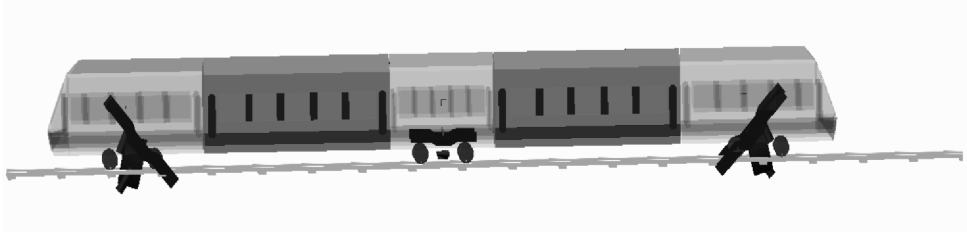

Fig. 8. Normal mode no. 28 – torsion of bogies for 13.50Hz

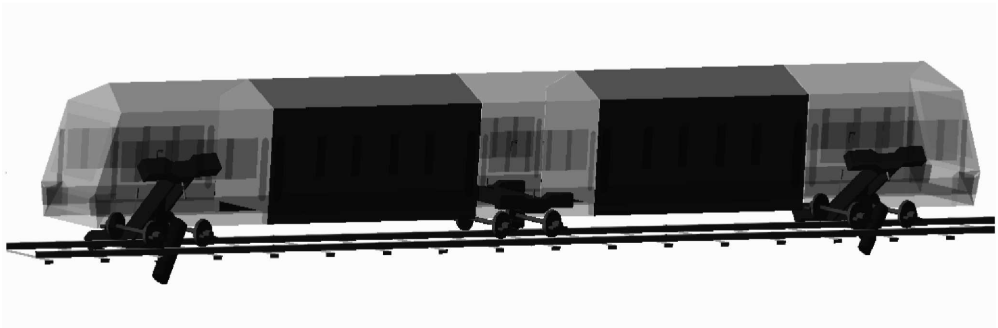

Fig. 9. Normal mode no. 32 – torsion of bogies along longitudinal axis for 14.22Hz

in bogies are very dangerous since they usually result in series accidents. There should be however mentioned the observation that not all dangerous normal modes are expected to appear especially with highest amplitudes of displacement as well. Some of them characterize quite high value of modal damping coefficients, e.g. 10% for mode number 32 (Fig. 9, for 14.22Hz), which results in considerable large damping of given mode although the frequency of kinematic excitation equals resonance frequency. Modes no. 19 and 28, in turn, seem to be more sensitive to external excitation at their resonance frequencies, both characterizing damping coefficients of about 6%.

4. Dynamic Behavior of Tram

There have been performed a series of transient analyses of tram run for different speeds on straight rail. For chosen components of model the accelerations of vibrations have been determined. Rail roughness has been randomly generated accordingly to the procedure implemented in MSC/Adams. There has been assumed uniform probability distribution of kinematic excitation which means equal contribution of waves of different lengths accordingly to formulas presented in section 2.

The tram runs have been performed for selected speeds from the range 40 km/h up to 75 km/h. Both vertical and horizontal transversal directions have been taken into account for the measurement of acceleration of vibrations. Fig. 10 presents contributions of vertical vibrations in frequency domain determined for the first bogie. Fig. 11 shows how the frequency distribution changes in the time domain.

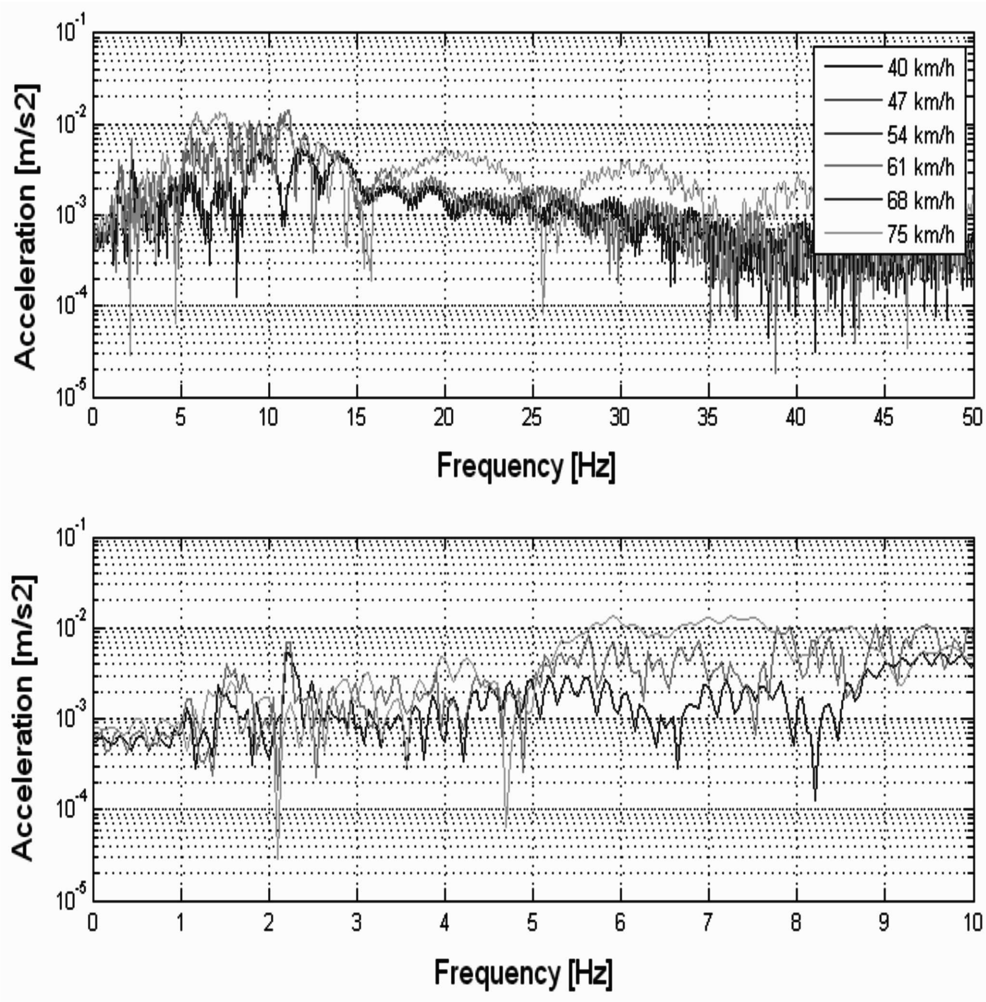

Fig. 10. Frequency characteristics of vertical vibrations present in the first bogie

For the lower speed, i.e. 40km/h, there have been identified two areas in frequency domain for which higher values of acceleration have been found. Related normal modes with resonance frequencies 2.05Hz (with low value of damping coefficient 3%) and 10.32Hz (damping coefficient - 6%) respectively stand for transversal vibrations of tram bodies and torsion of second bogie. The coupling of described modes may be dangerous especially in case of high amplitude of kinematic excitation. Mentioned situation disappears when increasing the speed up to 60km/h. It has

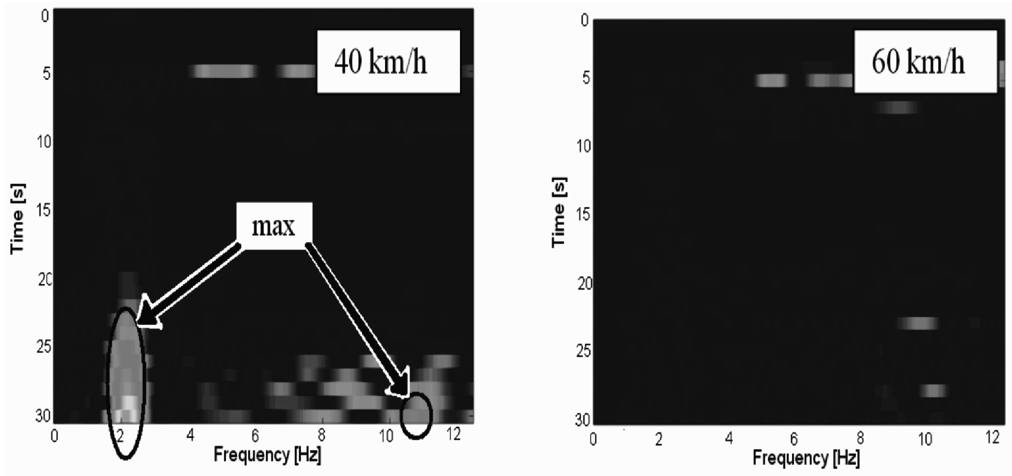

Fig. 11. Frequency distribution of acceleration of vertical vibrations in the first bogie in time domain for different speeds

turned out that there is no normal mode observed for which significant amplitude of vibrations could occur. Fig. 12 presents contributions of vertical vibrations in frequency domain determined for the first wagon. Fig. 13 shows how the frequency distribution changes in the time domain for both vertical and transversal directions.

From the analysis of amplitudes of vibrations presented in Fig. 12 there can be concluded their significant increase in the frequency range from 1Hz up to 2.5Hz for all checked speeds. Similar observations have been formulated for the other wagons as well. In given range of frequency there are contributions of many normal modes characterizing similar shapes and related to transversal vibrations of the tram bodies (as presented in Fig. 3). Some of included normal modes have low value of modal damping coefficient down to 1% which may result in considerable transversal motion of bodies even in case when relatively small roughness of rails is taken into account.

Scalograms presented in Fig. 13 make possible the analysis of influence of assumed kinematic excitation separately on vertical and transversal vibrations for exemplary speed 60 km/h. In case of transversal vibrations there is observed the decrease of frequencies for which maximal amplitudes of acceleration of vibrations appear (approximately 2Hz for vertical vibrations where as 1Hz for transversal ones). It results from larger number of normal modes defined as transversal vibrations of tram bodies found for the range of the lowest resonant frequencies, i.e. about 1Hz, rather than for about 2 Hz when the contribution of those ones related to vertical vibrations grows.

Finally the following main normal modes have been identified which characterize considerable acceleration of vibrations:

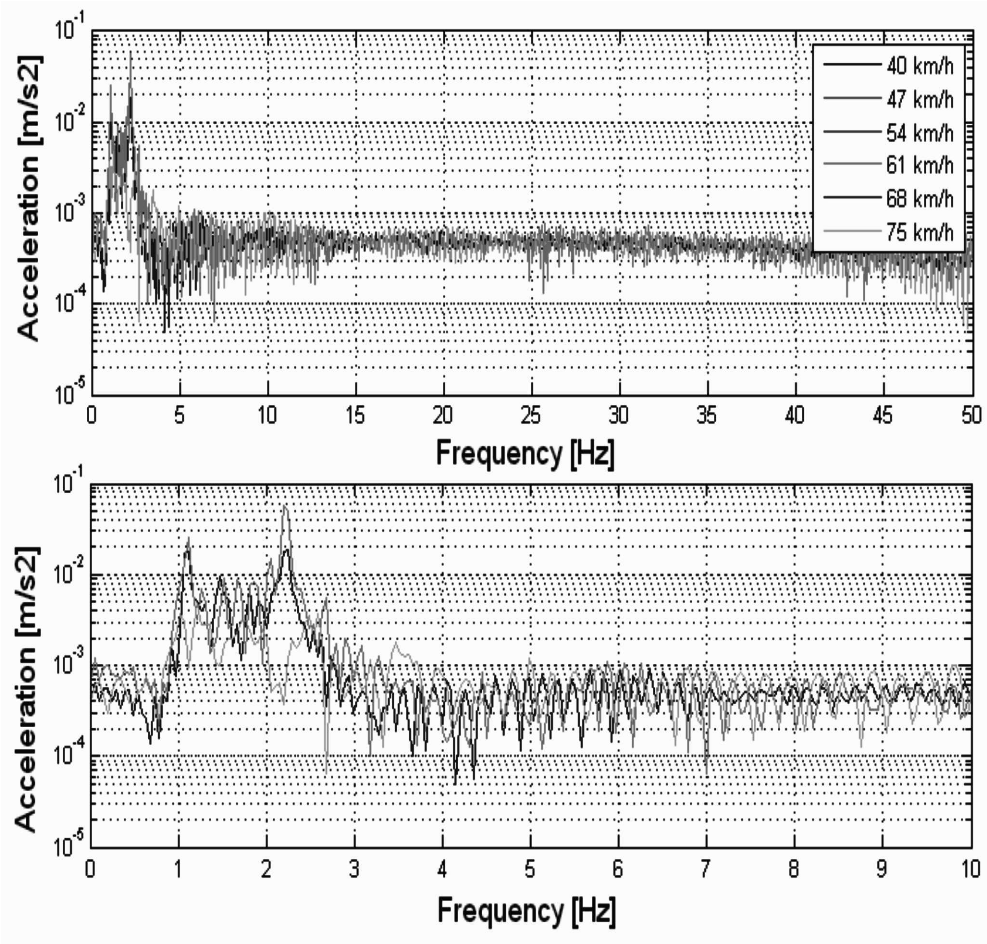

Fig. 12. Frequency characteristics of vertical vibrations for the first wagon

— vertical vibrations of tram body with the largest value of the accelerations measured in the middle wagon (Fig. 5); the shape of normal mode is not a symmetrical one and it has resulted from specificity of applied kinematic scheme, therefore there may appear some differences in accelerations of vibrations; mentioned mode appears twice for frequencies 1.24Hz and 1.7Hz which are also included in the area of maximal values of acceleration presented in scalogram in Fig. 13 on the left.

— transversal vibrations (the example is shown in Fig. 3) which are present for resonance frequencies from 0.5Hz up to 2Hz; related normal modes are responsible for transversal vibrations observed along the whole model of tram; they interfere with vertical vibrations and make them have larger amplitude of acceleration of vibrations; having similar resonance frequencies the transversal

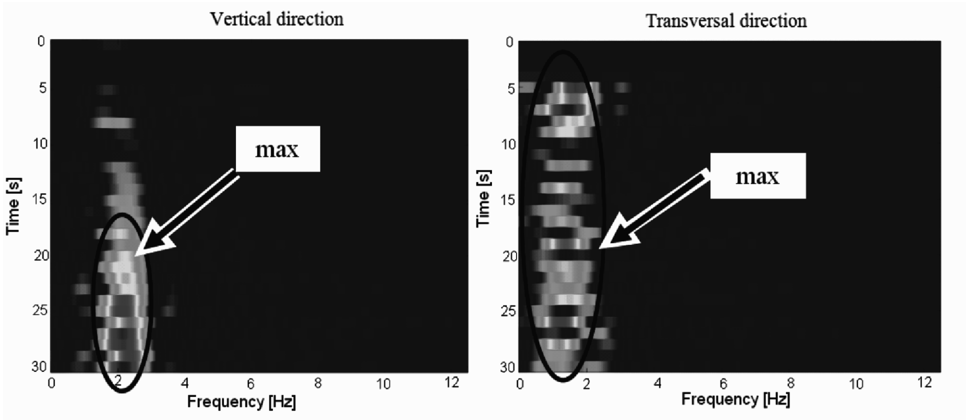

Fig. 13. Frequency distribution of acceleration of vertical and transversal vibrations in the first wagon in time domain for speed 60 km/h

and vertical normal modes of tram bodies may interact and therefore also vertical vibrations can grow simultaneously with transversal ones.

5. Summary and Concluding Remarks

In the paper there have been presented the results of analysis of dynamic behavior of multibody model of five-piece tram. In performed analysis the roughness of rail has been considered and its influence on acceleration of vibrations in selected components of the model has been studied. Obtained results have been used for the formulation of conclusions on potential sources of vibrations in vehicle in the context of assumed kinematic excitations resulting from rail roughness.

Taking into consideration the construction of vehicle and the results of both modal and transient analyses, the following components of the model should stand for an object of further analysis in the context of dynamic behavior. First, the localization of flexible joint in the middle bogie may result in large amplitudes of accelerations of vibrations of tram bodies for low frequencies especially in case of vertical curves and rail roughness with waves which length approximately equal spacing between bogies. For given conditions it may happen an unexpected grow of above mentioned vibrations. Similar phenomenon may appear for transversal direction, excluding that in the localization of the middle bogie there may occur both the node of vibrations and highest amplitude of acceleration. Moreover the coupling between different normal modes may act as additional excitation for all components mounted on bogie frames which should be checked anyway in the future study.

As reported in the paper the dynamic behavior of the tram may significantly differ over varying speed therefore overall rational analysis for the whole speed domain is needed to correctly conclude about real variation of dynamic properties for different conditions. Frequency range from 1Hz up to 2.5Hz seems to be critical in the context of large number of included normal modes characterizing both vertical and transversal vibrations. Assumed rail roughness has also resulted in dangerous coupling between different modes for mentioned frequency domain.

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