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Vertical Vibration of the Vehicle when Crossing over Transverse Speed Bumps

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The experimental investigation of the vertical vibration of the vehicle is presented. The vibration of the two axles vehicle is excited by crossing of the vehicle over the transverse speed bumps on the road. The methodology is oriented upon the quarter, half and full model solution. The full geometry model is the most suitable model for the vertical vibration of the vehicle. The full geometry model is capable to describe geometric and construction asymmetry of the vehicle. The methodology of an analytical solution of the symmetry and asymmetry distribution of the vehicle and load is presented. Also, symmetric and asymmetric excitation is involved to the solution. The Heaviside's function is applied for the excitation of vibrations. The presented methodology was applied to the experimental work with trolleybus Tr 21. The normalized speed bumps were used for the experimental setup. The vertical displacements, velocities and accelerations of axles, body of the vehicle. The acceleration of the vehicle chassis under driver seat was also recorded. This accelerations have significant effect on the comfort of driver and consequently on his/her fatigue and health condition.

Keywords: Vibration, Trolley Bus, Retarder, Axle, Self-Supporting Body

1 Introduction

The vertical vibrations of the vehicle excited by crossing over road speed bumps are investigated. The quarter or half geometry model is the most often applied model in the literature. There is lack information about investigation of the full geometry model [1]. Quarter model is common model with higher degree of freedom (symmetrical model alongside both planar axis). Less often, half models are used (symmetrical model alongside longitudinal axis) and the displacement of the center of gravity is neglected as differences between intensity of viscous dumping, springs stiffness and its combinations.

The effect of asymmetry on the vertical vibration of the vehicle requires detailed analysis especially for spatial model. Three basic tasks of asymmetry can be distinguished regarding axis location of geometrical symmetry. These axes are determined by mutually perpendicular axes of vehicle geometry. These tasks are:

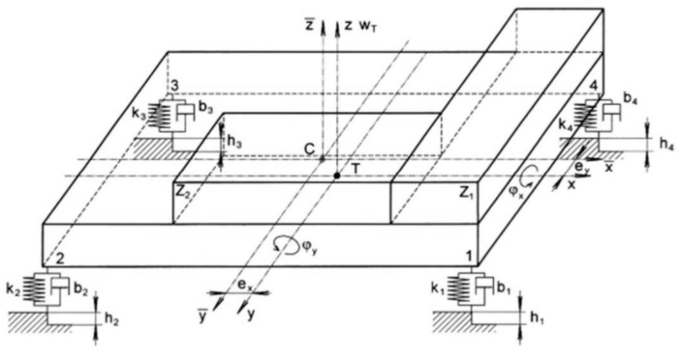
- asymmetrical distribution of vehicle mass with respect to geometric symmetry axes,
- asymmetrical allocations of elastic and dissipative elements and its mechanical properties (the linear relationships of particular quantities and small displacements and rotations of system parts are assumed),
- asymmetrical kinematics excitation, e.g. field of the roughness of the surface road, which defines the kinematic excitation of the system in the contact place wheel-road (or wheel-rail).

The essential condition for experimental and theoretical (analytic, numerical, simulation) investigation of the vertical vibration of the vehicle is choice and definition of suitable spatial model [1, 2, 4, 8].

2 Model definition

For systematic analysis of various cases of asymmetry loading and oscillation it was necessary to defined simple but sufficiently general 3D model of the vehicle, satisfying required assumptions and its variants. The proposed model is represented by rigid plate. The plate is supported on four springs. The springs have linear characteristics. The asymmetry of load is represented by two weights. The geometrical characteristics of the model (center of mass, position of the central axes of inertial mass) can be changed by spatial distribution of these weights, see fig. 1, [1, 4].

The center of gravity T is deflected far from geometric center C of model by the change of the weight position. In case of symmetric system $T \equiv C$, main center axes of inertial are coincident with axes of geometric symmetry. In case of asymmetric system $T \neq C$, the main center axes of inertia are not coincident with axes of geometric symmetry (offset from each other).



Legend: 1, 2, 3, 4 – position of the elastically dumping elements, C – geometrical center, T – gravity center, x, y, z – axes of symmetry, k – spring stiffness, b – dumping, e_x, e_y – eccentricity of the gravity center under asymmetric loading, h – height of the roughness (kinetic excitation), w – vertical displacement, φ_x, φ_y – rotations around axes of symmetry.

Fig. 1 *3D vehicle model*

The criteria for comparison of the particular cases of asymmetry (loading and excitation) has been chosen to be the time trend of the vertical displacement – the change of vertical coordinate of the gravity center $w_T(t)$, time trend of model rotation around central axes of inertia $\varphi_x(t)$ and $\varphi_y(t)$, and time trend of corresponding velocities and accelerations components. These three components define time trend of displacements, velocities $\dot{w}_T(t), \dot{\phi}_x(t), \dot{\varphi}_y(t)$ and accelerations $\ddot{w}_T(t), \ddot{\phi}_x(t), \ddot{\phi}_y(t)$ in the arbitrary points $A(x_A, y_A)$ of system with three degrees of freedom, e.g. in the location of the support

$$\begin{aligned} w_A(t) &= w_T(t) + x_A\phi_y(t) - y_A\phi_x(t), \\ \dot{w}_A(t) &= \dot{w}_T(t) + x_A\dot{\phi}_y(t) - y_A\dot{\phi}_x(t), \\ \ddot{w}_A(t) &= \ddot{w}_T(t) + x_A\ddot{\phi}_y(t) - y_A\ddot{\phi}_x(t), \end{aligned} \tag{1}$$

where index A = 1, 2, 3, 4.

In the place of rigid plate support. i. e. in the points A = 1, 2, 3, 4, the coordinates x, y are defined with regard to central axes of inertia

$$\begin{aligned} x_1 &= l_{x1} - e_x, & y_1 &= l_{y1} - e_y, \\ x_2 &= -\left(l_{x2} - e_x\right), & y_2 &= l_{y2} - e_y, \\ x_3 &= -\left(l_{x3} + e_x\right), & y_3 &= -\left(l_{y3} + e_y\right), \\ x_4 &= l_{x4} - e_x, & y_4 &= -\left(l_{y4} + e_y\right). \end{aligned} \tag{2}$$

After substation of eq. (2) into the eq. (1) and arrangement.

$$\begin{aligned} w_1(t) &= w_T(t) + \left(l_{x1} - e_x\right)\varphi_y(t) - \left(l_{y1} - e_y\right)\varphi_x(t) - h_1(t) \\ w_2(t) &= w_T(t) - \left(l_{x2} + e_x\right)\varphi_y(t) - \left(l_{y2} - e_y\right)\varphi_x(t) - h_2(t) \\ w_3(t) &= w_T(t) - \left(l_{x3} + e_x\right)\varphi_y(t) + \left(l_{y3} - e_y\right)\varphi_x(t) - h_3(t) \\ w_4(t) &= w_T(t) + \left(l_{x4} - e_x\right)\varphi_y(t) + \left(l_{y4} + e_y\right)\varphi_x(t) - h_4(t) \end{aligned} \tag{3}$$

The resulting motion equation [1, 3]

$$\mathbf{M}\ddot{q}_j(t) + \mathbf{B}\dot{q}_j(t) + \mathbf{K}q_j(t) = \mathbf{F}_j(t) \tag{4}$$

where **M** – mass matrix, **B** – dumping matrix, **K** – stiffness matrix, $\mathbf{F}_j$ – generalized function of kinetic excitation.
Complete derivation can be found in [1, 4].

The equation (4) was applied for analytical and experimental investigation of trolleybus Tr 21 (fig. 2). The asymmetry of mass distribution is given by trolleybus construction. Especially, vertical vibrations were recorded under condition of crossing irregularities on a real track [5, 7, 8]. The trolleybus driver is exposed to changing vibration of the vehicle during his/her working period. The irregularities on the road of a town cause the vibration variation. One of the investigation objective is therefore the analysis of vibration of trolleybus chassis in place of driver seat mounting (seat is dumped by air bellows spring and by two hydraulic linear dumpers. The vibration from the vehicle chassis is transmitted through driver's feet and the seat suspension to the entire driver's body. This leads to increased driver fatigue and, over time, to health problems (spine, internal organs, etc.) [6].

Trolleybus parameters are presented in tab. 1 and tab. 2.

Fig. 2 *Investigated trolleybus*

Tab. 1 *Trolleybus dimensions*

Dimension	*Unit*	*value*
Length/High/Width	mm	11 760/3 365/2 500
Curb Weight /Total	kg	10 950/16 900
Wheelbase	mm	5 700

Tab. 2 *Trolleybus description*

Parameter	*Description*
Front axle	Independent suspension, 2 bellows springs + 2 telescopic hydraulic shock absorbers
Rear axle	Rigid portal, 4 bellows springs + 4 telescopic hydraulic shock absorbers
Chassis	Self-supporting body with a frame welded from steel thin-walled profiles, covered with metal panels

3 Solution and discussion

Analytic and experimental methods were used for investigation of the trolleybus crossing over transverse speed bumps. The calculus model was assembled for analytical solution of proposed geometrical model, see fig. 1. In the article, experimental results are presented. Results are focused on vibration of floor in place of driver seat mounting. The vibrations were excited by front axle which had significant effect on the driver comfort.

3.1 Experiments

The experiments were carried out with the above-mentioned vehicle in the premises of the Public Transport Company in Ústí nad Labem. The road roughness was simulated by placing normalized retarders (width × height = 485 × 60 mm) on the road. The velocity of vehicle was 10, 15 and 20 km.hod^{-1} (according to the vehicle speedometer). Also the results for crossing speed bumps side-by-side are presented. First the front axle crossed speed bumps followed by rear axle, see fig. 3.

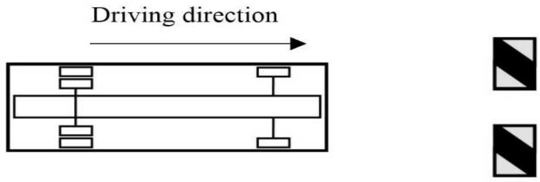


Fig. 3 *Location of speed bumps*

The location of transducers and experiment layout are presented in fig. 4.

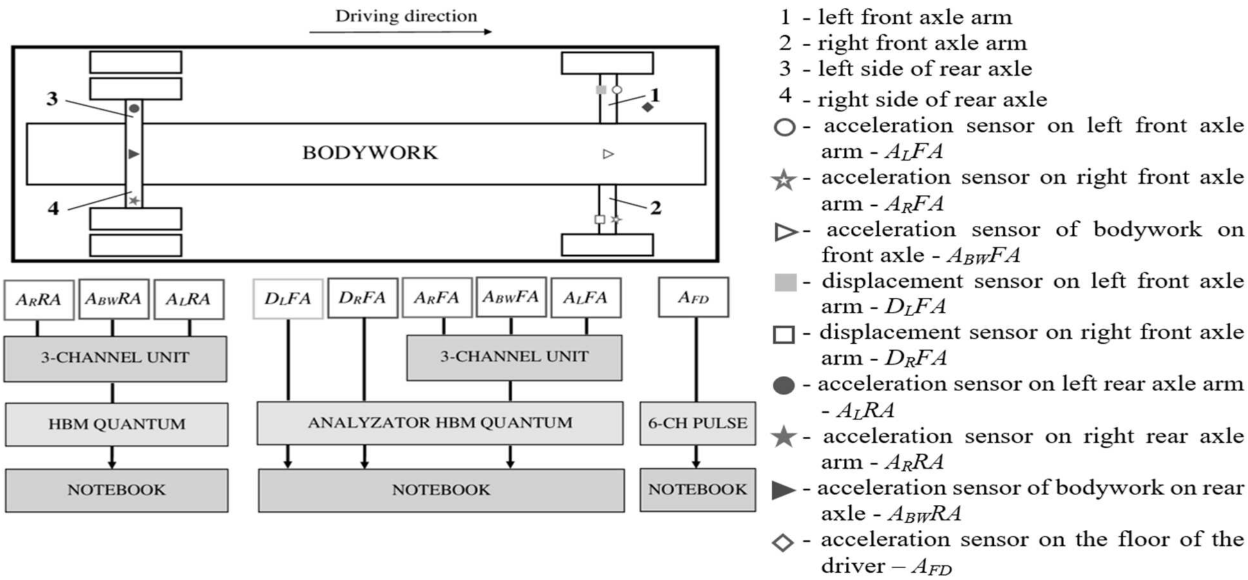


Fig. 4 *Location of transducers and experimental layout*

Vertical displacement – measured at front axles arms in the place of spring mounting from back side (near to middle part of the vehicle), optical transducers optoNCDT 1402, transducers were fixed on the body of the vehicle, see fig. 4.

Acceleration – recorded by one-axis piezoelectric transducer B12 connected to three-channel unit. Two transducers were placed on rear axle, the third one was placed on chassis above axle (in the middle). Accelerometers were located on both arms of front axle, at place of spring connection. The third transducer was located on vehicle chassis again, fig. 4. Transducers for displacement and acceleration measurement were connected to 16 channel control panel HBM, which was connected to PC. Acceleration at place of driver seat was recorded by piezoelectric transducer 4524-B connected to 6 channel control panel Brüel&Kjær PULSE.

Displacement trends measured at front axle arms are presented in fig. 5. The displacement was recorded by optic sensor at velocity 15 km.h^{-1} of vehicle crossing bumps. The speed bumps were placed side-by-side, see fig. 3.

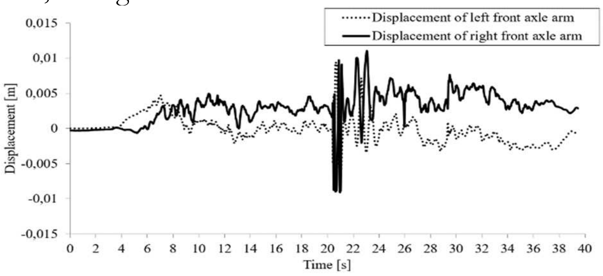


Fig. 5 *Displacements of left and right front axle arm*

After trolleybus start from 3.3 s, differences between left and right sensor record appear. It can be caused by road roughness or difference of spring stiffness and attenuation of dumpers (the trolleybus has been operating for several years). At the moment of speed bumper crossing at 20.8 s, the differences between recorder displacements are quite small circa 3.2 %. The differences after excitation of the system are probably caused by external reasons, i.e. differences between coefficient of dumping, pressure in tires and wear of both springs.

Acceleration record of left and right front axle arm is shown in fig. 6. Maximum value of acceleration can be found at time 20.7 s, at the moment, when wheel hit the speed bump.

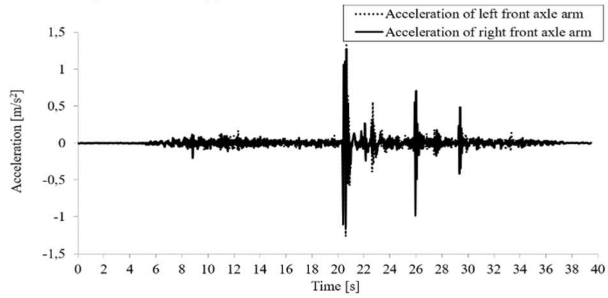


Fig. 6 *Acceleration of left and right front axle arm*

Acceleration value of left arm is 1.32 m.s^{-2} and right arm is 1.26 m.s^{-2} (difference is 14.5 %) at the moment of speed bump crossing. At returning period, the acceleration of left arm is 1.25 m.s-2 and fight arm is 1.09 m.s-2 (difference is 14.5 %). The difference is probably caused but difference between stiffness of springs and attenuation of dumpers. The differences of acceleration of the front axle has small and acceptable values.

In figs. 7 and 8, acceleration of the trolleybus floor in the place of driver seat is presented.

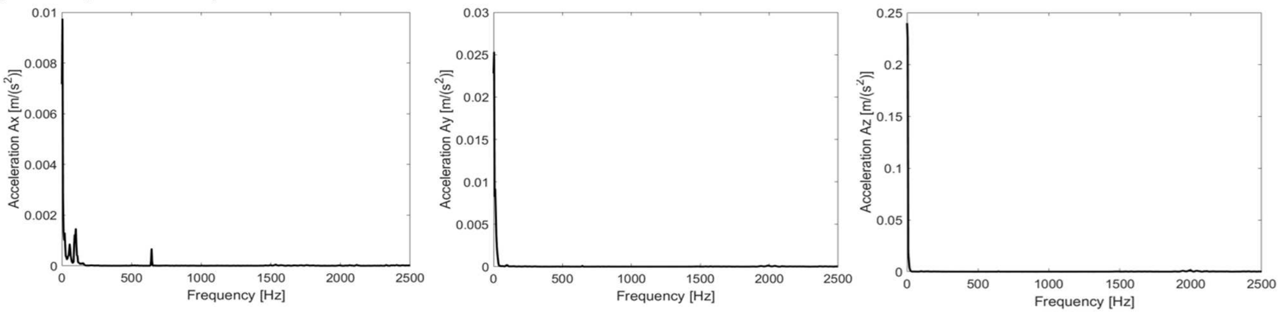


Fig. 7 *Dominant frequencies of trolleybus floor for vehicle velocity 10 km.h^{-1}*

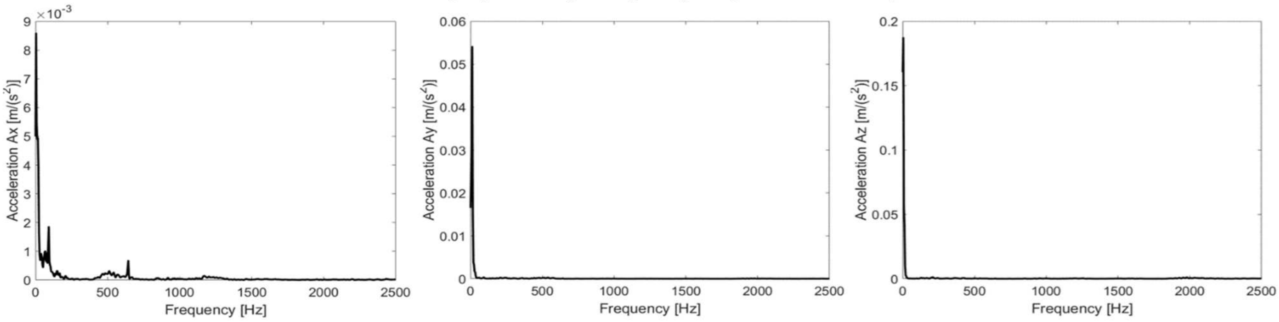


Fig. 8 *Dominant frequencies of trolleybus floor for vehicle velocity 15 km.h^{-1}*

Under velocity 10 km.h^{-1}, accelerations have the highest values in the *x* axis direction, is 0.0097 m.s^{-2}, in y axis direction, is 0.0253 m.s^{-2} and in *z* axis direction, is 0.2251 m.s^{-2}. The frequency of these accelerations is 4 Hz. Other accelerations are significantly lower under rest frequency field, and two order lower in z axis direction (see fig. 7).

Maximal value of accelerations in *x* axis direction is 0.0086 m.s^{-2} and in *z* axis direction, is 0.1871 m.s^{-2}, under velocity of vehicle 15 km.h^{-1}. The frequency of these accelerations is 4 Hz as well. Maximal value of acceleration in *y* axis direction is 0.541 m.s^{-2} at 12 Hz.

Other accelerations are significantly lower under rest frequency field, and two order lower in z axis direction (see fig. 8).

Under velocity 20 km.h^{-1}, accelerations have the highest values in the *x* axis direction, is 0.0366 m.s^{-2} at 16 Hz, in *y* axis direction, is 0.0776 m.s^{-2} at 12 Hz and in *z* axis direction, is 0.1769 m.s^{-2} at 4 Hz, see fig. 9.

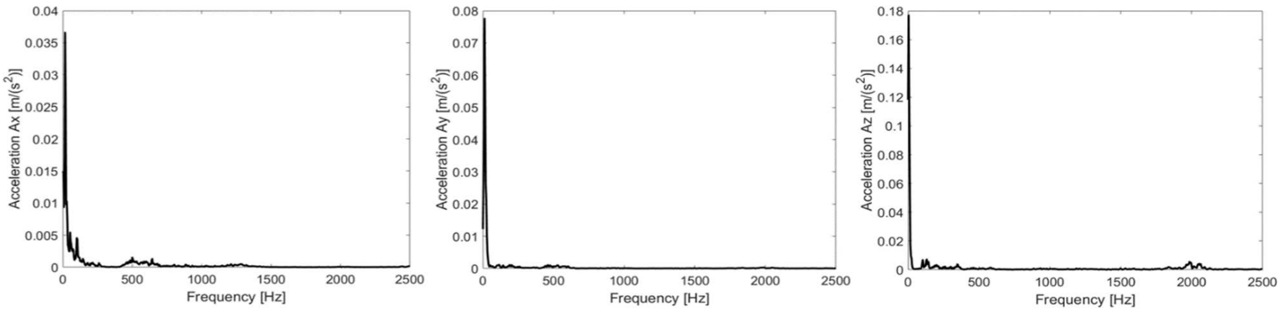


Fig. 9 *Dominant frequencies of trolleybus floor for vehicle velocity 20 km.h^{-1}*

4 Conclusion

The experimental investigation of the vertical vibration of the vehicle were carried out. The results of experiments, which mainly affect the fatigue and health of the driver, are presented in the article. The acceleration in the z-axis direction is crucial. Maximal value of acceleration are under same frequency 4 Hz for all three vehicle velocities. This frequency value is physiologically acceptable, but it is evident that at the lowest speed, i.e. 10 km.hr^{-1} acceleration value is the highest one. Acceleration is lower by 20 % at vehicle velocity 15 km.hr^{-1} and 27 % lower at vehicle velocity 20 km.hr^{-1}. The change in the acceleration value in the x-axis direction at higher speeds, i.e. 15 and 20 km.hr^{-1}, is insignificant and is around 5%. This implies that at a higher speed, there is no large oscillation of the axle and the vehicle overcomes these speed bumps with acceptable drive comfort. It also shows very good suspension and dumping of the front axle.

Acknowledgement

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