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INFLUENCE OF VEHICLE CHARACTERISTICS ON RIDE COMFORT

Rajko Radonjić[1], *Branislav Aleksandrović*[2], *Dragoljub Radonjić*[3], *Aleksandra Janković*[4], *Momir Praščević*[5]

[1]University of Kragujevac, Faculty of Engineering, Serbia, rradonjic@kg.ac.rs
[2]Technical College of Applied Studies Kragujevac, Serbia, banealeksandrovic@gmail.com
[3]University of Kragujevac, Faculty of Engineering, Serbia, drago@kg.ac.rs
[4]University of Kragujevac, Faculty of Engineering, Serbia, alex@kg.ac.rs
[5]University of Nis, Faculty of Occupational Safety, Serbia, momir.prascevic@znrfak.ni.ac.rs

Abstract - *Some actual problems related to examination of vehicle's oscillatory processes in terms of dynamic loads and ride comfort are discussed in this paper. A method for research of influence of the relevant vehicle's characteristics and interactions with the excitations from the environment are presented. An adequate simulation model and experimental system were developed as the base for results verification. The influence of the excitation type from road onto intensity of generated vibration during different weight conditions of the vehicle was analysed. The method for optimization of driver's seat vibratory parameters is proposed. The method is based on the vibrational comfort evaluation criteria.*

1. INTRODUCTION

The oscillatory processes arising from the movement of motor vehicles on the unevenness roadway generate dynamic forces that affect to life of the elements and components of vehicles but also lead to fatigue of the driver and passengers, which is reflected in their ability to work and health [1,2,3,4]. In this regard, a certain constructive measures are undertaken that lead to reduction of the level of dynamic forces, and aimed for optimisation of the characteristics of tires, the characteristics of primary system of elastic suspension, the elastic suspension of the trucks cabins and the commercial vehicles, as well as the characteristics of the seat [5,6,7,8]. The considerable efforts to improve of the oscillatory characteristics of the elastic suspension with passive components were made in the previous period [6,7,9]. However, the results achieved in this field almost reached the limit with respect to the actual requirements and standards relating to the life and the comfort of the vehicle. [5,7]. Further improvements are possible using of new types of suspension systems and elements, components with active and semi-active control, software support and integration of vital vehicle functions [7,9,10] etc.

Some specific features and limitations of passive systems of elastic suspension as well as some phenomena of vehicle's excitation from the environment that must be taken into account during selecting and developing appropriate solutions are discussed in this paper, bearing in mind the above presented problems. This approach is based on the aspect of isolation of human body from the harmful effects of oscillations and the possibilities gained by usage of new types of suspension systems with respect to the development of the trends in this area.

2. METHODOLOGY

This chapter will summarize the subject of work and used methodology based on the conducted analysis and conclusions given in the introduction. The first section deals with the problems of forming appropriate models for simulation-based investigation. In this sense, an oscillatory model of two-axle vehicle whose structure and parameters are shown in Fig.1. was chosen.

The marks in the figures have the following meaning: l - wheelbase, a, b, h - the coordinates of the mass center, c_1, k_1, c_2, k_2 - elastic - damping parameters of elastic suspension of front and rear axle, respectively, c_{1p}, k_{1p}, c_{2p}, k_{2p} - elastic - damping parameters of front and rear tires, respectively, $h(x)$ - the original function of uneven length profile of the soil, $z_o(x)$ - the equivalent function of longitudinal unevenness of the soil profile obtained by filtering the original function through H_1 and H_2 modules, which include the effects of the interaction of subsystems, rubber - ground. Apart from these, the marks of vehicle sprung mass, m, corresponding moment of inertia about the transverse axis of the vehicle, I, unsprung masses, m_1, m_2, the coordinates of the characteristic points, l_s and l_c, the direction and the orientation of velocity, v, are presented in Fig. 1.

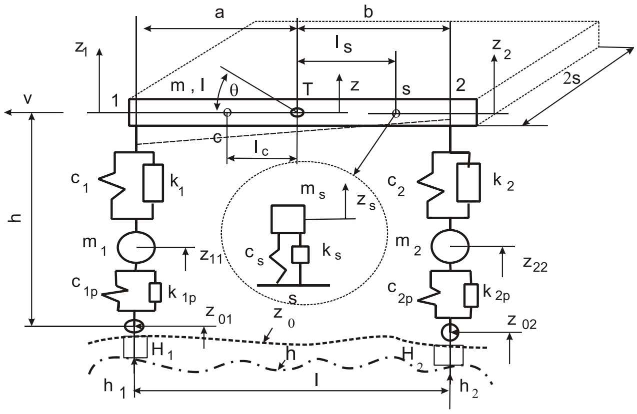


Fig. 1 *Oscillatory model of two-axle vehicle*

In general, the model has five degrees of freedom, two degrees of freedom of elastically sprung mass, m, marks of alternative, $z \rightarrow \theta$, vertical and angular displacement of sprung mass, or $z_1 \rightarrow z_2$, vertical movement of the front and rear side of the sprung mass, respectively. Then one degree of freedom of movement unsprung masses, m_1, m_2, therefore, their vertical displacements, z_1, z_2, respectively, and one degree of freedom of the basic, longitudinal movement of the tractor in the direction of coordinate, x, by velocity, v.

Differential equations in the summary form are written for presented model,

$$m\ddot{z} + F_{ck1} + F_{ck2} = 0 \tag{1}$$

$$-I\ddot{\theta} - F_{ck1}a + F_{ck2}b = 0 \text{ ’’} \tag{2}$$

$$m_1\ddot{z}_{11} - F_{ck1} + F_{pck1} = 0 \tag{3}$$

$$m_2\ddot{z}_{22} - F_{ck2} + F_{pck2} = 0 \tag{4}$$

$$F_{pck1} = F_{zo1}, \qquad F_{pck2} = F_{zo2} \tag{5}$$

$$F_0 = R_f + R_v + R_j + R_\alpha + R_P \tag{6}$$

In addition to the marks given in Fig. 1, the following marks are also used in the above expressions: F_{cki}, i=1,2, the resultants of elastic and damping forces of elastic suspension of the front axle, i=1, and the rear axle, i=2; F_{cki}, i=1,2, the resultants of elastic and damping forces at the front tires, i=1, and the rear tires, i=2; F_{zoi}, i=1, 2, excitation forces in the tire and road contact of the front wheels, i=1, and rear wheels i=2; F_0 – vehicle’s tractive force; R_f, R_v, R_α, R_j, R_p, the rolling resistance of air, slope, inertia and towed vehicles, respectively.

A detail of oscillatory sub-model of the seat mounted at a distance l_s from the center of mass, T, with marked parameters: m_s - the total mass of the seat and the driver, c_s, k_s - characteristics of stiffness and damping seat, respectively, S - binding site of the seat on the vehicle's chassis and the appearance of vertical oscillations is also shown in Fig. 1.

Levels of oscillations in some points of the vehicle's chassis depend on their position in relation to the reference points or axes, for example the center of mass, the axis of front, i.e., rear wheels, center of oscillation. Therefore, when the location of the seats, l_s, is being choosing, these levels should be taken into account, but the impact will also express also some other factors: structural constraints, the position of the controls, the requirements of specific working and transport operations. The adopted vehicle model structure with the sub-model of seat suspension passive system (Fig. 1) provides further simplification of simulation models and reduces the time needed. In fact, considering that the mass of the seat with the driver is significantly less than the total mass of the vehicle, then this mass exerts a negligible effect on overall levels of oscillation of systems. This allows the system shown in Fig, 1, to spread into two oscillatory models; the first is two-dimensional model of the vehicle with mass, m, and one sub-model of the seats with concentrated mass, m_s, on a mobile platform. In this way, the vehicle model can be analysed separately from the model of the seats, and thus it is possible to describe its relevant oscillatory processes depending on influential parameters, then the same could be imported into a database and used in all cases of seat's selection or replacement. On the other hand, the partial sub-model shown in Fig. 1, excited by oscillations of the moving platform, i.e., vehicle's chassis, which can be quickly generated or used from database, is used for the analysis of oscillatory processes of the seats, in terms of its filtering properties.

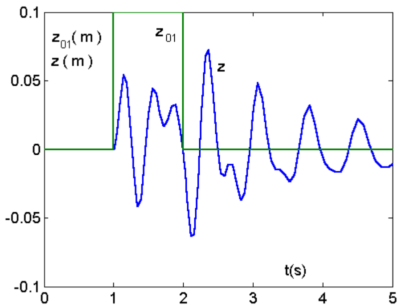
Fig. 2 *Self-driven measuring platform with equipment.*

The platform of a truck with its own drive and measuring equipment, shown in Fig. 2, was used in simulation studies. This is universal purpose platform, for stationary, quasi-stationary and dynamic testing of road, off-road, commercial vehicles, their aggregates, sub-assemblies and components. For the purposes of this study, i.e., for the proposed methodology, this platform can be used for testing and identifying of sub-models of elements and components of the elastic suspension of the road vehicles and other working machines. With regard to the concept of open supporting structure and the lateral position of the wheels, the wheels with tires of different types and sizes can be set up and examined on the platform. The part of measuring equipment is shown in Fig. 2, including measuring dynamometer of traction force and speed sensor Leitz-Correvit LG 2, and three-axial acceleration sensor, drive encoder HBM, eight channel measuring system and data measuring acquisition system, HBM Spider 8 and notebook.

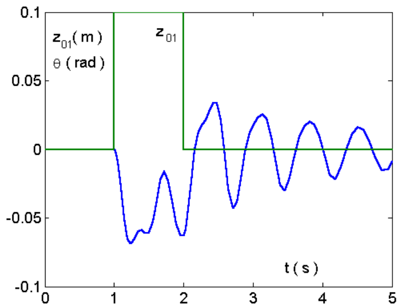
3. RESEARCHING RESULTS

Simulation studies shown in this paper were carried out on the basis of vehicle oscillatory model, shown in Fig.1, which is reduced from spatial to planar model assuming the longitudinal symmetry and the corresponding mathematical model described by differential equations from (1) to (6). Thus, two weight conditions of the vehicle and four characteristic impacts from uneven road were observed. The required input data for simulation studies, elastic-damping characteristics of the tires and the primary elastic suspension of the vehicle were obtained using the measuring platform presented in Fig. 2. Some typical results for the above-mentioned cases are shown in Fig. 3 to Fig. 10.

Fig. 3 shows the time history of the vertical displacement of center of gravity, z and angular swing of sprung mass of the vehicle around the transverse axis, θ at impulse excitation of the front axle by individual impulse of the road, with the height of 0.1m and duration 1 second.

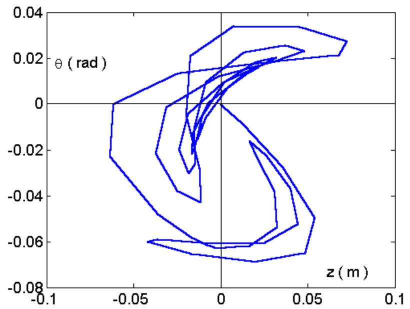


a)



b)

Fig. 3 *The change of vehicle sprung mass coordinates, a) vertical displacement, z (m), b) angular swing θ (rad), during impulse excitation of front wheels by road at the given parameters, z_{01} / t=0.1m/1s.*



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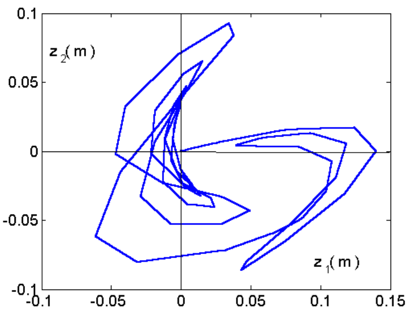
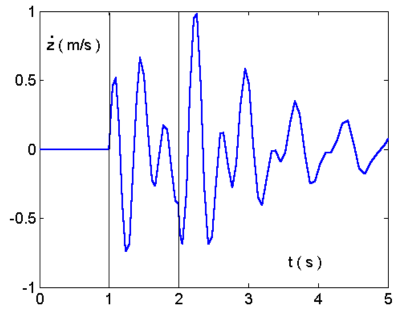


b)

Fig. 4 *Lissajous plot of coordinates, a) $\theta \rightarrow z$, b) $z_2 \rightarrow z_1$ Impulsive excitation through the front wheels z_{01}/t=0.1m/1s*



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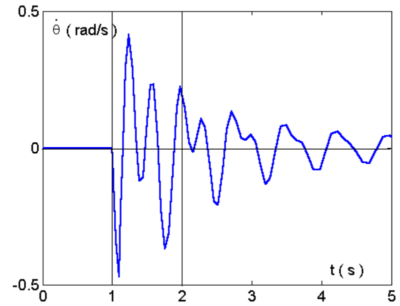

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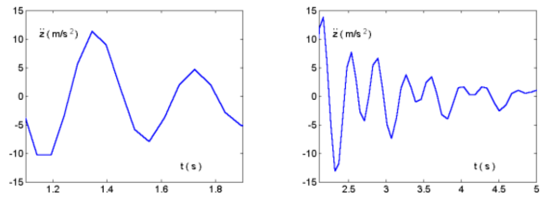
Fig.5 *a) Vertical, B) Angular speed of vehicle sprung mass. Impulsive excitation same as in Fig. 4.*



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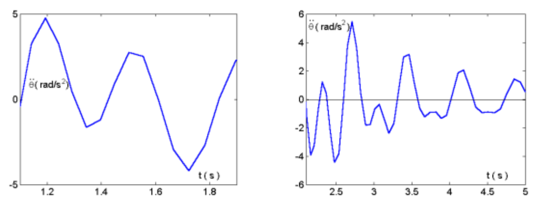

b)

Fig. 6 *a) Vertical, b) Angular acceleration of vehicle sprung mass. Impulsive excitation same as in Fig. 4.*

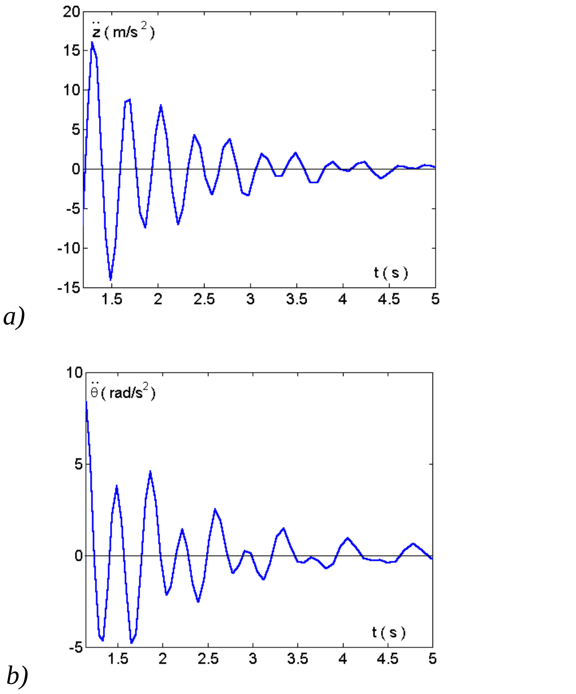

Fig. 7 *a) Vertical, b) Angular acceleration of vehicle sprung mass. Impulsive excitation of the front wheels from uneven road at the given the parameters, z_{01} / t =0.1m/0.1s*

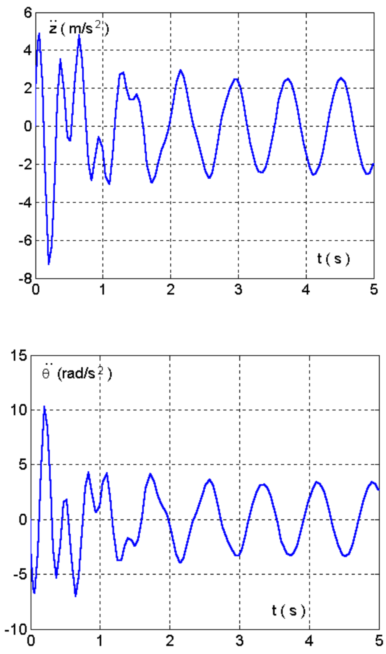

Fig. 8 *a) Vertical, b) Angular acceleration of vehicle sprung mass at the sinuous excitation of the front wheels from uneven road at the given the parameters, $z_{0\,1}$/ λ =0.1m / 0.5m*

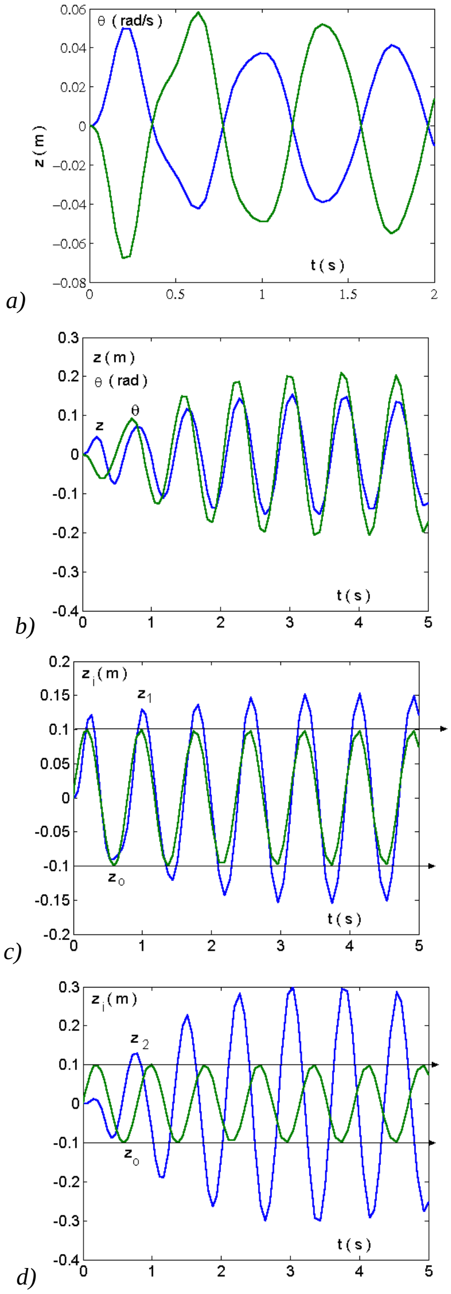

Fig. 9 *Oscillation of sprung mass of the vehicle at the sinuous excitation front wheels from uneven road at the given the parameters, z_{01} / λ / v = 0.1m/0.5m. a) unladen vehicle, coordinate $\theta \rightarrow$ z,;b) laden vehicle, coordinate $\theta \rightarrow$ z; c) and d) laden vehicle coordinates, z_1 ,z_2 in relation to the excitation of the front wheels, z_{01}, respectively.*

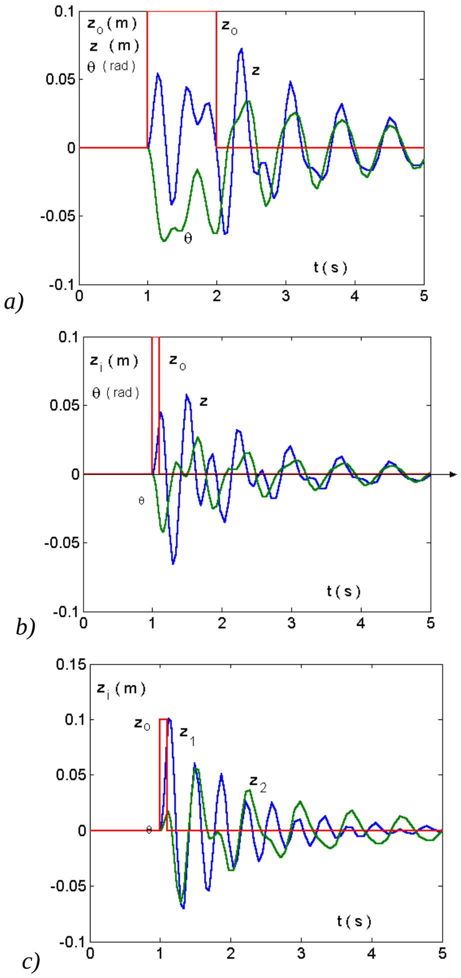

Fig.10. *Transient processes of vehicle sprung mass movement, a) at impulse excitation type 1, time histories of the coordinates z, θ, and excitation, z_0, b) at impulse excitation type 2, time histories of the coordinates, z, θ and excitation z_0, c) at impulse excitation type 2, time histories of the coordinates, z_1, z_2 and excitation z_0.*

Fig. 10a provides a comparative overview of time histories from which it is obvious that the observed transient processes are phase-asynchronized during the interval of impulsive excitation of the front wheels. Thereby, the vertical displacement of center of gravity, *z*, is changing sign, while the swing angle *θ* retains a negative sign for the same time. After the excitation is off, the processes are oscillatory damped and phase-synchronized.

Lissajous loops (Fig. 4) show the mutual dependence of the observed coordinates of the vehicle sprung mass, $\theta \rightarrow z$, $z_2 \rightarrow z_1$, respectively, as an alternative position coordinates of sprung mass. The shape and position of these diagrams indicate two characteristic phases of the observed processes that are presented also in Fig. 10a.

The time histories of the translational and angular velocity during these transitional processes are shown in Fig. 5 a, b, respectively, and the corresponding accelerations in Fig. 6 a, b, respectively. Thus, acceleration time history shown in Fig. 6 a, b, have breakpoints in the time-points of 1 second and 2 seconds, respectively, i.e., at the occurrence of pulses and the disappearance of pulses, respectively, with extremely high values of the shock acceleration. In addition; the high values of vibratory accelerations are marked during the impulse excitation interval and at the beginning of an damping process interval (for the considered case shown in Fig. 6 - vertical acceleration range from -10m/s^2 to 10m/s^2 and -14m/s^2 do 14m/s^2, respectively and angular acceleration range from 4.8 rad/s^2 to -4.2 rad/s^2 and from -4.2 rad/s^2 to 4.8 rad/s^2, respectively).

Higher levels of the vertical and angular accelerations are achieved in the case of impulse excitation type 2, i.e., the same height, the same impulse intensity, but the shorter duration, shown by a transition process in Fig. 10 b, c, and levels of the acceleration in Fig. 7 a, b. It can be observed from Fig. 6 and 7 that the angular acceleration is in a greater degree asymmetric in relation to horizontal axis than vertical acceleration.

A simulation of the process for the case of sinusoidal excitation was carried for the purpose of comparative analysis of road excitation impact on vehicle oscillatory processes and determination of impact on the levels of vertical and angular accelerations relevant for the evaluation of ride comfort of the vehicle [10,11,12]. Thereby, the model of front wheels excitation was used, described in the spatial domain by expression, $z(x) = Asin\Omega x = Asin(2\pi/\lambda)x$, and in the time domain by expression, $z(t) = Asin\omega t = Asin(2\pi/T)t$. Using dependence of spatial and time coordinates, the mathematical model of periodic excitation of front wheels from the road is obtained in the form: $z(t) = Asin(2\pi/\lambda)vt$, which establishes a relationship between the height of road roughness *A*, their wavelength, λ and speed *v*. Simulation results presented in Fig. 8 and 9 were obtained for the following values: *A*=0.1m, λ=0.5m, which approximates the roughness parameters of lower quality road.

The researching results presented in Fig. 8 a, b, show the vertical acceleration time history of vehicle sprung mass and its angular acceleration during rotation over the transverse axis, respectively. The transitional process of the acceleration takes about 2 seconds, and then it is establish a process of forced oscillation with frequency that coincides with frequency of sinusoidal excitation. Maximum accelerations are achieved at the beginning of the transitional period: maximum vertical acceleration of - 7.3m/s^2 and the maximum angular acceleration of - 10.2 rad/s^2. It is obvious from above presented pictures that vibratory acceleration levels at steady, forced oscillation process are significantly lower compared to the above extreme values. In addition, this type of excitation does not generate shock acceleration, which occurs in previously presented impulsive excitation.

The time history of coordinates, $z \rightarrow \theta$ of sprung mass of the unladed vehicle with the basic parameters of the corresponding variant of vehicle with decoupled oscillation of the front and rear axles are given in Fig. 9a. The results presented in Fig. 9 b, c, d show time history of position coordinates, $z \rightarrow \theta$, the coordinates of the front part of the sprung mass and excitation $z_1 \rightarrow z_0$, the coordinates of the rear part of the sprung mass and excitation $z_2 \rightarrow z_0$, for loaded vehicle. Whereby, load growth has led to the coupling oscillation of the front and rear axles, i.e., the oscillation of the rear side of the vehicle at excitation of the front wheels only. That was not the case in the previous example of unloaded vehicle.

From point of view of the vehicle's ride comfort, all three variables of state of oscillatory process, above presented, are important: displacement, velocity and acceleration observed in the time and frequency domain [5,10]. However, the previous works in the domain of vehicle oscillations and development of criteria for their evaluation in terms of comfort, usually were based on correlation between acceleration levels and frequency, subjective assessments, identification of vibration exposure criteria curves (with equal observations of human factor) [1,2,3,4]. An example of these curves proposed in the ISO for the vibration frequency range 0.63 to 100 Hz is presented in Fig. 11 [13,14].

The vibration levels of the curves shown in Fig. 11 are given as RMS acceleration levels that produce equal fatigue – decreased job proficiency.

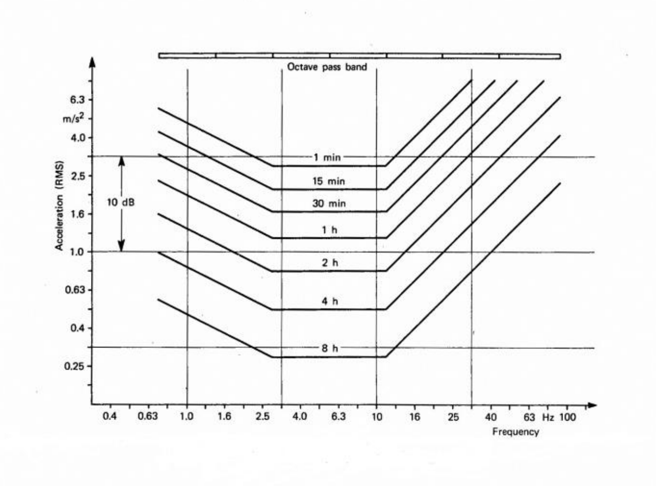

Fig. 11 *Vibration exposure criteria curves.*

The curve shapes in Fig. 11. indicate human sensitivity to vibration magnitude and frequency. The range of the RMS levels is 20 dB, in the middle frequency domain for exposure time from 1 min to 8 h. The RMS acceleration levels, obtained in this paper, for observed road disturbance, crossover the levels exposure curves by corresponding frequency. The vehicle seat must reduce acceleration overshoot. An approach to optimal design of the vehicle seat request the inverted exposure curves in Fig. 11, then identified oscillatory processes of sprung mass by means above presented method and with these data can estimation of the transfer function of seat with respect of vehicle ride comfort.

4. CONCLUSIONS

Oscillation processes generated by vehicle movements and unevenness of road produce dynamic forces that impact to the driver and passengers, which is reflected in their ability to work and health in general. The base features of the uneven road, movement state, the characteristics of the vehicle - especially a the weight condition, then primarily elastic support of the vehicle, seats, cab affect on the intensity of these processes. Change of the weight conditions of the vehicle, i.e. its useful payload, reflects to increased total weight, weight distribution, the position of the center of mass, the oscillation coupling degree of the front and rear axles. The baseline characteristics exert significant influence, i.e. potential excitation properties of road as a deterministic functions, impulse, step, ramp, sinuous as stochastic excitations of the real profile of the road. The results of this research have shown that negative effects especially manifest impulse excitation of individual obstacles of the road. They generate shock acceleration in characteristic transient spots of impulse generation. Generated acceleration depends on the level of the impulse, its duration, and the regime of movement and structural characteristics of the vehicle. The performance of the intensity of the impact processes can be obtained by the comparison with those processes obtained with sinusoidal excitations of the road, in terms of the extreme values of acceleration and transient, damped and forced oscillation phases. The obtained results are the basis for qualification of influential system parameters on comfort during the ride for a specific vehicle and its current situation, and for the optimal choice of the seats for the platform on the bases of inverted iso-comfort curves and identified vibration processes of the vehicle chassis, as a platform on which the seat will be placed.

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