SHOCK-IMPULSE DIAGNOSIS OF RAILWAY

Janat Musayev[1], Yerbol Yelzhanov[2], Madina Aliyarova[3], Aliya Yelemanova[3], Algazy Zhauyt[3]
[1]Kazakh Academy of Transport and Communications named after M. Tynyshpayev, Kazakhstan;
[2]Kazakh Leading Academy of Architecture and Civil Engineering, Kazakhstan;
[3]Almaty University of Power Engineering and Telecommunication, Kazakhstan
ali84jauit@mail.ru

Abstract. The paper suggests application of an experimental method for the assessment of the dynamic effect of a locomotive underframe on the railway track. The assessment is to be based on the spectral analysis of the response of structural elements of the railway track undergoing a shock pulse. Application of digital measuring systems for monitoring of motion of the train is also proposed. Shock absorbers are intended for developing forces providing the elimination or reduction of the amplitude of cars or their parts fluctuations. On the CIS railroads, the wedge-type shock absorber became the most wide-spread in trucks of freight cars. In friction shock absorbers, the friction force arises in vertical and horizontal movements of the absorber wedges rubbing on friction plates fixed on the columns of the truck sidewalls. Given the above, to increase the level of safety of railway operation, it is necessary to apply progressive methods of diagnostics and monitoring aimed at the assessment of the vibration level occurring during operation and of the current status of all elements of the railway track. The need and urgency of implementing innovative systems and means of diagnostics and monitoring of the railway facility are reflected in programs for the development of railway system until 2020 in the Republic of Kazakhstan.

Key words: shock-impulse, determination, railway track, cross-sleepers, experimental.

Introduction

Since the path is a discrete system, resistance to its vibrational effects is manifested primarily through frictional forces and cohesion between its individual elements and particles (between rails and sleepers, sleepers and ballast, between gravel, particles and sand grains of the roadbed) [1]. The changing nature of oscillations passed from the point of excitation to a certain point of the medium is determined exclusively by the properties of the medium in the propagation path fluctuations [2]. It is obvious that the energy of mechanical vibrations in any environment is transmitted through the physical interaction of the structural particles constituting the medium [3]. The patterns of this interaction for bodies of a certain shape are ultimately determined by the properties of the material in the medium and do not depend on other factors that do not affect these properties [4]. If the rail support has a discrete structure of cross-sleepers, the elastic supports above the concrete basis (as opposed to the rails embedded in concrete) and the wheel while running on the rail “feel” the change in the support stiffness [5; 6]. Variable elastic forces create vibrations of the wheel and rail at a frequency that depends on the speed of the rolling stock and spatial discreteness of the support. Other discreteness (and excitation frequencies corresponding to it) is characterized by the distance between the wheelsets and bogies [7; 8]. If the excitation frequencies coincide with the natural frequencies of the railway track, the vibrations of the track and surrounding soil can be quite considerable [9; 10]. The frequency of the impact f_k (Hz), corresponding to the n-th characteristic distance l_v (m) is determined via the rolling stock speed v ($\mathrm{m \cdot s^{-1}}$) according to the formula.

Materials and Methods

Completed model and full-scale experimental studies of the flexural vibrations of multilayer elastic plates and oscillations of the upper structure of the path of various structures have made it possible to identify the main regularities of railway track oscillations and to develop a technique for shock-impulse diagnostics [1]. This method is based on the fundamental provisions of the theory of elasticity and the theory of mechanical oscillations and allows with sufficient accuracy for practice to evaluate the ability of railway track structures to suppress vibrations arising from the movement of rolling stock. In addition to the roughness on the surface of wheels and rails, there can be also observed rough defects occurring during the rail-track operation. The major defects are associated with the presence of wheel flats and corrugations of the rails [2; 3]. Moreover, the wheels are subjected to such defects as out-of-roundness, out-of-balance and eccentricity. In the course of time, the defects

DOI: 10.22616/ERDev2018.17.N551

accumulate, especially if the track is not provided with a timely and proper technical care and maintenance.

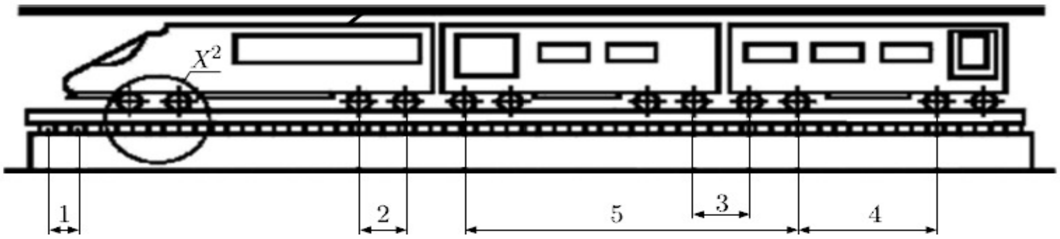

Fig. 1. **Characteristic parameters of vibration source:** 1 – distance between supporting elements of the track; 2 – distance between wheelsets of the bogie carriage; 3 – distance between the adjacent bogie carriages of the adjacent railcars; 4 – distance between bogie carriages of one railcar; 5 – distance between railcars

First, the impact on the design of the track should be characterized by the optimal value of the duration of the shock-impulse τ. Impulse duration τ, in the first approximation is determined on the basis of the theory of Hertz impact: There is a relationship for U_n depending on time t_n, $n = 0, \ldots, N - 1$, where N is the number of counts, $T = t_{N-1}$ is the time length of signal realization. The actual vibration speed is calculated per voltage values:

$$\upsilon_n = \frac{U_n}{K_n}, \tag{1}$$

where K_n – sensor transformation coefficient.

The vibration sensors are preliminarily calibrated on vibration table ESE 201 to determine the transformation coefficients [4]. The transition frequency domain is carried out using the direct Fourier transformation:

$$V_k = \sum_{K_n}^{N-1} \vartheta_n \exp\left(\frac{-2\pi i k_n}{N}\right).$$

In order to obtain the amplitude-time dependence of the vibrational displacement (oscillograms), it is necessary to integrate the function υ_n. For this, let us use the property of the direct Fourier transformation (DFT):

$$(\hat{F}') = 2\pi i f \hat{F} \tag{2}$$

where $\hat{F}$ – DFT of the function F with respect to V_k.

Then, by the inverse of Fourier transformation (IFT), we obtain:

$$S_k = \frac{V_k}{2\pi i f_k + \pi \ V_O \Delta f},$$

where $\Delta f = 1/T$ (signal sampling by frequency), $f_k = \Delta f k$.

Fig. 5 shows the oscillogram spectrum obtained on the rail base during passage of the locomotive VL80. Let us make IFT to return to the time domain:

$$S_n = \frac{1}{N} \sum_{k=0}^{N-1} S_k e^{\frac{2\pi i k_n}{N}}. \tag{3}$$

Hence, we obtain the amplitude-time dependence of the vibrational displacement on time. For obtaining of amplitude-time dependence of vibrational accelerations (accelerograms), it is necessary to differentiate υ_n and consider elimination of distortion of the sensor transformation coefficient in the frequency domain over 1000 Hz. For differentiation [5; 6], let us again make use of DFT property represented by expression Eq. (2), and use the Butterworth filter for the elimination of distortion in the low frequencies domain:

$$A_k = V_k H_k (2\pi i f_k), \tag{4}$$

where H_k – Butterworth filter:

$$H_k = \frac{1}{\sqrt{1+\left(\frac{f_k}{f_B}\right)^{2p}}} \cdot \frac{1}{\sqrt{1+\left(\frac{f_H}{f_k+1}\right)^{2p}}},$$

where f_H, f_B – range of frequencies passing through the filter;
p – filter order.

The Butterworth filter characteristics are given in Fig. 4. For the obtained amplitude A_k let us make IFT. As a result, Eq. (4), we obtain the amplitude-time dependence of vibrational accelerations filtered in the low frequencies domain in the following form:

$$a_n = \frac{1}{N}\sum_{k=0}^{N-1} A_k \exp\left(\frac{2\pi i k_n}{N}\right). \tag{5}$$

The modulus of spectrum density has the following form:

$$|S(f)| = \frac{2S_O\tau}{\pi}\frac{\cos(\pi f\tau)}{1-(2\pi f\tau)^2}, \tag{6}$$

where S_O – initial amplitude vibrations of the track superstructure under the impact force;
τ – impulse length collision of the impact system with the object, which in the first approximation is determined according to the Hertz impact theory, i.e.:

$$\tau = \frac{4.531}{\sqrt[5]{V_O}}\left(\frac{M\left(\frac{1-\mu_1^2}{E_1}+\frac{1-\mu_2^2}{E_2}\right)}{\pi\sqrt{R}}\right)^{\frac{2}{5}}, \tag{7}$$

where E_1, E_2 and μ_1, μ_2 – Young's moduli and Poisson's ratios, respectively, of the hammer and rail steel material;
R is the radius contact surface of the impact system;
V_O – linear velocity of the hammer;
M – reduced mass of the colliding bodies equal to

$$M = \frac{m_1 \cdot m_2}{m_1 + m_2}, \tag{8}$$

where m_1 and m_2 – masses of the hammer and railway track structure, respectively.

In addition to the optimal choice of the quantity τ, the energy of the external action on the control object must be sufficient to excite the path of the own bending vibrations in the fundamental mode in the structure of the upper structure [7]. And the initial amplitude of these oscillations S_O should be such that, given the sensitivity of the receiving vibration sensor ζ, it was possible to conduct a qualitative spectral analysis of the recorded responses to the impact ($S_O > 10\zeta$).

$$S_O = \frac{F}{2\pi m f_O},$$

where f_O – frequency of the first bending mode railway track (at the calculation it is assumed not less than 100 Hz);
$m = m_1 + m_2$.

Thus, for example, the Young's modulus of a shock system made of rubber material has a certain discrete value and is equal to $1.8\cdot10^6$ N·m^{-2}. The mass of the striker excitation system can vary from 0.2 to 6 kg. The same can be said about the radius of the drummer [8], which can vary in the range from 0.005 to 0.1 m. The most highly variable quantity is the collision velocity, although the actual variation range V_O is relatively small (from 1 to 10 m·s^{-1}). For the case shown in Fig. 5a, an analytic expression for the intersection curve of a three-dimensional surface and a plane $\tau = 0.001$ s has the form:

$$V(m_1) = \frac{2\times10^{17}\left(\dfrac{m_1\cdot m_2}{m_1+m_2}\right)^2\left(\dfrac{1-\mu_1^2}{E_1}+\dfrac{1-\mu_2^2}{E_2}\right)^2}{R}. \tag{9}$$

The curve corresponding to the expression Eq. (3) is shown in Fig. 5b. From this curve it follows that with the chosen parameters of the impulse system $R = 0.05$ m; $E_1 = 1.8\cdot10^6$ N·m^{-2}, which corresponds to the contact layer of rubber, with the mass impactor $m = 3$ kg, the required speed V_O should be 5 m·s^{-1}.

Results and discussion

In the paper, the topical issues of an experimental method assessment of dynamic impact of the locomotive body on the track are studied. The proposed method is based on spectral analysis of the railway track structural element responses to the impact impulse and monitoring the train motion with the use of digital measuring equipment (see Fig. 2-5).

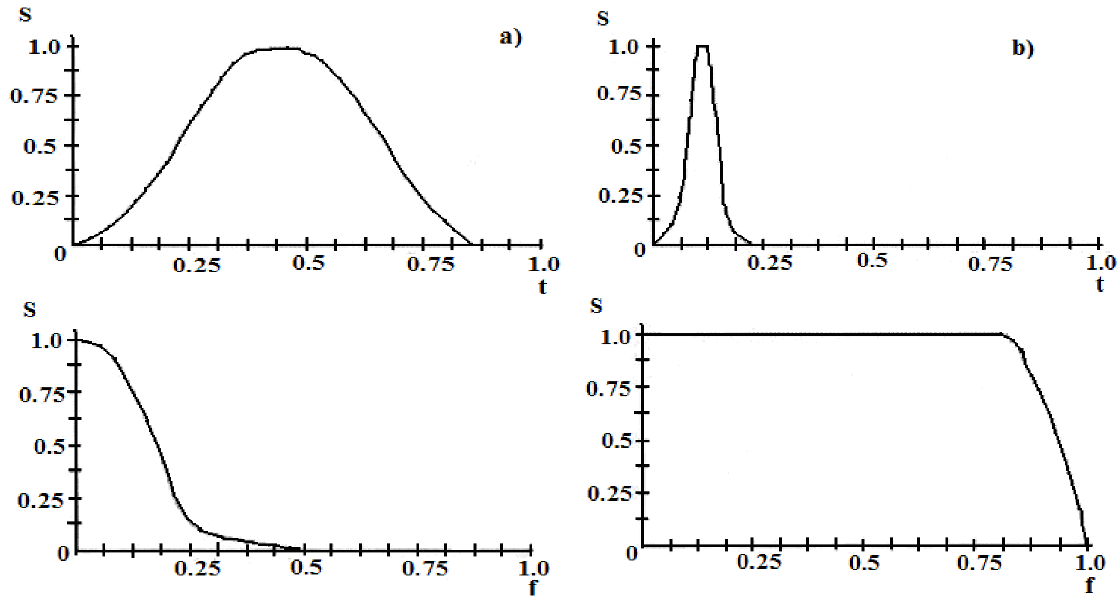

Fig. 2. **Dependence of frequency spectrum impact on duration of shock-impulse:** when the shock duration is longer than the optimal one (a), at the optimum duration of the shock-impulse (b)

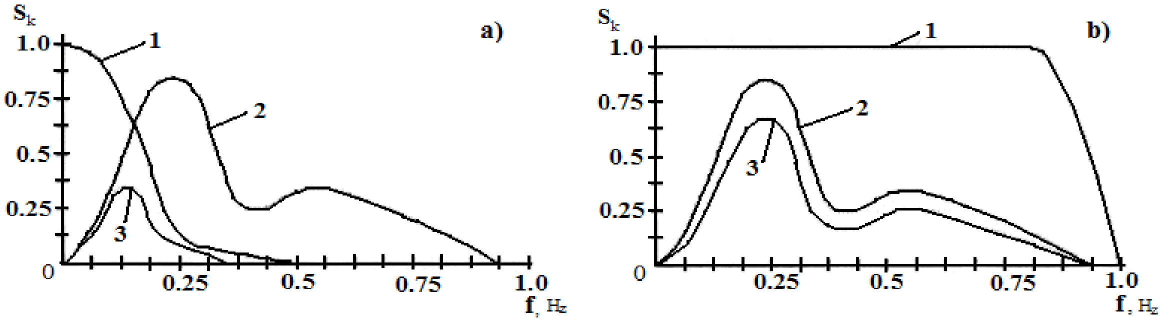

Fig. 3. **Dependence response spectrum of system on frequency response characteristics with a shock duration greater than the optimal (a), optimal duration of the shock-impulse (b):**
1 – spectrum of impact on the system; 2 – own spectrum of the system; 3 – spectrum of the response system to impact

Conducting a complex monitoring railway track allows one to carry out an objective assessment of the condition of track elements on the level of vibrations occurring during motion of the rolling stock [9].

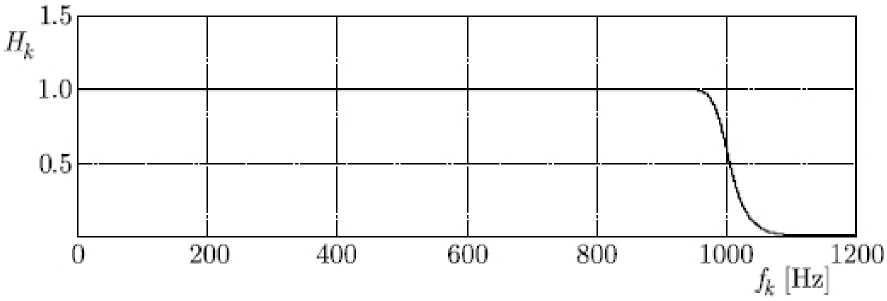

Fig. 4. **Butterworth filter characteristics in low frequencies domain at f_H = 0.1 Hz, f_B = 1000 Hz, p = 50**

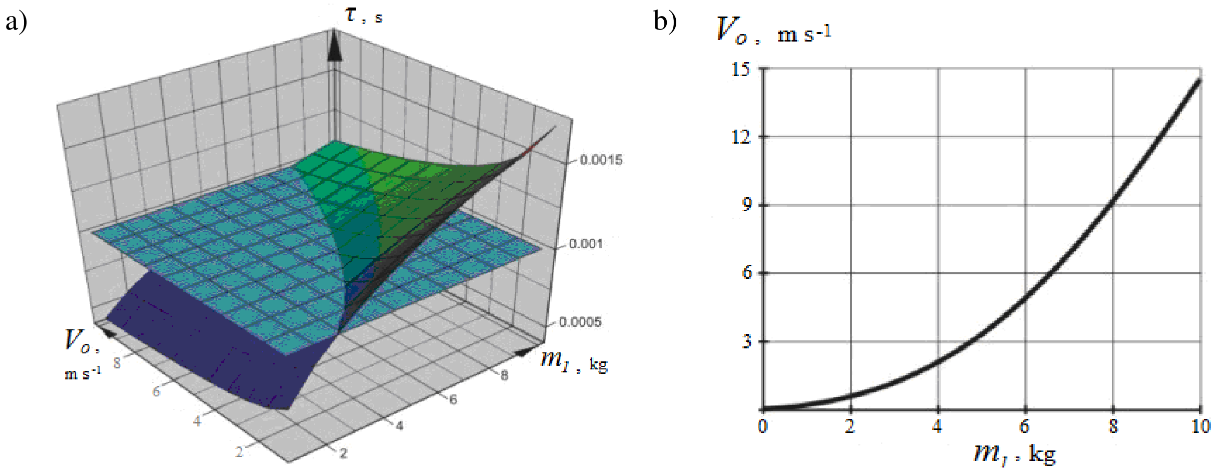

Fig. 5. **Result of construction of three-dimensional surface of impact function $\tau = F(m_1, \vartheta_0)$, at R = 0.05 m, $E_1 = 1.8 \cdot 10^6$ N·m^{-2}: three-dimensional function surface and plane τ = 0.001 s (a), intersection curve of three-dimensional surface and plane τ = 0.001 s (b)**

Thus, it is possible to select very simply the optimal ranges for changing the parameters of the impulse system, when designing it to diagnose a particular design of the upper structure of the path or for the possibility of adjusting these parameters directly at the object of investigation [10], for example, by changing the collision velocity V_O or the radius of the contact surface lining impact device.

Conclusions

1. The review of the works related to the improvement of the dynamic qualities of freight cars showed that over the last years a large number of models of bogies have been developed and among them the most promising is the model 18-9945, since the characteristic feature and advantage of this bogie is the use of a separate scheme for suppressing vertical and horizontal vibrations and the use of special devices (longitudinal leashes and diagonal ties). The carried out analysis of foreign experiments on the use of such devices in the construction of bogies also shows the prospects of this direction.
2. To investigate the possibility of applying a separate scheme for damping vertical and horizontal oscillations and using longitudinal leashes and diagonal links in spring suspension, the refined mathematical models of long-distance freight cars on test bogies (models 18-9945) were first developed describing the motion along real road irregularities.
3. Theoretical and experimental studies were carried out to determine the angular stiffness of the experimental railway track. Multivariate calculations have been carried out to identify the parameters of the model of the bogie and its structural connections. The results of computer

simulation are close to the experimental data, which show the adequacy of the selected parameters. Divergences do not exceed 3 %.

4. Comparison of the results of calculation and experiments on the stability factor, frame forces and the dynamic coefficient showed their satisfactory coincidence, which indicates the reliability of the developed mathematical models of cars. The discrepancy does not exceed 15 %.

References

[1] Bodare A. Evaluation of track stiffness with a vibrator for prediction of train-induced displacement on railway embankments. Soil Dynamics and Earthquake Engineering, Vol. 29, Issue 8, 2009, pp. 1187-1197.

[2] Choi J.Y. Influence of track support stiffness of ballasted track on dynamic wheel-rail forces. Journal of Transportation Engineering, Vol. 139, 2013, pp. 709-718.

[3] Berggren E.G., Kaynia A.M., Dehlbom B. Identification of substructure properties of railway tracks by dynamic stiffness measurements and simulations. Journal of Sound and Vibration, Vol. 329, 2010, pp. 3999-4016.

[4] Alves Costa P., Calc-ada R., Cardoso A. S., Bodare A. Influence of soil non-linearity on the dynamic response of high-speed railway track. Soil Dynamics and Earthquake Engineering, Vol. 30, Issue 4, 2010, pp. 221-235.

[5] Abdelkrim M., Bonnet G., Buhan P. A computational procedure for predicting the long term residual settlement of a platform induced by repeated traffic loading. Computers and Geotechnics, Vol. 30, 2003, pp. 463-476.

[6] Solonenko V., Mahmetova N., Musayev J., Kvashnin M., Zhauyt A., Buzauova T. Modeling of dynamic characteristics of freight car with optimized parameters of wedge-type shock absorber. Journal of Vibroengineering, Vol. 19, Issue 2, 2017, pp. 1197-1213.

[7] Abilkaiyr Z., Musayev J., Kaiym T., Alpeisov A., Alimbetov A., Zhauyt A. The interaction of the freight car and way taking into account deformation of assembled rails and sleepers. Vibroengineering Procedia, Vol. 8, 2016, pp. 269-274.

[8] Yefimov V. P., Pranov A. A. Development and carrying out of complex testing of truck of model 18-578. Problems of Mechanics of Railway Transport, Dnepropetrovsk, 2004, pp. 123-135.

[9] Pranov A. A., Yefimov V. P. Modernization of truck of model 18-100 as effective way of increasing traffic safety of trains. Heavy Mechanical Engineering, Vol. 12, 2003, pp. 75-85.

[10] Houthis V. D., Anisimov P. S. Power characteristics of frictional wedge-type shock absorbers in mathematical models of studying freight cars. VNIIZhT Bulletin, Vol. 4, 2005, pp. 127-133.