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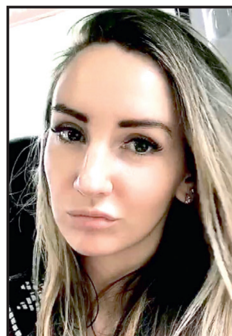
The Influence of the Stiffness of the Elements of Rail Damping on the Parameters of the Multi-Mass Wagon–Track Vibratory System



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ABSTRACT

The task of achieving the increase in the weight and velocity of trains being core objective for most railways is inherently associated with solving problems of improving the dynamic qualities of wagons and locomotives during their interaction with the upper structure of the railway track. The strength and stability of rolling stock against derailment in various climatic zones should be ensured together with minimizing operating costs. Analysis of the reliability and performance of the wagon–track system can be conducted based on multivariate dynamic calculations of mathematical models and through experimental studies of dynamic vibrations.

Currently, the issues of the influence of changes in the elastic-hysteresis properties of various impact absorber materials under the action of temperature factors on the elements of a multi-mass vibratory system remain insufficiently studied.

The purpose of the research described in the article was to analyze the dynamic processes taking place in a multi-mass wagon–track vibratory system under the influence of changes in stiffness and internal friction coefficients of a damping rail component depending on a type of structural material and ambient temperature.

The research resulted in elaborating model of multi-mass vibratory wagon–track system. Multivariate model calculations were carried out regarding various stiffness and internal friction parameters of damping elements situated under the rail; the parameters being selected from experimentally constructed dynamic hysteresis. The work presents results of calculations of the reaction forces and deviations in the elements of the vibratory system depending on temperature, type of material, thickness and design of damping elements.

Keywords: railway, railway track, wagon, multi-mass vibratory system, generalized wagon–track model, damping material, sandwich plates, pads, elastic-hysteresis properties, dynamic secant stiffness, dynamic hysteresis, thermal action, multivariate calculations, reaction force, deviation.

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Introduction

The increase in train weight and velocity is inseparably associated with solving the problems of improving the dynamic qualities of wagons and locomotives during their interaction with the upper structure of the railway track. However, it is important to ensure the strength and protection of the rolling stock against derailment in various climatic zones at minimum operating costs. It is possible to conduct an analysis of the reliability and performance of the general system based on multivariate dynamic calculations of mathematical models, built using experimental data. The results of the calculations determine optimal values of the parameters of the elements of this general system.

Many Russian scientists were engaged in modelling the interaction forces arising between vehicle and track followed by derivation and analysis of the basic differential equations of motion, based on the general variational principles of analytical mechanics for specific practical problems [1–4].

Several international works were also devoted to theoretical modelling and experimental studies of dynamic vibrations, for example, to the effects of frequency, stiffness, damping factors, velocity of moving harmonic load on the intensity of emergence of corrugated rail and wheel wear [5–9].

However, the issues dedicated to the influence of changes in the elastic-hysteresis properties of dampers made of different materials under the action of temperature factors on the elements of a multi-mass vibratory system are still insufficiently studied.

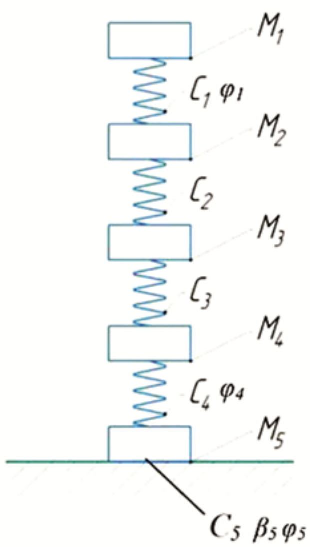
In view of the above mentioned, the *objective* of the research was to analyze dynamic processes occurring in a multi-mass wagon–track vibratory system under the influence of changes in stiffness and internal friction factors of a the elements of rail damping depending on the type of structural material and thermal effect.

Method of the research

Railway track and rolling stock compose a complex thermodynamic system, where they get into interaction with each other and with the environment. The movement of a wagon along the railway track can be represented as the vibration of the multi-mass system shown in Pic. 1.

Usually the d'Alembert's principle is used to describe the spatial motion of a body. According to that principle the constraints between bodies are disregarded and their action is replaced by reactions, inertia forces and moments of inertia forces' moments are applied [10; 11]. The calculation model of interaction of moving parts of multi-mass vibratory system is shown in Pic. 2.

At each moment of mechanical system motion, the work of active forces, of constraint reaction forces and of inertia forces for any



M_1 – wagon sprung mass per 1 wheel.

C_1 – stiffness of spring suspension.

ϕ_1 – friction factor of friction shock absorber (damper).

M_2 – mass of unsprung part of the wagon (1/2 of the lateral frame journal-box, ½ of the wheelset axis).

C_2 – wheel rim stiffness.

M_3 – wheel rim mass.

C_3 – rail rim mass.

M_4 – reduced mass of the rail.

C_4 – rail pad stiffness.

ϕ_4 – internal friction factor of the rail pad.

M_5 – reduced mass of the undertrack foundation (1/2 of track sleeper).

C_5 – track stiffness.

β_5 – ballast dry friction factor.

ϕ_5 – ballast viscosity factor.

Pic. 1. Generalized model of interaction between the running parts and the superstructure of the railway track.

possible movement from the occupied position is equal to zero. In this regard, the system of differential equations describing such system can be represented as shown below:

$$\begin{cases} M_1 \ddot{z}_1 + R_1 = 0 \\ M_2 \ddot{z}_2 + R_2 - R_1 = 0 \\ M_3 \ddot{z}_3 + R_3 - R_2 = 0 \\ M_4 \ddot{z}_4 + R_4 - R_3 = 0 \\ M_5 \ddot{z}_5 + R_5 - R_4 = 0. \end{cases} \qquad (1)$$

Constraint reactions in calculation model are directed in such a way as if they acted under positive deformation of elastic and viscoelastic elements. Compression deformation is assumed as positive deformation.

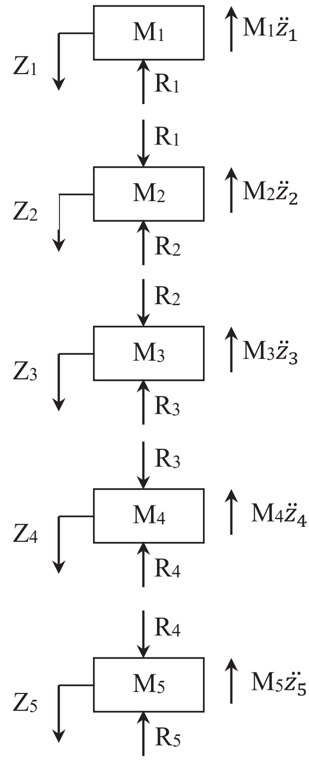
Equations of constraints reactions forces are the functions of stiffness, coordinates, and their derivatives:

$$\begin{cases} R_1 = C_1[1 + \phi_1 \mathrm{sign}(\dot{z}_1 - \dot{z}_2)](z_1 - z_2) \\ R_2 = C_2(z_2 - z_3) \\ R_3 = C_3(z_3 - z_4) \\ R_4 = C_4(z_4 - z_5) + \beta_4(\dot{z}_4 - \dot{z}_5) \\ R_5 = C_5[1 + \phi_5 \mathrm{sign}(\dot{z}_5)]z_5. \end{cases} \qquad (2)$$

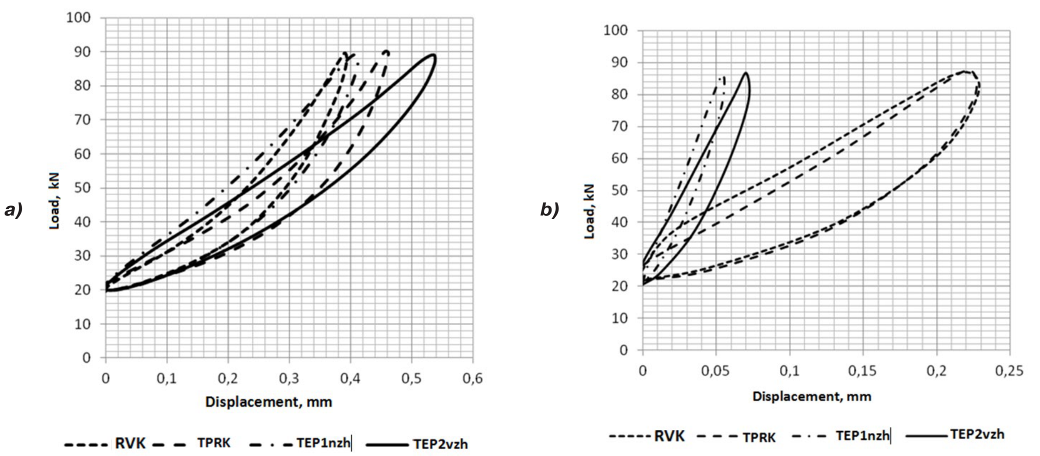
Masses (M_i) and stiffness values (**C_i**) of the system elements are determined from the reference data for the freight semi-wagon with axle loading 25 tf, R65 rail, steel reinforced concrete sleeper and broken-stone ballast. Each calculation variant operated different stiffness parameters and internal friction values of rail pad, which were selected among experimentally determined dynamic hystereses at temperatures +23°C and -40°C (Pics. 3, 4) [12] for the two types of compound material and sandwich plates:

• Rubber compounds: non-reinforced (TPRK) compound of 10 mm and 14 mm thickness, and aggregated PA-cord reinforced compound (RVK) of 14 mm thickness.

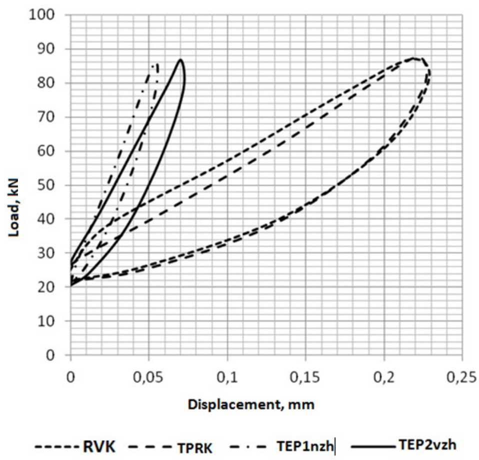
• Thermoplastic elastomers (mixture of polyurethane and non-rigid PVC), which differ in parameters of static stiffness – TEP1nzh type was of 14 mm thickness and TEP2vzh type was of 14 mm thickness.



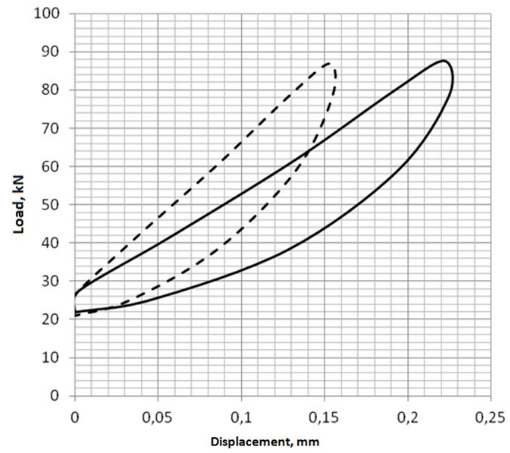
Pic. 2. The calculation model of interaction between the running parts with the superstructure of the railway track.



Pic. 3. The dependence of the compressive load on displacement under dynamic loading with 10 Hz frequency of TPRK, RVK, TEP1nzh, TEP2vzh samples at temperature +23°C (a) and temperature -40°C (b).



a)



b)

Pic. 4. The dependence of the compressive load on displacement under dynamic loading with 10 Hz frequency of TPRK samples of 10 mm thickness (dotted line) and 14 mm thickness (solid line) at temperature +23°C (a) and temperature -40°C (b).

Table 1

Values of dynamic secant stiffness of the samples depending on material and temperature

Name and specification of sample	Dynamic secant stiffness, kN/mm, at temperature	
	+23°C	-40°C
Rubber compound TPRK, 10 mm thick	204,4	426,4
Rubber compound TPRK, 14 mm thick	150,1	296,1
Rubber-fiber compound RVK, 14 mm thick	179,2	297,9
Thermoplastic elastomer TEP1nzh, 14 mm thick	126,5	903,6
Thermoplastic elastomer TEP2vzh, 14 mm thick	169,0	1186,9
Sandwich plates R9-P5 composed of RVK material, 9 mm thick and TEP1nzh, 5 mm thick	170,6	389,2

• Sandwich plate of R9-P5 type, composed using calculation method of RVK material of 9 mm thickness and of TEP1nzh type elastomer of 5 mm thickness.

Values of dynamic secant stiffness of the samples depending on material, thickness and temperature are shown in the Table 1 below.

Upon the wheel impact on the rail the initial conditions will be as follow:

$$t = 0: z_1 = z_2 = z_3 = z_4 = z_5 = 0;$$
$$\dot{z}_1 = \dot{z}_2 = \dot{z}_3 = V_0\, \dot{z}_4 = \dot{z}_5 = 0. \qquad (3)$$

The second-order differential equations were integrated by numerical methods by means of specially developed software.

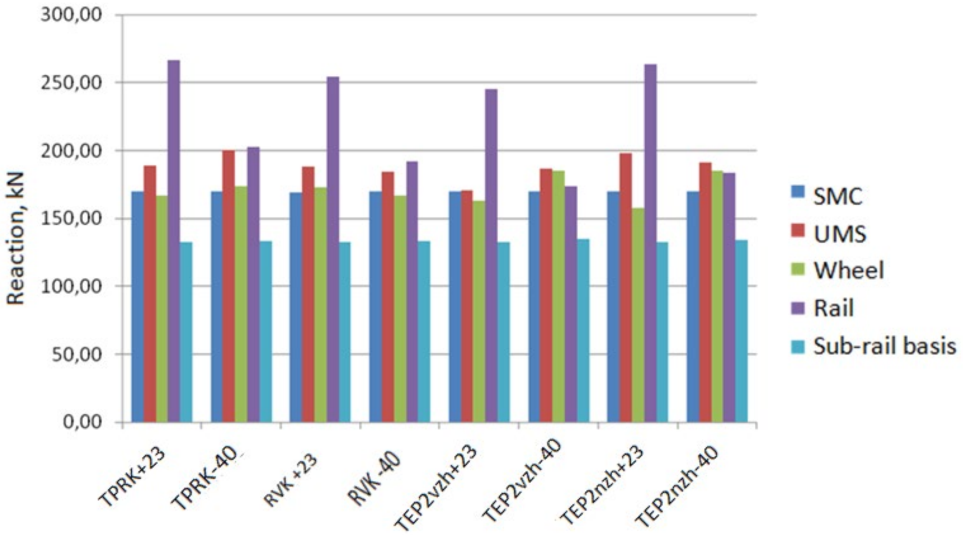
Multivariate calculations of the developed mathematical model were conducted determining the reaction forces (constraint reactions) and deviations following the changes in stiffness and internal friction parameters of one of the system elements located between the rail and the rail supporting elements.

Results and discussion

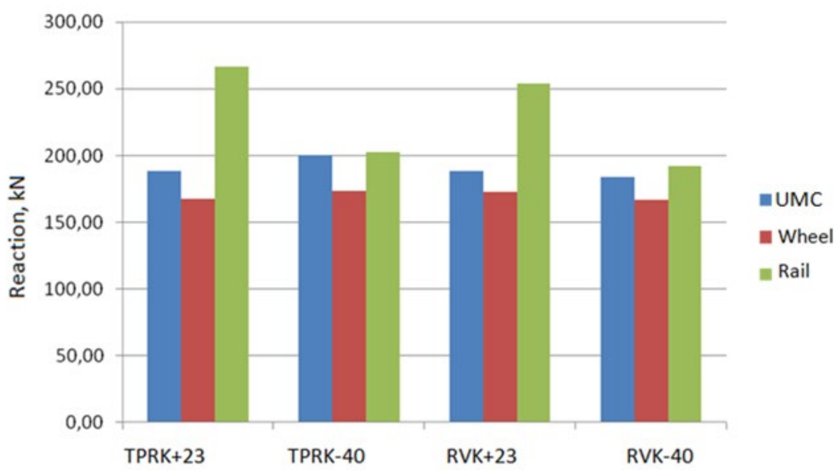
The analysis of the distribution of constraint reactions, equivalent by the module to the reaction forces, of a multi-mass wagon–track system during wheel rolling by a rail revealed that a change in the elastic-hysteresis parameters of a rail damping element depending on the type of material used has a slight effect on the sprung part of the wagon and the rail supporting basis (Pic. 5).

Further, the following abbreviations are assumed for Pics. 5–13: SMC – sprung mass of the railcar (railway wagon), UMC – unsprung mass of the wagon, «+23» – temperature +23°C; «-40» – temperature -40°C, sub-rail basis – rail supporting system.

The increase of dynamic stiffness of rail damping element of rubber compounds at -40°C (Pic. 6) compared with the stiffness at +23°C temperature results in decrease of reaction force of the rail by 24 % due to its lower



Pic. 5. Distribution of constraint reaction forces within multi-mass system elements.



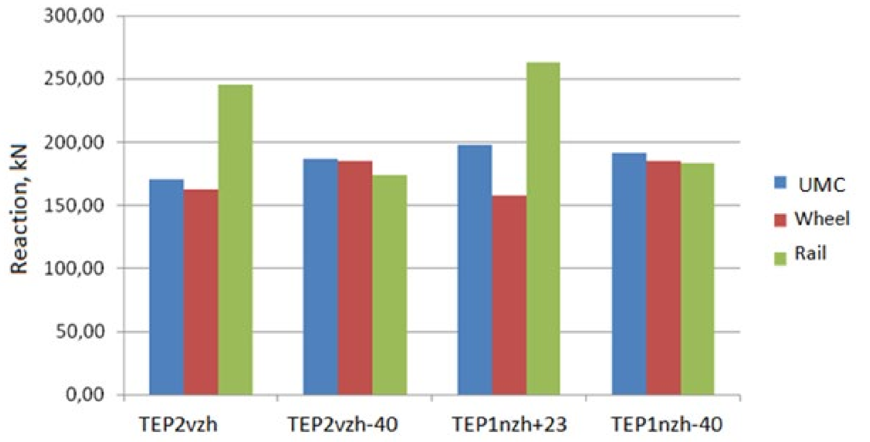
Pic. 6. Distribution of constraint reaction forces in case the rubber compounds are installed as rail damping elements.

deflection. At the same time, the reaction forces in the rest of the system elements (wheel, unsprung mass of the wagon, foundation supporting the rail) vary inconsiderably. Application of rubber-fiber compound at sub-zero temperatures in comparison with the rubber one is most expedient, since it reduces the load on the unsprung mass of the wagon by 8 %, and on the foundation supporting the rail by 1,37 times.

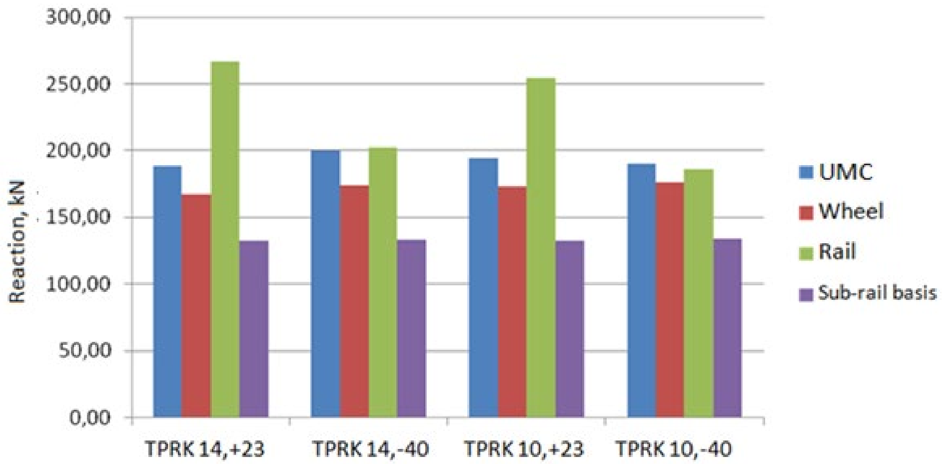
In the case of using thermoplastic elastomers under study as rail damping elements at -40°C temperature compared with normal temperatures (Pic. 7), the reaction force on the rail decreases by 30 %; and that on the wheel increases by 14–17 %. Therefore, when operating under temperature range of -40°C and lower, the wheel load will increase by 26 kN, which will exceed by 8 % the maximum permissible axial load of 25 tf per wheel.

In case the rubber compounds 14 mm and 10 mm thick are applied, the increase of stiffness of rail damping elements for stabilization of rail gauge parameters due to reducing of their thickness results in insignificant changes (± 5 %) of the reaction forces in all system elements (Pic. 8).

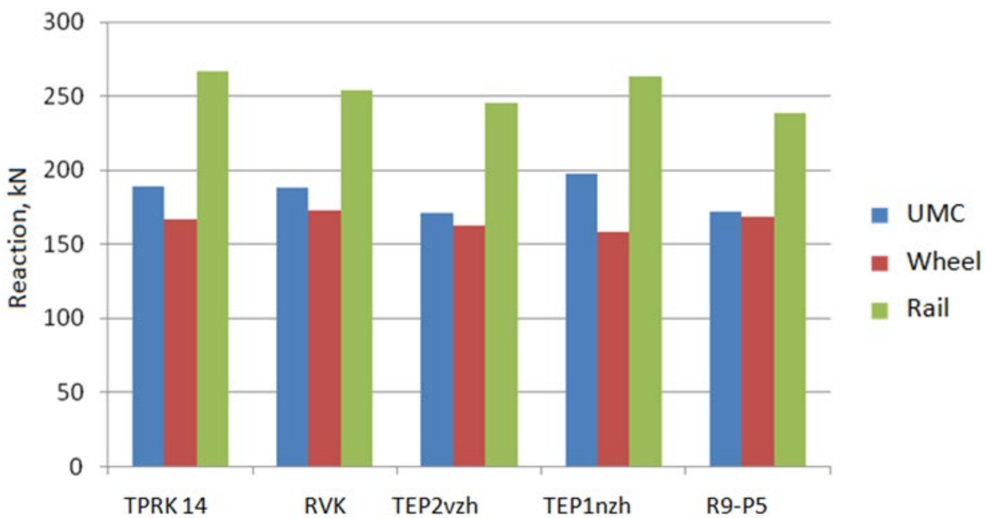
Analysis of the reaction forces' distribution when using 14 mm thick rail damping elements (TPRK, RVK, TEP2vzh, TEP1nzh and R9-P5 sandwich plates, Pic. 9) demonstrated that in R9–P5 sandwich plates the forces were



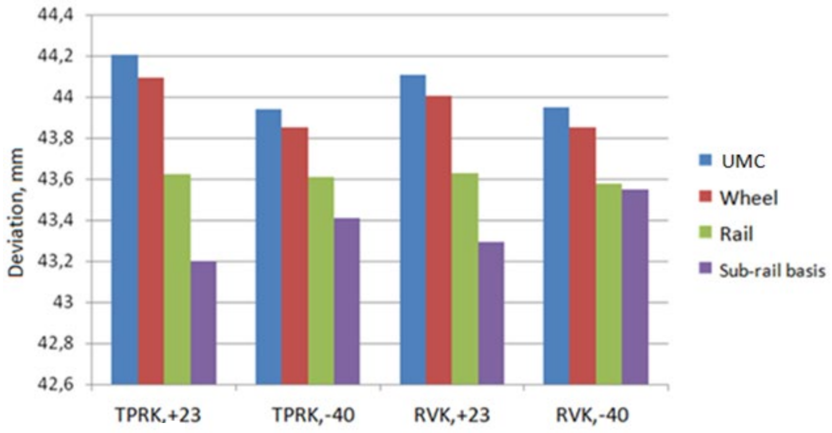
Pic. 7. Distribution of constraint reaction forces in case the thermoplastic elastomers are installed as rail damping elements.



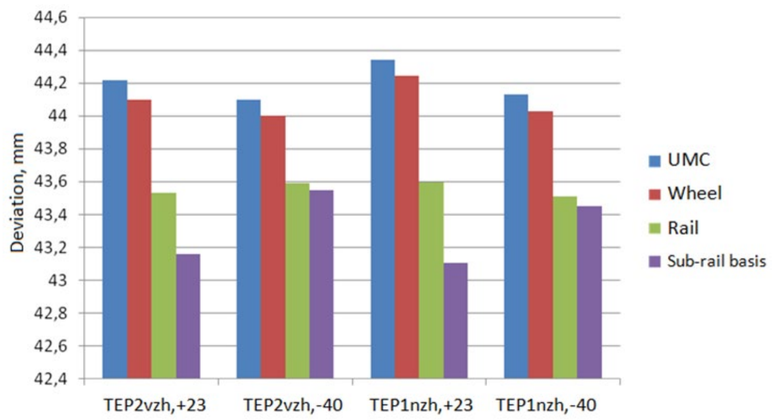
Pic. 8. Distribution of constraint reaction forces in case if rail damping elements of various thicknesses are applied.



Pic. 9. Distribution of constraint reaction forces in case if single-type rail damping elements and 14 mm thick sandwich plates are used.



Pic. 10. Distribution of deviations in case if rubber compounds are used as rail damping elements.



Pic. 11. Distribution of deviations in case if thermoplastic elastomers are used as rail damping elements.

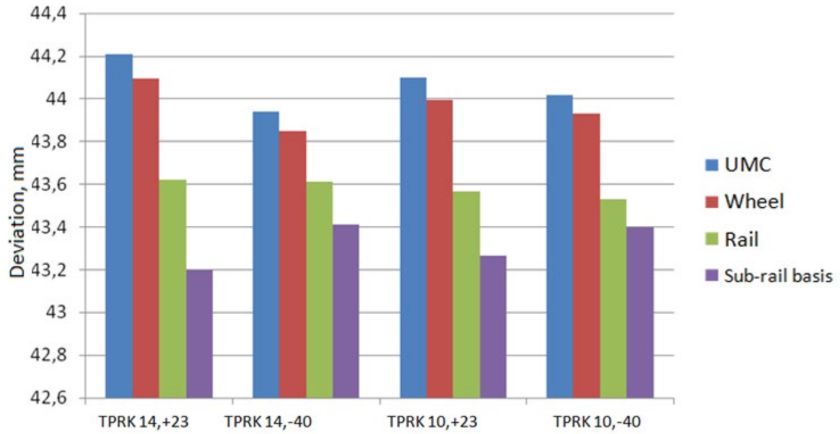
distributed in the most optimal way. While the reaction forces of the wheel and of the unsprung part of the wagon were almost equal to or smaller than those in other variants, the reaction force of the rail was considerably less.

Analysis of deviations' distribution (displacements from the initial position) of the wagon–track multi-mass system elements demonstrated that change in the elastic-hysteresis parameters of rail damping components depending on the type of applied material has an inconsiderable impact on the sprung part of the wagon.

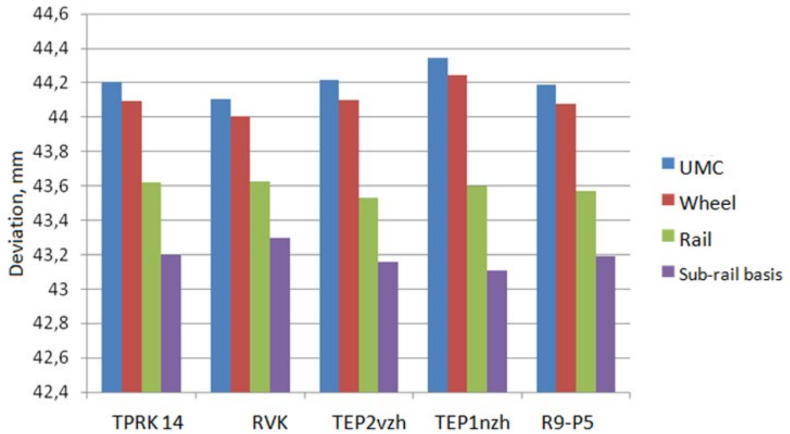
As the stiffness of rubber compounds increases at a temperature of -40°C, and, hence, the damping properties decrease, the wheel deviations are reduced (when using TPRK by 0,24 mm; with RVK by 0,15 mm), and the deviations of the unsprung part of the wagon are also reduced (with TPRK by 0,27 mm; with RVK by 0,16 mm), while rail deviations experience only slight changes (Pic. 10). However, the deviations in the rail supporting system increase (with TPRK by 0,21 mm; with RVK by 0,25 mm).

In the case of using thermoplastic elastomers with stiffness increase at a temperature of -40˚C compared with normal temperature, wheel deviations decrease (when using TEP2vzh by 0,1 mm; with TEP1nzh by 0,21 mm), deviations of the unsprung part of the wagon also decrease (with TEP2vzh by 0,12 mm; with TEP1nzh by 0,21 mm), while deviations of the rail supporting foundation increase (with TEP2vzh by 0,39 mm; with TEP1nzh by 0,34 mm) (Pic. 11).

It is important to mention, that the level of deviations of the wheel and of the unsprung part of the wagon is 1,6 times higher in case if thermoplastic elastomers are used compared with the rubber compounds, that can be explained by the significant increase of stiffness



Pic. 12. Distribution of deviations in case if various thickness of rail damping elements is used.



Pic. 13. Distribution of deviations in case if single-type rail damping elements and 14 mm thick sandwich plates are used.

of these materials due to work within the brittle temperature range.

An increase in stiffness of rubber rail damping components due to reduction of their thickness results in a slight decrease in the deviations of the wheel and the unsprung part of the wagon at a positive temperature (0,1 mm) and in a slight increase at a negative temperature (Pic. 12). As the stiffness increases, the rail deviations decrease, whereas the deviations of rail supporting basis increase at a positive temperature, and practically do not change at a negative temperature thanks to the beginning of the compound crystallization process, which has a predominant influence.

In the case sandwich plate R9-P5 is used (9 mm RVK; 5 mm TEP1nzh), it is observed that the deviations of the wheel and of the unsprung part of the wagon decrease to a level characteristic of the rubber compound. At the same time the deviations of the rail and rail supporting basis decrease by 0,2 mm compared to deviations of RVK dominant component (Pic. 13).

Conclusions

1. The developed calculation model of multi-mass vibratory (wagon–track) system makes it possible to determine with high accuracy and minimum expenses the reaction forces and deviations in the system elements depending on the change of elastic-hysteresis properties of rail damping elements made of various materials, variously designed, and used at different operating temperatures.

2. It is observed that with the increase of rail damping elements' stiffness associated with structural peculiarities, material type and operating temperature, the reaction force of the rail decreases, whereas the reaction force on the wheel and the unsprung mass of the wagon on the

contrary increases. That may constitute an important cause of their early wear and destruction.

3. It is more expedient to apply rubber-fiber compound at low temperatures instead of the rubber compound, since it reduces the load on the unsprung mass of the wagon by 8 %, and the load on rail supporting system by 1,37 times.

4. The application of thermoplastic elastomers as rail damping elements at a temperature of -40°C results in considerable increase of the reaction force acting on the wheel, exceeding by 8 % maximum permissible axial load per one wheel.

5. Deviations of the wheel and of the unsprung part of the wagon when using thermoplastic elastomers as rail damping elements at a temperature of -40°C are 1,6 times higher as compared with the case of the use of rubber compounds; it is explained through considerable stiffness increase of thermoplastic elastomers due to operation within brittle temperatures range.

6. If rubber compounds 14 mm and 10 mm thick are used, the reduction of their thickness in search for an increase in the stiffness of rail damping elements for rail track stabilization will result in inconsiderable changes of the reaction forces and deviations of all system elements.

7. Optimal distribution of the reaction forces and deviations in overall system is achieved by the application of sandwich plates combining various types of components, which in a certain proportion have positive effect on each other. Thereby, the multi-mass vibratory system becomes more balanced, which results in enhancement of the dynamic processes, travel smoothness, reduced risks of resonance phenomena, and the rolling stock stability against derailment.

These conclusions are considered to be quite universal if applied to various types of the rolling stock and infrastructure; however, the appropriate experimental studies of the parameters are required to confirm the suggested conclusions.

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