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New possibilities to reduce wheel and rail wear in the operation of metro wagons on curved track with small curve radii without considering propulsion and braking systems

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Highlights

- Operation of metro wagons on straight and curved tracks with small curve radii.
- Non-linear lateral dynamics of rail vehicles.
- Passive control of wheelsets with very high steering rigidity.

Abstract

This paper analyses the parameters relevant to the operation of underground cars in traffic on straight and curved tracks with small curve radii. Current knowledge of non-linear lateral dynamics of rail vehicles and the specialist computer programme Vi-Rail were used to build models and perform simulation calculations. A solution was proposed to enable passive control of wheelsets with very high guiding stiffnesses, without the need for systems to force the axles of the wheelsets to align radially in a curved track. This effect was achieved by varying the guidance stiffness in the horizontal plane of the frame of each bogie. The adopted course of action offers simplicity of solution, low manufacturing costs, high critical speed on straight track and extended service life of the metro wagons resulting from reduced mechanical wear of wheels and rails. A multi-criteria evaluation was carried out, confirming improved bogie controllability and reduced impact of the metro wagon wheelsets on the curved track. Comparisons were made of the design volumes for metro model wagons with bogies with symmetric and unsymmetric axle-guide stiffness arrangements of the wheelsets.

Keywords

critical speed, lateral dynamics, wheel wear, bogie steerability, curved track interaction

1. Introduction

Rail vehicles play an important role in the global transport system. In operation, they are required to be reliable and sustainable, which means that the running surfaces of the wheels should wear down as slowly as possible over time, enabling the vehicle to cover hundreds of thousands of kilometres without the wheelsets having to be replaced. This problem is still relevant today and is closely linked to the lateral dynamics that affect both wheel wear and the critical speed of vehicles, as pointed out in [13]. In the case of rail vehicles, however, the conflict between the running characteristics of straight and curved track is well known for vehicles in which the wheels are rigidly connected to the axle. This classic design solution, which is commonly used in railway vehicle construction, can only ensure satisfactory wheel wear in curved track operation with a small curve radius if the rigidity of the axle guidance of the wheelsets in the bogie frame is sufficiently low. This results in limitations in the operational characteristics of the rail vehicle on straight track, as the critical speed of the vehicle decreases drastically. To prevent this, the stiffness of the primary suspension in the horizontal plane would have to be increased significantly. This, in turn, will cause undercutting of the flanges running up against the outer rails of a track with a small curve radius. Possible lubrication of the inner surface of the rail and the flange of the wheel running up against it does not remove the main cause of wear. A fairly straightforward way to alleviate this problem is to increase the ability of the bogies to fit passively into the curved track. This can be achieved for example by introducing an additional mechanical system to connect the wheelsets outside the bogie

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frame [1, 3]. In such cases, lightweight sub-frames rigidly connected to the wheelsets and elastically between each other or radial arms connecting the wheelset bearing housings on either side of the bogie frame are often used. An example of a lever-type mechanical system for controlling a single axle in a bogie can be found in the 1000 metro train on the Ginzaw Tokyo line [11]. In the case of technically advanced methods of controlling the position of bogies in curved track, the rotation of the wheelsets relative to the frame is forced in the horizontal plane by electro-mechanical actuators [8, 9]. However, this requires continuous measurement of the lateral displacements and running angles of the wheelsets on the rails, as well as the forces generated by the actuators on the first and second suspension stages of the vehicle. An important group of rail vehicles are metro cars, which have two suspension stages and use electric traction motors to drive the wheelsets individually. This type of drive significantly increases the stiffness of the first suspension stage and thus causes faster wear of the wheel running surfaces on curved track. This is a major operational problem, as metro routes in urban areas consist of both straight track sections and a large number of curved track sections with small curve radii. The following article presents the comprehensive results of simulation studies of metro car models using both current knowledge of rail vehicle dynamics and a specialised computer tool in the form of the Vi-Rail software. The results obtained confirmed the appropriateness of the chosen course of action for reducing wheel wear in metro wagons with rigid axle guidance of the wheelsets in the bogies. The proposed solution enables passive control of the bogie wheelsets in a four-axle wagon with a very rigid primary suspension, without the need for any systems to force the wheelsets to align radially on a curved track. It has been proven that in the case of a four-axle metro car equipped with two-axle H-frame bogies without a swing bolster, even with very high primary suspension stiffnesses, remarkable results can be obtained. Providing an adequate difference between the axle stiffnesses of the wheelsets in the bogies makes it possible to achieve a critical speed of 130 km/h on straight track, which is more than enough to meet the requirements of the metro trains currently in use in Europe and around the world. It has been shown that on a curved track with a curve radius of 300 m, the proposed solution will allow at least an 80% reduction in wheel wear in the bogies of a metro car compared with a contemporary metro car with very high axle-steer rigidity of the wheelsets. In the case of extremely tight access curved tracks with a curve radius of 150 m, a reduction in wheel wear of at least 50% is possible. It has also been demonstrated that the proposed solution provides significantly improved bogie steerability on curved track and, as a result, contributes to a reduction in transverse tangential forces in the wheel-rail contact areas. The content presented in this article is a completely new perspective on the existing problems of metro wagons with rigid axleboxes in terms of minimising wear on the running surfaces of the wheels on curved track and ensuring that the vehicle operates at a high enough speed on straight track. The adopted course of action offers simplicity, low implementation costs and an increase in service life. In the available literature, only in [4] can one find partially similar test results. However, they concern freight wagons with three-piece bogies and high-speed traction vehicles with a swing bolster in the bogie design, which, however, use an additional mechanical system to assist in the control and entry of the bogies into the curved track.

2. Reference model of a metro wagon

Based on the structure of a typical modern metro car, the authors of this article created its 'reference model'. Two suspension stages are used in the design of metro railcars in operation today, using structural elements with elastic and damping characteristics. The first stage of suspension usually referred to as primary suspension is usually formed by steel or metal-rubber springs and hydraulic dampers located between the axle bearing bodies of the wheelsets and the bogie frame. The reference model of a metro wagon is characterised by the symmetry of the stiffness of the primary suspension in the horizontal plane. This means that in each bogie the wheelset guiding stiffnesses are the same. A vehicle was considered whose body is connected directly to the bogies by two pairs of gas springs, mounted on an H-frame without a swing bolster. The only additional structural elements are the traction bars mounted along the wagon between the bogie frames and the body. The components of the reference model are shown in Fig. 1. The wheelsets were assigned the four indices $i = 1, 2, 3, 4$ while the two wheels of a single axle were identified by the indices $j = 1, 2$. The phenomenological model of the metro wagon, including the primary suspension springs and the elastic components of the traction rods, is shown in Fig. 2. The upper and lower case letters were used to describe the stiffness and damping parameters of the first and second stage suspension, respectively. The longitudinal and lateral stiffnesses of the axle guide of the wheelsets in the bogie frame are denoted by the symbols k_{xi}, k_{yi}, while the longitudinal stiffnesses of the traction rods of the front and rear bogies are denoted by the symbols k_{xf} and k_{xr}.

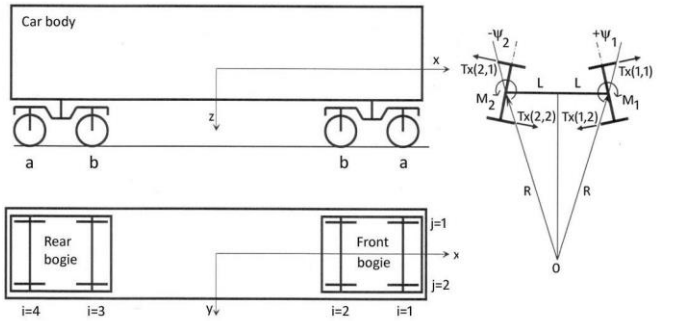

Fig. 1. Elements of the reference metro wagon model and interpretation of signs of angles of attack of wheelsets and moments M_i generated by the longitudinal tangential forces $Tx(i, j)$ on curved track in case of the front bogie.

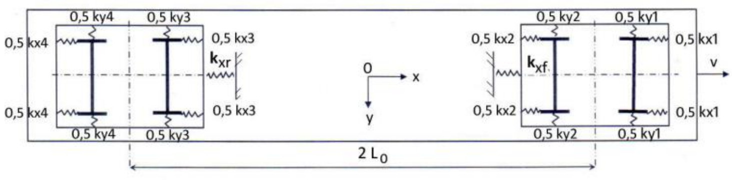

Fig. 2. Plan view of the phenomenological model of the primary suspension and of the traction rod.

Table 1. Main parameters of the reference metro wagon model.

Designation	Parameter	Value
$2L_0$	Wagon base	12.6 m
$2L$	Bogie base	2 m
M	Mass of wagon with passengers	29 610 kg
M_C	Car body mass	17 960 kg
M_W	Wheelset mass	1756 kg
r	Nominal wheel radius	0.43 m
k_Z	Vertical stiffness of the primary suspension	1.88 MN/m
K_Z	Vertical stiffness of the secondary suspension	0.690 MN/m
k_X, k_Y	Longitudinal/lateral guide stiffness of the primary suspension	6 or 14 MN/m
K_X, K_Y	Longitudinal/lateral guide stiffness of the secondary suspension	0.45 MN/m
k_{Xf}, k_{Xr}	Longitudinal stiffness of the traction rod in the front and rear bogies	10 MN/m
c_{Xf}, c_{Xr}	Longitudinal damping of the traction rod in the front and rear bogies	25 kNs/m
$c_X, c_{Y,} c_Z$	Damping coefficients in x, y, z directions for primary suspension	3 kNs/m
$C_X, C_{Y,} C_Z$	Damping coefficients in x, y, z directions for secondary suspension	2 kNs/m
μ	Coefficient of friction between wheel and rail	0.36

The stiffness and damping parameters of all primary and secondary suspension components in the vertical plane were also taken into account. In the following, the letters **a** and **b** were used to describe the axle guiding stiffness system of the wheelsets in the bogie frame in the longitudinal and lateral directions. According to Fig. 1, letter **a** identifies the stiffnesses of wheelsets numbered 1 and 4, whereas letter **b** is assigned to wheelsets numbered 2 and 3. Based on the design of the metro wagons manufactured by global rolling stock manufacturers such as Alstom and Siemens, two sets of parameters describing the numerical values of the stiffness and damping of the primary and secondary suspension of the reference model were prepared. The data for the geometric, mass and inertia parameters of all elements of the metro wagon model are listed in Table 1. A distinction is made between the case of moderately stiff axle guidance of the wheelset in the bogie frame, when the stiffnesses $k_{X,i}, k_{Y,i}$ take values equal to 6 MN/m, and the case of very stiff suspension, when each of these stiffnesses reaches a value of 14 MN/m in the longitudinal and transverse directions.

3. Simulation model of a metro wagon

A simulation model of a metro wagon was built using Vi-Rail, a specialised program designed to study the dynamics of three-dimensional rail vehicle models. Vi-Rail is one of the MBS computer tools for the dynamics of mechanical multi-body systems, where the mass elements are treated as perfectly rigid bodies, connected by kinematic pairs. The position of each member in the simulated metro wagon model was described by 3 Cartesian coordinates and 3 Euler angles relative to a global, stationary reference system. The result is a simulation model of a metro wagon with 46 degrees of freedom. Knowing the coordinates of the centres of gravity of the individual members of the model, their masses and mass moments of inertia, the Vi-Rail software generated and solved a system of differential-algebraic equations in each calculation cycle. The calculations took into account non-linear, real rail outlines with a gauge of 1.435 m and a slope of 0.025 rad, and track parameters in accordance with the operational requirements contained in [5]. The curved track consisted of three sections: a straight track section of 50 m length, a transition curve of the same length and a full curve of 200 m length with a cant of 150 mm for a 300 m radius curve. For the 150 m radius curve, the rail spacing in the full curve section of the curved track was widened by 15 mm while reducing the cant from 150 mm to 66 mm, according to [2]. In the description of the wheel-rail contact, the non-linear contours of the S1002 transverse wheel profiles and the UIC60 rails were taken into account, based on the simplified rolling theory of Kalker [7]. The FASTSIM procedure [10] was used to calculate the dynamic quantities describing the behaviour of the simulated metro wagon model on straight and curved track. The tangential forces were calculated as a function of the sliding slips and spins between wheels and rails [7]. For this purpose, a table of contact parameters was used, from which the difference of the current rolling radii of the wheels, the contact angles of the wheels with the rails and the half-axes of the contact ellipse of a given wheel with the rail were read out in each calculation step [12].

4. Criteria for assessing the dynamic properties of a simulation model of a metro wagon with unsymmetric stiffness system for guiding the wheelsets in the bogie frame

The loss of stability of the bogies of a wagon on straight track is a consequence of the occurring vibrations of the wheelsets. These are self-excited oscillations caused by the interaction between the wheels of the moving vehicle and the rails. The speed at which such oscillations occur is called the critical speed. If the contact model takes into account the non-linear transverse profiles of the wheels and rails, we are dealing with the non-linear critical speed of the wagon. The dynamic properties of the simulated metro wagon model were investigated on straight and curved track, assuming that an arc with a radius of 300 m represents a tightly curved track under operating conditions and a radius of 150 m characterises rail access curved track. The permissible speeds of the wagon model, for the above curve radii, were 20 m/s and 15 m/s, respectively. The reference model of the metro wagon described earlier, with a symmetric stiffness system for guiding the axles of the wheelsets in the bogie frame and the parameters collected in Table 1, was used for comparative analyses and evaluation of the dynamic properties of the model with an unsymmetric stiffness system. For this purpose, the following criteria were adopted:

- the non-linear critical speed of the wagon model in straight track traffic should not be less than 120 km/h;
- the wear index for each wheel of the model with unsymmetric stiffness system should be lower compared to the corresponding wheel index of the model with symmetric system;
- the controllability of the bogies on curved track for the model with the unsymmetric stiffness system should be better than for the symmetric system.

In the Vi-Rail programme, wear index are referred to as wear number. This term will be used later in the article. An energy model (the so-called “T-gamma model”) based on the assumption that the amount of material lost is a function of the energy dissipated in the wheel-rail contact area was used to assess wear on the wheel rolling surface [6]. The wear number $W_n(i,j)$ was calculated for a given wheel as the sum of the products of the tangential and slip forces, according to the following relationship:

$$W_n(i,j) = |T_x(i,j) \cdot \gamma_x(i,j)| + |T_y(i,j) \cdot \gamma_y(i,j)| \qquad (1)$$

The wear number $W_n(i,j)$ physically is interpreted as the amount of energy dissipated over a distance of one metre by the longitudinal $T_x(i,j)$ and lateral $T_y(i,j)$ tangential forces acting in the wheel-rail contact area when the vehicle is in motion. In formula (1), the symbols $\gamma_x(i,j)$ and $\gamma_y(i,j)$ denote the longitudinal and lateral slip of the wheel in question, respectively. The principle of assigning the respective letter indexes to the axles of the wheelsets and the individual wheelsets is shown in Fig. 1. The moments exerted by the longitudinal tangential forces $T_x(i,j)$ and acting on the given wheelset are described by the symbols M_i. The steerability Θ_k of a bogie on a curved track is defined according to the following relation:

$$\Theta_k = \frac{1}{2}(\Psi_i + \Psi_{i+1}) \qquad (2)$$

In formula (2), the symbols Ψ_i and Ψ_{i+1} denote the attack angles of the wheelsets on the rails of the curved track, as interpreted in Fig. 1. The index $k = 1,2$ indicates the number of the bogie in the wagon. According to the above formula, perfect bogie steerability occurs when $\Theta_k = 0$.

5. Results of simulation studies

5.1 Symmetric arrangement of the wheelset guide stiffness in the bogie frames

In the case of the symmetric system, all guidance stiffnesses kx_i, ky_i of the axles of the wheelsets within the bogies were equal to each other. The numerical values of these stiffnesses were varied from 2 MN/m to 14 MN/m. Based on the bifurcation diagrams prepared for each wheelset, the numerical values of the non-linear critical speeds of the metro wagon model with empty $v_{cr,n}(E)$ and full $v_{cr,n}(L)$ body were determined. Analysis of the results, shown in Fig. 3, led to the conclusion that a non-linear critical speed of 120 km/h in straight-track operation can only be achieved by a wagon model with a symmetric stiffness system if the leading stiffness of each wheelset is at least 4 MN/m. The numerical values of the wheel wear numbers of the reference metro wagon model in curved track traffic with a radius of 300 m are shown in Fig. 4.

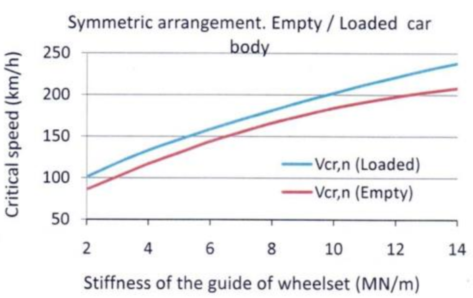

Fig. 3. Nonlinear critical speeds $v_{cr,n}(E)$ and $v_{cr,n}(L)$ in function of the guide stiffness of wheelsets in bogie frame of the metro wagon model with empty and loaded car body for symmetric arrangement.

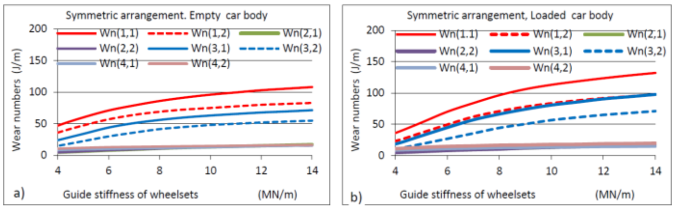

Fig. 4. Wear numbers for wheels of metro wagon models with symmetric arrangement on the full arc of curved track with a radius of 300 m: a) empty car body, b) loaded car body.

The case of a curved track with a radius of 150 m is illustrated in Fig.5. In each of the bogies of the metro wagon model, the wheel wear numbers were many times higher for the front wheelsets than for the rear axle wheelsets. Further analyses looked at the guiding stiffnesses $k_{X,i}$, $k_{Y,i}$ of the wheelsets in the range of 4 MN/m to 14 MN/m for the fully passenger-loaded wagon model.

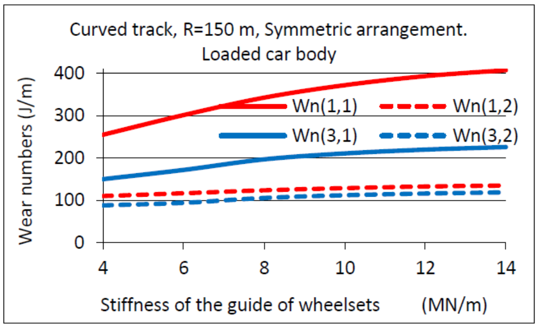

Fig. 5. Wear number values for the wheels of leading wheelsets of metro wagon model with loaded car body on curved track with the radius of 150 m.

5.2 Unsymmetric arrangement of the wheelset guide stiffness in the bogie frames

The unsymmetric stiffness system means that the axle-guiding stiffnesses of the wheelsets differ by the same amount in the two bogies of the wagon model. The bogies are positioned under the body in such a way as to ensure that the running characteristics of the wagon are the same regardless of the direction of travel. If the successive wheelsets in the wagon are assigned indexes 1, 2, 3, 4 then in the asymmetric arrangement of the axle-guiding stiffness of the **a-b-b-a** wagon, the letter **a** represents the stiffness of wheelsets numbered 1 and 4, while the letter **b** is assigned to wheelsets numbered 2 and 3. Taking into account the conclusions of the previous calculations, it was determined that the axle-guiding stiffness of a single axle of a wheelset could not be lower than 4 MN/m. Two calculation scenarios were considered for the wagon model. According to the first scenario, the axle-guidance stiffnesses of wheelsets 2 and 3 were changed with a step of 2 MN/m, while the axle-guidance stiffnesses of wheelsets 1 and 4 were fixed and unchanged. The second scenario was implemented in reverse. The values of the non-linear critical speeds of the metro wagon model in straight track traffic are graphically presented in Fig.6 as a function of the difference between the axle guiding stiffnesses of the wheelsets in the bogie frame.

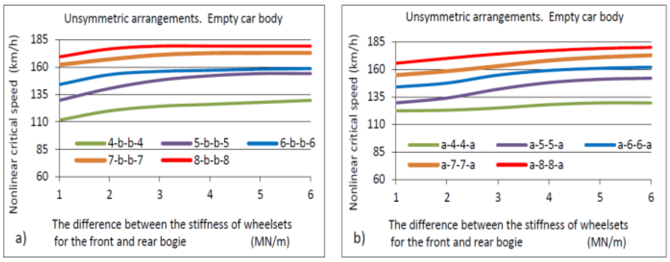

Fig. 6. Changes of nonlinear critical speed of metro wagon model with empty car body in function of difference between the guide stiffnesses of wheelsets for the front and rear bogie with unsymmetric arrangements; a) first scenario, b) second scenario.

Obtained calculation results indicate that in an unsymmetric axle-guiding arrangement of the wheelsets, the difference between the stiffnesses of the two axles in the bogie frame should be at least 2 MN/m, while the guiding stiffness of a single axle should not be less than 4 MN/m in order for the non-linear critical speed of the wagon without passengers to reach 120 km/h. The calculation of the non-linear critical speed for a wagon model with passengers (i.e. with full body load) were made for configurations **4-b-b-4** and **a-4-4-a** and shown in the Fig. were7. In the **4-b-b-4** configuration, the guiding stiffnesses of axle numbers 1 and 4 were constant and equal to 4 MN/m, while the guiding stiffnesses of axle numbers 2 and 3 were varied from 6 MN/m to 14 MN/m. In the **a-4-4-a** configuration, the guiding stiffnesses of axle numbers 2 and 3 were constant and equal to 4 MN/m, while the guiding stiffnesses of axle numbers 1 and 4 were varied from 6 MN/m to 14 MN/m.

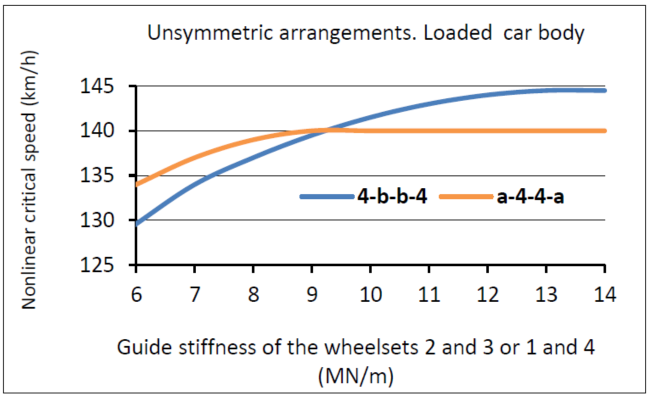

Fig. 7. Comparison of nonlinear critical speed values of metro wagon model with loaded car body in function of guide stiffness of wheelsets for **4-b-b-4** and **a-4-4-a** arrangements.

More promising was the **4-b-b-4** arrangement for the axle-guiding stiffness of the wheelsets in the bogies of a metro wagon model. A comparison of the magnitudes of the wear numbers for the wheelsets in the leading wheelsets of the front and rear bogies on curved track with radii of 300 m and 150 m is shown in Figures 8, 9, 10 and 11. The results for the variants possible in the **4-b-b-4** arrangement were compared with the corresponding results for the symmetric **b-b-b-b** configurations, marked in red in the figures. The best results were obtained for the **4-14-14-4** arrangement, in which the differences between the guidance stiffnesses of the wheelsets within the bogies were 10 MN/m.

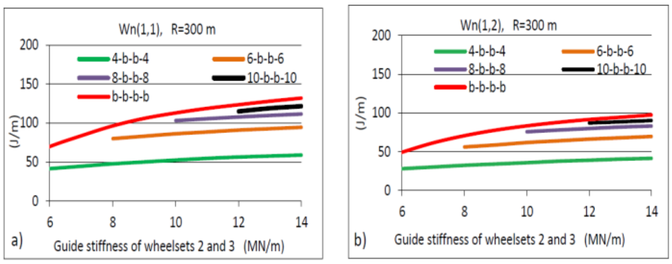

Fig. 8. Comparison of wear number values for the wheels of leading wheelset of front bogie on curved track with the radius of 300 m for symmetric **b-b-b-b** and unsymmetric **4-b-b-4** arrangements; a) left wheel, b) right wheel.

In order to better illustrate the calculation results obtained, a coefficient $p(i,j)$, calculated as a percentage, was introduced to compare the value of the wear number of a wheel in the unsymmetric wheelset axle stiffness system with the value of the wear number of the corresponding wheel in the symmetric system. The values of the coefficients $p(i,j)$ were determined according to the following formula:

$$p(i,j) = \frac{W_{n,unsym}(i,j) - W_{n,sym}(i,j)}{W_{n,sym}(i,j)} \cdot 100\% \,, \tag{3}$$

where $W_{n,sym}(i,j)$, $W_{n,unsym}(i,j)$ denote the numerical values of the compared wheel wear numbers in the symmetric and unsymmetric axle-steering stiffness arrangement of the wheelsets, respectively. A negative sign of $p(i,j)$ indicates a reduction and a positive sign an increase in the amount of wear of the respective wheel. The calculation results for the **4-6-6-4** and **4-14-14-4** arrangements were compared with each other with the corresponding symmetric **6-6-6-6** and **14-14-14** configurations. The relevant results are summarised in Table 2 and Table 3. In both cases, the asymmetric layouts offer lower values of wheel wear rates compared to the symmetric layouts. On a curved track with a radius of 300 m, the **4-6-6-4** arrangement reduces the wear numbers of the outer wheels by approx. 40% for the first bogie and by approx. 30% for the second bogie. On a track with a 150 m radius, the corresponding wheel wear numbers were reduced by 11% and 31%.

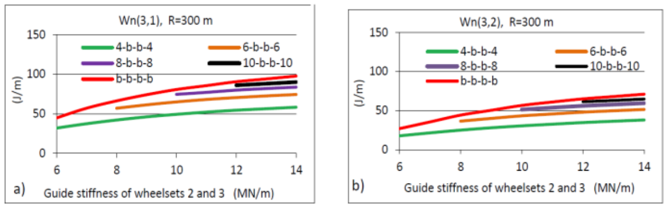

Fig. 9. Comparison of wear number values for the wheels of leading wheelset of rear bogie on curved track with the radius of 300 m for symmetric **b-b-b-b** and unsymmetric **4-b-b-4** arrangements: a) left wheel, b) right wheel.

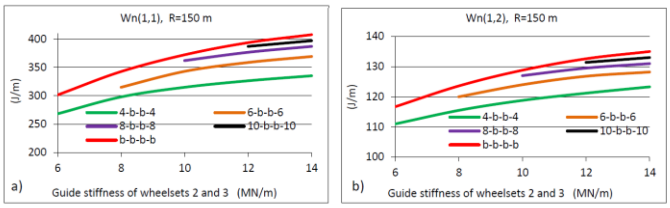

Fig. 10. Comparison of wear number values for the wheels of leading wheelset of front bogie on curved track with the radius of 150 m for symmetric **b-b-b-b** and unsymmetric **4-b-b-4** arrangements: a) left wheel, b) right wheel.

The **4-14-14-4** arrangement offers on curved track with a radius of 300 m, a reduction in wheel wear of 56% and 43% on average in the first and second bogies, respectively, compared to the symmetric **14-14-14-14** arrangement. On a track with a radius of 150 m, the reduction in wear is 14% and 26% respectively. Even better results, in terms of wheel wear on curved track, were obtained when the unsymmetric **4-14-14-4** arrangement was modified to reduce the longitudinal axle guiding stiffnesses of the wheelsets with the numbers 2 and 3, from 14 MN/m to 2 MN/m and leaving the lateral guiding stiffness of these wheelsets at 14 MN/m.

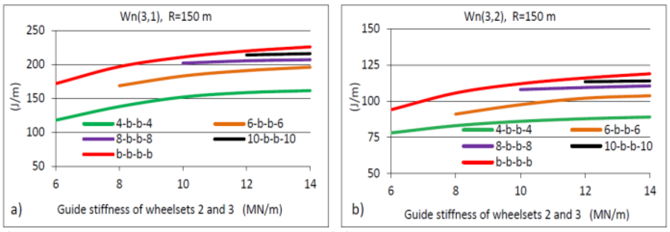

Fig. 11. Comparison of wear number values for the wheels of leading wheelset of rear bogie on curved track with the radius of 300 m for symmetric **b-b-b-b** and unsymmetric **4-b-b-4** arrangements: a) left wheel, b) right wheel.

Table 2. The values of the $p(i,j)$ coefficients on a curved track for wheelsets no. 1 and 3 of the **4-6-6-4** axle guidance system in the metro wagon model.

Arrangement	R = 300 m				R = 150 m			
	Position of the bogie in the metro wagon model							
	Front bogie		Rear bogie		Front bogie		Rear bogie	
	p(1,1)	p(1,2)	p(3,1)	p(3,2)	p(1,1)	p(1,2)	p(3,1)	p(3,2)
	%	%	%	%	%	%	%	%
4-6-6-4	-40	-43	-29	-34	-11	-5	-31	-17

Table 3. The values of the p(i, j) coefficients on a curved track for wheelsets no. 1 and 3 of the **4-14-14-4** axle guiding stiffness in the metro wagon model.

Arrangement	R = 300 m				R = 150 m			
	Position of the bogie in the metro wagon model							
	Front bogie		Rear bogie		Front bogie		Rear bogie	
	p(1,1)	p(1,2)	p(3,1)	p(3,2)	p(1,1)	p(1,2)	p(3,1)	p(3,2)
	%	%	%	%	%	%	%	%
4-14-14-4	-55	-57	-40	-46	-18	-9	-28	-25
modified **4-14-14-4**	-78	-81	-94,5	-96,5	-53	-30	-78	-78,5

For such a modified guide stiffness arrangement of the wheelsets, the non-linear critical speed of the metro wagon model on straight track was 129 km/h. At the same time, on a curved track with a radius of 300 m, a reduction in wheel wear of 80% and 95% on average was achieved in the first and second bogies compared to the symmetric **14-14-14-14** arrangement. For a curved track with a radius of 150 m, wear was reduced by 41% and 78% respectively. Table 4 collects numerical values evaluating bogie steerability for the cases discussed above. On a curved track with a radius of 300 m, the **4-6-6-4** arrangement of guide stiffness of the wheelsets in the metro wagon model provides three times better steering for the front bogie and two and a half times better steering for the rear bogie compared to **6-6-6-6** arrangement. In the case of the **4-14-14-4** arrangement, the steerabilities Θ_1 and Θ_2 on the curved track with the radius of 300 m are 17 and 80 times better compared to **14-14-14** arrangement. On the track with the radius of 150 m they are 2 and 36 times better in comparison with the symmetric **14-14-14** configuration.

Table 4. Steerabilities Θ_1 i Θ_2 of metro wagon model bogies on a curved track for selected arrangement of guide stiffnesses.

Arrangement	R = 300 m		R = 150 m	
	Position of the bogie in the metro wagon model			
	Front bogie	Rear bogie	Front bogie	Rear bogie
	Θ_1(mrad)	Θ_2(mrad)	Θ_1(mrad)	Θ_2(mrad)
6-6-6-6	0,9	0,5	4,8	2,6
4-6-6-4	0,3	0,2	4,3	1,9
14-14-14-14	2,3	1,6	6,2	3,6
4-14-14-4	0,6	0,7	5,1	2,4
modified **4-14-14-4**	0,13	0,02	3,1	0,1

The steerability of the bogies on curved track is determined by the forces acting in the wheel-rail contact areas. Figures 12 to 14 show the wheelset positions of a metro wagon model on curved tracks with radii of 300 m and 150 m, for **14-14-14** and **4-14-14-4** arrangements. These figures also show the senses and values of moments $M_i (i = 1, \ldots, 4)$, as well as the values of angles of attack of wheelsets on the rails. Values of lateral tangential forces $T_y(i,j)$ acting on the wheels of wheelsets of the front and rear bogies on the curved tracks with radii of 150 and 300 meters are presented in figures from 15 to 17. The results summarised in Table 5 apply to a curved track of radius 150 m. They were used to compare the dynamic properties of the metro wagon model in **4-14-14-4** configuration with those of **4-4-4-4** and **14-14-14**. The magnitudes of the lateral displacements of the wheelset axles, angles of attack, the moments generated by the longitudinal tangential forces in the wheel-rail contact plane and the magnitudes of the lateral tangential forces were compared with each other. In the case of metro wagon model with symmetric arrangement of the wheelset axle guiding stiffness in the bogie frame, the angles of attack and lateral displacements of the wheelsets numbered 1 and 3 are greater than those of the wheelset axles numbered 2 and 4. This promotes large lateral tangential forces acting on the leading wheelsets in each bogie. These forces are further increased as a result of the so-called steering moments generated by the longitudinal tangential forces acting on the wheels of wheelsets numbered 2 and 4, whose axles are much closer to the centre of the track than the axles of wheelsets 1 and 3. In case of symmetric arrangement these are the main reasons for the high wear of the wheels in the leading wheelsets when primary suspension is hardly stiff. It, therefore, seems obvious that the reduction of wheel wear in a metro wagon model with highly rigid primary suspension should be possible if the angles of attack of the leading wheelsets are reduced and the difference between the lateral displacements of the wheelset axles in the respective bogie is reduced. In the case of a metro wagon model with a symmetric **14-14-14-14** stiffness arrangement, such an effect was achieved by reducing the guiding axle stiffness of wheelsets No. 1 and No. 4 from 14 MN/m to a value of 4 MN/m and leaving the axle stiffness of wheelsets No. 2 and No. 3 unchanged at 14 MN/m.

When the metro wagon model ran over a curved track with a radius of 150 m, the resulting **4-14-14-4** configuration reduced the difference between the lateral displacements of the wheelset axles in the first bogie by 4.1 mm and in the second bogie by 4.9 mm compared with the symmetric **14-14-14-14** configuration. Angles of attack of the leading wheelsets were also reduced in the front and rear bogies by 1.9 mrad and 2.7 mrad, respectively, and the magnitude of the lateral tangential forces in the wheel-rail contact areas was also reduced by 0.6 kN for each leading wheelset. This had the resultant effect of reducing the wear of the wagon running on tight curved track, while at the same time maintaining very rigid guidance of one of the wheelset axles within each bogie.

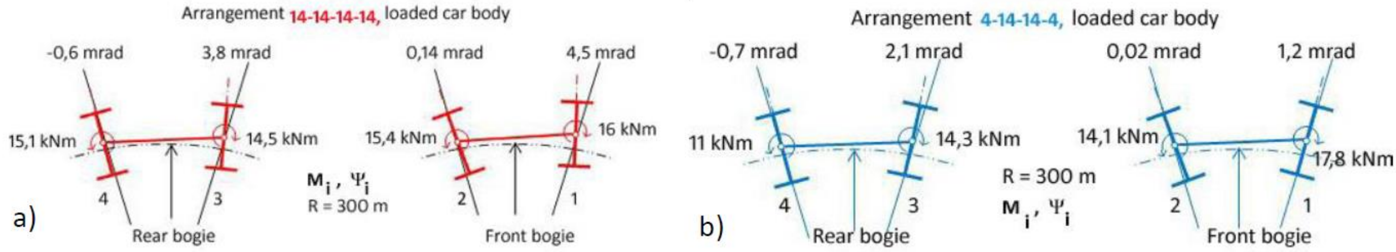

Fig.12. Alignments of leading and trailing wheelsets of the front and rear bogies on curved track with radius of 300 m: a) symmetric arrangement **14-14-14-14**, b) unsymmetric arrangement **4-14-14-14.**

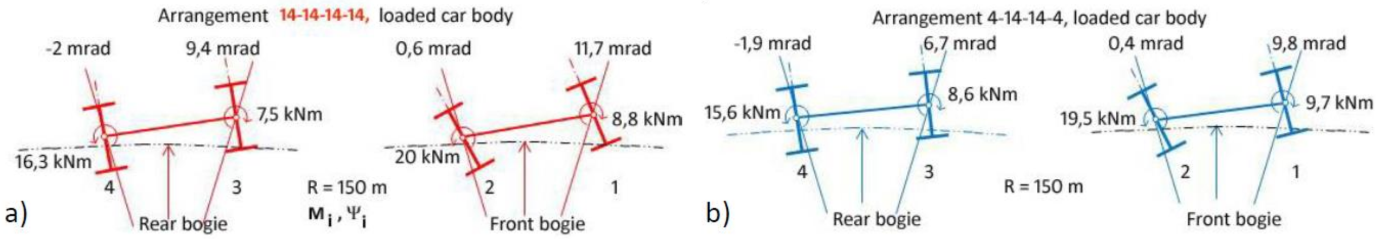

Fig. 13. Alignments of leading and trailing wheelsets of the front and rear bogies on curved track with radius of 150 m: a) symmetric arrangement **14-14-14-14**, b) unsymmetric arrangement **4-14-14-14.**

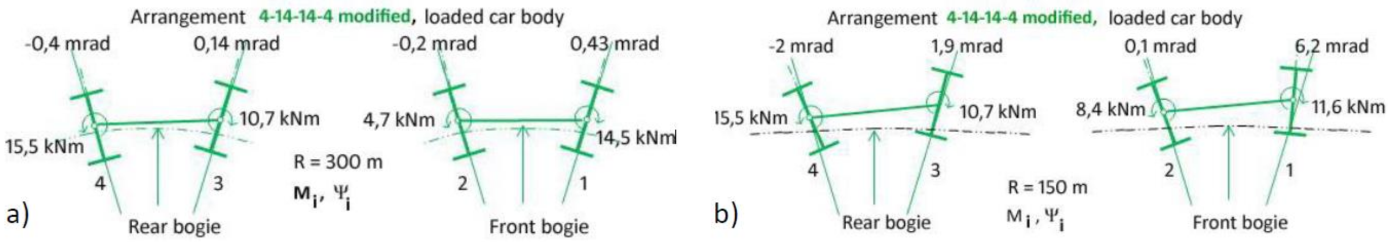

Fig. 14. Alignments of leading and trailing wheelsets of the front and rear bogies on curved track for modified unsymmetric arrangement **4-14-14-14:** a) curved track with radius of 300 m, b) curved track with radius of 150 m.

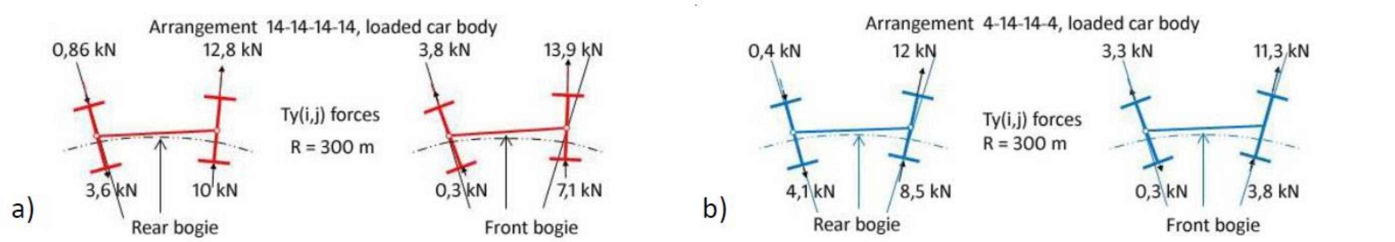

Fig. 15. Values of lateral tangential forces $T_y(i,j)$ acting on the wheels of wheelsets of the front and rear bogies on the curved track with radius of 300 m: a) symmetric arrangement **14-14-14-14**, b) unsymmetric arrangement **4-14-14-14.**

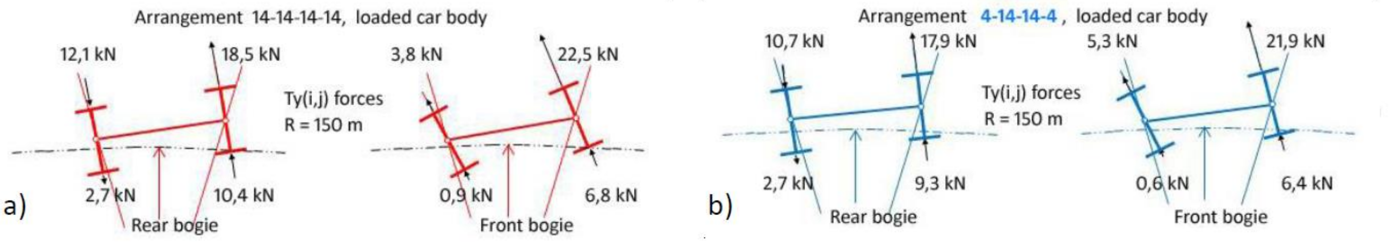

Fig. 16. Values of lateral tangential forces $T_y(i,j)$ acting on the wheels of wheelsets of the front and rear bogies on the curved track with radius of 150 m: a) symmetric arrangement **14-14-14-14**, b) unsymmetric arrangement **4-14-14-14.**

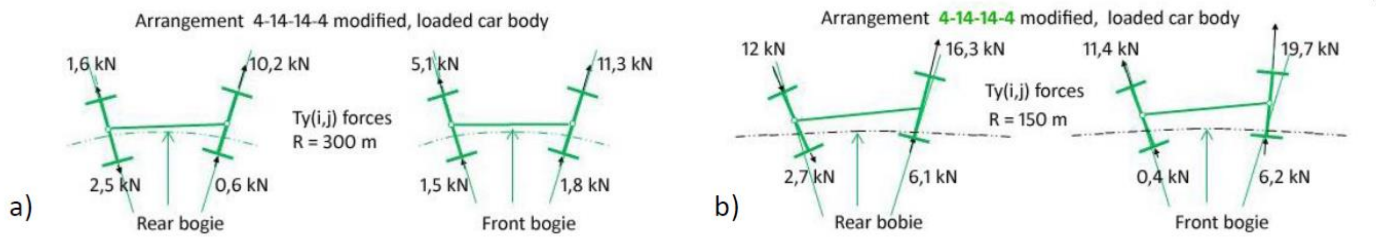

Fig. 17. Values of lateral tangential forces $T_y(i,j)$ acting on the wheels of wheelsets of the front and rear bogies on curved track for modified unsymmetric arrangement **4-14-14-14**: a) curved track with radius of 300 m, b) curved track with radius of 150 m.

Table 5. Comparative analysis of three arrangements **4-4-4-4**, **14-14-14-14** and **4-14-14-4** on the curved track with radius of 150 m.

Arrangement											
4-4-4-4				**14-14-14-14**				**4-14-14-4**			
Lateral displacements of wheelsets towards outer rail (mm)											
y1	y2	y3	y4	y1	y2	y3	y4	y1	y2	y3	y4
13.3	12.3	13.2	8.4	13.6	4.2	13.3	5.8	13.6	8.3	13.2	8.3
Angles of attack (mrad)											
Ψ_1	Ψ_2	Ψ_3	Ψ_4	Ψ_1	Ψ_2	Ψ_3	Ψ_4	Ψ_1	Ψ_2	Ψ_3	Ψ_4
7.2	0.33	4.7	-2	11.7	0.6	9.4	2.1	9.8	0.4	6.7	-1.9
Moments acting on wheelsets in the horizontal plane (kNm)											
$\lvert M_1\rvert$	$\lvert M_2\rvert$	$\lvert M_3\rvert$	$\lvert M_4\rvert$	$\lvert M_1\rvert$	$\lvert M_2\rvert$	$\lvert M_3\rvert$	$\lvert M_4\rvert$	$\lvert M_1\rvert$	$\lvert M_2\rvert$	$\lvert M_3\rvert$	$\lvert M_4\rvert$
11.1	14.4	10.2	15.6	8.8	20	7.5	16.3	9.7	19.5	8.8	15.9
Lateral tangential forces in the wheel-rail contact areas (kN)											
$T_y(1,1)$	$T_y(2,1)$	$T_y(3,1)$	$T_y(4,1)$	$T_y(1,1)$	$T_y(2,1)$	$T_y(3,1)$	$T_y(4,1)$	$T_y(1,1)$	$T_y(2,1)$	$T_y(3,1)$	$T_y(4,1)$
20.6	9.6	17.3	11.3	22.5	3.8	18.5	-12.1	21.9	5.3	17.9	-10.7

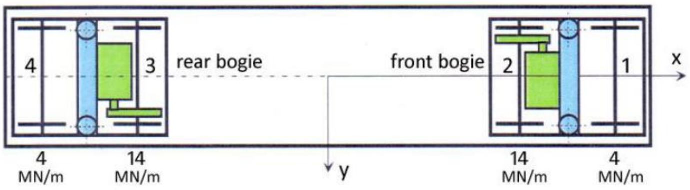

Fig. 18. Visualization of traction motor alignment for the subway car model with unsymmetric primary suspension stiffness arrangement **4-14-14-4.**

This is important because of the propulsion system, which often requires high steering rigidity. In the case under consideration, it would be possible to use a drive system with only one axle in each bogie, as shown schematically in Fig. 18. This is analogous to the solution used in the 1000 series metro cars, with the difference that the solution proposed in the article is much simpler and does not require the introduction of a special system to control the wheelsets in the bogies of the car.

6. Final conclusions

This paper presents the results of numerical simulation studies that have confirmed the expectations of the possibility of reducing wheel and rail wear on curved track by metro wagon with rigid axle guidance of wheelsets in H-frame two-axle bogies without swing bolster. The prerequisite for this is that the primary suspension takes into account a sufficiently large difference between the stiffnesses of the axle guidance of the adjacent wheelsets in the horizontal plane of the bogie frame. A simple solution has been proposed that allows passive control of the bogie wheelsets in a four-axle metro wagon with very high primary suspension stiffness, without the need to introduce additional systems to force radial alignment in the curved track. The essence of this solution, as applied to the metro wagon, is to replace the symmetric arrangement of axle-guidance stiffnesses of the wheelsets in such a way that the axle-guidance stiffnesses of axle numbers 2 and 3 are significantly higher compared to the axle stiffnesses of wheelsets numbered 1 and 4. In the proposed asymmetric **4-14-14-4** arrangement, the difference between the axle-guide stiffnesses of the wheelsets in each bogie was 10 MN/m, and this was the optimum magnitude in the case under consideration. The asymmetric **4-14-14-4** configuration allowed, on a curved track with a radius of 300 m, taken as a representative curved track with a small curve radius, to reduce wheel wear by an average of 56% and 43% in the first and second bogies, respectively, compared with the symmetric **14-14-14-14** arrangement.

On the 150 m radius track, which is normally a rail access curved track, the reduction in wear was 14% and 26% respectively. For the **4-14-14-4** arrangement, the front and rear bogie steerabilities of the metro wagon model on the 300 m radius curved track were four times and two times better than the bogie steerabilities with the symmetric 14-14-14-14 guide stiffness. For the 150 m radius track, these rates were 20% and 50% better, respectively, compared with the symmetric layout. This demonstrates the propensity of bogies with unsymmetric guide stiffness arrangement of the wheelsets to adopt near-radial positions in curved track. The proposed solution decisively removes the main causes of high wheel wear in the bogies of a wagon with a very rigid primary suspension. The solution reduces the wear of the wagon's wheels in tight curved track traffic when one of the axles of the wheelset is very rigidly guided within each bogie. This is important in view of the propulsion system used in metro cars, which requires a high stiffness in the guidance of the axles of the wheelsets within the bogie frame. In the case under consideration, only one axle in each bogie can be driven. This is analogous to the solution used in the 1000 series metro cars, with the difference that the one proposed in the article is much simpler and does not require an additional system to control the wheelsets in the bogies of a car running on a curved track.

References

1. Bruni, S., Goodall, R., Mei, T.,Tsunashima, H.: Control and monitoring for railway vehicle dynamics. *Vehicle System Dynamics,* Vol. 45, Nos. 7–8, July–August 2007, 743–779, doi.org/10.1080/00423110701426690.
2. European Document EN 14363:2016. Railway applications - Testing and Simulation for the acceptance of running characteristics of railway vehicles - Running Behaviour and stationary tests.
3. Illingworth,R., Pollard, M.: The use of steering axle suspensions to reduce wheel and rail wear in curves. Proc InstnMechEngrsVOl, 196, 1982, 379-385, doi.org/10.1243/PIME_PROC_1982_196_040_02.
4. Illingworth, R.: The use of unsymmetric plan view suspension in rapid transit steering bogies. *Vehicle System Dynamics.* Volume 12, 1983, Issue 1-3, 75-88, doi.org/10.1080/00423118308968724.
5. Iwnicki, S. D.:The Manchester benchmarks for rail vehicle simulation. Suppl.Veh. Syst. Dyn., 1999.
6. Jendel, T., Berg, M.: Prediction of wheel profile wear. Vehicle System Dynamics, Supplement 37(2002), pp. 502–513.https://doi.org/10.1080/00423114.2002.11666258.
7. Kalker, J.J.: Three-Dimensional Elastic Bodies in Rolling Contact. Kluwer Academic Publishers. Dordrecht/Boston/London, 1990, doi.org/10.1080/00423114.2002.11666258.
8. Kamoshita,S.,Ishige,M.,Sato,E.,Tanifuji, K.: Development of Assist Steering Bogie System for Reducing the Lateral Force. Railway Technology Avalanche, July 17, 2008, No 22-23.
9. Liu, X., Goodall, R., Iwnicki , S.: Yaw compensation and yaw relaxation controls for active steering of railway wheelsets via electromechanical actuators. Proc IMechE Part F: J Rail and Rapid Transit, 2022, Vol. 236(1) 70–79, doi.org/10.1177/09544097211004322.
10. Piotrowski, J.; Kik, W.: A simplified model of wheel/rail contact mechanics for non-Hertzian problems and its application in rail vehicle dynamic simulations. Vehicle System Dynamics, vol. 46, January-February 2008, pp. 27-48, doi.org/10.1080/00423110701586444.
11. Shimokawa, Y., Mizuno, M.: Development of the New Concept Steering Bogie. Nippon Steel & Sumitomo Metal Technical Report no. 105, December 2013, doi.org/10.1299/jmtl.6.124.
12. VI-Rail 15.0 Documentation. VI-grade engineering software & services, 2013.
13. Žuraulis V, Sokolovskij E, Matijošius J. The opportunities for establishing the critical speed of the vehicle on research in its lateral dynamics. Eksploatacja i Niezawodnosc – Maintenance and Reliability 2013; 15 (4): 312–318.