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On the problems of lateral force effects of railway vehicles in S-curves

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ABSTRACT

This paper deals with the lateral force effects of conventional four-axle railway vehicles in conditions of S-shape curves, which are represented especially by switches negotiated in the diverging direction. With the use of a multi-body model of an electric locomotive, a sensitivity analysis of selected vehicle parameters on the maximum guiding force in the S-curve is performed. Subsequently, a definition of so-called equivalent guiding force is proposed. This simplified parameter can be used for the purposes of a quantification of the damaging effects of a vehicle in switches, e.g. in wear-dependent track access charge systems. Attention is also paid to the particularities of systems for active radial bogie or wheelset steering in conditions of tight S-curves and relevant requirements on these systems. The problem of a punctual on-board recognition of the S-curve is demonstrated using experimental data measured on an electric locomotive.

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S-curve; equivalent guiding force; four-axle vehicle; multi-body simulation; active radial wheelset steering; active yaw dampers

1. Introduction

In the framework of the acceptance process of new railway vehicles, attention is paid to running safety, track loading and ride characteristics as well as to stationary tests. The relevant requirements for the evaluation of the tests and assessment of their results are defined in the European standard EN 14363 [1]. Verification of the track loading effects is based on the evaluation of force interaction in wheel/rail contact (vertical wheel force Q and guiding force Y) in defined track sections. These track sections cover the tangent track and curves with a minimum radius of 250 m (test zone 4, so-called very small radius curves, is defined by the range 250 m $\leq R <$ 400 m). However, the requirements on the track loading assessment given in Ref. [1] do not cover the conditions of curves with radius smaller than 250 m as well as S-curves (e.g. diverging parts of switches and crossings). Problems related to the vehicle/track interaction in 'extra tight curves' typically occur at infrastructure managers in mountainous countries, e.g. in Switzerland [2]. Because the vehicle approval process covering the tests in accordance with EN 14363 is embedded in the relevant technical specifications for interoperability, a full interoperability of the vehicles does not have to be guaranteed in such conditions. For example, the track access to the relevant part of Swiss

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railway network is regulated by means of the guideline SBB R I-50127 defining so-called test zone 5 (i.e. curves with radius smaller than 250 m) and relevant limit values of the vehicle/track force interaction (see, e.g. Ref. [2]).

The second problem is related to the vehicle running through S-curves (S-shaped curves or reverse curves). In the actual version of EN 14363 [1], only a reference to the requirement on verification of safety against derailment under longitudinal compressive forces in S-curves (according to EN 15839) is mentioned. The problem of longitudinal forces in freight trains is also discussed in some papers – see, e.g. Ref. [3,4]. However, the question of track loading in conditions of S-curves is solved in the standard [1] only marginally, in the informative Annex F. It means that the standard EN 14363 does not specify any requirements for the assessment of vehicle behaviour in switches and crossings nowadays. This situation does not correspond to the fact that the S-curve with a radius of 190 m is represented by a common (and widely spread) single crossover negotiated at a speed of 40 km/h as well as to the fact that an important part of infrastructure maintenance costs is usually spent for the maintenance of switches, damaged by means of the force effects of running vehicles. This is one of the reasons why some infrastructure managers (typically SBB [5]) want to reflect the damaging effects of running vehicles on switches in calculations of track access charges, although these effects are not investigated in the framework of vehicle testing according to EN 14363.

Generally, the problem of damaging effects of running railway vehicle in switches (and crossings) can be divided into two groups – vertical dynamic effects related especially with the unsprung mass in the running gear in combination with vehicle speed and lateral force effects related with run of the vehicle through a switch in the diverging direction. This paper deals with the problem of lateral force effects.

2. Lateral force interaction between vehicle and track in S-curve

A typical S-curve is defined by two reverse small radius curves without transitions and cant and an intermediate tangent track section. On the top of Figure 1, a nominal alignment of the track axis in the S-curve according to the Annex F of EN 14363 [1] is shown. This case corresponds to a single crossover designed for the speed of 40 km/h. The combination of this speed value and the non-superelevated curve with a radius of 190 m leads to a cant deficiency value of 100 mm or an unbalanced lateral acceleration of 0.65 m s^{-2}.

An example of the characteristic time history of guiding forces acting on wheels of the first wheelset of a four-axle railway vehicle equipped with two bogies is presented in Figure 1. These data were gained as results of a multi-body simulation of a four-axle modern electric locomotive with a total weight of 90 tons negotiating the S-curve according to the scheme presented on the top of Figure 1 (curve radius: 190 m, length of the intermediate tangent track section: 6 m) at the speed of 40 km/h. The multi-body simulation tool ‘SJKV’, which is being developed at the Faculty of Transport Engineering of the University of Pardubice since 1990s, was used for the calculations; an ideal track without irregularities and dry rails was considered. From the point of view of the lateral force effects of the vehicle on the track in such track conditions, the highest intensity of the guiding force occurs on the outer wheel of the first wheelset in both curves. Especially, the peaks of the guiding force occurring immediately after the entrance of the wheelset into the curves can achieve

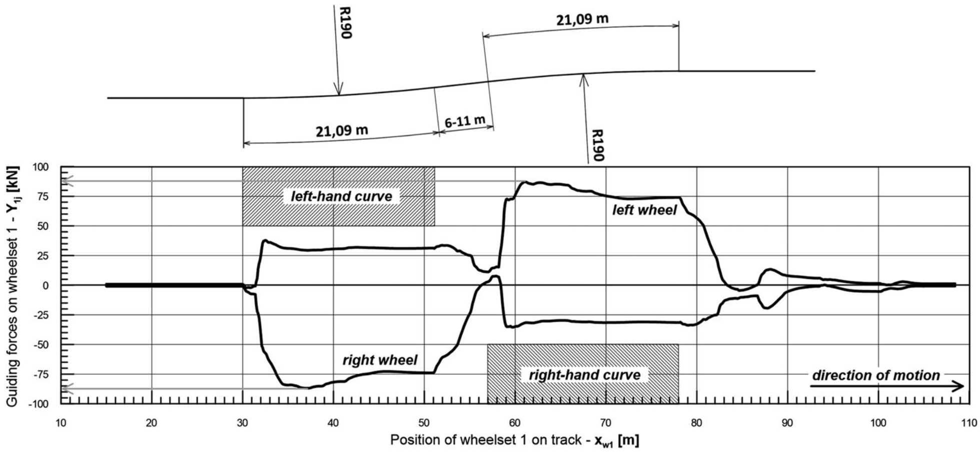

Figure 1. Nominal track alignment of a reference S-curve defined in the Annex F of EN 14363 [1] (on the top) and guiding forces on wheels of the first wheelset of a four-axle locomotive model running through this S-curve with a 6 m long intermediate tangent track section (multi-body simulation results).

very high values. In a real situation (i.e. a vehicle runs through a switch in the diverging direction), the peak of the guiding force represents an intensive load of a switch rail. In operational conditions, the running vehicles induce fatigue loading of the switch rail, which can lead to a fracture of this constructional part of the switch in a limit case.

It is evident that the achieved peak values of the guiding force are influenced by many parameters. Besides the vehicle speed, track parameters (nominal track alignment in the S-curve, track gauge, etc.) and environmental conditions (friction coefficient), vehicle design parameters can be very significant. Therefore, the influence of selected vehicle parameters on the achieved maximum value of the guiding force of the vehicle running through an S-curve was investigated by means of multi-body simulations under defined conditions.

3. Influence of vehicle parameters on the maximum guiding force in S-curve

To quantify the influence of the selected parameters of a four-axle vehicle on the maximum value of guiding force in S-curve, the reference conditions for multi-body simulations were determined. Therefore, the following conditions were considered:

- nominal track alignment according to Figure 1 ($R = 190$ m, 6 m of tangent track);
- ideal track without irregularities;
- theoretical rail profile 60E1/1:40, track gauge: 1435 mm;
- theoretical wheel profile S1002/e32.5, wheelset gauge: 1425 mm;
- friction coefficient in wheel/rail contact: 0.4;
- vehicle speed: 40 km/h.

Relevant simulations were performed using a multi-body model of a four-axle modern electric locomotive with a total weight of 90 tons implemented into the simulation tool ‘SJKV’. The influence of the following parameters was observed:

- characteristics of the rotational joint between the vehicle body and bogie to the vertical axis (bogie yaw stiffness and characteristics of bogie yaw dampers);
- stiffness of wheelset guiding in longitudinal and lateral directions;
- dimensional parameters (bogie wheelbase and distance between bogie pivots);
- mass inertia moment of vehicle body to the vertical axis.

The used vehicle model consists of four wheelsets, two bogie frames (including fixed, fully suspended traction motors) and a vehicle body. All the bodies are considered as rigid, coupled with flexible and damping joints with relevant – in a general case non-linear – characteristics, representing the relevant suspension elements (flexi-coil springs, hydraulic dampers, longitudinal guiding rods, bump stops, etc.). Considered values of the basic model parameters are evident from the following subsections. The model also allows simulation of a system of active yaw dampers or an active radial wheelset steering system (ARWS) (see Section 5). In case of the active radial wheelset steering simulation, the vehicle model is supplemented with additional four bodies, representing the individual active elements (replacement of the 'passive' longitudinal guiding rods – one piece per each wheelset). From the point of view of wheel/rail contact modelling, the system 'SJKV' is using a rigid, one-point contact model (described by pre-prepared tables of wheel/rail contact geometry characteristics) and the adhesion model by prof. Polách [6]. Further information about the system 'SJKV' can be found, e.g. in the documentation of the basic academic version 'SJKV-V4N' [7].

3.1. Characteristics of the rotational joint vehicle body/bogie

In case of conventional railway vehicles equipped with two bogies, the rotational joint between the vehicle body and individual bogies can be described with two characteristics:

- bogie yaw stiffness leading to a reverse moment at the bogie yaw motion;
- passive moment against bogie rotation.

A non-zero bogie yaw stiffness value is typical for bolsterless bogies. If the bogie yaw angle has a non-zero value, a reverse moment between the vehicle body and bogie acts around the vertical axis as a consequence of deformation of secondary suspension elements (flexi-coil springs, air springs, etc.) and corresponding forces in these elements. The reverse moment M_{rev} can be expressed as a function of the bogie yaw angle:

$$M_{\mathrm{rev}} = \gamma \cdot \psi, \tag{1}$$

where γ (N m/rad) is the value of bogie yaw stiffness (bogie yaw resistance) and ψ (rad) is the bogie yaw angle. The passive moment against bogie rotation M_{pas} represents the second part of a total moment acting in the rotational joint vehicle body/bogie around the vertical axis. This passive moment occurs only if the bogie is rotating, i.e. the velocity of the bogie yaw motion $\dot{\psi}$ has a non-zero value, and its existence follows especially from the forces acting in the bogie yaw dampers, side bearers, centre plate or bowl, etc. The total moment M_z can be expressed as a sum of the above-mentioned two parts:

$$M_z = M_{\mathrm{rev}}(\psi) + M_{\mathrm{pas}}\left(\dot{\psi}\right). \tag{2}$$

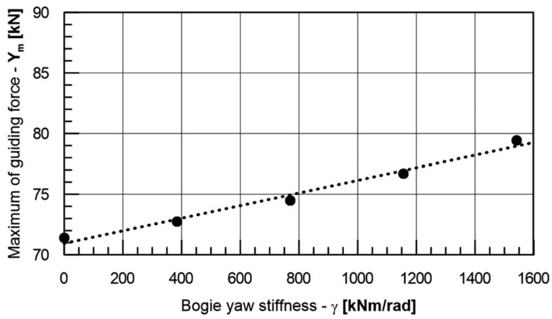

Figure 2. Maximum value of guiding force of the four-axle locomotive model in the S-curve in dependency on the bogie yaw stiffness value.

This total moment is usually assessed in the framework of the stationary tests according to EN 14363 [1] by means of so-called X-factor (see also Ref. [8]), as well.

To quantify the influence of both components of the total moment M_z on the maximum value of guiding force Y_m acting on wheels of the first wheelset of the investigated locomotive model in the S-curve, two sets of simulations were performed. At first, the bogie yaw stiffness value was changed in few steps in the range of 0–1540 kN m/rad using the locomotive model without yaw dampers. Relevant simulation results are presented by points in the graph in Figure 2; the dotted line represents a linear approximation of these results.

Subsequently, the influence of passive moment against bogie rotation was observed. For these purposes, simulations with a locomotive model equipped with four yaw dampers (i.e. two yaw dampers per bogie – see the scheme in Figure 4) were performed. The model with a nominal value of the bogie yaw stiffness of 1150 kN m/rad (representing the real vehicle used for measurements discussed in Section 5) was used and the original (nominal) characteristics of the yaw dampers were reduced in three steps. To characterise the (nonlinear) characteristics of bogie yaw dampers, these are expressed by means of a reference value of the passive moment. The methodology for estimation of this reference value is described in Section 4 in detail. The resulting dependency of the maximum guiding force on the passive moment value is shown in Figure 3 (the points represent simulation results; the dotted line is a linear approximation).

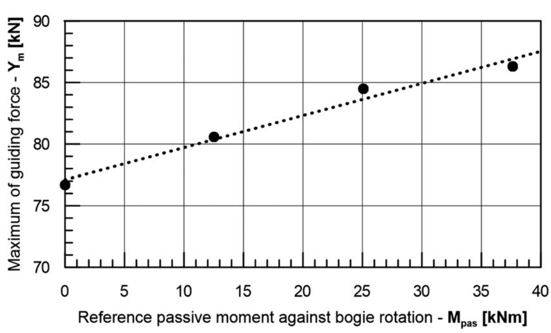

Figure 3. Maximum value of guiding force of the model in the S-curve in dependency on the reference value of the passive moment against bogie rotation.

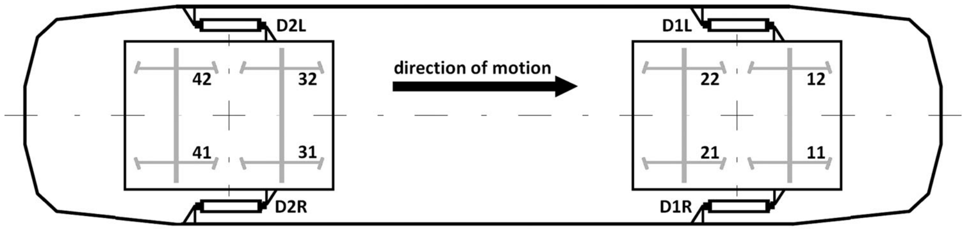

Figure 4. Scheme of the locomotive and identification of wheels and yaw dampers.

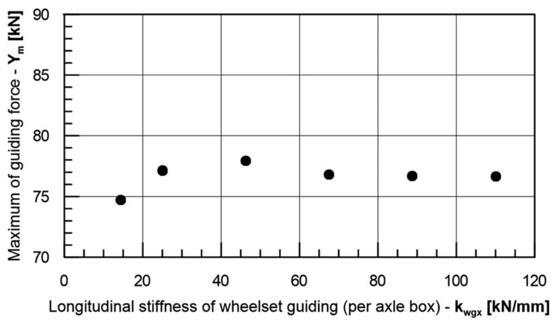

Figure 5. Maximum value of guiding force of the four-axle locomotive model in the S-curve in dependency on the longitudinal stiffness of wheelset guiding (for the lateral stiffness value of ca. 4 kN/mm).

3.2. Stiffness of wheelset guiding

In the next stage, the influence of stiffness of wheelset guiding in longitudinal as well as lateral direction was investigated. Relevant simulations were realised with the locomotive model with a nominal value of the bogie yaw stiffness of 1150 kN m/rad and without bogie yaw dampers. In the case of the longitudinal wheelset guiding stiffness, the value of this parameter was changed in the range of 14 up to 110 kN/mm (this value is related to one axle box). The lateral wheelset guiding stiffness ranged from 1.0 up to 5.8 kN/mm per one axle box. The simulation results are presented in the form of the achieved maximum values of the guiding force in Figure 5 (the influence of longitudinal stiffness) and Figure 6 (the influence of lateral stiffness).

In both observed cases, the simulation results do not show any significant trends. An exception is represented by a very low longitudinal stiffness of the wheelset guiding leading to lower peak values of the guiding force in the S-curve. However, a practical application of the very low stiffness of wheelset guiding is questionable, especially in the case of locomotive running gears with higher requirements on traction force transmission and running stability.

3.3. Basic dimensional parameters of vehicle

In the field of basic dimensional parameters of the investigated railway vehicle model, attention was paid to the bogie wheelbase and the distance between bogie pivots. Both parameters were changed in the range of 90 % up to 120 % of their nominal values; these

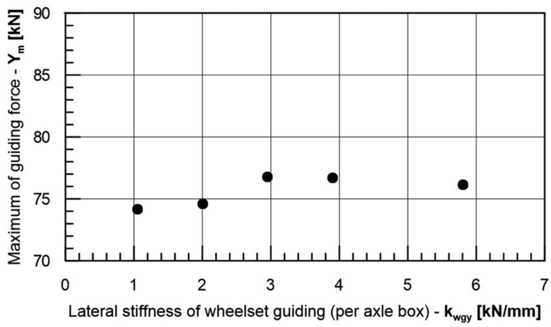

Figure 6. Maximum value of guiding force of the four-axle locomotive model in the S-curve in dependency on the lateral stiffness of wheelset guiding (for the longitudinal stiffness value of ca. 90 kN/mm).

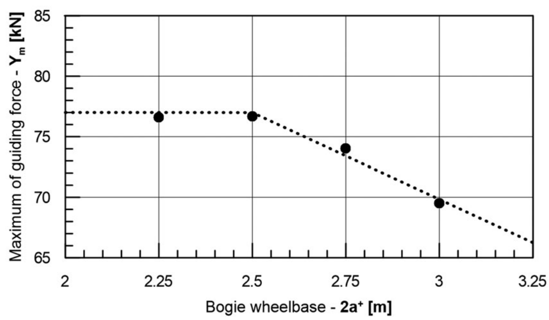

Figure 7. Maximum value of guiding force of the four-axle locomotive model in the S-curve in dependency on the bogie wheelbase (for the distance between bogie pivots of 8.7 m).

nominal values are following: 2.50 m (bogie wheelbase) and 8.70 m (distance between bogie pivots). For the relevant simulations, the locomotive model with the bogie yaw stiffness of 1150 kN m/rad and without yaw dampers was used again. Simulation results showing the influence of the bogie wheelbase are presented in the graph in Figure 7. It can be stated that the shorter bogie wheelbase leads to higher peaks of the guiding force in the S-curve. However, the increase of the maximum guiding force is not observed for very short bogies. Therefore, the approximation (dotted line) of the simulation results (points) is considered as folded. In Figure 8, the influence of the distance between bogie pivots is demonstrated. The observed dependency has a convex shape; a minimum value can be found in the investigated range. The reason for that can be found in a combination of the following two opposing factors:

- a longer distance between bogie pivots leads to a higher value of the bogie yaw angle of the vehicle running through a curve. Therefore, the vehicles with a non-zero bogie yaw stiffness show a higher value of the reverse moment (see Eq. (1)) leading to higher peaks of the guiding force (see Section 3.1);
- a longer distance between bogie pivots leads to a lower value of a moment of inertia forces, which has to be counterbalanced by the lateral forces between the vehicle body and individual bogies at the entrance of the vehicle into the curve. This principle is described in more detail in Section 4.

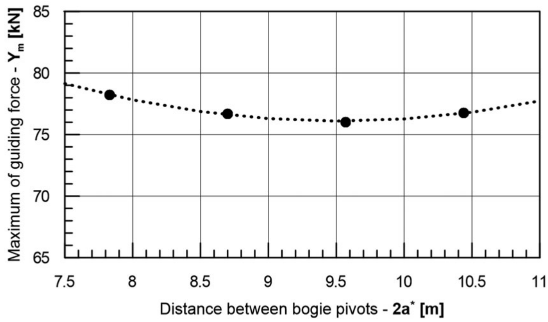

Figure 8. Maximum value of guiding force of the four-axle locomotive model in the S-curve in dependency on the distance between bogie pivots (for the bogie wheelbase of 2.5 m).

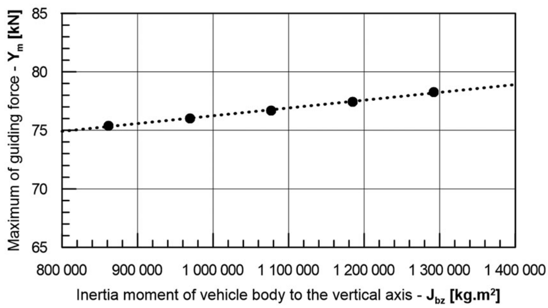

Figure 9. Maximum value of guiding force of the four-axle locomotive model in the S-curve in dependency on the inertia moment of the vehicle body to the vertical axis (for the distance between bogie pivots of 8.7 m and the bogie yaw stiffness of 1150 kN/m).

3.4. Mass inertia moment of vehicle body to the vertical axis

In the last stage, the influence of the mass inertia moment of the vehicle body to the vertical axis was investigated. The locomotive model without yaw dampers and with nominal values of the other parameters was used again. Relevant simulation results as well as their linear approximation are presented in Figure 9. The observed positive trend is related to the moment of inertia forces acting at the entrance of the vehicle into the curve, similarly to the second aspect of the effect of the distance between bogie pivots (see Sections 3.3 and 4).

4. Introduction of equivalent guiding force in a switch

The performed analysis (see Section 3) shows that some parameters of railway vehicles can influence the maximum value of guiding force in the S-curve significantly. Because this value characterises the damaging effects of the vehicle on a switch, it seems to be a suitable reference parameter which can be used as an input for the calculation of wear-dependent track access charges. This approach is used, for example, in the Swiss track access charge system and the relevant maximum value of the guiding force in S-curve is determined by means of a multi-body simulation under defined conditions [5]. With

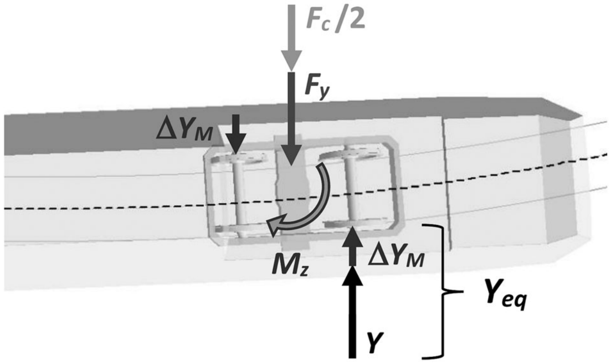

Figure 10. A simplified scheme of acting forces for derivation of the equivalent guiding force definition.

respect to the observed effects of the selected vehicle parameters, the simplification based on a definition of so-called equivalent guiding force is proposed. The equivalent guiding force Y_{eq} expresses the maximum value of the guiding force acting on wheels of the first wheelset of a four-axle vehicle equipped with two bogies negotiating a defined tight S-curve. The simplification of this proposal is based on the fact that it is not necessary to perform computer simulations. For purposes of estimation of the equivalent guiding force, a knowledge of basic vehicle parameters is only needed.

The equivalent guiding force definition is based on the idea that the lateral force acting on the outer wheel of the first wheelset has to equilibrate:

- a part of a centrifugal force F_c acting on the vehicle running through the curve;
- a lateral force F_y occurring at the entrance of vehicle into the curve as an effect of its inertia moment to the vertical axis;
- a moment against bogie rotation M_z (represented by the couple of forces ΔY_M).

This idea is presented in a simplified scheme of the forces acting on the front bogie of relevant vehicle in Figure 10. Therefore, the equivalent guiding force can be defined as

$$Y_{\mathrm{eq}} = \frac{F_c}{2} + F_y + \Delta Y_M, \tag{3}$$

where the meaning of the individual components of this equation is explained in the next.

The first component F_c represents the inertia (centrifugal) force occurring at the run of the vehicle on a circular trajectory. Because a non-superelevated (S-)curve is considered, the centrifugal force F_c is directly equal to the unbalanced lateral force acting on the vehicle, i.e.

$$F_c = m \cdot \frac{v^2}{R}, \tag{4}$$

where m(kg) is the total mass of the vehicle, v(m/s) is the vehicle speed and R(m) is the curve radius. The proposed definition of the equivalent guiding force in a switch assumes an even distribution of the total unbalanced lateral (centrifugal) force between both bogies, i.e. a half of the force F_c is considered on the front bogie.

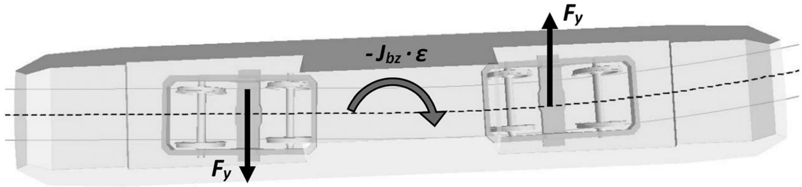

Figure 11. Moment of inertia forces acting on vehicle body at the entrance into the (S-)curve.

The second component of the equivalent guiding force is represented by the lateral force F_y acting between the vehicle body and relevant bogies. This inertia force occurs at the entrance of the vehicle into a curve because the vehicle body has to start rotating around its vertical axis while it is negotiating from the tangent track to the circular curve without transitions. This effect is depicted in Figure 11 and related to the inertia moment of the vehicle body to the vertical axis. A relevant equation of motion can be expressed as

$$J_{bz} \cdot \varepsilon = F_y \cdot 2a^*, \tag{5}$$

where J_{bz}(kg m^2) is the inertia moment of vehicle body to the vertical axis, ε(rad/s^2) is the angular acceleration of the vehicle body around the vertical axis and $2a^*$(m) is the distance between bogie pivots or bogie centres.

To express the lateral force F_y, the angular acceleration ε of the vehicle body at the entrance into the curve can be specified in a following way. At the run on a circular trajectory (i.e. in the curve), the angular velocity ω of the vehicle body around the vertical axis can be expressed by means of the vehicle speed v(m/s) and the curve radius R(m):

$$\omega = \frac{v}{R}. \tag{6}$$

If a linear increase of the angular velocity is considered at the entrance of the vehicle into the curve, this value of the angular velocity has to be achieved during the time Δt which corresponds to the distance between bogie pivots $2a^*$(m) negotiated by the vehicle at the speed v(m/s):

$$\Delta t = \frac{2a^*}{v}. \tag{7}$$

Finally, the value of the lateral inertia force F_y acting between the vehicle body and the bogie at the entrance of the vehicle into the curve can be expressed as

$$F_y = \frac{J_{bz} \cdot v^2}{(2a^*)^2 \cdot R}. \tag{8}$$

The third component of the equivalent guiding force – expressed by the force ΔY_M – represents the effects of the moment against bogie rotation and therefore it depends on the design of the joint between the bogie and the vehicle body. Generally, the relevant force component can be expressed as

$$\Delta Y_M = \frac{M_z}{2a^+}, \tag{9}$$

where M_z(N m) is the total moment acting in the rotational joint vehicle body/bogie around the vertical axis and $2a^+$(m) is the bogie wheelbase. In coherence with Eq. (2), the moment against bogie rotation M_z can be divided into two parts – the reverse moment M_{rev} and the passive moment M_{pas}. The reverse component of the total moment against bogie rotation is defined by means of Eq. (1), using the bogie yaw stiffness γ. For purposes of the equivalent guiding force definition, the reference value of the bogie yaw angle ψ can be expressed on the theoretical basis of the geometry of a vehicle in a curve, i.e.

$$\psi = \frac{2a^*}{2 \cdot R}, \tag{10}$$

where $2a^*$(m) is the distance between bogie pivots (centres) and R(m) is the curve radius; an influence of a gauge clearance is neglected for these purposes. The passive moment M_{pas} against bogie rotation can be expressed in a general form as

$$M_{\text{pas}} = \sum_{(i)} r_i \cdot F_i, \tag{11}$$

where F_i(N) represents forces acting against the bogie rotation (i.e. damping forces in yaw dampers, frictional forces in side bearers, eventually damping forces in lateral secondary dampers, etc.) and r_i(m) is a relevant arm length of the force F_i to the bogie centre. In case of some types of the passive moments (e.g. frictional moments in centre plates or bowls), these effects can be characterised by means of an effective value of the arm r_i. Because the individual forces F_i (as well as the total passive moment M_{pas}) are usually dependent on the velocity of bogie yaw motion $\dot{\psi}$, it is necessary to determine their reference values. The idea for determination of the reference velocity of bogie yaw motion $\dot{\psi}$ is based on the fact that the nominal alignment of the considered S-curve does not include any transitions. It means that the reference value of the bogie yaw angle ψ (see Eq. (10)) has to be achieved during the time Δt corresponding to the entrance of the vehicle into the curve, i.e. the distance between bogie pivots $2a^*$ negotiated by the vehicle at the speed v (see Eq. (7)). Therefore, the reference value of the velocity of bogie yaw motion $\dot{\psi}$ can be defined as

$$\dot{\psi} = \frac{v}{2 \cdot R}. \tag{12}$$

Then, in case that the force F_i depends on a velocity of deformation v_i of the relevant suspension element (i.e. especially in case of the hydraulic dampers), the reference value of the velocity of bogie yaw motion $\dot{\psi}$ has to be used for estimation of the reference velocity of deformation v_i; i.e.

$$v_i = \dot{\psi} \cdot r_i, \tag{13}$$

where $\dot{\psi}$(rad/s) is the reference value of the velocity of bogie yaw motion (see Eq. (12)) and r_i(m) is the relevant arm length of the force F_i (acting in the relevant suspension element) to the bogie centre.

With using the above-mentioned equations, the equivalent guiding force (see Eq. (3)) can be modified into the following general form:

$$Y_{\text{eq}} = \frac{v^2}{R} \cdot \left(\frac{m}{2} + \frac{J_{bz}}{(2a^*)^2} \right) + \frac{1}{2a^+} \cdot \left(\gamma \cdot \frac{2a^*}{2 \cdot R} + \sum_{(i)} r_i \cdot F_i \right). \tag{14}$$

If this parameter should be used for the quantification of damaging effects of a railway vehicle on the track, a practical application would be complicated, especially by these facts:

- in case of freight wagons, the inertia moment of the vehicle body to the vertical axis is not known because of its dependency on the loading;
- in case of frictional joints, the value of frictional moment against bogie rotation can be only roughly estimated.

The problem of unknown value of the inertia moment J_{bz} of the vehicle body can be solved in a simplified way, approximating the vehicle body by a prismatic beam. Then, the relevant inertia moment can be estimated by means of basic vehicle parameters as

$$J_{bz} \approx \frac{1}{12} \cdot m \cdot l^2, \tag{15}$$

where m(kg) is the total vehicle mass and l(m) is the length of the vehicle over the buffers or couplers. In case that this simplification as well as the nominal parameters of run through the S-curve/switch according to the Annex F of EN 14363 [1] (i.e. curve radius: $R = 190$ m, vehicle speed: $V = 40$ km/h) is considered for purposes of calculation of a nominal value of the equivalent guiding force, Eq. (14) can be modified as

$$Y_{\text{eq}} = 0.65 \cdot m \cdot \left(\frac{1}{2} + \frac{1}{12} \cdot \left(\frac{l}{2a^*} \right)^2 \right) + \frac{\gamma}{380} \cdot \frac{2a^*}{2a^+} + \frac{1}{2a^+} \cdot \sum_{(i)} r_i \cdot F_i. \tag{16}$$

The problem of frictional rotational joints can be solved by means of a categorisation of vehicles (in case of freight wagons, for example, in dependency on the actual axle load and type of bogies) and a definition of a reference value of the passive moment against bogie rotation M_{pas} for the individual categories.

5. Effects of advanced technologies and their limitations

In relationship to the lateral force effects of railway vehicles in the S-curves, the question of application of some advanced technologies in running gears (e.g. mechanical or hydraulic bogie couplings [9], active yaw dampers [10,11], systems for radial wheelset steering [11,12]) should be mentioned. Generally, these systems are designed to reduce the lateral force effects between the vehicle and the track in (small radius) curves and related wear of wheels and rails. The results (presented in Refs. [12,13]) show that the reduction of guiding forces by means of active elements can be very significant in small radius curves. However, the situation in S-curves is specific, different from the 'standard curves'. Whereas the curves are negotiated by the vehicles in a stabilised position (e.g. from the point of view of the bogie yaw angle) and the entrance into the curve is usually smooth (because of the transitions), the S-curves (and the diverging parts of switches) require a quick turning of the bogies into the very small curve radius. This is the reason why the passive moments against bogie rotation (see Section 3.1) as well as the inertia effects of the vehicle body (see Section 3.4) increase the maximum value of guiding force in the S-curve, although these parameters do not influence the lateral force interaction between the vehicle and the track in 'standard curves' significantly.

In relationship to the advanced technologies for radial wheelset or bogie steering, the above-mentioned fact leads to the basic requirements for these systems: a sufficiently quick

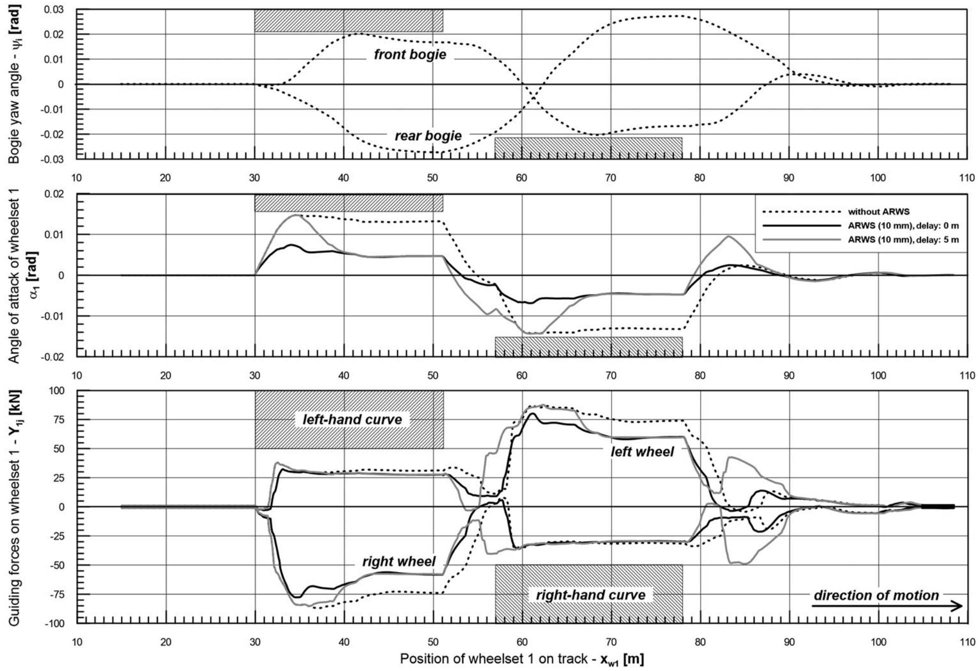

Figure 12. Influence of the ARWS and its inaccuracy (delay) on the guiding forces acting on wheels of the first wheelset (bottom graph) and on the angle of attack of the first wheelset (middle graph) in the S-curve (2× R190, 6 m tangent track, 40 km/h) and corresponding behaviour of bogie yaw angle of both bogies of the locomotive without ARWS (upper graph) – multi-body simulation results. *Note*: The position of the curves corresponds to the *x*-coordinate of 30–51 m and 57–78 m.

and accurate reaction. In case that these basic requirements are not fulfilled, it is practically impossible to achieve an effective reduction of the lateral force effects in the S-curve. The required promptitude of reaction of the system is related to the position of peak of the guiding force. As it is shown in Figure 1 for the reference case (the S-curve according to Ref. [1]), the peaks of the guiding force on the leading wheel are observed only a few metres behind the entrance of the first wheelset into the curve. Therefore, the relevant active steering systems should prove the reaction time ranging in tenths of seconds.

Besides to the reaction time, the accuracy of the system has to be solved. It means that it is necessary to know the position of the vehicle on the track with a high precision because the peak of the guiding force on the leading wheel occurs at a very short distance (in the order of metres). Simulation results for a multi-body model of the four-axle electric locomotive equipped with an ARWS negotiating the S-curve according to the Annex F of EN 14363 [1] with a 6 m long intermediate tangent track at the speed of 40 km/h are presented in Figure 12 in order to demonstrate the influence of reaction inaccuracy. For these purposes, the vehicle model created in the system 'SJKV' and described in Sections 2 and 3 was used, again. The ARWS system is considered so that the standard longitudinal rods ensuring guiding of the axle boxes of the wheels 12, 21, 32 and 41 (see the scheme in Figure 4) are replaced by active elements (actuators). These actuators allow a change of their length by ±10 mm at the velocity of 35 mm/s; the maximum force exerted by the actuator is ±76 kN.

If the applied control algorithm evaluates it as desirable, the individual actuators start to steer the wheelsets in individual bogies into the radial position (if possible). In this case, three versions of the model are considered:

- a locomotive without the ARWS system – see the dotted lines in Figure 12;
- a locomotive equipped with the ARWS system which uses a theoretical control strategy assuming that the position of the vehicle on the track is exactly known; it means that the actuators on each bogie start acting exactly at the time of entrance of the front wheelset of the relevant bogie into the curve – see the black lines in Figure 12;
- a locomotive equipped with the same ARWS system which shows a delayed reaction (a start of action of the actuators is delayed by 0.45 s, i.e. 5 m behind the theoretically required position) – see the grey lines in Figure 12.

For the demonstration of the influence of reaction inaccuracy, only the middle and the bottom graphs are important. It is evident that the accurate function of the active system with the considered parameters (stroke of actuators up to 10 mm) can reduce the angle of attack of the first wheelset by more than 50%. A significant reduction of the guiding force acting on the leading wheel of the first wheelset in the curves is also observed. However, in the case of the slightly delayed reaction of the active system, the maximum (peak) values of the guiding force as well as the angle of attack reach immediately after the entrance of the vehicle into the curve the same peak values as in case of the locomotive without the ARWS. This problem is also analysed in Ref. [14], where a more detailed description of the ARWS system model can be found and an assessment of influence of an active radial bogie steering system (a system of active yaw dampers) is discussed.

A specific problem of the wheelset/bogie steering technologies (ARWS systems, bogie couplings) in the S-curves is related to the control of these systems, especially in case that the intermediate tangent track section is shorter than the distance between bogie pivots. For example, in the case of coupling between bogies [9], the reaction of the system depends on the difference between yaw angles between both bogies; in the case of the system of active yaw dampers [10], its function is controlled on the basis of a signal describing the actual bogie yaw angles (using integrated deformation sensors in the longitudinal yaw dampers). However, these control strategies using on-board signals (characterising the bogie yaw angles) can have a serious problem with a recognition of the reverse (second) curve of the S-curve with a short intermediate tangent section as well as they are usually not able to detect the entrance of the vehicle into the curve punctually. The first problem is demonstrated in Figure 13 as well as in the upper graph in Figure 12 (using the multi-body simulation results). If the distance between bogie pivots is longer than the length of the intermediate tangent track, the yaw angle of the front bogie (relatively to the vehicle body) at the time of its entrance into the reverse curve corresponds to the orientation of the bogie yaw angle in the first curve. Therefore, the bogie yaw angle is opposite than desired, and it is practically impossible to detect the entrance of the vehicle into the reverse curve only by means of the signals of the bogie yaw angle soon enough. If the function of the bogie or wheelset steering systems was controlled only on the basis of these signals, their reaction would be delayed or it could be even contrary. The second problem (a punctual recognition of the vehicle entrance into the first curve) is related to the typical time behaviour of the bogie yaw angles. As it is evident from the upper graph in Figure 12, a significant increase

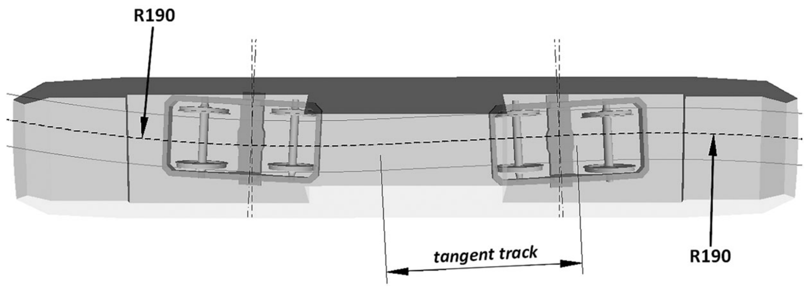

Figure 13. Position of a vehicle in the S-curve with intermediate short tangent track section.

of the yaw angle of the front bogie is shifted by a few metres behind the entrance of the first wheelset into the first curve. And as it was mentioned above, these few metres are crucial from the point of view of the guiding force peaks.

To verify the above-mentioned conclusions following from the results of multi-body simulations, the behaviour of a real railway vehicle in the S-curve was also observed through measurements in operation. For these purposes, the results of measurements performed with a modern four-axle electric locomotive in conditions of the Czech railway network were used (the relevant measurements were realised in the framework of another R&D activities of the Faculty of Transport Engineering of the University of Pardubice, focussed especially on the vertical dynamic effects of the vehicle on switches – see, e.g. Ref. [15]).

The measurement results are presented in Figure 14. For the purposes of the assessment of vehicle behaviour in the S-curve, a single crossover consisting of a pair of switches 'J60-1:12-500' (turnout A and turnout B) with a 15 m long intermediate tangent section (see the track layout scheme on the top of Figure 14) was chosen. In the observed case, the switches were negotiated at the speed of 50 km/h and the locomotive (see the scheme in Figure 4) was equipped with accelerometers on all axle boxes (measurement of the vertical acceleration near wheels 11 and 12 was used for the evaluation) and instrumented yaw dampers on the rear bogie (measurement of the damper deformation of the dampers D2R and D2L).

To compare the simulation results with the experimental (measured) data, a synchronisation of the relevant signals had to be done at first. For this purpose, the signals of the vertical acceleration on the first wheelset were used. In these signals, a transition of wheels 11 and 12 over both turnout frogs in the investigated track section is detectable (see graphs 1 and 2 in Figure 14). Subsequently, it is possible to compare the time behaviour of damper deformation of the instrumented yaw dampers (D2L and D2R) with relevant simulation results. The comparison is performed in graphs 3 and 4 in Figure 14. It is evident that the measurement and the multi-body simulation provide similar results from the point of view of their shape as well as the reached values. A smoother character of the simulation results is related to the fact that the simulation was performed on the track without irregularities. Besides to the yaw damper deformation, the difference between the deformation of the left and the right yaw damper was also evaluated. The yaw damper deformation difference Δd is just the quantity which can be used for the on-board detection of curve radius. The results presented in graph 5 in Figure 14 show a good correspondence between the measurement and the simulation again.

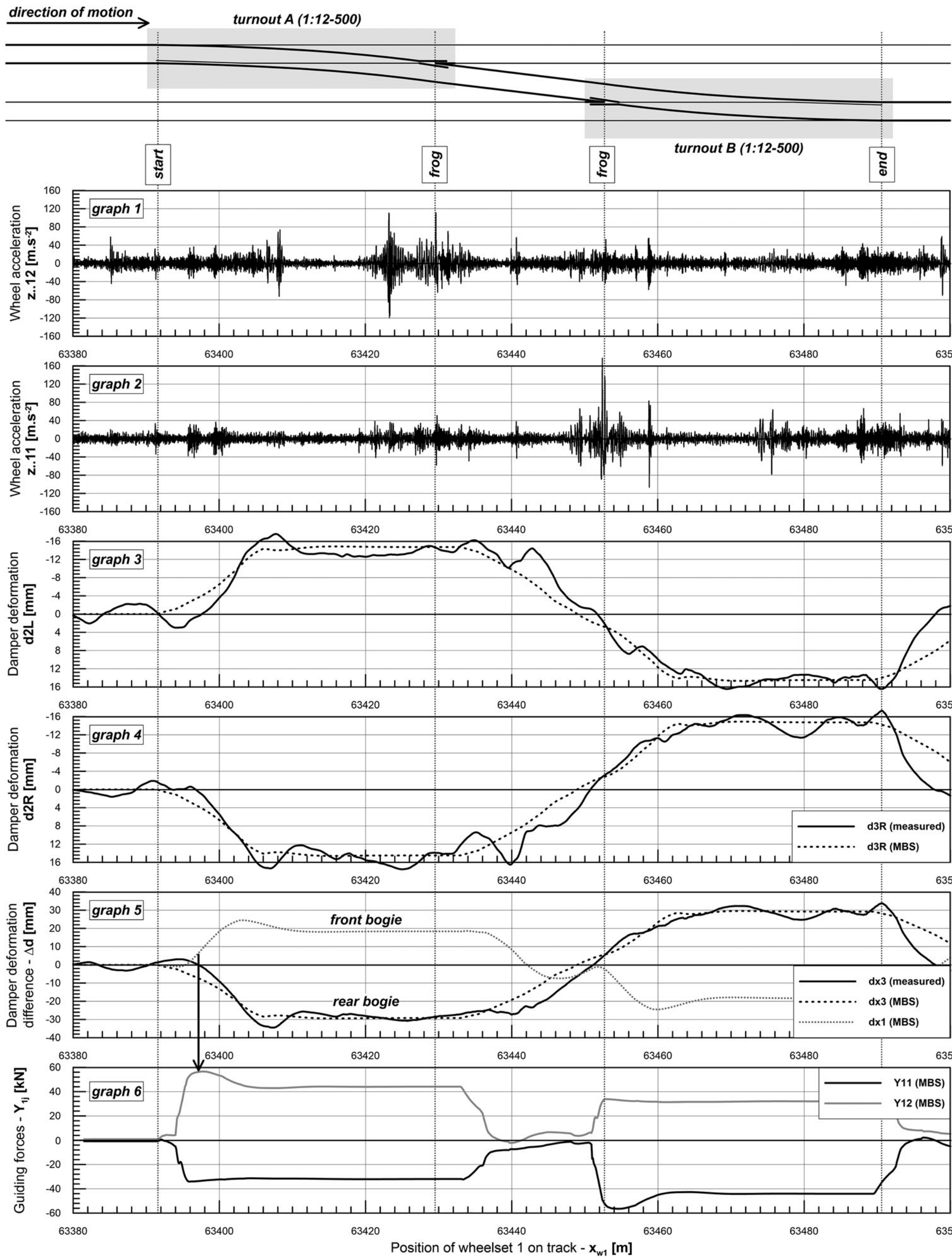

Figure 14. Track layout in the investigated track section (scheme on the top); signals of vertical acceleration measured on the axle boxes of the left (graph 1) and the right (graph 2) wheel of the first wheelset of the locomotive; a comparison of measured and simulated deformation of the left (graph 3) and the right (graph 4) yaw damper on the rear bogie; a difference between yaw damper deformation on the left and right side of the rear bogie and on the front bogie (graph 5) and corresponding simulated guiding forces acting on both wheels of the first wheelset of the multi-body locomotive model (graph 6).

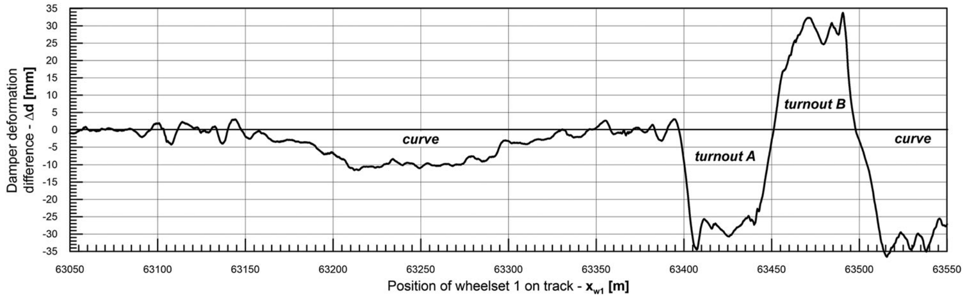

Figure 15. Example of the yaw damper deformation difference signal at a run of the investigated locomotive; results of measurement on the rear bogie.

If the yaw damper deformation difference signal is verified, the connection between the yaw damper deformation difference and the lateral force effects of the vehicle on the track in the conditions of the S-curve can be investigated. Therefore, the simulated signal of the yaw damper deformation difference of the front bogie and the relevant simulated guiding forces on the wheels of the first wheelset (i.e. wheels 11 and 12) is presented in graphs 5 and 6 in Figure 14. These results confirm the basic problem of the on-board detection of the vehicle entrance into the S-curve and show it very illustratively: the maximum value of the guiding force on the outer wheel of the first wheelset (i.e. wheel 12 in turnout A and wheel 11 in turnout B) occurs at the same time when the relevant yaw damper deformation difference (and the bogie yaw angle) still has a very small value.

For the purposes of the on-board detection of the curve radius with using the evaluation of yaw damper deformation difference, a limit value of this quantity should be defined for application in control algorithms of active radial steering systems. This limit value must ensure that the system will not react on a random kinematic excitation of the bogie due to the track irregularities or in curves with large radii. In Figure 15, an example of the measured signal of yaw damper deformation difference on the rear bogie is presented. In comparison with the run on a 'regular track' (besides the S-curve, the depicted track section includes a tangent track and a curve with a larger radius), it is evident that the absolute reached values of the yaw damper deformation difference are significantly higher just in the S-curve. However, the results presented in graphs 5 and 6 in Figure 14 lead to a conclusion that the punctual reaction of the active radial steering system is preconditioned by a limit value of the damper deformation difference only in the order of millimetres (in the context of dimensional parameters of the investigated locomotive). And as it can be seen in Figure 15, this range of values can be commonly reached at the run of the vehicle in curves with large radii as well as in the tangent track with ordinary track irregularities.

6. Conclusions

The S-curves and diverging parts of the switches represent a specific track alignment. In the framework of running tests according to EN 14363 [1], railway vehicles are not commonly subjected to investigation of their damaging effects on the track in these conditions. However, the infrastructure costs related to the maintenance of switches are usually very significant. For example, in conditions of Swiss railway infrastructure, the switches represent 48 % of damage-related maintenance costs (see Ref. [5]). Therefore, it is important to pay attention to the lateral force effects of railway vehicles in the S-curves.

An introduction of wear-dependent track access charges seems to be an effective way to support the development and operation of so-called track-friendly vehicles. For the purposes of quantification of different categories of damaging effects in the framework of the track access charge systems, suitable parameters have to be defined. However, this vehicle pricing system should be as simple as possible. Generally, it is undesirable to burden railway operators with further administration and costs, harming the competitiveness of the railway system.

To quantify the lateral force effects of a conventional four-axle railway vehicle negotiating a reference S-shape curve, a definition of so-called equivalent guiding force is proposed by the authors and presented in this paper. This definition (see Section 4) is based on the basic mechanical principles, uses only the basic parameters of vehicles, and reflects the most important results of multi-body simulations performed in the framework of sensitivity analysis of the influence of selected vehicle parameters on the lateral force effects of the vehicle in the S-curve. Therefore, the equivalent guiding force can be applied to the relevant wear-dependent track access charge systems for quantification of the lateral force effects of vehicles in switches.

Besides to the analysis of lateral force effects of conventional vehicles in the S-curves, attention is paid to the 'advanced technologies' in vehicle running gears (e.g. systems for radial wheelset steering, active yaw dampers, couplings between bogies) in these track conditions. Whereas these systems can bring a very significant improvement of the guiding behaviour of relevant vehicles in (small radius) curves; in Section 5, it is demonstrated that the effectiveness of these systems in conditions of tight S-curves can be significantly limited. One of the most problematic factors is a sufficiently accurate as well as punctual reaction of these systems on the vehicle entrance into the curve, especially in the case of the on-board detection of the curve based on (indirect) measurement of the bogie yaw angles. This problem is analysed on the basis of multi-body simulation results, supplemented with a partial model validation, which was performed using data measured in regular operation on an electric locomotive.

Finally, it is possible to give a suggestion for a more systematic verification of the vehicle/track interaction of the vehicles equipped with these 'advanced technologies' in the relevant track sections, minimally in the range of the Annex F of EN 14363 [1]. The lateral force effects of railway vehicles on the track in the S-curves seem to be an important aspect of the assessment of their 'track friendliness'.

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