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Modelling of flexi-coil springs with rubber-metal pads in a locomotive running gear

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Abstract

Nowadays, flexi-coil springs are commonly used in the secondary suspension stage of railway vehicles. Lateral stiffness of these springs is influenced by means of their design parameters (number of coils, height, mean diameter of coils, wire diameter etc.) and it is often suitable to modify this stiffness in such way, that the suspension shows various lateral stiffness in different directions (i.e., longitudinally vs. laterally in the vehicle-related coordinate system). Therefore, these springs are often supplemented with some kind of rubber-metal pads. This paper deals with modelling of the flexi-coil springs supplemented with tilting rubber-metal tilting pads applied in running gear of an electric locomotive as well as with consequences of application of that solution of the secondary suspension from the point of view of the vehicle running performance. This analysis is performed by means of multi-body simulations and the description of lateral stiffness characteristics of the springs is based on results of experimental measurements of these characteristics performed in heavy laboratories of the Jan Perner Transport Faculty of the University of Pardubice.

Keywords: flexi-coil spring, lateral stiffness, tilting rubber-metal pad, railway vehicle, multi-body simulations

1. Introduction

Flexi-coil springs are steel spiral springs which allow their lateral loading besides to the axial (vertical) loading. This property is often used in design of modern railway vehicles, especially in their secondary suspension stage (i.e., between the vehicle body and bogie frame). Sometimes we talk about a *bolsterless bogie concept* in such cases. Besides to the vertical suspension of the vehicle body (realized by means of the axial deformation of the springs), the secondary springs ensure by means of their lateral deformation also the lateral suspension of the vehicle body and yawing of the bogies (especially during the run of the vehicle through a curve). For the lateral suspension, the lateral deformation (in the vehicle-related coordinate system) is used; at the yawing of bogies, especially the longitudinal deformation (in the vehicle-related coordinate system) of the springs is significant.

The flexi-coil springs (and also the bolsterless bogie concept) have been used in the design of locomotive bogies for higher speeds approximately since 1960’s; for example bogies of the legendary German six-axle electric locomotive Class 103 for speed 200 km/h or so-called Henschel bogies (see e.g. [1]) are well known in this context. At the development of modern railway bogies for high speed operation, a requirement on detailed knowledge of stiffness characteristics and credible modelling of these springs has arisen together with a significant progress in computer simulations in the branch of railway vehicle dynamics. For calculation

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of the lateral stiffness of the flexi-coil springs, several theories were formulated and many experiments were realized by different authors (see e.g., report [7] or paper [8]); also the FEM calculations can be used for these purposes (see [8]). Nowadays, some models of flexi-coil springs, which allow their description on basis of the knowledge of their parameters, are implemented into the commonly used multi-body simulation tools as SIMPACK, ADAMS/VI-Rail etc. A state-of-the-art of modelling of suspension elements in multi-body vehicle dynamics is summarized e.g. in paper [2].

Sometimes it is suitable to modify the lateral stiffness of the flexi-coil springs in a desirable way. That modification is usually related with the requirement on a minimization of the resistance against bogie rotation (i.e., on a reduction of the lateral stiffness of the springs in longitudinal direction) at a preservation of the relatively high lateral stiffness of the springs in the lateral direction because of a needful value of the stiffness of lateral suspension of the vehicle body. This directionally dependent modification of the lateral stiffness of the flexi-coil springs can be reached for example by means of an application of tilting rubber-metal pads.

2. Influence of lateral stiffness of flexi-coil springs on parameters of running gear

The lateral stiffness of a flexi-coil spring is influenced by means of its design parameters, i.e., especially height, number of coils, mean diameter of coils and wire diameter. Values of these parameters follow from a proposal of the spring which must reflect the required vertical (axial) stiffness as well as conditions of state of stress in the spring. Therefore, the lateral stiffness of the spring, which is intended for a particular application, cannot be simply modified by means of a change of the design parameters of this spring. As it was mentioned in the introduction, the lateral stiffness of secondary flexi-coil springs determines in case of vehicles with the bolsterless bogie concept these two important basic parameters of their running gear:

- *lateral stiffness of the secondary suspension* which is determined by means of the total lateral stiffness of the secondary springs in lateral direction (in the vehicle-related coordinate system) and possibly also by means of the additional lateral stiffness of a mechanism for transmission of forces between the vehicle body and bogie frame;
- *resistance against bogie rotation* which is determined especially by means of the lateral stiffness of the secondary springs in longitudinal direction (in the vehicle-related coordinate system) because of their dominant component of deformation at the rotation of bogie around its vertical axis (see the scheme in Fig. 1).

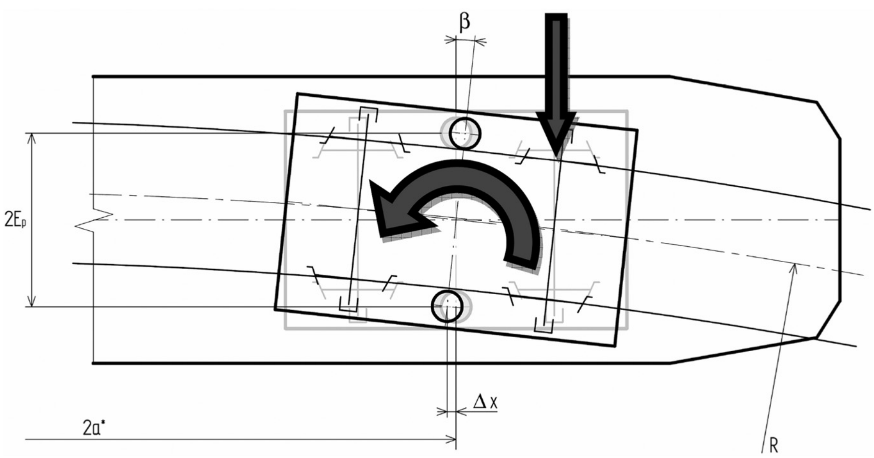

Fig. 1. Principle of acting of the moment against bogie rotation on the front bogie of railway vehicle during its run through a curve

Especially the resistance against bogie rotation is a very important quantity from the point of view of the vehicle running performance because of its influence on lateral force interaction between the vehicle and the track during the run through curves as well as on stability of vehicle run at higher speeds in a straight track. During the run of the vehicle through a curve with a radius R, the bogie turns around its vertical axis about the angle β. If the lateral distance of centres of the springs of secondary suspension on the bogie is marked as $2E_p$, a moment against bogie rotation (which is outlined in Fig. 1) can be simply estimated as:

$$M_z = 2 \cdot k_x \cdot E_p^2 \cdot \beta, \tag{1}$$

where k_x is a lateral stiffness of the flexi-coil springs situated on one side of the bogie in longitudinal direction. Then, the resistance against bogie rotation can be expressed as:

$$\gamma = \frac{M_z}{\beta} = 2 \cdot k_x \cdot E_p^2. \tag{2}$$

2.1. *General requirements on lateral stiffness of secondary flexi-coil springs*

A negative consequence of higher values of the resistance against bogie rotation is a higher value of the *quasistatic guiding force* which acts on the outer wheel of the first wheelset of the vehicle during its run through a curve (and which is also outlined in Fig. 1). This force is related with wear of wheels and rails in curves and its value is assessed in framework of the approval process of new railway vehicles according to relevant standards (e.g. the Technical Specifications for Interoperability which refer to the European Standard EN 14363 [3]). Because it is usually problematic to meet the requirements of the standards on a limit value of the quasistatic guiding force in small-radius curves, the resistance against bogie rotation should be as low as possible. On the other hand, higher values of the resistance against bogie rotation contribute to a better *riding stability* of the vehicle at higher speeds during the run in a straight track. Therefore, a proposal of the secondary suspension from the point of view of the resistance against bogie rotation — i.e., from the point of view of the lateral stiffness of the secondary springs, as well — is always a certain compromise.

In the design stage, the lateral stiffness of a flexi-coil spring can be estimated with using of some of existing empirical formulae, e.g., according to Gross, Wahl, Sparing, Timoshenko-Ponomarev or British Standard (see e.g. the overview in paper [8]). The lateral stiffness of the spring itself is practically directionally independent. A certain influence of orientation of the end coils of the spring can be observed; however, this effect (in relation with the absolute magnitude of the lateral stiffness of the springs used in design of running gears of railway vehicles) is not too significant. However, it is desired to reach a low lateral stiffness of the flexi-coil spring in longitudinal direction (because of the minimization of the resistance against bogie rotation) and simultaneously preserve a higher value of the lateral stiffness of the flexi-coil spring in lateral direction (because of its function as an element of the lateral suspension of the vehicle body). Therefore, the application of special rubber-metal pads is practically the only effective way how to reach the directionally dependent modification of the lateral stiffness of the flexi-coil springs.

2.2. *Influence of tilting rubber-metal pads on lateral stiffness of flexi-coil springs*

A frequently used type of the pads, which is used for purpose of the directionally dependent modification of the lateral stiffness of flexi-coil springs, is a tilting rubber-metal pad. This pad (see e.g. the visualisation in Fig. 2) can be placed on the upper or bottom end of the spring. At the lateral deformation of the assembly spring/pad, these pads allow tilting of the adjoining end

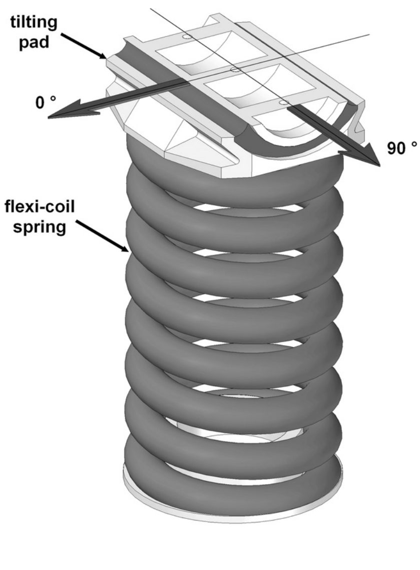

Fig. 2. Flexi-coil spring with tilting rubber-metal pad GMT

coil of the spring in relevant direction. This tilting effect is related with a significant change of the lateral stiffness of the assembly spring/pad as well as a reduction of stress in the spring. In the perpendicular direction, the tilting of the end coil of the spring is practically impossible. Therefore, the influence of the pad on the resulting lateral stiffness of the assembly is marginal in this direction.

Because the above mentioned formulae for calculation of the lateral stiffness of flexi-coil springs assume fixed end coils, it is not possible to use them for estimation of the lateral stiffness of a spring with the tilting pad. Therefore, a new analytical formula considering an angular stiffness of the pad was derived on the basis of the Gross' formula in following form:

$$k_x^{sp} = \left[\frac{\frac{1}{k} \cdot \tan kH - H - \gamma_p \cdot \left(1 - \frac{1}{\cos kH}\right) \cdot \frac{1 - \cos kH - k \cdot H \cdot \sin kH}{F_z + \gamma_p \cdot k \cdot \sin kH}}{F_z \cdot \left(1 + \gamma_p \cdot k \cdot (1 - \cos kH) \cdot \frac{\tan kH}{F_z + \gamma_p \cdot k \cdot \sin kH}\right)} + \frac{H}{S} \right]^{-1}, \tag{3}$$

where F_z is a vertical load of the assembly spring/pad, H is a length of the loaded spring, S is a shear stiffness of the spring according to Gross, γ_p is the angular stiffness of the tilting pad and k is a constant, which is defined as:

$$k = \sqrt{\frac{F_z}{B \cdot \left(1 - \frac{F_z}{S}\right)}}, \tag{4}$$

where B is a bending stiffness of the spring according to Gross. A detailed description of derivation of the formula (3) can be found in paper [4]. An analysis of this formula shows that decreasing angular stiffness and increasing vertical load of the assembly spring/pad lead to decreasing lateral stiffness of the assembly. Lower values of the angular stiffness of the pad in combination with a higher vertical load can even lead to a negative value of the resulting lateral stiffness. However, the description of the lateral stiffness of the assembly by means of the formula (3) is applicable only for the relevant direction of the lateral loading and also only in case of pads with linear angular characteristics.

In order to verify the results of the formula (3) and describe the lateral stiffness characteristic of the assembly spring/pad (see Fig. 2) completely, a measurement of this lateral characteristic was performed on the dynamic test stand in heavy laboratories of the Jan Perner Transport Faculty of the University of Pardubice in 2013. Results of the measurement, which was carried out with using of a secondary flexi-coil spring from a bogie of a modern electric locomotive supplemented with a tilting rubber-metal pad GMT, show following facts:

- the direction of loading has a dominant influence on the resulting lateral stiffness;
- in the direction, in which the pad allows tilting of the end coil of the spring (i.e., the direction, which is defined as 0° in Fig. 2), the reduction of the lateral stiffness of the assembly is most significant;
- in the direction, in which the pad allows tilting of the end coil of the spring, the measurement results confirm the analytically calculated trend that increasing vertical load of the assembly leads to decreasing resulting lateral stiffness;
- because of a non-linear angular characteristic of the real tilting pad, the lateral stiffness of the assembly spring/pad depends on its lateral deformation, as well.

The performed measurement allows to describe the lateral characteristic of the investigated assembly spring/pad completely. Especially in the directions which are not parallel or perpendicular to the direction, in which the pad allows tilting of the end coil of the spring, an analytical description of the resulting lateral stiffness would be too complicated. On basis of the measurement results, the lateral characteristic of the assembly was approximated by means of mathematical functions — in a general form in dependency on the vertical load F_z, direction of lateral loading β_p (relative to the orientation of the tilting pad — see Fig. 2) and resulting lateral deformation of the assembly Δ as:

$$k_\Delta = f\left(F_z, \beta_p, \Delta\right). \tag{5}$$

A graphical representation of this characteristic is depicted for a chosen value of the vertical load F_z (which corresponds to the static load of the spring on the locomotive) in Fig. 3. This characteristic was used as an input to the multi-body model of an electric locomotive.

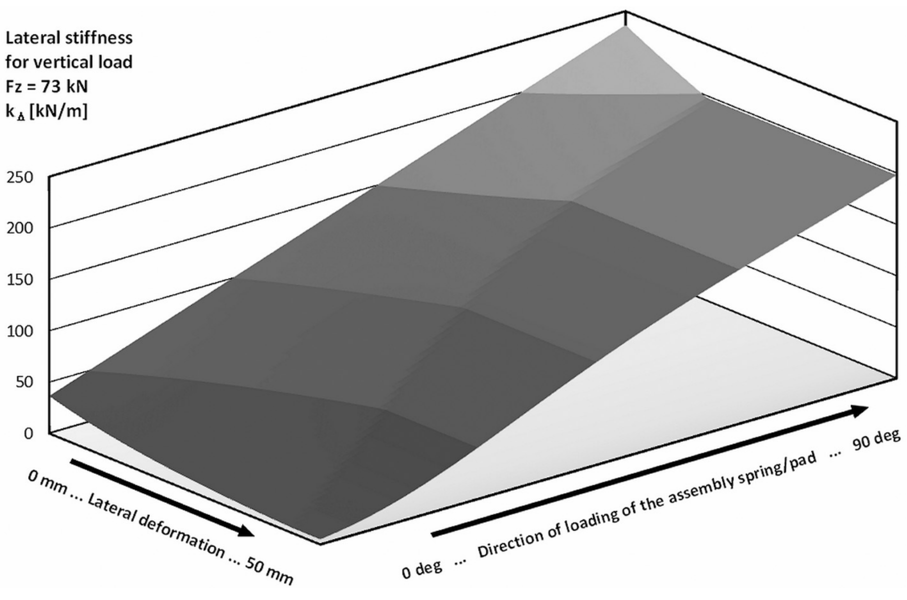

Fig. 3. Approximation of measured lateral characteristic of flexi-coil spring with tilting rubber-metal pad for the static vertical load 73 kN; the basic directions of loading (0° and 90°) are defined in Fig. 2

3. Modelling of flexi-coil springs with tilting rubber-metal pads in multi-body models for simulations of railway vehicle dynamics

The measured characteristic of lateral stiffness of the flexi-coil spring supplemented with the tilting rubber-metal pad (see Fig. 3) was implemented into a multi-body simulation model of an electric locomotive. The aim of the subsequently performed simulations was an assessment of influence of the newly arranged secondary suspension on dynamic properties of the vehicle. The simulation model was created in the multi-body simulation tool "SJKV" which is being developed at the Detached Branch of the Jan Perner Transport Faculty in Česká Třebová. A more detailed description of this original simulation tool is presented e.g. in the paper [5].

For purposes of modelling of the flexi-coil springs with tilting pads, several modifications of the simulation software had to be carried out. The first change is related with a level of detail of the dynamic model. Newly, the model of the locomotive consists of 15 rigid bodies and has 58 degrees of freedom. This requirement on the more detailed computational model is connected especially with the new principle which was applied at the modelling of joints between individual bodies. Originally, some joints were characterized with reduced parameters. It means that e.g. lateral stiffness of wheelset guiding was described by means of only one characteristic which covered relevant stiffness of all elements forming the wheelset guiding (primary springs, longitudinal rods, bump stops etc.). Similarly, the resistance against bogie rotation was usually characterized with a constant value, at least in case of the bolsterless bogies without friction elements. This approach to modelling of joints is very useful especially in such cases, in which the needed characteristics of these joints can be measured (e.g. on a prototype of the railway vehicle on a special test stands).

In the new version of the program system "SJKV", each joint element is modelled separately. The reason for that way of modelling is the knowledge of the characteristics of these individual joint elements. Especially in case of the flexi-coil springs supplemented with the tilting rubber-metal pads, the lateral stiffness of these elements shows a strong dependency on the direction of loading and therefore, the new way of joint modelling seems to be more correct than the original one. The process of computing of the forces acting in the springs with tilting pads in horizontal plane was algorithmized into following steps:

- a couple of coordinates is assigned to each of 4 spring/pad assemblies on each bogie (see the scheme in Fig. 4) — these coordinates determine the position of the upper and bottom end of these assemblies, i.e., their position on the bogie frame $[x_{fi}, y_{fi}]$ and on the vehicle body $[x_{bi}, y_{bi}]$; these coordinates are coupled with the vehicle body-related reference frame with the origin in the centre of the bogie pivot;
- for a general position of the bogie relative to the vehicle body, i.e., for the longitudinal displacement R_{bfxi}, lateral displacement R_{bfyi} and angle of bogie rotation β_i (see Fig. 4), actual coordinates can be calculated for individual spring/pad assemblies as:

$$[x_{bi}, y_{bi}] = [\pm V_p, \pm E_p], \tag{6}$$

$$x_{fi} = R_{bfxi} \pm \sqrt{\frac{E_p^2 + V_p^2}{1 + \tan^2\left[\arctan\left(\frac{E_p}{V_p}\right) \pm \beta_i\right]}}, \tag{7}$$

$$y_{fi} = R_{bfyi} \pm \tan\left[\arctan\left(\frac{E_p}{V_p}\right) \pm \beta_i\right] \cdot (x_{fi} - R_{bfxi}), \tag{8}$$

where V_p is a longitudinal distance of a centre of the spring/pad assembly from the lateral bogie axis and E_p is a lateral distance of the centre of the assembly from the longitudinal

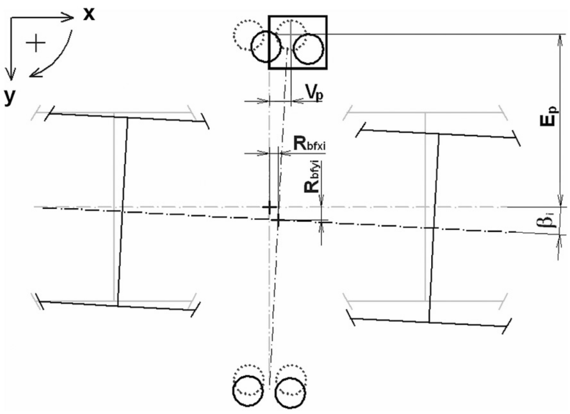

Fig. 4. General position of a bogie under a vehicle body

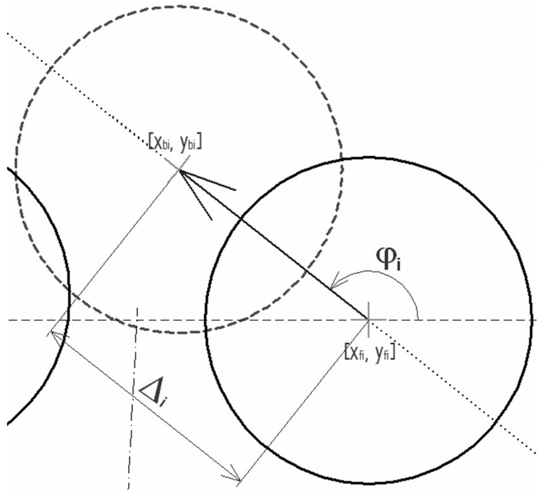

Fig. 5. Detail view on a part of Fig. 4

bogie axis (see Fig. 4); the expressions (7) and (8) can be derived by means of analytical geometry with using of the condition that the actual position of the observed points on the bogie frame is given as intersection points of a circle, defined with a shifted centre $[R_{bfxi}, R_{bfyi}]$ and a radius corresponding to the distance of the spring centres from the bogie centre, and straight lines, connecting these spring centres in the turned position, i.e., with respect to the angle of bogie rotation β_i;

- on basis of the actual position of the upper and bottom end of the spring/pad assemblies, the resulting horizontal deformation Δ_i and the direction of this deformation φ_i can be calculated for individual assemblies according to the scheme in Fig. 5:

$$\Delta_i = \sqrt{(x_{bi} - x_{fi})^2 + (y_{bi} - y_{fi})^2}, \tag{9}$$

$$\varphi_i = \arctan \frac{y_{bi} - y_{fi}}{x_{bi} - x_{fi}}; \tag{10}$$

- the direction of deformation φ_i of individual spring/pad assemblies is transformed (with respect to the orientation of the tilting pads) to the direction of lateral loading of these assemblies β_{pi} (i.e., relative to the orientation of the tilting pad);
- on basis of knowledge of the vertical load F_{zi} of the spring/pad assemblies, the horizontal forces acting in the assemblies in relevant direction, which is given by means of the angle φ_i, can be calculated with using of the mathematical description of the lateral stiffness characteristic of the assembly (5), which is depicted in Fig. 3, as:

$$F_{\Delta i} = \int_{(\Delta_i)} k_{\Delta i}(F_{zi}, \beta_{pi}, \Delta_i)\, \mathrm{d}\Delta_i. \tag{11}$$

4. Simulations of dynamic behaviour of locomotive with modified secondary suspension

The new model of an electric locomotive with a total weight of 90 t was subjected to investigation of running and guiding behaviour. The simulation results show that the tilting pads influence the resistance against bogie rotation of the vehicle in a very specific way with all consequences on the vehicle running performance. Because of the new approach to modelling of joints, this resistance is no more an input quantity into the simulation. Newly, the *resistance against bogie rotation* is an output quantity which depends on specific situation and which can be approximately calculated for the actual value of angle of bogie rotation β with using of the horizontal forces $F_{\Delta j}$ acting in the spring/pad assemblies on each bogie as:

$$\gamma \doteq \frac{1}{\beta} \cdot E_p \cdot \sum_{(j)} F_{\Delta j} \cdot \cos \varphi_j . \tag{12}$$

The influence of implementation of the tilting pads into the secondary suspension on the *stability of vehicle run* is negative. Under given conditions, the critical speed of the locomotive is (significantly) lower than in case of a locomotive without the tilting pads in secondary suspension. This effect is related with a significant reduction of the resistance against bogie rotation and corresponds to general conclusions presented in the paper [6].

The influence of the tilting pads on the *guiding behaviour in small-radius curves* is more complicated because of its dependency on the actual cant deficiency. The cant deficiency is an important quantity which characterizes the unbalanced lateral force acting on the vehicle in curves and which can be calculated on the basis of actual speed and constructional-technical parameters of relevant curve (see e.g. [9]). Similarly to the run of the locomotive in a straight track, the resistance against bogie rotation has a very low value in case of run of the vehicle through a curve with low values of the cant deficiency. A consequence of this fact is a similar behaviour of both bogies in such situations. It can be demonstrated on the example of distributions of the quasistatic guiding forces on individual wheels of the locomotive at the simulations of its run through a 300 m curve which are presented in Fig. 6. However, in case of run of the vehicle through curves with higher (absolute) values of the cant deficiency, the resistance against bogie rotation has higher values as a consequence of the strong dependence of lateral stiffness of the spring/pad assemblies on the direction of their loading (see Fig. 3) and the fact, that the increasing unbalanced lateral force acting on the vehicle body causes higher lateral deformations of the secondary suspension leading to a significant change of the direction of horizontal loading of the individual spring/pad assemblies in comparison with the cases corresponding to the small cant deficiency or to the run in a straight track. From the graph in Fig. 6, it is evident that especially the difference of the guiding forces acting on the leading wheels of both bogies (i.e., on wheels 12 and 32) increases with increasing value of the cant deficiency.

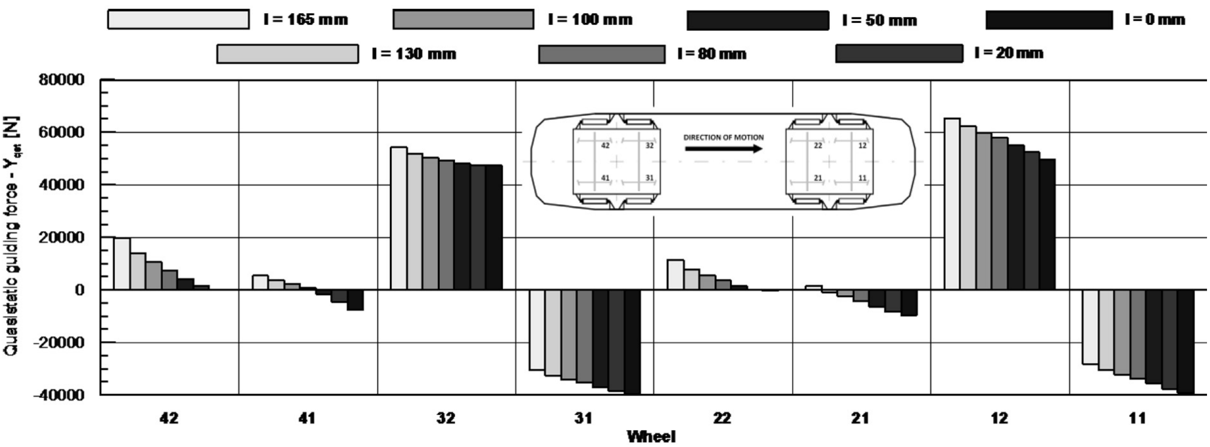

Fig. 6. Quasistatic guiding forces acting on individual wheels of a 90 t locomotive with the secondary suspension supplemented with tilting pads during the run through a 300 m curve at various values of cant deficiency

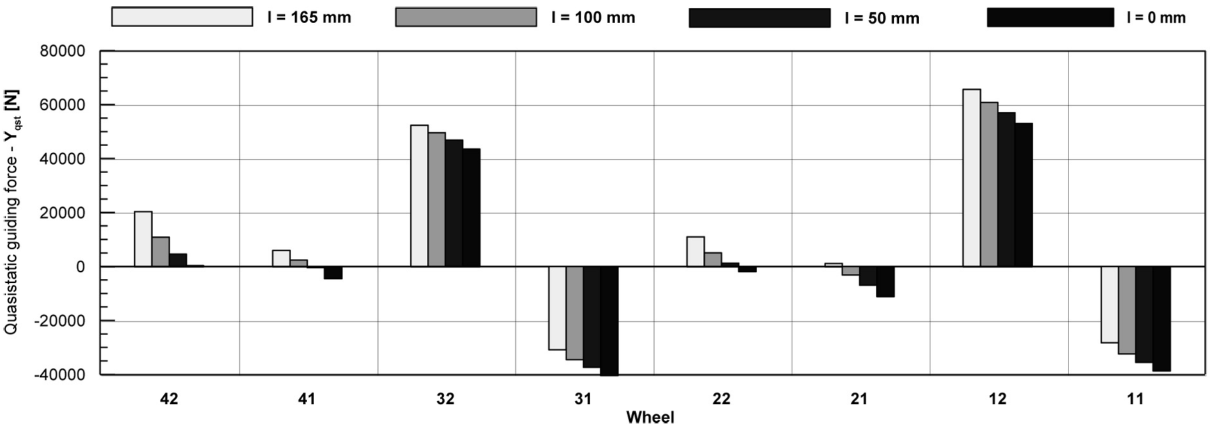

Fig. 7. Quasistatic guiding forces acting on individual wheels of a 90 t locomotive with the secondary suspension without the tilting pads during the run through a 300 m curve at various values of cant deficiency

For comparison, simulation results of a similar locomotive with the same parameters, which is not equipped with the tilting pads in the secondary suspension, can be shown for analogous situations defined with chosen values of cant deficiency at the run through the 300 m curve. The obtained distributions of quasistatic guiding forces on individual wheels of the locomotive are demonstrated in Fig. 7 for that case. Identically to the case of locomotive equipped with the tilting pads, these simulations were also performed for conditions of dry rails (i.e., for a value of friction coefficient in wheel/rail contact $f = 0.40$) and characteristics of wheel/rail contact geometry corresponding to theoretical wheel profiles S1002 and rail profiles 60E1 at nominal values of wheelset gauge (1 425 mm) and track gauge (1 435 mm).

From the graph in Fig. 7, it is evident that the quasistatic guiding force on the leading wheel of the front bogie (i.e., on the wheel 12) has a higher value in case of the locomotive without tilting pads in all cases of the simulated cant deficiency. However, this difference between the locomotive with the tilting pads and without them decreases with increasing cant deficiency. At the zero cant deficiency, the locomotive without tilting pads shows by 3.3 kN higher force Y_{qst12} than the locomotive with pads; at the cant deficiency of $I = 165$ mm, this difference is only 0.3 kN. The second well observable effect is the dependency of difference of the guiding forces acting on the leading wheels of both bogies (i.e., on wheels 12 and 32) on the cant deficiency. In case of the locomotive without tilting pads, a change of this difference with increasing cant deficiency is not so significant. At zero cant deficiency, the difference $Y_{qst12} - Y_{qst32}$ has a value of 6.5 kN (in case of the locomotive with tilting pads, it is only 2.3 kN); for the cant deficiency of $I = 165$ mm, the difference is 13.3 kN (and 11.1 kN for the locomotive with tilting pads). These results confirm the dependency of the resistance against bogie rotation of the locomotive with secondary suspension supplemented with the tilting pads on the cant deficiency. While the results of simulations of guiding behaviour of both observed computational models are similar at higher values of the cant deficiency; at lower values, the above mentioned differences can be observed.

5. Conclusion

This paper deals with a possibility of application of tilting rubber-metal pads into the secondary suspension of an electric locomotive where these pads are able to modify the lateral stiffness of the currently used flexi-coil springs in dependency on their horizontal loading. Section 2 deals with general requirements on the secondary suspension of railway vehicles from the point of view of their running performance; a theoretical calculation of influence of the tilting pad on the lateral stiffness of the spring/pad assembly and results of measurement of complete lateral characteristics of that assembly are presented in this section, as well. Section 3 describes the implementation of model of joint representing the flexi-coil springs with tilting pad into the multi-body model of the locomotive created in the simulation tool "SJKV". Results of the performed simulations of running performance of the locomotive are presented in section 4. These results point out very specific properties of the secondary suspension consisting of the flexi-coil springs with tilting pads. Especially the resistance against bogie rotation, which depends on the cant deficiency, shows following important properties:

- at the run with a low cant deficiency and in a straight track, the resistance is lowest which has a negative influence on the stability of the vehicle at higher speeds;
- at the run through curves with higher values of cant deficiency, the desired softening of the resistance is not too significant, and therefore its contribution to the decrease of the guiding force acting on the leading wheel of the first wheelset is limited;
- the dependency of the resistance against bogie rotation on the lateral deformation of the secondary suspension (i.e., on the cant deficiency and on the speed in curves) complicates the possibility of using of bogie couplings as well as the assessment of safety against derailment; to this point, an attention will be paid in the next research.

Acknowledgements

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