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OPTIMUM AXLEBOX-GUIDANCE STIFFNESSES FOR TRADITIONAL RUNNING GEARS OPERATING ON A GIVEN RAILWAY LINE

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Abstract

On the basis of dynamical simulation of the lateral motion of railway vehicles, the increase of the mileage expressed by the distance covered between two tyre-profile renewal turnings of the wheel sets on lathe is examined as a function of the axlebox-guidance stiffnesses in longitudinal and lateral directions with the given railway line and the stochastic lateral track unevennesses taken into consideration. With the help of applying a numerical optimization procedure, the numerical value of the stiffnesses of the axlebox guidance ensuring the maximum mileage are determined.

1. Introduction

The tribological processes of the metallic rolling/sliding contact characterizing the connection of the rails and the wheel sets built-in the running gears of railway vehicles take a considerable influence on the increase in wear of type profiles with time, and consequently on the mileage expressed by the distance covered between two tyre profile renewal turnings of the wheel set on lathe, and through it, on the life of them, as well as on the maintenance demand occurring in the course of operation, too.

The advanced stage of the vehicle system dynamics renders possible the thorough analysis of the contact and wear relations between the wheel and rail with the help of computer simulation. With the application of dynamical models, the contact forces determining the motion of wheel sets within the track gauge, and through it, the wear loads developing on the contact surface of wheel and rail, as well as their distribution along the tyre profile (or rail profile, respectively) can be determined in the case of both straight line sections and sections of varying curvature. The above models take into consideration the lateral deviations of rails from the nominal position, too, in the form of track unevenness functions [1, 2, 3].

With the knowledge of the wear load distribution of the tyre profiles, the alteration in the shape of the tyre profiles due to wear can be predicted, and this altered tyre profile can always be compared with the geometrical limit values of wear prescribed with respect to the running safety aspects, and after reaching those, the total distance covered up to the reaching of the limit values, i.e. the mileage related to the wheel set/vehicle will be determined [4, 7].

As a matter of course, the mileage determined according to those said above will depend on the geometrical and dynamical parameters of the vehicle examined, e.g. on the initial geometry of the tyre and rail profiles, and the stiffness, damping and inertial characteristics of the track and the wheel-set guidance system. On the basis of this dependence relation, the variation of the mileage will be examined as a function of the axlebox guidance stiffnesses of the wheel sets in longitudinal and lateral directions with the given railway line and the stochastic lateral track unevennesses taken into consideration. With the help of a numerical optimization procedure (method of gradients), the range of the axlebox guidance stiffnesses ensuring the maximum mileage will be determined [5, 6].

2. Dynamical Simulation

To determine the lateral motions of a four-axle, two-bogie railway vehicle, the dynamical model of the track–vehicle system was used as shown in *Fig. 1.* From among the 30 free coordinates of the model, there are 22 ones which describe the motion of the vehicle, while 8 of them characterize the lateral motion of the equivalent track masses placed under the individual wheels. In the case of the vehicle body and the bogie frames, the considered motions are represented by the lateral displacements and the angular (yawing) displacement around the vertical, gravity-centre axle, while in the case of wheel sets, in addition to the motions mentioned above, the angular displacements around their own axle, as well as the longitudinal displacements relative to the bogie frame were also taken into consideration.

According to *Fig. 1*, the connections symbolized by springs between the individual elements of the model represent non-linear, dissipative force connections. The non-linearity of force connections can be given in the form of force-displacement diagrams, while the dissipative property is expressed by the linear damping force proportional to the relative velocity.

From the viewpoint of the dynamical simulation of the lateral motion of the vehicle, the appropriate modelling of the wheel–rail contact is of a decisive importance. In this respect, our dynamical simulation procedure is based on the solution to the geometrical problem of contact between arbitrary tyre and rail profiles given by their contact points. Prior to dynamical simulation, with the knowledge of the shapes of the tyre and rail

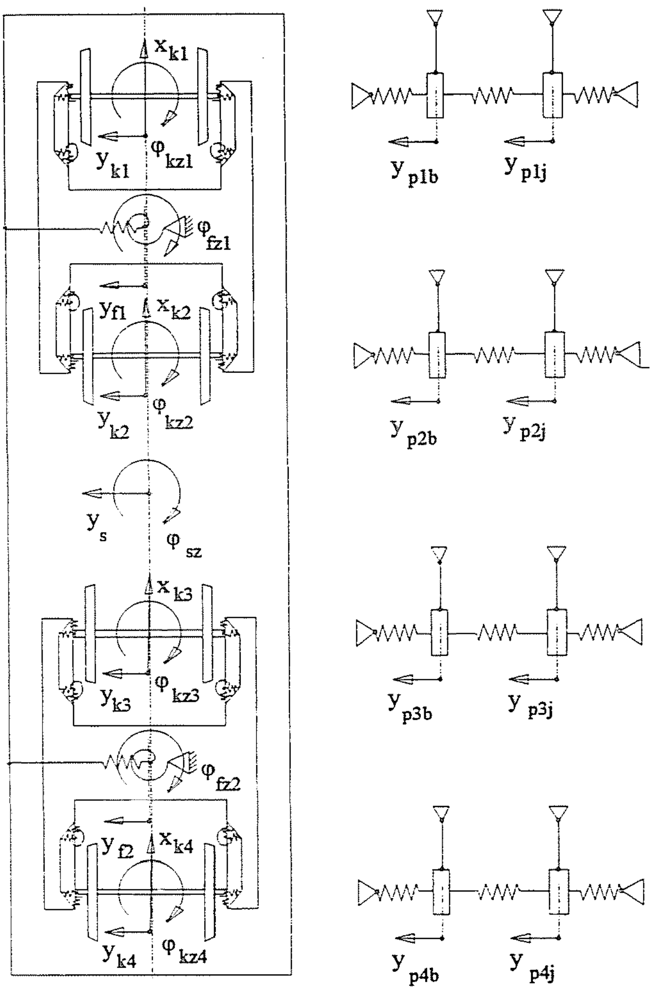

Fig. 1. Dynamic model

profiles, the contact points of the two profiles in the possible different relative positions of the vehicle and rail, as well as the contact characteristics required for additional calculations in contact points are determined. The relative position of the wheel and rail is considered as given by the relative lateral position of the two profiles, or by the angle subtended by the plane of rotation of the wheel and the longitudinal tangential direction of the rail, respectively, which is the so-called angle of attack [1].

To determine the force actions arising on the contact surfaces of the wheel and rail, in case the contact point is on the tread of the tyre profile, our calculations are based on the Kalker's linear theory, while in case the contact point is on the wheel flange, the normal contact force is derived from elastic deformation, and the force components arising in the contact plane are deduced with the help of the relationship of sliding friction. In the contact points of the wheel tread, the 'creep forces' determined with the help of Kalker's linear theory were confined by the frictional force limits associated with wheel sliding.

The lateral motion of the vehicle is excited by the always present lateral track unevennesses. In our model, the track unevenness exerts its influence through the fact that the relative position of the rolling tyre profile performing translatory motion and the contracted rail profile is changed due to the track unevenness. As a natural consequence of the change in relative position, the position of the contact points of the two profiles will also be changed, and so will behave the forces arising in the contact points. The changed force components, in turn, will result in the lateral acceleration of the wheel sets, and through them, in that of the bogie frames, as well as the vehicle body. Naturally, the changed contact forces will exert an influence — through the elastic supports — on the lateral motion of the equivalent track masses, too. In our simulation example, the realizations of the left- and right-hand side track unevenness functions were derived from the spectral density functions of the measured gauge and direction errors [3, 4].

The dynamical simulation procedure was elaborated in a way that it should be suited to simulate the lateral motion of vehicles travelling at a high speed along both a straight and a curved track of arbitrary curvature. Accordingly, the description of the vehicle's motion will take place in a coordinate system moving together with the centre of gravity of the vehicle body and adapted on the actual tangent to the theoretical centre line of the track.

3. Determination of Wheel Wear

For the numerical determination of the wheel wear observed on the tread-profile of the wheel, the relationship widespread in the literature [2, 4]

$$\frac{\partial w}{\partial L} = k_t \left(F_x \gamma_x + F_y \gamma_y \right) \tag{1}$$

was used, where $\partial w / \partial L$ represents the specific wear related to the unit of the covered distance in mg/m, γ_x and γ_y are the longitudinal and lateral creepages, respectively, occurring in the given contact points, F_x and F_y are the creep forces corresponding to the creepages, while k_t is the wear constant related to the wheel tread given in mg/Nm. To determine the wheel wear occurring in the contact points on the wheel flange of type profile, relationship:

$$\frac{\partial w}{\partial L} = k_n \left(F_s \frac{\Delta v}{v} \right) \tag{2}$$

was used by the analogy of formula (1), where F_s is the force component lying in the contact plane of the wheel flange, Δv is the relative (sliding) velocity component of the two contacted profiles rising in the contact plane of those, while k_n is the wear constant associated with the wheel flange.

For the determination of wear distribution along the wheel-profile curve, the examined section of the tyre-profile curve was divided into intervals of 1 mm width. Simultaneously with the dynamic simulation, the instantaneous values of the wheel wear were determined according to the above formulae with the knowledge of the position of contact points, as well as the arising force actions with respect to all the wheels of the vehicle in each step of integration [3]. The obtained wear values were distributed along the contact area between the wheel and rail characterized by the contact point as centre, and in this way, the wear values allotted to the individual distribution intervals of the wheel profile were obtained. Parallel with the dynamical simulation, with the wear values totalled with respect to the individual distribution intervals, the distribution of the wear along the tyre profile of the individual wheels of the vehicle travelling along the given track section has been determined. With the topology of a given railway line, with the characteristics of its straight and curved sections taken into consideration, with the dynamical simulation and wear calculations performed for all sections and with the summation of the results obtained, the resultant wheel wear developed after the coverage of the whole distance of the given line will be yielded. This wheel wear and its distribution along the tyre profile provides a basis for the determination of the new shape of the profile changed due to wear.

After performing the material removal in the given distribution intervals in proportion to wear obtained for the given interval, the wheel profile changed due to wear will be determined. Of course, in the course of the material removal, the characteristics of the facing rail profile being in contact with the tyre and causing the wear should be taken into consideration [5, 6].

With the changed wheel profile, the dynamical running simulation can be repeatedly performed, the latest wear of wheel and wear distributions can be determined, and the shape of the wheel-profile can be modified additionally. This procedure can be repeated in case the obtained latest wheel profile has reached the wear limits determined on the basis of safety considerations. By the summation of the distances covered by the vehicle up to the reaching of wear limits, the mileage performance of the vehicle is obtained.

4. Optimization of the Axlebox-Guidance Stiffnesses

Our procedure developed to determine the numerical values of the axlebox guidance stiffnesses of the wheel sets advantageous with respect to wheel wear is based on the above determination of the mileage performance associated with the given dynamical parameter system of the vehicle and with those of the given railway line. The calculation of the mileage performance according to those said above is carried out with different pairs of stiffness values of the axlebox guidance, and the pair of values is searched for which the obtained mileage performance will have the highest value.

Since the determination of the mileage performance according to those said above is a computation-time-consuming process, therefore to determine the optimum axlebox guidance values, such a procedure based on the gradient method was elaborated which requires the calculation of the functional values in relatively few points. In the following, also an example of the optimization process will be given [6].

5. Computation Results

From among the results of an actual stiffness-optimization process, *Figs. 3 – 8* will be presented. The stiffness optimization is related to a four-axle, two-bogie vehicle of rubber-bell wheel-set guidance (Clouth system) and with a line section of about 30 km in length, where the minimum track curve radius was 450 m, and the maximum radius was 3000 m, while the total length of the straight line sections amounted to 12 km, see *Fig. 2*. The initial wheel profile is identified with K5, while the rail profile with UIC 54 [6].

In *Fig. 3*, the effect of wheel profile shape on the variation in wear distribution is shown. Here, the variation of the resultant specific wheel set wear allotted to the profile division intervals of 1 mm width is represented along the profile in the case of a new wheel profile (plotted in continuous line) and a worn one (plotted in dashed line). In the Figure, the shapes of both the new and the worn profiles are represented for informative purposes. As it can be seen in the Figure, in the case of the new wheel profile, three nearly identical wear peaks can be observed, the first of which shows the wear peak of the wheel-flange contacts. As a result of wear, a trough-like medium part has developed on the running tread, and this resulted in a certain rearrangement of wear distribution. The leading

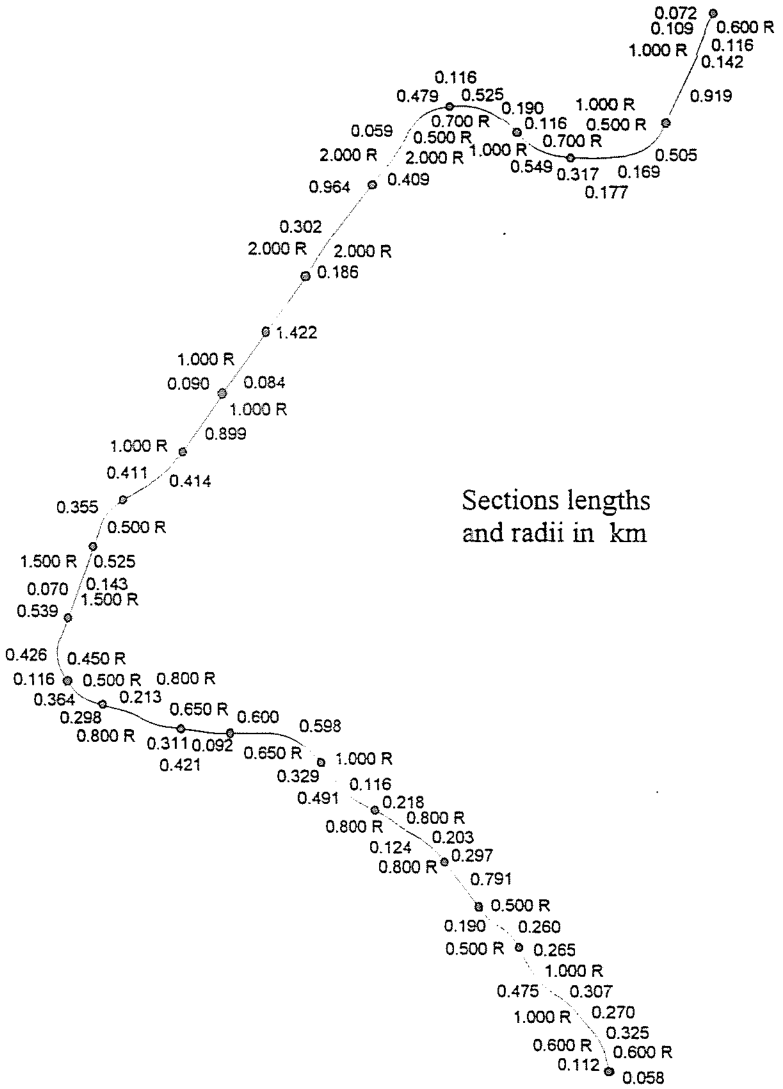

Fig. 2. Topology of the railway line (Section lengths and radii in km)

role of the wheel-flange was taken over nearly totally by the edges of the trough, and accordingly, the wear maxima were re-arranged to the edges of those troughs. The wear peak of the wheel tread moved away slightly from the medium part of the profile, which fact refers to the slight increase of the motion amplitude of the wheel set.

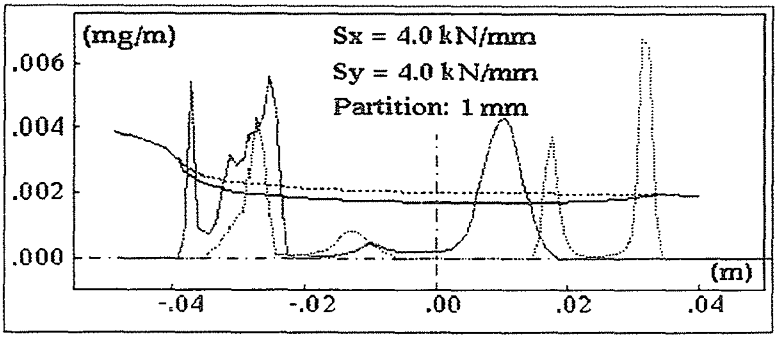

Fig. 3. Resultant specific wear (with new and worn profiles)

In *Fig. 4*, the same wheel-wear values were plotted in the case of a pair of stiffness values proved to be more favourable in the course of optimization. In this case, the wear peak observed with the wheel-flange is much more marked in the case of a new profile, and the trough characteristic of the worn profile will also be narrower on the edge towards the wheel-flange, and the wear maximum will have a higher numerical value. However, at the same time, the wear-maximum values allotted to the wheel-flange part show a declining tendency with both the new and the worn profiles as compared to the former ones. In *Fig.5*, with the same stiffness values, the variation of the specific wear along the profile as allotted to the division intervals is plotted also at some intermediate stages of the profile wear. As it can be seen in the Figure, in the course of profile changes, the peak values of profile wear practically pass around the profile and the wear peaks occurring near the wheel-flange show very high values during the initial phase of the profile wear.

In *Fig. 6*, the variation of the wheel profile due to wear is visualized at five stages up to reaching the wear-limit values. As it can be seen, the initial wear is more intensive on the wheel-flange part of the profile, then starting from the third wear stage shown in the Figure, the formation of the trough mentioned before will begin on the tread part of the wheel profile. With the propagation of wear, the wheel wear and the profile change are increasing as moving nearer towards the parts of the trough being opposite to the wheel-flange.

In *Fig. 7*, the variation of the mileage performance can be followed up in the case of different axlebox-guidance stiffnesses as function of the profile wear. On the horizontal axis of the diagram, the deviation of the worn tread-profil from the wear-limit value — as the characteristic of the

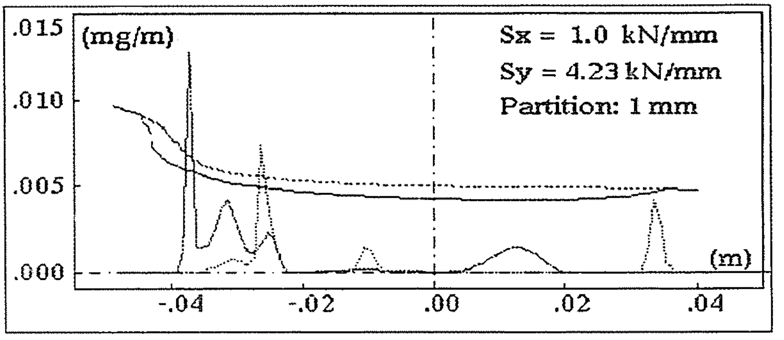

Fig. 4. Resultant specific wear (with new and worn profiles)

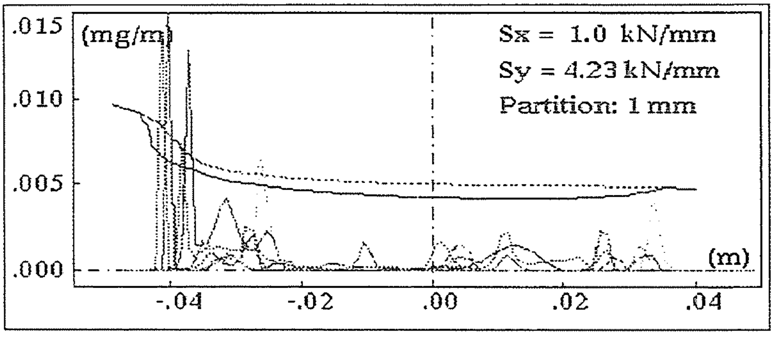

Fig. 5. Variation of wear distribution in the course of profile wear

wear propagation on the tyre — is plotted. The curves start from –5 mm — according to the limit size of 5 mm permitted for the tread wear and where they intersect the vertical line associated with zero value, there has the wheel wear reached the wear-limit position. With all the curves, an initial section of a milder slope can be observed, which fact refers to intensive wear, then the increase in the slope of the curves represents an ever increasing covered distance for a wheel wear of the same size, i.e. the reduction of the wheel wear allotted to the unit of the distance covered. It can also be read in the Figure that in the course of optimization process, the mileage performance of the vehicle has grown to about a 2.77-fold value of the initial one.

Similarly, the process of stiffness optimization can be followed up in *Fig. 8*, where the mileage performance of the vehicle is plotted as a

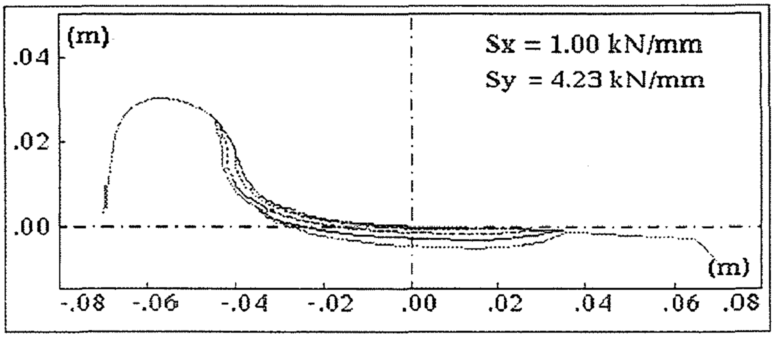

Fig. 6. Variation in the wheel profile

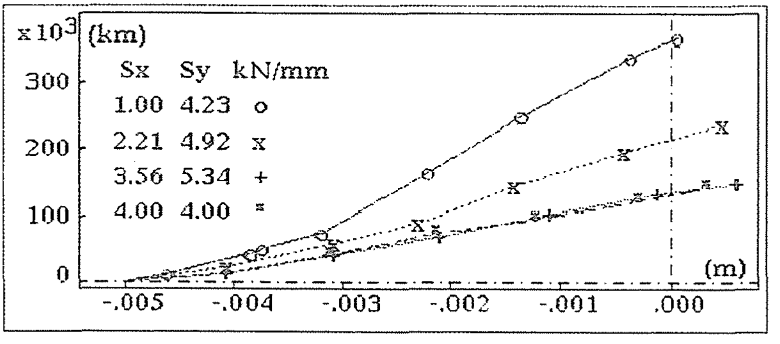

Fig. 7. Mileage performance as a function of profile wear

function of the longitudinal and lateral stiffnesses of the axlebox guidance. With respect to both the longitudinal and lateral guidance stiffnesses, the most favourable pair of the stiffness values concerning the wheel wear was investigated in the 1–10 kN/mm range, starting from the value of 4 kN/mm near to the actual stiffness values of the vehicle examined. The diagram shows the main steps of finding the optimum, and as it can be read, the maximum of the mileage performance was yielded with a more or less constant lateral stiffness (about 4 kN/mm) and with a considerably reduced longitudinal guidance stiffness of a value of 1 kN/mm.

As a matter of course, during the variation of the wheel-set guidance stiffness, neither can other problems be ignored, e.g. those associated with

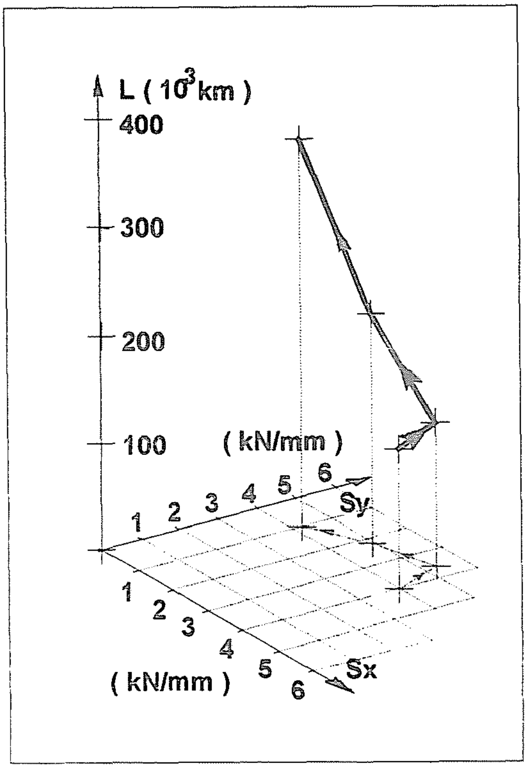

Fig. 8. Process of axlebox-guidance stiffness optimization

running stability. In this respect, without the details of our examination imparted, it can be stated only that in the course of optimization process, no signs of unstable running could be encountered within the examined velocity range of the vehicle ($v_{\max} = 70\,\mathrm{km/h}$).

6. Concluding Remarks

In connection with our procedure elaborated for maximizing the mileage performance associated with the specified wheel wear of the two-axle railway vehicles equipped with a bogie of traditional layout, and based on dynamical simulation, globally the following can be stated:

1. The elaborated dynamical simulation is suitable for determining — with due details — the lateral motion of the four-axle, two-bogie rail-

way vehicles travelling along a straight or curved track, respectively, as developed by the exciting effect of lateral track unevennesses, as well as the current relative position of the contacting wheel and rail.
2. With the help of the elaborated wear-simulation procedure, and with the determination of the mileage performance of a railway vehicle travelling along a well-specified railway line, the vehicle parameters favourable with respect to wheel wear (e.g. axlebox-guidance stiffnesses) can be determined.
3. The results obtained for the case of the vehicle examined show that the basic case of the bell-spring layout of axlebox guidance, when the longitudinal and lateral stiffnesses of the axlebox guidance coincide, meets less the requirements of constructing a wear-starved running gear. The wheel wear will slow down in case stiffness s_x is reduced.

As a matter of course, when the vehicle parameters are changed with the goal of optimization, attention should be paid — in addition to the development of wheel wear — also to other aspects as, e.g. the running safety and running stability.

In connection with the development of this method, additional effects influencing essentially the wheel wear should also be taken into consideration. Here, first of all, the consideration of the variation in the vertical wheel loads, as well as the more detailed investigations into the wheel-rail contact and their wear processes is recommended.

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