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Impact of Contact Geometry, Rail and Wheelset Structural Dynamics on the Dynamic Behaviour on an Irregular Track

I. Kaiser[1], F. Badea Romero[1], J. Á. Pérez Fernández[2], S. Strano[3], M. Terzo[3] and C. Tordela[3]

[1]Escuela Politécnica Superior, Universidad Antonio de Nebrija, Madrid, Spain
[2]Área de Ingeniería Mecánica, Universidad de Oviedo, Gijón, Spain
[3]Dipartimento di Ingegneria Industriale, Università degli Studi di Napoli Federico II, Napoli, Italy

Abstract

The running behaviour of a passenger coach on a track with irregularities is simulated using an enhanced multibody model, which takes the structural flexibilities of both wheelsets and rails into account and which uses a highly detailed wheel-rail contact model. The results show that the strength of the impact of the structural flexibilities depends on the wheel-rail contact geometry. This combined impact affects the running behaviour as well as the stress distribution within the wheel-rail contact.

Keywords: vehicle-track interaction, flexible wheelset, flexible rail, wheel-rail contact, contact geometry, track irregularities.

1 Introduction

The running stability of a railway vehicle is a key issue for save operation. In order to assure that the vehicle meets all requirements like, e.g., running stability, its dynamic behaviour is simulated using computer-based models. This enables the choice of an appropriate mechanical design of the vehicle, in particular of its running gear.
Usually, multibody models are used for the simulation of a railway vehicle's dynamic behaviour. In the original multibody approach, a technical system is described by bodies, joints and force elements. Here, bodies are elements, which have an inertia, and their motions are determined by joints. In the model of a railway vehicle, bodies typically represent components of high mass like the carbody, the bogie frames and

the wheelsets. Force elements are massless elements, which act between the bodies, whereby forces are calculated from the relative kinematics using specific force laws. In the model of a railway vehicle, force elements typically represent the springs and dampers of the suspension and the wheel-rail contacts.

In the original multibody approach, bodies are considered to be rigid, i.e., regardless of the forces acting on the body, no deformations occur. This idealization is justified in many cases, but also has its limitations. One evident limitation is that the description as a rigid body completely omits the structural dynamics of the component so that for an excitation of the components by forces close to the structural eigenfrequencies a model using rigid bodies may produce unreliable results. Another limitation occurs, when a force element is very sensitive to the relative kinematics, i.e., when even a small change of the relative kinematics can cause a big change of the forces. This can be the case for the wheel-rail contact. For certain combinations of wheel and rail profiles, it is known that a small change of the position of the wheel rim relative to the rail head can cause a strong change of the location and the shape of the wheel-rail contact area. In Central Europe, probably the best-known example is the combination of the wheel profile S1002 and the rail profile UIC60 using a rail inclination of 1:40.

Due to the high mass of railway vehicles, the wheelsets and the rails are subjected to high forces, which are transmitted by a relatively small contact area. These high forces are causing structural deformations of the wheelsets and the rails, which in turn can strongly affect the wheel-rail contact because of its aforementioned high sensitivity.

In several earlier works, in particular those by Kaiser and Popp [1] and by Kaiser [2], an enhanced modelling for a coupled vehicle-track system is presented. In [1], the modelling of wheelsets and rails as flexible and the coupling of the flexible bodies with the wheel-rail contact is described. In [2], the coupled vehicle-track model is enhanced by integrating a more detailed wheel-rail contact model, which uses an iterative calculation instead of just estimating the contact area and the stress distribution.

While the earlier works [1] and [2] are focused on the modelling, in more recent works like those by Kaiser, Poll, Voss and Vinolas [3] and by Kaiser, Poll and Vinolas [4] the enhanced vehicle-track model is used for investigating the behaviour of the system under different operating conditions like different track geometries. The track geometry is defined by several factors including the rail profile, the rail inclination and the exact track gauge, which determines the lateral play of the wheelset within the track. The crucial point is that in the railway networks of different European different track geometries are used. These different track geometries can result in different equivalent conicities so that vehicles, which are used in cross-border traffic, can show very different running behaviours in different countries, even under ideal conditions. In the works [3] and [4], the impacts of the structural flexibilities of wheelsets and rails and of the different track geometries are investigated. It turns out that the impact of the structural flexibilities is particularly strong for track geometries characterized by high equivalent conicities, i.e., for such track geometries significant changes of the motions of the wheelset and of the wheel-rail forces are caused by the structural flexibilities, while for low equivalent conicities these effects are rather weak, but still recognizable. The different dynamic running behaviour on tracks with different geometries is an important aspect of the general issue of interoperability.

In the previous works [1], [2], [3] and [4], the immanent behaviour of the coupled vehicle-track system, its “eigenbehaviour” so to speak, is investigated. An important aspect of this “eigenbehaviour” is the limit cycle behaviour, i.e., at which running speed a permanent hunting motion sets in, which hunting frequency occurs, whether flange contact occurs. The limit cycle behaviour is important, because a permanent hunting at higher frequencies, also known as “bogie instability”, can cause high dynamic forces, which can seriously damage the track and thereby cause accidents. Therefore, an important goal of the mechanical design is to avoid the occurrence of such a dangerous operating status under regular operating conditions.
For the analysis of the “eigenbehaviour” of the vehicle-track system, an ideal system with no track irregularities were used. After having gained some insight into the system’s immanent behaviour, the next logical step is the introduction of track irregularities as an external factor. Therefore, the line of research described above is continued by investigating the behaviour of the coupled vehicle-track system for the scenario of running on a straight track with irregularities. As already done for the investigation of the limit cycle behaviour, the calculations are carried out for different model variants, i.e., for the description of wheelsets and rails either as rigid bodies or as flexible bodies, and for different track geometries.

2 Modelling of the vehicle-track system

The vehicle model describes a passenger coach of conventional design having two bogies and four wheelsets. It represents a coach used for intercity traffic in Germany, which is equipped with bogies of the type MD 52. The vehicle data is taken from the doctoral thesis by Diepen [5]. The track model, which is based on the model by Ripke [6], describes a track consisting of two rails supported by 128 discrete sleepers. The bodies of the coupled vehicle-track system are shown in Figure 1, whereby the bodies displayed in dark grey, i.e., the wheelsets and the rails, are modelled as flexible bodies, while the other bodies displayed in light grey are considered to be rigid.

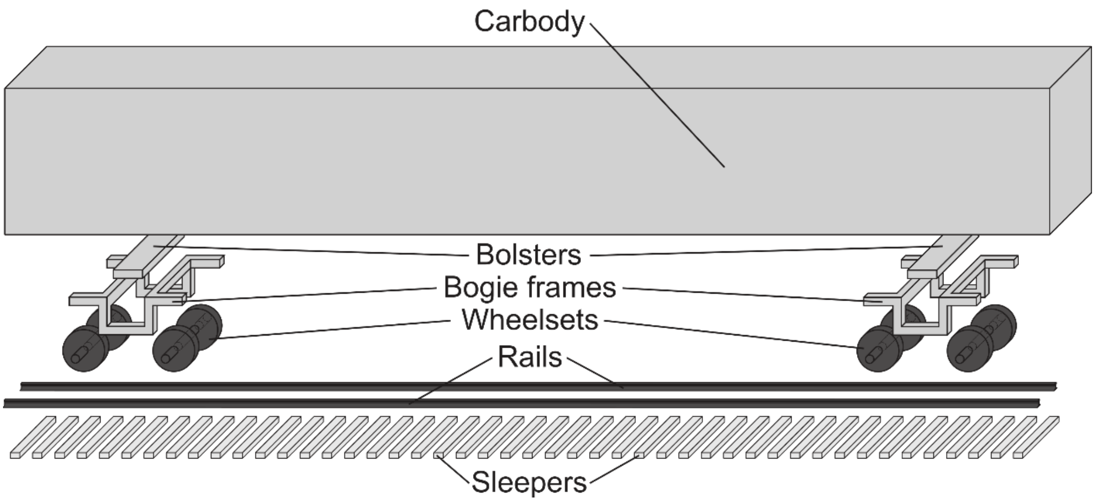

Figure 1: Bodies of the coupled vehicle-track system. Dark bodies are modelled as flexible bodies, light bodies are modelled as rigid bodies.

Both the wheelsets and the rails are modelled as three-dimensional flexible bodies using semi-analytical finite element models, which consider the wheelsets as solids of revolution and the rails as solids of extrusion.
The wheelsets and the rails are connected by eight wheel-rail modules in total. Each wheel-rail module performs the three subsequent steps of analyzing the contact geometry, solving the normal contact problem and solving the tangential contact model. The analysis of the contact geometry is based on the work by Netter [7]. The contact mechanics models for the normal and tangential contact problem are boundary element models, which are based on a discretization the Boussinesq-Cerruti equations. The contact modelling follows the fundamentals of rolling contact modelling given by Kalker [8]. The nonlinear boundary conditions of the contact equations require a solution by an iterative method based on the Gauss-Seidel algorithm, which has been successfully introduced by Vollebreght [9] for problems of such kind. The iterative resolution of the contact equations is carried out during the simulation of the multibody system, i.e., the contact model uses an actual calculation instead of just estimating the contact area or replacing it by an equivalent ellipse.
In order to investigate the impact of the structural flexibilities, two different variants of the vehicle-track model are used. The model variant, which will be addressed as "flexible model", is the full model described before, in which both wheelsets and rails are modelled as three-dimensional flexible bodies. In the model variant, which will be addressed as "rigid model", the wheelsets are modelled as rigid bodies, and a simplified track model using a rigid body as shown in Figure 2 is employed, since a completely rigid track would be highly unrealistic,

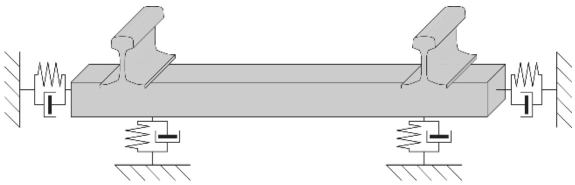

Figure 2: Simplified track model referred as "rigid rails".

A more detailed description of the methods and models used for the coupled vehicle-track model and its components can be found in [1], [2] and [4].

3 Scenario

The results, which will be presented in the following sections, are calculated for the scenario of the running of the passenger coach on a straight track at a constant running speed of $v_0 = 200$ km/h. For all eight wheel-rail contacts, a friction coefficient of $\mu = 0.35$ is used. The irregularities are described by the vertical and lateral offset of the left and the right rail using measured data from a railway line homologated for regular traffic with speed up to 200 km/h.
For all eight wheels, the original S1002 profile according to the standard UIC 510-2 is used. In order to investigate the impact of different equivalent conicities, the following four track geometries are used, which differ with respect to the rail profile, to the rail inclination and to the track gauge.

Track geometry	Rail profile	Rail inclination	Track gauge	Country
60E1/1:40/1435	60E1	1:40	1.435 m	Poland
60E1/1:40/1437	60E1	1:40	1.437 m	Austria, Switzerland
60E1/1:20/1435	60E1	1:20	1.435 m	France, Italy, Spain, Belgium
60E2/1:40/1436	60E2	1:40	1.436 m	Germany

Table 1: Track geometries used for the investigation.

4 Results

Using the two model variants, i.e., the flexible model and the rigid model, and the four track geometries listed in Table 1 the running of the passenger coach at 200 km/h is simulated over a track length of 2 km. The results are evaluated for the wheelset 1, i.e., the leading wheelset of the front bogie. Here, the lateral displacement y_{WS1} of the centre of gravity of the wheelset 1 are considered as a representative quantity for the running behaviour. In Figure 2 the results obtained for the different calculations are shown. Furthermore, the lateral irregularities of the left and the right rail under the wheelset 1 are displayed. For all calculations, the same irregularities are used.

The comparison of the results shown in the four lower diagrams shows the strong impact of the contact geometry. For the track geometry 60E1/1:40/1435, high-frequent components of the lateral motion can be recognized. This can be explained by the fact that the combination of the wheel profile S1002 with the track geometry 60E1/1:40/1435 is characterized by a relatively high equivalent conicity. For the track geometry 60E1/1:40/1437, these high-frequent components are weaker, because the equivalent conicity of this contact geometry is slightly lower. For the track geometry 60E1/1:20/1435, which in combination with the wheel profile S1002 produces a very low equivalent conicity, the lateral motions are rather small for $t < 25$ s, before the coach enters a section with stronger track irregularities. For the track geometry 60E2/1:40/1436, relatively large lateral motions of the wheelset can be seen. In the previous investigation [3] it is shown that a low-frequent limit cycle, commonly known as "carbody instability" was obtained for the rail profile 60E2 with an inclination of 1:40 and a track gauge of 1.435 m at 200 km/h for both the flexible and the rigid model; this limit cycle may explain the strong lateral oscillations. Furthermore, the oscillations have a relatively low frequency, because the combination of the wheel profile S1002 with the track geometry 60E2/1:40/1436 is characterized by a relatively low equivalent conicity.

The comparison of the two curves of one diagram shows the impact of the structural flexibilities of wheelsets and rails. For the track geometry 60E1/1:20/1435, deviations between the black curve and the magenta curve are hardly visible for $t < 25$ s. This indicates that for this track geometry the impact of the structural flexibilities is very weak. A possible explanation is that for a contact geometry characterized by a low equivalent conicity the lateral wheel-rail forces, which guide the wheelset, are very

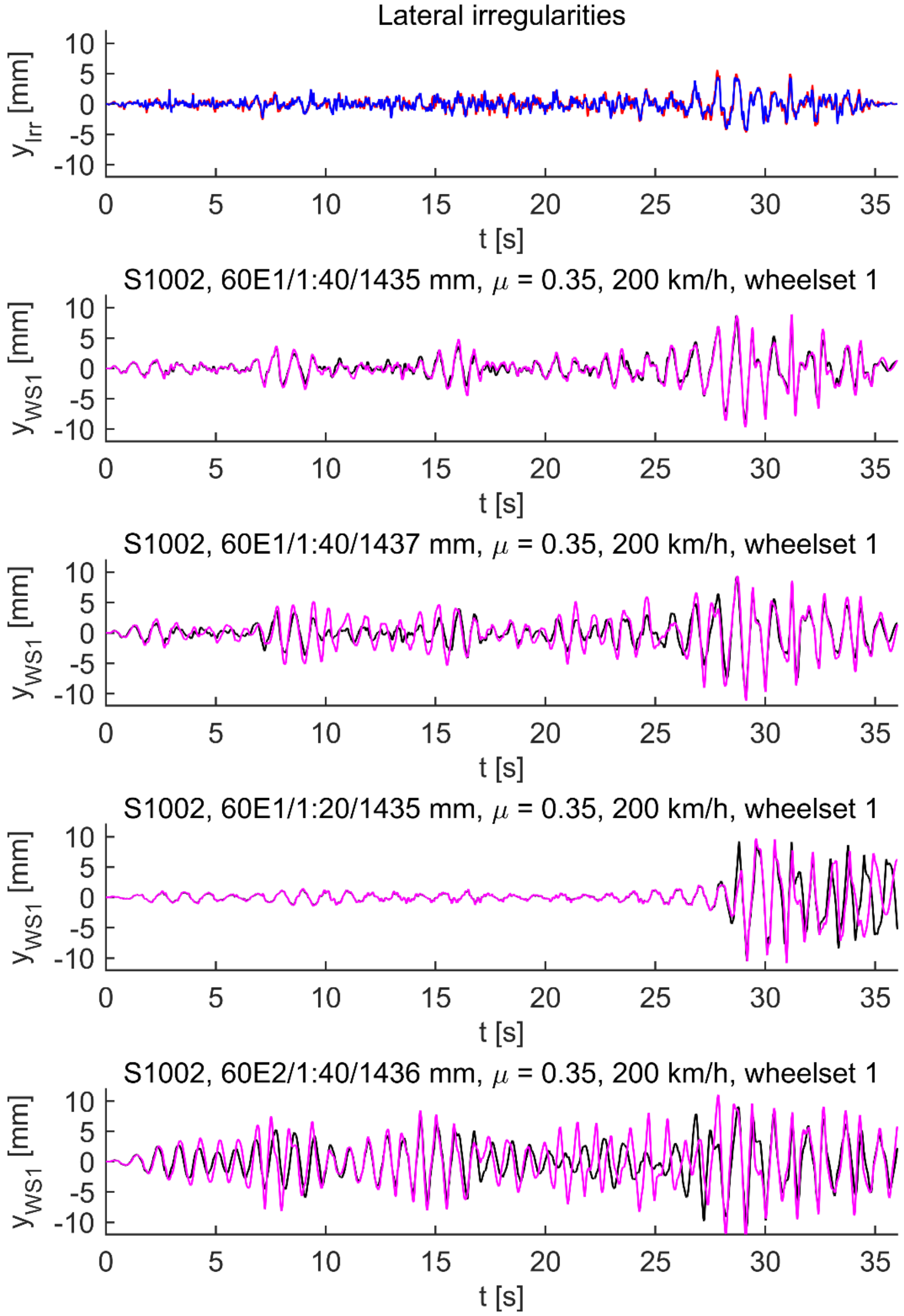

Figure 3: Lateral track irregularities and lateral displacements of the wheelset 1. Red line: left rail. Blue line: right rail. Black line: model variant "rigid model". Magenta line: model variant "flexible model".

low so that the structural deformations are very small. In contrast, for the track geometries 60E1/1:40/1437 and 60E2/1:40/1436, considerable deviations between the black curve and the magenta curve can be seen, which indicates a notable impact of the structural flexibilities of wheelsets and rails. As described in Section 2, the coupled-vehicle track model uses a very detailed model for the wheel-rail contact.

This enables a precise calculation of the stress distribution in the contacts as a direct result of the simulation of the entire system. In Figures 4, 5, 6, and 7, the pressure distributions over the running surface of the rail are displayed for the right contact of the wheelset 1 so that the “trace” of the wheel-rail contact becomes visible. These results were calculated using the flexible model and the four different track geometries. The interval $100\ \mathrm{m} \leq x \leq 900\ \mathrm{m}$ corresponds to the time interval $1.6065\ \mathrm{s} \leq t \leq 16.00668\ \mathrm{s}$ of the diagrams shown in Figure 2.
The comparison of the figures shows the strong impact of the track geometry on the pressure distribution. In Figures 4 and 5, short wavelengths can be observed for the

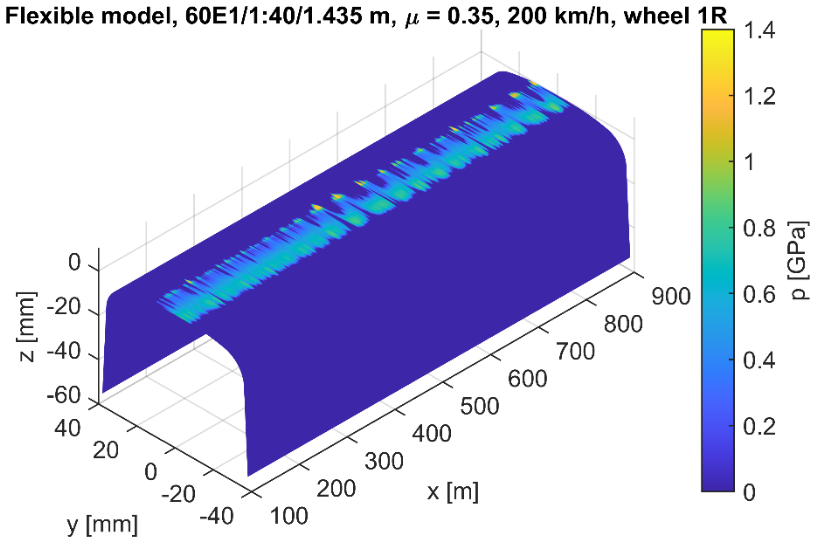

Figure 4: Pressure distribution over the rail head for the model variant “flexible model” and the track geometry 60E1/1:40/1435.

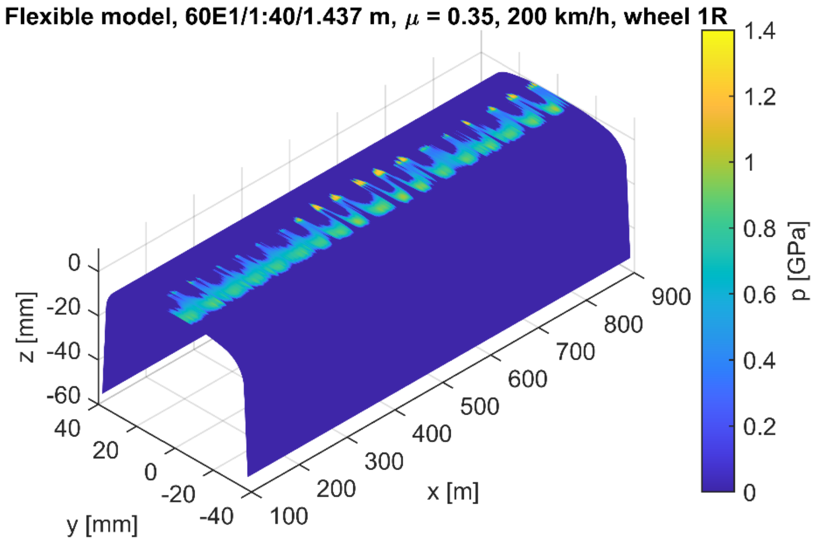

Figure 5: Pressure distribution over the rail head for the model variant “flexible model” and the track geometry 60E1/1:40/1437.

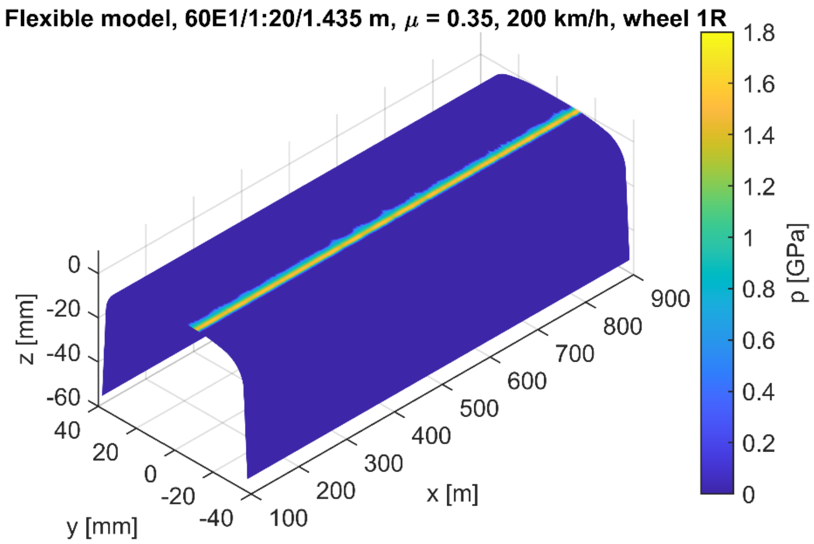

Figure 6: Pressure distribution over the rail head for the model variant "flexible model" and the track geometry 60E1/1:20/1435.

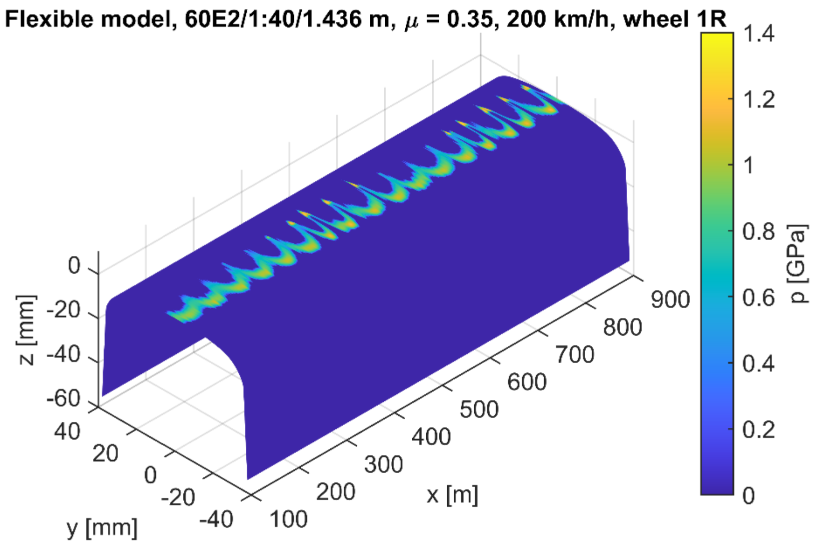

Figure 7: Pressure distribution over the rail head for the model variant "flexible model" and the track geometry 60E2/1:40/1436.

track geometries 60E1/1:40/1435 and 60E1/1:40/1437, which correspond to the high-frequent components of the wheelset's lateral motion shown in Figure 2. While in Figures 4, 5, and 7 a distinct "shifting" of the contact within a wider zone can be recognized, Figure 6 shows that for the track geometry 60E1/1:20/1435, the contact always occurs in a very narrow zone of the rail head. Because of the narrow contact area obtained for the track geometry 60E1/1:20/1435, a high contact pressure up to $p_{max} = 1.609$ GPa is obtained. For the other track geometries, wider contact areas occur so that the maximum pressure reaches just $p_{max} = 1.278$ GPa for

60E1/1:40/1435, p_{max} = 1.296 GPa for 60E1/1:40/1437, and p_{max} = 1.301 GPa for 60E2/1:40/1436.

The detailed wheel-rail contact model enables also the calculation of the friction energy density W_F/A, i.e., the density of the energy dissipated by sliding friction within the contact. This quantity is an important input for wear calculations. In Figure 2, the strongest impact of the structural flexibility can be observed for the track geometries 60E1/1:40/1437 and 60E2/1:40/1436. Therefore, Figures 8 and 9 show the distributions of the friction energy density obtained for these two geometries combined with the two model variants. The interval 350 m $\leq x \leq$ 650 m corresponds to the time interval 6.1065 s $\leq t \leq$ 11.5065 s of the diagrams shown in Figure 2, where deviations between the two curves are observed. Again, the right wheel-rail contact of the wheelset 1 is considered.

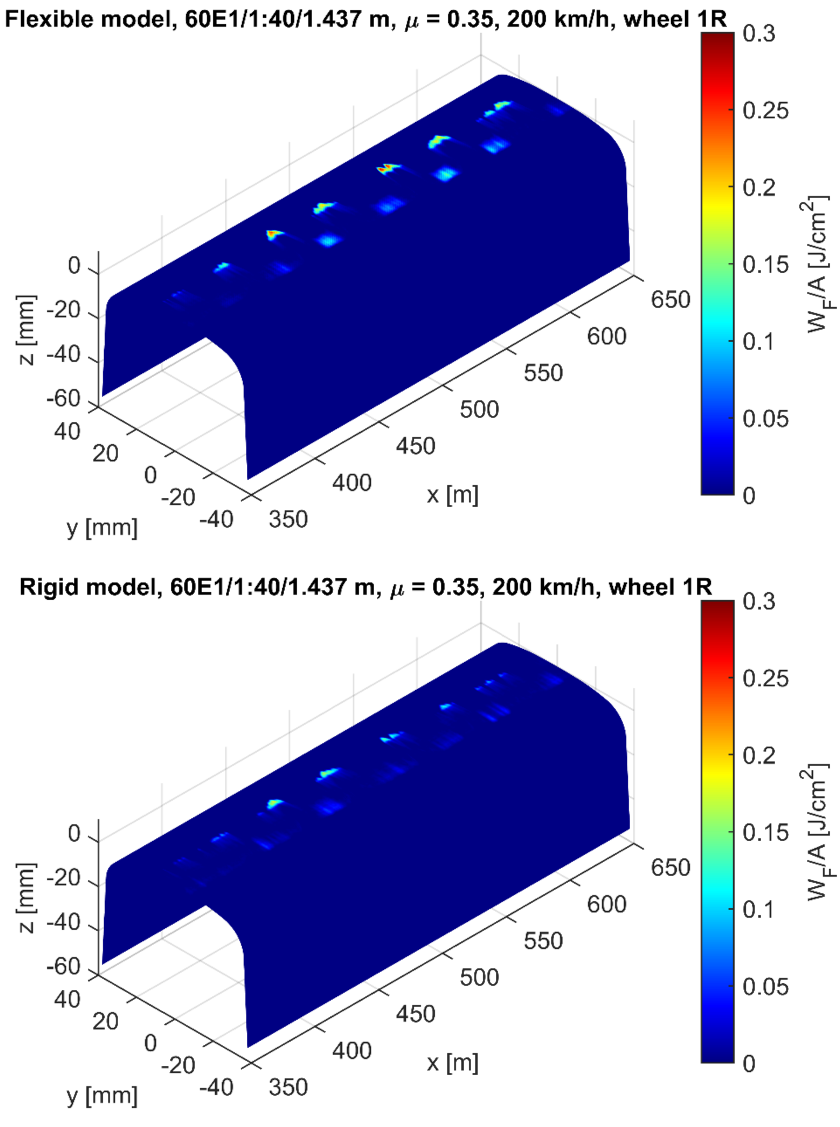

Figure 8: Distribution of the friction energy density over the rail head for the track geometry 60E1/1:40/1437.

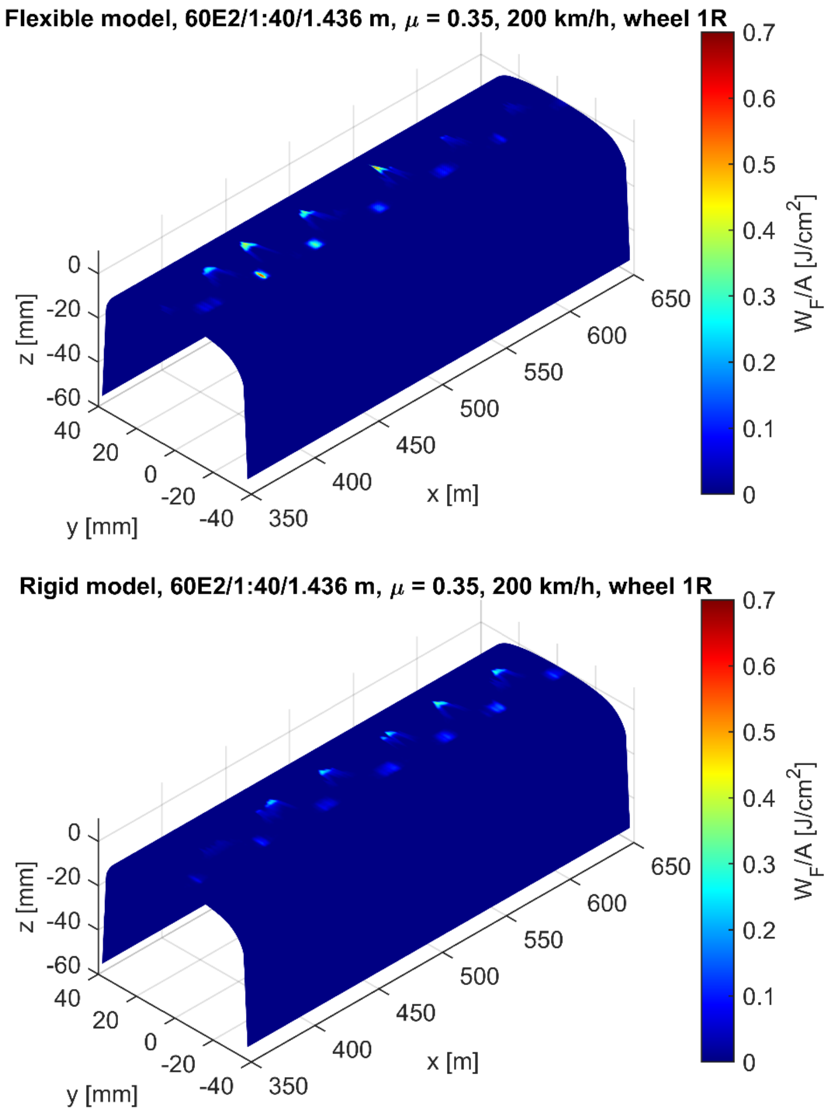

Figure 9: Distribution of the friction energy density over the rail head for the track geometry 60E2/1:40/1436.

Generally, the highest values for the friction energy density occur in the turning points for the wheelset's lateral displacement, since at these points the lateral forces, which guide the wheelset, are the highest. – In both Figures 8 and 9 it can be recognized that the red colours occur only for the results obtained using the flexible model, but not for those obtained using the rigid model. This indicates that higher values of the friction energy density are obtained for the flexible model than for the rigid model. For a direct comparison, the maximum values for the friction energy density obtained at the right contact of the wheelset 1 within the interval 350 m $\leq x \leq$ 650 m for the eight combinations of the two model variants with the four track geometries are listed in Table 2.

Track geometry	Flexible model	Rigid model
60E1/1:40/1435	$\max(W_F/A) = 0.2064$ J/cm^2	$\max(W_F/A) = 0.1906$ J/cm^2
60E1/1:40/1437	$\max(W_F/A) = 0.2800$ J/cm^2	$\max(W_F/A) = 0.1776$ J/cm^2
60E1/1:20/1435	$\max(W_F/A) = 0.0101$ J/cm^2	$\max(W_F/A) = 0.0081$ J/cm^2
60E2/1:40/1436	$\max(W_F/A) = 0.6416$ J/cm^2	$\max(W_F/A) = 0.3527$ J/cm^2

Table 2: Maximum values for the friction energy density obtained at the right contact of the wheelset 1 within the interval 350 m $\leq x \leq$ 650 m.

The comparison of the maximum values obtained for the flexible model and for the rigid model confirms the observation taken from Figures 8 and 9 that the friction energy density is generally higher for the flexible model than for the rigid model. For the track geometries 60E2/1:40/1436 and 60E1/1:40/1437 the increase of the friction energy density is particularly high, while it is moderate for the track geometry 60E1/1:20/1435 and it is small for the track geometry 60E1/1:40/1435. The moderate increase of the maximum value for the track geometry 60E1/1:20/1435 corresponds with the small deviations between the two curves depicted in Figure 2 within the considered interval.
The increase of the maximum values indicates that the structural flexibilities cause an increase of the friction energy density. A plausible explanation is that due the structural deformations of the wheelset and the rail deformational velocities occur, which contribute to the creepages in the contact.

4 Conclusions and Outlook

The enhanced coupled vehicle-track model, which has been developed in [1] and [2], has been extended in order to simulate the running of the vehicle on a track with irregularities. The successful calculation of the results presented in this work confirm the feasibility of a more detailed simulation of vehicle-track interaction using a more detailed model than the original multibody approach, in which the wheelsets and the track are modelled by rigid bodies and where usually far simpler wheel-rail contact models are employed.
The results show a strong impact of the track geometry on the running behaviour of the railway vehicle and on the wheel-rail contact itself including its location on the running surface of the rail. The issue that one and the same vehicle can show a very different dynamic behaviour on tracks with different geometries is an important aspect of the more general issue of interoperability.
Moreover, the results show that the structural flexibilities of the wheelsets and rails, too, can have a strong impact on the dynamic behaviour of the vehicle. As already shown in previous investigations [3], the strength of this impact depends on the wheel-rail contact geometry. This can be explained by the fact that the contact geometry strongly influences the wheel-rail contact forces, which guide the wheelsets and also cause structural deformations of the wheelsets and the rails. In turn, these structural deformations affect the contact kinematics. This, again, has an impact on the friction energy density, which is a relevant physical quantity for the wear occurring in the wheel-rail contact.

The model presented in this work is a useful base for further investigations. Besides enabling a more detailed analysis of the running behaviour and of the wear behaviour, it is also a useful base for developing more accurate monitoring methods like the estimation method of the equivalent conicity presented by Kaiser, Strano, Terzo and Tordela [10].
Of course, the irregularities of the track are an important factor, which strongly influences the running behaviour of the vehicle. Although the investigation presented in this work already shows certain trends of how the wheel-rail contact geometry and the structural flexibilities affect the dynamic behaviour and the wheel-rail contact, the irregularity profiles used in this work are just one example. Therefore, more calculations using different irregularity profiles are necessary in order to confirm the trends identified in this work.

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