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A Mixed Numerical Approach to Evaluate the Dynamic Behavior of Long Trains

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Abstract

The evaluation of the longitudinal forces exchanged between the wagons composing a long train is very complex due to the large number of d.o.fs to be considered and due to the non linearities introduced by the coupling elements. The most common approach to simulate long trains is the use of simplified wagon models realized considering only the longitudinal d.o.f. In this way the number of d.o.fs used for the full vehicle model is equal or a little greater than the number of the connected cars. The efficiency of this approach, in calculating the in train forces during traction and braking operations, has been demonstrated by several authors in the literature. In particular the long train simulators have been developed with the aim to evaluate the longitudinal forces during the braking operations in order to optimize the braking strategy and the mass distribution along the train. This method is efficient to optimize the train configuration in order to minimize the in train forces, but it does not allow to evaluate the vehicle safety indexes (such as derailment, wheel unload and lateral force) because the wheel-rail contact forces are completely neglected. This work shows a novel approach where the long train numerical model, realized using the Simpack multibody code, is developed considering both simplified wagon models, with few d.o.fs and no contact module, and detailed wagon models, which include several d.o.fs and the algorithm for the contact forces evaluation. In particular this mixed technique allows to evaluate both the longitudinal train dynamic and the behavior of some of the wagons when the train is running on curve. The position of the detailed wagon models along the train combination can be selected by the user in order to evaluate the influence of a particular wagon position on the vehicle safety.

Keywords: long train simulation; longitudinal dynamics; mixed numerical simulation; multibody; wheel-rail interaction

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1. Introduction

In the last years the study of the longitudinal train dynamic has involved several researches with the aim to develop numerical models able to evaluate the forces generated on the connection systems. The evaluation of long train dynamics involves different research topics such as large d.o.fs problems, friction modeling and connection system simulation. Recently the International Benchmarking of Longitudinal Train Dynamics Simulator has been published by Wu et al. (2018) with the aim to compare the long train simulators developed by researchers coming from 6 different countries. Cole et al. (2017) demonstrate that the evaluation of the in-train forces on long trains is a very complex problem due to the numerous phenomena that are involved, such as friction on coupling elements, traction and braking operations, resistance forces and load distribution along the train. Furthermore the calculation of these forces is made more complex by the huge number of wagons that usually compose the train. A state of the art regarding the long train dynamic (LTD) simulators is shown in by Wu et al (2016). The work highlights that the interest in evaluating the dynamic performance of long train exists from the beginning of the previous century and that the evolution and improvement of the numerical models is strictly related with the increasing of the computing capabilities. Wu et al. (2016), Cole et al. (2017) and Qi et al. (2012) demonstrate that the element which more affects the performance of the LTD simulators is the coupling device, which plays a fundamental role for the estimation of the in-train forces. The friction draft gear coupler, which is the more widely used, is composed by elastic and friction elements that give a nonlinear characteristic force with different loading and unloading behavior as shown by Wu et al. (2015), West et al. (1978), Cole (1998) and Qi et al. (2012). The role of this component is to transmit the load between adjacent wagons and to dump the relative longitudinal vibrations. A detailed description and state of the art of the friction draft gear has been made by Wu et al. (2014).

The numerical tools usually used to investigate the dynamics of the railway vehicle, such as Multibody codes, are optimized for short trains and/or for single vehicles, focusing their attention on detailed vehicle models and wheel-rail contact models. They are, therefore, not suitable for the simulation of long trains due to the huge number of vehicles composing the train, for which simplified and specialized codes have been developed. One of the activities more studied by researchers is the development of specific mathematical models capable of simulating the dynamic behavior of the vehicle. These are of considerable complexity since many degrees of freedom are required, non-linear elements (automatic coupler model) and discontinuity of forces due to traction and braking actions. Due to the complexity of the system, the LTD simulators usually consider only the longitudinal vehicle dynamics and the wagons, composing the train, are simulated as single rigid bodies with the only longitudinal degree of freedom. One example of LTD simulator, which only considers the longitudinal train dynamic, is TrainDy developed by Cantone et al. (2011), which allows to evaluate the in-train forces during the braking operations. The numerical model includes one module for the simulation of the brake pneumatic system and a second model for the simulation of the longitudinal train dynamics. Another example of LTD simulator, which considers the only longitudinal d.o.f., is TDEAS proposed by Wu et al. (2014). In this case the numerical model was used to evaluate the energetic efficiency of the vehicle during traction and braking operations, paying particular attention to the energy wasted by the coupling system.

All these models consider straight tracks and the resistances due to slopes and curves are modeled as longitudinal concentrated loads directly applied on the centre of mass of the wagon. This simplification, in some cases, could be inaccurate, in fact, when the vehicle is running in curve, the effective distance between the connection systems of two consecutive wagons is greater. Furthermore, during this situation, a relative rotation occurs between the wagons. For this reason Wu et al. (2012) developed a coupler model with 9 d.ofs, which is able to consider the relative rotation between the vehicles around the vertical axis (required when the train is running on curve) and the lateral axis (required when the vehicle runs on track gradients). The simulation approach of considering simplified wagons, modeled with a single d.o.f., has another important restriction, in fact, this method allows to evaluate the in-train forces with a good precision, as shown by Wu et al. (2018) and Massa et al. (2012), but it does not provide any information about the dynamic behavior of the vehicle, such as stability, derailment, wheel unloading. These phenomena can be evaluated only if the wheel-rail contact forces are known and this is possible only if the numerical model includes a specific wheel-rail contact module, such the one proposed by Bosso et al. (2012). The Universal Mechanism (UM) multibody code allows to develop detailed long train numerical models that include the module for the contact force calculation. An example of a long train numerical model developed with UM software is described by Petrenko (2016), where the author proposes a case study to correlate the longitudinal in-train forces with the risk of derailment during traction and braking operations. Another approach, which allows to obtain the contact forces only on selected wagons of the train, was recently proposed by Bosso et al. (2017). The work proposes the use of a mixed approach to model the train, where the wagons or locomotives are simulated in detail, considering many d.o.fs and the wheel-rail contact forces, while, the other wagons are modeled as single rigid bodies with the only longitudinal d.o.f. The work is limited to the study of the dynamic behavior of the second locomotive of a train composed by two leading locomotives and 50 freight wagons.

The work proposed in this paper evaluates two main aspects: the first one is the effect of considering a curved track instead of a straight track in the in-train forces, while the second one is the influence of the vehicle position on the risk of derailment. Considering the first case all the vehicles, composing the train, are simulated considering one d.o.f., while in the second case two wagons are simulated in detail while the others are modeled with one d.o.f. The models proposed in this work were developed using the Simpack 2017 multibody software.

Nomenclature

$F_{T/B}$	traction/dynamic braking force
v	vehicle speed
t	simulation time
N	spline describing traction/dynamic braking notch level
H	spline describing the traction/ dynamic braking characteristic
$F_{R,P}$	propulsion resistance force
m_a	axle-load
m_w	vehicle mass
Q	frontal resistance factor
$F_{R,C}$	curving resistance force
R	curve radius
$F_{R,G}$	gradients resistance force
g	gravitational acceleration
G	spline describing the track gradients
s	vehicle position on track
Y	lateral wheel force
Q	vertical wheel force
Y/Q	derailment coefficient
DQ	wheel vertical load/unload

2. Simulation scenario

The simulation scenario adopted in this work corresponds to the first configuration proposed by Spiryagin et al. (2017) in the International Benchmarking of Longitudinal Train Dynamics Simulator. The work considers three different numerical models:

- Simplified train model running on straight track: all the locomotives and wagons composing the train are simulated only considering the longitudinal d.o.f. and the track is composed by a single straight section. This model corresponds to the PoliTo model described in the International Benchmarking of Longitudinal Train Dynamics Simulator
- Simplified train model running on curved track: all the locomotives and wagons composing the train are still simulated only considering the longitudinal d.o.f., but in this case the track is simulated considering straight and curve sections connected by clothoid type curve transition
- Mixed train model running on curved track: the model is composed by both simplified and detailed vehicles. In particular the first and the 25th wagons composing the train are simulated in detail, while the other vehicles are

modeled with one d.o.f. The track is simulated considering straight and curve sections connected by clothoid type curve transition.

2.1. Track

The track adopted for the first model is composed by a single section of straight track 52 km long. The track used for the second and third model is composed by six different curve radii in both the directions for a total of 12 curves. In this case the track was modified with respect to the original one by inserting curve transitions, which are necessary to guarantee a realistic curve negotiation when considering the detailed vehicle model. In addition to curve transitions, superelevations were added to the track on the basis of the curve radius and the vehicle maximum speed. Tab. 1 shows an half of the track since the other part has the same layout, but with the curves in the opposite direction. The vertical layout of the track is not considered and the resistance forces due to track gradients are simulated by means of a concentrated load applied on each wagon and locomotive.

Table 1. Horizontal and superelevation layout of the curved track (half of the track).

Type	Length (*m*)	Initial radius (*m*)	End radius (*m*)	Initial superelevation (*m*)	End superelevation (*m*)	Maximum speed (*km/h*)
Straight	4020	-	-	-	-	80
Clothoid	30	0	1000	0	0.08	80
Curve	350	1000	-	0.08	-	80
Clothoid	30	1000	0	0.08	0	80
Straight	3590	-	-	-	-	80
Clothoid	30	0	800	0	0.08	80
Curve	350	800	-	0.08	-	80
Clothoid	30	800	0	0.08	0	80
Straight	3590	-	-	-	-	80
Clothoid	30	0	600	0	0.10	80
Curve	350	600	-	0.10	-	80
Clothoid	30	600	0	0.10	0	80
Straight	3590	-	-	-	-	80
Clothoid	30	0	400	0	0.12	60
Curve	350	400	-	0.12	-	60
Clothoid	30	400	0	0.12	0	60
Straight	3590	-	-	-	-	60
Clothoid	30	0	300	0	0.14	60
Curve	350	300	-	0.14	-	60
Clothoid	30	300	0	0.14	0	60
Straight	3590	-	-	-	-	60
Clothoid	30	0	200	0	0.16	60
Curve	350	200	-	0.16	-	60
Clothoid	30	200	0	0.16	0	60

2.2. Train layout

The numerical models consider a short head-end train composed of 50 wagons hauled by two head locomotives. Tab. 2 shows the main characteristics of locomotives and wagons.

Table 2. Principal characteristics of locomotives and wagons.

Vehicle type	Axle-load (*tonne*)	N. of Axles (-)	Length (*m*)	Vehicle mass (*tonne*)
Locomotive	22.33	6	22.95	134
Wagon	32	4	15	128

The vehicles are connected by means of couplers and bars, the first ones are composed by two draft gears and a coupler, while the second ones are composed by two draft gears and a rigid bar. The main difference between the two systems is that the first system allows a slack of 10 mm between the connected wagons, while the second works as a rigid connection without slack. The draft gear is a complex system composed by elastic elements and friction surfaces and in this work it is simulated using the characteristics defined by Spiryagin et al. (2017). The equivalent force-displacement characteristic that simulates the connection between wagons and/or locomotives is given by the series of the two draft gears and a coupler or a bar depending on the type of connection adopted for the two wagons. The train model adopted in this work is realized connecting the wagons in wagon pairs by means of bar elements. The locomotives and the wagon pairs are instead connected using coupler elements.

The diesel-electric locomotives have traction and dynamic braking characteristics with notch control. In particular eight notch levels are used both for traction and dynamic braking. According to the benchmark published by Spiryagin et al. (2017), used as a reference for this work, the traction/braking effort has been set as a function of time, in order to maintain a pre-selected train speed profile. The tractive and braking efforts have been modelled with a longitudinal force applied to the locomotive carbody in the same way adopted for the resistance force. The traction force is defined by the function described in Eq. 1, where N represents the notch level imposed by the driver, and it is defined as a function of the time, using integer values in the range [−8,8]. In particular positive values are used for traction end negative for braking. Obviously a notch level equal to zero is adopted for no traction or braking operation.

$$F_{T/B} = H\left(v, N\left(t\right)\right) \tag{1}$$

The value of the index allows to choose the correct motor characteristic, which is defined by a series of splines H as a function of the vehicle velocity v, measured at each time step from the track joint. N and H splines have been defined using Simpack input function sets. Since the two leading locomotives have radio-based communication a delay of three seconds has been simulated between the first and second locomotive. The delay is simulated by shifting the notch N level characteristic of the remote locomotive of three seconds.

The numerical model considers the resistance forces due to propulsion and curving resistance. These forces have been modeled as concentrate loads applied in longitudinal direction. Propulsion load of both locomotives and wagons are modeled according to Eq. 2.

$$F_{R,P} = Qm_w\left(2.943 + \frac{89.2}{m_a} + 0.0306v + \frac{0.122v^2}{m_w}\right) \tag{2}$$

In Eq. 2 Q is the longitudinal resistance factor, which is equal to 3.2 for the leading locomotive and equal to 1 for the other vehicles, m_a is the axle-load in tonne, m_w is the vehicle mass in tonne and v is the vehicle speed in km/h.

The curving resistance is simulated for all the vehicles (both locomotives and wagons) according to Eq. 3, where R is the curve radius.

$$F_{R,C} = m_w\left(\frac{6116}{R}\right) \tag{3}$$

The track does not include gradients since they are simulated by means of a longitudinal resistance force applied on the vehicle carbody. This force is modeled according to Eq. (4) for all the vehicles composing the train. The spline *G* describes the track gradients as a function of the vehicle position on track *s*, which is evaluated directly from the carbody joint during the simulation.

$$F_{R,G} = m_w g G(s) \tag{4}$$

The vehicle speed profile is hence controlled during the simulation by the equilibrium between the traction force and the resistance forces. Fig. 1 shows the vehicle speed during the simulation and the vehicle maximum speed on the considered track.

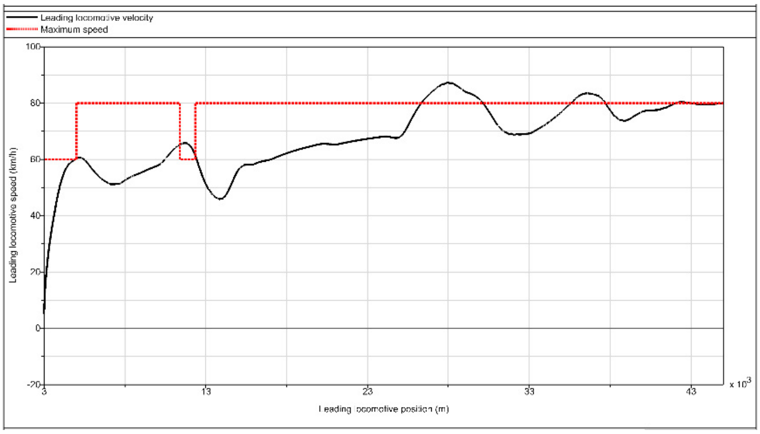

Fig. 1. Leading locomotive speed profile during the simulation.

Numerical model

This section describes the numerical models of wagons and locomotives used to build the whole train model according to the composition described in the previous section. As already described the vehicles are connected each other by means of couplers and bars. The vehicle models used in this work are three: simplified locomotive model, simplified wagon model and detailed wagon model.

2.3. Simplified locomotive/wagon model

The scheme shown in Fig. 2 describes the simplified vehicle model, which is used both for locomotives and wagons. In order to keep into account the different behavior of the two types of vehicle the relevant parameters, such as mass and dimensions, were modified.

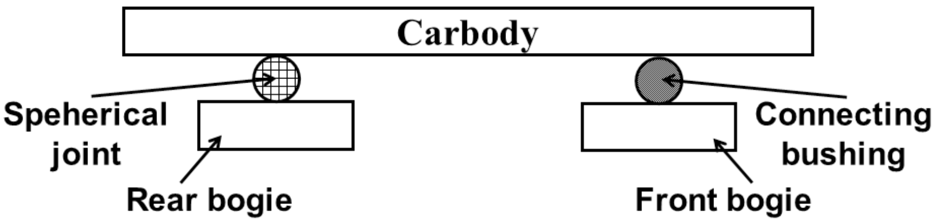

Fig. 2. Scheme of the simplified vehicle model.

The numerical model is realized considering three rigid bodies, which represent the two bogies and the carbody. The first two bodies are constrained to the track by means of the Simpack "general rail track" joint, but only allowing the longitudinal d.o.f.. The carbody is, instead, constrained to the rear bogie by means of a spherical joint, which allows the three rotations. The connection between the front bogie and the coach is instead modeled using a bushing force element, which has high stiffness translational components in order to simulate a rigid connection. This strategy is required since the adopted code (Simpack 2017) only allows one joint per body. Although this model is intended for longitudinal dynamic studies, the height from the top of the rail of the parts of the model and of the center of mass has been defined realistically, since this model will be integrated with a detailed wagon model. Since the simplified model of the vehicle does not include the antiroll system, two rotational stiffnesses have been applied between each bogie frame and the carbody. As regards the front bogie this stiffness is modeled using the bushing element used for the connection of the bogie to the carbody, while for the rear bogie a bushing element, with only the rotational stiffness around the longitudinal axis, is used to connect this bogie and the carbody. The anti-roll stiffness has been chosen in order to allow a roll angle less than 6 deg with a lateral acceleration of 1 m/s^2. The resistance forces due to propulsion, curve resistance and gradients are modeled by means of a point to point force element defined between the carbody center of mass and a marker belonging to the main reference system. The last one is defined as a "follow track joint" marker, which has the characteristic to modify its orientation according to the orientation of a specific joint. This type of marker is necessary when considering a curved track in order to assure that the resistance force is always parallel to the track center line.

2.4. Detailed wagon model

The freight wagon, modeled in detail, is composed by a carbody and two three-piece bogies. The bogie consists of three essential parts, the two side frames and the bolster. It is therefore not equipped with a rigid bogie frame, on the contrary, this particular structure allows the bogie to overcome slants and to easily negotiate small radius curves. The wheelsets of the bogie are connected to the side frames or rigidly or by means of adapter elements that usually has a concentrate stiffness. The two side frames support the bolster by means of the secondary suspension level that is composed by a series of helical springs that work in lateral and vertical direction. The vertical oscillations of the bolster are controlled and damped by means of friction elements that are called friction wedges. The damping is obtained by means of two wedges, each one connected to the side frame by a set of two or three concentric helical springs. The two wedges have friction surfaces both on the vertical and on the oblique sides. These ones slide on friction surfaces realized on the bolster, generating a damping effect on the bolster. The friction wedges play a fundamental role on the dynamic behavior of the vehicle. The bogie is connected to the carbody by means of a cylindrical center pin and the roll motion is controlled by two friction side bearers, which are also used to damp the yaw oscillations of the bogie.

The vehicle model includes 19 rigid bodies: 1 carbody, 2 bolsters, 4 side frames, 4 wheelsets and 8 axle-boxes. All the bodies with the exception of the axle-boxes are connected to the main reference system by means of the Simpack "general rail track" joint, which allows 6 d.o.fs. The axle-boxes are constrained to the wheelsets by means of a revolute joint that only allows the rotation around the lateral axis. Tab. 3 shows the inertial properties of the rigid bodies.

Table 3. Inertial properties of the rigid bodies adopted for the detailed wagon model.

Inertial property	Value	Unit
Coach mass	122500	Kg
Coach inertia I_{XX}	57200	Kgm^2
Coach inertia Iyy	4033777	Kgm^2
Coach inertia Izz	4072636	Kgm^2
Bolster mass	365	Kg
Bolster inertia I_{XX}	175	Kgm^2
Bolster inertia Iyy	10	Kgm^2

Bolster inertia Izz	176	Kgm^2
Side frame mass	448	Kg
Side frame inertia I_{XX}	100.4	Kgm^2
Side frame inertia Iyy	116	Kgm^2
Side frame inertia Izz	116	Kgm^2
Wheelset mass	1500	Kg
Wheelset inertia I_{XX}	563	Kgm^2
Wheelset inertia Iyy	134	Kgm^2
Wheelset inertia Izz	563	Kgm^2

Tab. 4 shows the principal geometric characteristics of the wagon.

Table 4. Principal geometrical characteristics of the wagon.

Geometrical characteristic	Value	Unit
Gauge	1435	mm
Bogie wheelbase	1.7	m
Wheel nominal radius	0.46	m
Coach length	15	m
Distance between the bogies	9.15	m

The friction wedges are modeled using “stick-slip” force elements. The "stick-slip" phenomenon regards the sliding friction and can be described as the motion that develops between two contact surfaces, alternately adhering (stick) and sliding (slip) between them, with a corresponding change in the friction force. In fact, typically the static friction coefficient is higher than the dynamic one. If the force applied in the tangential direction with respect to the two contact surfaces is high enough to overcome the static frictional force, then the reduction of the frictional force can generate a speed discontinuity. The presence of the friction contact and the consequent "stick-slip" phenomenon make the dynamics of the secondary suspension difficult to predict a priori. For this reason a model of only the secondary suspension was developed with as many details as possible to describe the dynamics with greater accuracy, see Fig. 3. The model was used to characterize the friction wedge model and to develop an equivalent and simpler force element to introduce in the complete vehicle model. The detailed model of the friction wedge it was not directly included in the complete vehicle model due to the complexity an the huge number of markers necessary to model the friction surfaces.

The detailed vehicle model has been included in the long train model by replacing two of the simplified wagons. In particular the first and the 25th wagons of the composition were replaced and included in the long train model. Fig. 4 shows the 25th detailed model of the wagon included in the long train model.

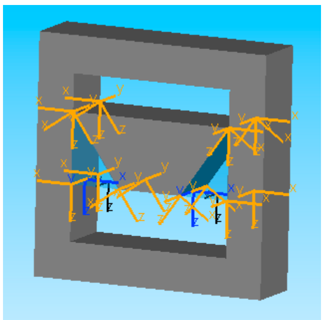

Fig. 3. Scheme of the detailed friction wedge model.

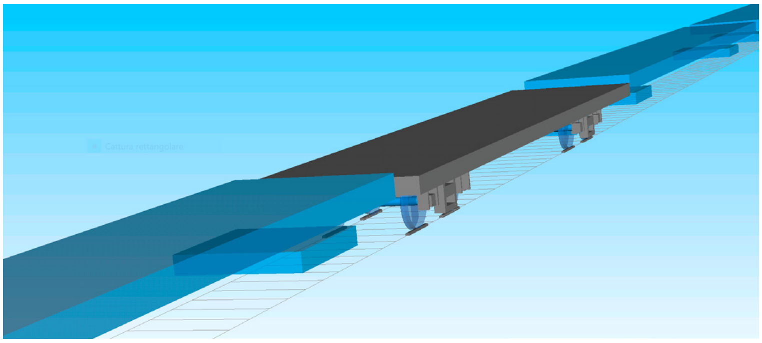

Fig. 4. Detail of the long train model included in the long train composition.

3. Numerical results and discussion

The results shown and discussed in this section evaluate two principal aspects related to the simulation of long trains. The first analysis evaluates the effect of considering a curved track instead of a straight track in the in-train forces. The main difference is related to the relative rotation between the draft gears connecting two wagons when the train is running on curve. As regards this analysis all the vehicles are modeled with the simplified approach and two simulations are performed: one considering a straight track and the other one considering a curved track.

The second case study evaluates the strategy of using a mixed simulation technique to get information about the dynamic behavior of selected wagons belonging to a long train. The strategy consists of replacing some of the simplified wagons of the train with a detailed wagon model. In this way it is possible to correlate the longitudinal in-train forces with the risk of derailment and to evaluate the effect of vehicle position in the wagon dynamic behavior.

3.1. Longitudinal dynamic

This section shows the results involving longitudinal dynamics and focuses on the analysis of the in-train forces generated by the coupler and bar elements. A different numerical sequence is used to identify coupler and bar elements. The first coupler connects the leading locomotive to the remote one, the second coupler connects the remote locomotive to the first wagon pair, the third coupler connects the first wagon pair to the second one and so on. As regards bar elements, which are used to joint two wagons in a wagon pair, they are numbered starting from the wagon pair closer to the remote locomotive.

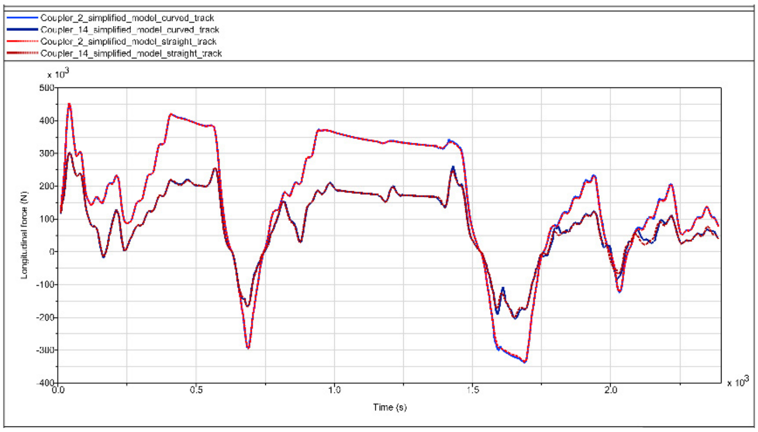

Fig. 5. Longitudinal in-train forces generated by couplers 2 and 14 when considering straight track (solid line) and curved track (dashed line).

Being the train composed by two locomotives and fifty wagons a total of 26 coupler elements and 25 bar elements are used. Fig. 5 shows a comparison of the longitudinal force generated by coupler 2 (connecting the remote locomotive and the first wagon pair) and coupler 14 (connecting the 12th and 13th wagon pair) considering the simplified model when considering straight track (solid line) and curved track (dashed line). The plot shows that, as expected, the forces generated by the second coupler are always higher than the ones generated by the coupler 14. This is obviously due to the head-end train configuration which causes a traction effort gradually lower from the first to the last coupler. Comparing the curves obtained considering the two tracks not remarkable differences can be observed.

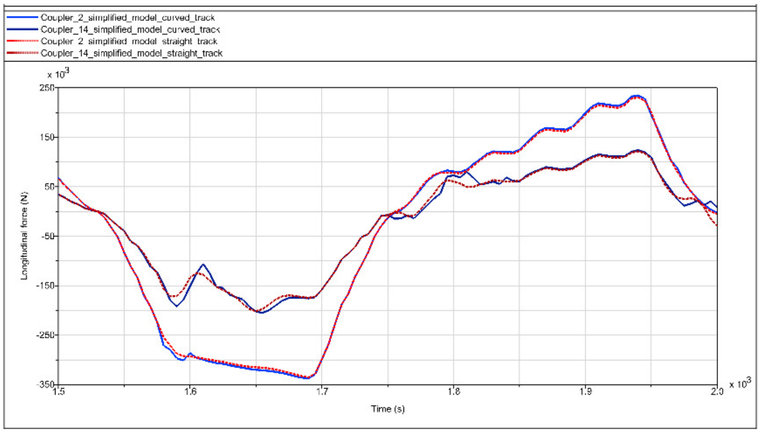

Fig. 6. Detail of the longitudinal in-train forces generated by couplers 2 and 14 when considering straight track (solid line) and curved track (dashed line).

Some more important differences can be detected at the end of the track, where the vehicle speed is higher. These differences can be observed in Fig. 6, which has been obtained by zooming Fig. 5.

The points where higher differences occur correspond to curve transition section, where the wagons composing the vehicle modify their orientation. Furthermore, the in-train forces on curved track are higher than in straight track, since the relative rotation between the wagons modifies the distance between the markers where the coupler is defined. In particular the distance between the markers is higher when the vehicle is running on curve and this generates an higher force value. A further comparison is shown in Fig. 7 where the simplified (solid line) and detailed (dashed line) model are compared in term of in-train forces.

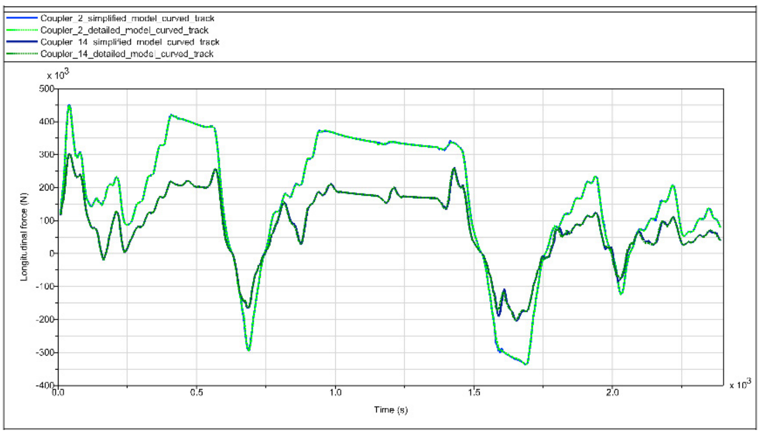

Fig. 7. Longitudinal in-train forces generated by couplers 2 and 14 considering the detailed (dashed line) and simplified (solid line) train model.

Also in this case the agreement between the two models is very good. Fig. 8 shows a zoom of Fig. 7 where some differences between the two models are evident. In this case some differences occur in curve transition, but the two models agree in curve sections, since both the models consider a curved track and consider the same distance between the markers used to define the coupler force. The results shown in this section allow to conclude that the use of a simplified model, considering one d.of., allows to estimate the in-train forces with a good precision. Some small differences can be appreciated only when the vehicle is running in curve transitions.

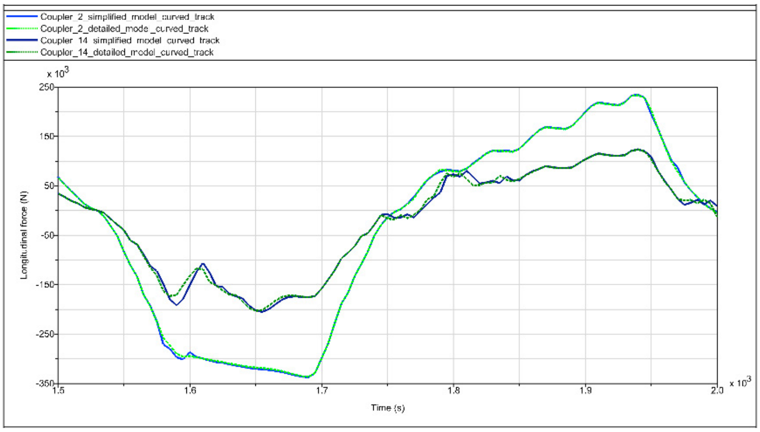

Fig. 8. Detail of the longitudinal in-train forces generated by couplers 2 and 14 considering the detailed (dashed line) and simplified (solid line) train model.

3.2. Vehicle track interaction

The use of a mixed simulation technique allows to have information about the dynamic behavior of selected vehicles composing the train. In this way, it is possible to investigate the effect of the train composition on a specific vehicle. We can observe that usually a vehicle is designed by the dynamic point of view, using multibody codes and considering the vehicle as a single entity, but the composition of the train can have an influence on the vehicle dynamic, and therefore on the vehicle safety. Fig. 9 shows the derailment coefficient *Y/Q* measured on the detailed wagon which is installed near the remote locomotive (1st wagon).

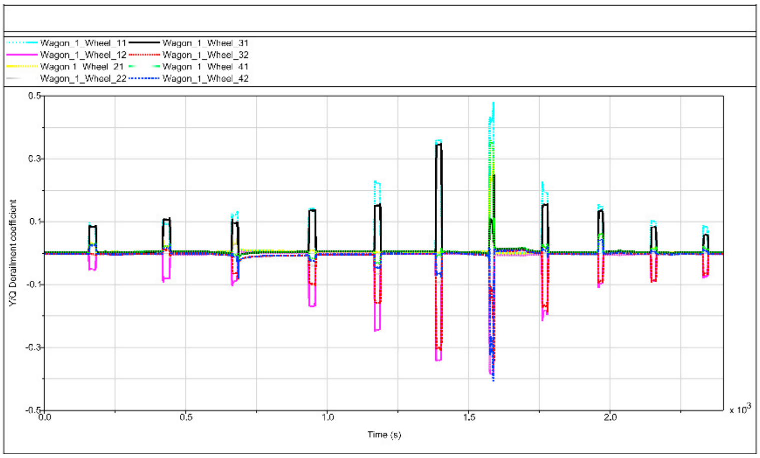

Fig. 9. Derailment coefficient *Y/Q* on the detailed wagon near the remote locomotive (1st wagon).

As regards the notation adopted in the plot legend the first index indicates the axle number and the second index describes the wheel side, 1 is used for the right side and 2 for the left side. Fig. 10 shows the *Y/Q* derailment coefficient measured in the detailed wagon model installed in the middle of the train (25th wagon). Comparing the results it is possible to observe that the *Y/Q* ratio is almost the same for the two vehicles although some difference can be observed, especially in the fifth and sixth curves.

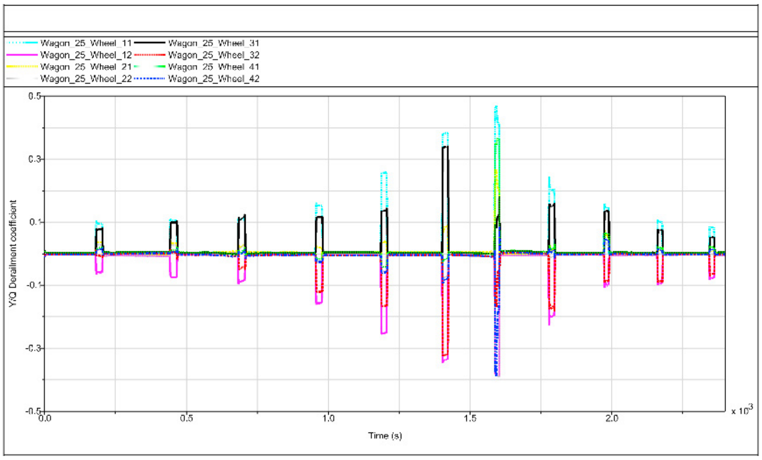

Fig. 10. Derailment coefficient *Y/Q* on the detailed wagon in the middle of the train (25th wagon).

Fig. 11 and Fig. 12 show the unloading ratio *DQ/Q* measured on the two wagons modeled in detail. In this case, instead, the results are different, in fact, the vehicle in the middle of the composition shows an higher unloading ratio in the critical curve (7th curve). Another important effect of the vehicle position is that the critical wheel for wheel unloading changes and depends from the position of the vehicle inside the composition, as can be seen for example in the 5th and 6th curve.

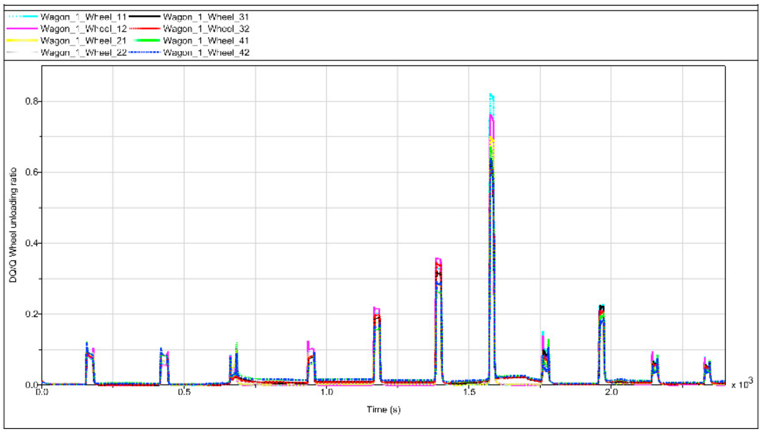

Fig. 11. Wheel unloading ratio *DQ/Q* on the detailed wagon near the remote locomotive (1st wagon).

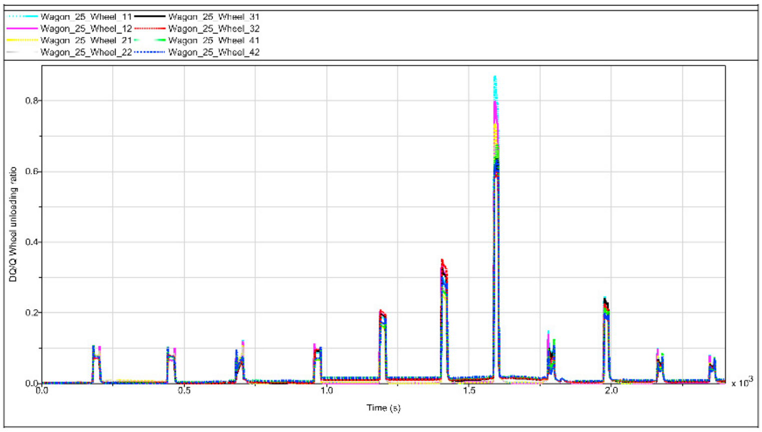

Fig. 12. Wheel unloading ratio *DQ/Q* on the detailed wagon in the middle of the train (25th wagon).

4. Conclusions

The work analyses a long train using the multibody code Simpack. The benefit of the proposed approach consists in the possibility to realize mixed models of the train, with some vehicles modeled using a simplified approach, and some others modeled in detail. The results of this paper show that the strategy adopted for the model allows to investigate the effects on a single vehicle inside a long train reducing the computation effort. Furthermore the results show that the in-train forces are few influenced by curves and the number of d.o.fs used to describe the vehicle composing the train. The results support the model actually used for long train simulation that are usually based on simplified model with one d.o.f. Another important result of this work is that the position of the vehicle inside the composition can have an important role in the dynamic behavior of the vehicle and can modify the critical wheelset or wheel of the wagon. This aspect should be considered during the design of railway vehicles, that are usually based on the simulation of the single vehicle, and particular attention should be paid to the different positions that the vehicle can assume during the train composition.

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