Simulation analysis of the short-term impact of acceleration and braking of a railway traction unit on its guiding forces in a curved track

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**Abstract:** Travel time by rail is related to the achievable commercial speed on a given route; it is also related to the appropriate use of the available acceleration and braking capabilities of the vehicle. Assuming different values of longitudinal accelerations and decelerations of a light rail vehicle achieved in practice in the range of 0.4-1.2 m/s2, possible differences in travel time and commercial speed were estimated. Then, an attempt was made to examine the impact of changing the driving speed with the maximum allowable acceleration on the dynamic interactions between the vehicle wheels and the track while the vehicle is moving in the curve with a radius of 300 m. The values of contact forces and the derailment index of the leading wheelset were taken into account when driving along a curve with uniformly variable speed and constant longitudinal acceleration resulting from traction and braking. The configuration of the vehicle model includes three connected units with the outermost motor cars and the middle rolling car. Acceleration and braking is carried out by the action of torques on the appropriate axles of the wheelsets with a magnitude that ensures a constant value of acceleration or deceleration. The computer model of the traction unit was implemented in a commercial package for the simulation of multi-body systems.

**1. Introduction**

In rail transport, as in other modes of transport, there is a link between travel time, energy consumption, passenger comfort and safety. Safety has strict requirements that must be met, but consideration of other factors depends on the priorities or trade-offs made in the transport process. From the perspective of train motion theory, travel time depends on the accelerations and speeds that can be achieved. A graph of changes in train speed over time is a convenient illustration of the different phases of a train journey. A typical speed-time curve (Fig. 1) consists mainly of the following elements: 1. Initial constant acceleration, 2. Further acceleration, 3. Running at constant speed (free running), 4. Coasting, 5. Braking. To simplify calculations, the speed change graph can be approximated by a trapezoidal curve with constant acceleration and deceleration values. Since the area under the curve represents the distance travelled, the area of the trapezoidal curve should be chosen to be equal to the area of the actual curve. The travel time between stops can be calculated using the formula (Eq. 1):

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| --- | --- |
| $$t=\frac{d}{v\_{m}}+\frac{1}{2}v\_{m}\left(\frac{1}{a}+\frac{1}{b}\right)+t\_{w},$$ | (1) |

where: $t$ – travel time (s), $d$ – distance between stations (m), $v\_{m}$ – maximum speed (m/s), $a$,$b$ – absolute acceleration and braking deceleration (m/s2), $t\_{w}$ – waiting time at the train stop (s).

t (h)

v (km/h)

2

1

3

4

5

0

t1

t2

t3

Fig. 1. Train speed-time profile and its approximation with a trapezoidal model. 1. Initial acceleration, 2. Further acceleration, 3. Free run, 4. Coasting, 5. Braking

We will determine the commercial speed by dividing the total distance by the total travel time (including waiting time) $d$/$t$. The maximum acceleration is limited by the available engine power and the maximum traction force that can be transferred by the wheels to the rails (Fig. 2, Eq. 2):

|  |  |
| --- | --- |
| $$a\_{max}\left(v\right)=\frac{P}{m∙v}\leq \frac{F\_{max}\left(v\right)}{m},$$ | (2) |

where: $P$ – engine power transmitted to wheel sets (kW), $v$ – speed (m/s), $m$ – train mass (kg), $F\_{max}$ – maximum traction force (N). The maximum braking deceleration with a friction brake is limited mechanically by the adhesion conditions (Eq. 3):

|  |  |
| --- | --- |
| $$b\_{max}\left(v\right)\leq μ\left(v\right)g,$$ | (3) |

where: $μ$ – friction coefficient depending on the speed (e.g. Parodi's formula), $g$ – acceleration due to gravity.

Speed(km/h)

Force

(N)

0,0

*F(*$v$*)*

0

$v$max

Adhesion limit

Train motion resistance

Fig. 2. General characteristics of the driving force *F(v)* on the axles of a rail vehicle

Another limitation of the longitudinal accelerations and decelerations permissible values is related to the passengers' driving comfort and their individual safety, which may be threatened due to loss of balance in a standing position or loss of support in a sitting position. Research conducted on the permissible values of these accelerations uses various subjective methods (Guo, Gan, & Fang, 2015; Powell & Palacín, 2015). Therefore, they only provide general guidance on what may be considered acceptable or unacceptable acceleration levels. It is worth noting that the results of these studies are susceptible to individual opinions and interpretations. Nevertheless, research confirms that both jerk and acceleration impact on passenger comfort and stability. Passengers standing without support and facing the vehicle's acceleration are particularly sensitive to these factors. This means that people in this position have the least tolerance for acceleration fluctuations that can negatively impact their travel experience. In practice, there are passenger rail vehicles of various types and purposes, which are operated with acceleration of 0.3–1.3 m/s2 and service braking decelerations of 0.7–1.5 m/s2. In special situations of higher necessity, higher emergency braking deceleration values of up to 3 m/s2 are declared, but it is stated that they are used only as a last resort because that they carry a high risk of individual injuries to passengers. The provisions of the TSI (Komisja Europejska, 2019) state that the greatest average deceleration obtained after applying all brakes, including the wheel/rail brake independent of adhesion, must be less than 2.5 m/s2; this requirement is related to the longitudinal resistance of the track. The maximum jerk resulting from using the brakes must be less than 4 m/s3 (Komisja Europejska, 2019). For comparison, in passenger road transport, the actual values occurring during bus operation may be much higher. The average values reported in (Frej, Grabski, Jurecki, & Szumska, 2023) may be 2.1 m/s2 for acceleration manoeuvres and 2.7 m/s2 for braking, while the maximum values may be 4.9 m/s2 for acceleration and 6 m/s2 for braking. In passenger air transport, during take-off, longitudinal accelerations of the order of 3.5 m/s2 do not cause discomfort to passengers sitting facing the take-off direction, although getting up from the seat may be difficult.

Taking into account typical values of operational accelerations in rail transport, the difference in travel time was estimated for the selected permissible speed and the selected length of the track section between stations. The trapezoidal velocity profile model described earlier was used. The input data was adopted based on the table: average waiting time 30 s, maximum speed of metro trains 70 km/h, average distance between stations 1.1 km.

Tab. 1. Train service parameters, based on (Madej, 2004) and utk.gov.pl

|  |  |  |  |
| --- | --- | --- | --- |
| Train type | Vmax(km/h) | Distance between stops (km) | Range of longitudinal accelerations (m/s2) |
| Freight | 60$÷$80 | - | 0.15-0.2 |
| Passenger express | 120$\rightarrow $ | 40$÷$60 | 0.35$÷$0.5 |
| Passenger local | 70$÷$120 | 2$÷$8 | 0.3$÷$0.4 |
| Long-distance train set | 100$\rightarrow $ | 20$÷$60 | 0.4$÷$0.6 |
| Local train set | 80$÷$120 | 1.3$÷$3 | 0.6$÷$0.8 |
| Metro train | 60$÷$90 | 0.6$÷$1.1 | 0.8$÷$1.2 |

The calculation results are presented in Fig. 3. With acceleration values lower than 0.4 m/s2, the vehicle does not achieve the assumed maximum speed over the assumed length of the 1.1 km section, taking into account the need to maintain the braking distance. Considering relationship (Eq. 1), the impact of the acceleration value on the travel time is greater the smaller the distances between stations.

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Fig. 3. Example of calculations of the longitudinal acceleration influence on the travel time and commercial speed of metro vehicles. Average distance between stations 1.1 km, maximum speed 70 km/h, average stopping time 30 s

**2. Modelling a railway vehicle moving with longitudinal acceleration**

Ignoring other resistances to movement, the uniform change in vehicle speed results from the operation of the braking or drive system. Advanced train braking modelling requires the simulation of a dynamic hydraulic system that runs in parallel to the simulation of the train itself. The output from the braking system simulation is the force in the brake cylinder, which is converted into a retarding force. There are many types of brakes (Fig. 4) and many methods of modelling the braking system, one of them is the most commonly used pneumatic model.



Fig. 4. Brakes in rail vehicles

The model of the pneumatic braking system takes into account parameters such as pressure in the air conduit, force in the brake cylinder, friction coefficients of brake pads, etc. These parameters affect the retarding force, which is one factor in the sum of retarding forces during braking (Cole, 2019). The braking system model should also include accurate brake characteristics (such as brake pad friction coefficients vs temperature), to ensure an accurate representation of the actual braking behaviour of the train. It may also be important to limit the movement of the wheelset suspension relative to the bogie frame due to the pressure of the brake pads, in particular the yaw angle (Yang, Kang, Luo, & Fu, 2015). In the most commonly used pneumatic brake, the speed of impulse propagation in the main conduit is important for longitudinal and transverse interactions between wagons, especially on a curved track. In the case of long trains, especially when transporting heavy loads, where the length of the train may be several kilometres, the propagation of the brake signal may take several seconds, which favours the occurrence of high compressive forces in the train, which in a curve may lead to the wagon being pushed laterally off the track (Durali & Shadmehri, 2003; Günay, Korkmaz, & Özmen, 2020; Zbieć, 2018; Zhou, Ye, & Hecht, 2022). Actual deceleration and braking distance values have significant tolerances, even with adequate traction. This is influenced by the variable friction of brake pads and brake discs, depending on the temperature of the discs and external parameters such as temperature and humidity (Gerlici, Gorbunov, Kravchenko, Prosvirova, & Lack, 2017; Kukulski, Wolff, & Walczak, 2023). Additionally, brake system tolerances, brake actuator efficiency and wheel wear also affect braking power. Therefore, these tolerances are considered in the design of braking systems to ensure that the train stops within the maximum permitted emergency braking distance, even in the most challenging conditions. Reducing the impact of these tolerances can be achieved by the deceleration control function, which immediately responds to differences between the actual and desired braking value, which significantly reduces the variability of the braking distance and prevents it from being exceeded (Fig. 5).

Speed (km/h)

Braking deceleration

 (m/s2)

1.5

0.0

Target value

0

vm

0.2

 without acceleration control

 with acceleration control

Fig. 5. General characteristics of the braking delay of a rail vehicle

In order to ensure effective braking, modern passenger trains use braking systems in which the braking process is controlled by an electric signal, which enables quick and uniform activation of the brakes along the entire length of the train (Wolfram, 2003).

In practice, for short trains and short braking periods, braking modelling can be simplified because there is no need to take into account long brake signal propagation times or complex interactions between multiple wagons, and the effect of temperature in the short term will also not be significant. In such a case, simpler braking models may be sufficient to simulate and analyse the train's braking behaviour.

As in the case of the braking system model, advanced modelling of train acceleration requires taking into account many factors, such as locomotive power, track characteristics, air resistance, rolling resistance, characteristics of traction motor (Paul, Han, Chang, Chun, & Lee, 2022). The motive power of a locomotive is usually described as a function of speed, and drag characteristics are usually described as functions of speed and train mass. A simplified train acceleration model can only consider the basic factors affecting the train's movement. One example of a simplified train acceleration model is the uniform acceleration model, in which the driving force is constant, and resistance is described as a linear function of the speed and mass of the train. In this model, the train’s acceleration has a constant value, and the train's speed is described as a function of time. In this work, simplified models of braking and accelerating the train were used, in which the torque acting on the appropriate wheelsets is a smooth and continuous function of time, including sections of increasing and decreasing values with a constant gradient and a section with a constant value. Smoothing was applied to the section joints. The maximum torque value (Eq. 4) is in the range of 5200-7000 Nm per wheelset, and is selected to ensure a specific value of the vehicle's longitudinal acceleration.

|  |  |
| --- | --- |
| $$M\_{max}=\frac{F\_{max}∙r}{η∙i},$$ | (4) |

$M\_{max}$– torque, $r$ – wheel rolling radius, $η$ – efficiency, $i$ – gear ratio.

When braking the wheelset, its yaw angle is not limited by the interaction of the braking system with the wheels of the wheelset.

**3. Simulation analysis of contact interactions between vehicle wheels and the track**

Simulation calculations were carried out on a route consisting of a section of straight track 110 m long, a transition curve 20 m long, a regular curve with a radius of 300 m, a superelevation of 0.1 m and a length of 100 m, an initial transition curve 20 m long and a straight track section 74 m long. The transition curves are slightly shortened compared to the values required by the regulations to shorten the simulation process because the analysis concerns the circular arc section itself. The choice of travel speed is dictated, on the one hand, by safety regulations regarding train traffic on a track containing a curve with a small radius of 300 m, on which, in such conditions, the speed of the traction unit may be limited in the range of 40-60 km/h. On the other hand, the desire was to conduct research where the wheel-rail interaction forces are relatively high and allow for the analysis of phenomena in ranges close to the limit values. In the case of acceleration, the driving speed increased to 72 km/h, which can be considered the permissible limit (Komisja Europejska, 2019; PKP PLK, 2017), while during braking the speed drops to 48 km/h.

Initially, the vehicle moves at a constant speed of 60 km/h, and after 5 seconds, it begins to change its speed due to the torque acting on the wheelsets of the motor cars, the value of which was selected so that the longitudinal acceleration of the vehicle is $1$ m/s2. During braking, the braking torque is slightly lower at a deceleration of $-1$ m/s2 because all the wheelsets are braking. The torque action lasts for 3.5 seconds and ends at 8.5 seconds of the simulation. The dry friction coefficient between wheels and rails is 0.4. Some technical data of the vehicle are presented in Tab. 2, the suspension parameters were partially based on the ERRI B176 test vehicle. The vehicle model was implemented in the SIMPACK package environment. The configuration of the vehicle model includes three connected units with the outermost power wagons and the middle rolling wagon (Fig. 6). Between the wagons, there is a centrally placed coupling containing a spring and damping element. The force acting in the coupling depends on the value of the relative velocity vector and the relative displacement of points located at the ends of adjacent wagon bodies.

Fig. 6. Model of a three-unit train set

Tab.2. Vehicle technical data

|  |  |
| --- | --- |
| **Parameter** | **Value** |
| Total weight of the trainset | 133,5 t |
| Spacing of the turning pins | 19 m |
| Wheelbase in bogies | 2,56 m |
| Wheel radius (wheel/rail profile S1002/UIC60) | 0,46 m |
| Number of driven axles | 8 |
| Total length | 74 m |

The total force acting in the plane of contact between the vehicle wheel and the rail can be presented as components acting along ($T\_{x}$) and across the track ($T\_{y}$). The values of these components can be treated as independent of each other until the total tangential force reaches the saturation level. After reaching the saturation level, the relationship (Eq. 5) between the values of these components means, according to Coulomb's law, that they are mutually dependent and limited by the adhesion conditions (Eq. 6). The values of constant coefficients and the shapes of friction characteristics in rolling contact in the presence of relative slip may vary significantly depending on the condition of the surface and other conditions, these issues are still the subject of research (Vollebregt, Six, & Polach, 2021).

|  |  |
| --- | --- |
| $$T=μ\_{t}∙N; μ\_{t}=\frac{\sqrt{T\_{x}^{2}+T\_{y}^{2}}}{N},$$ | (5) |

$T$ – total tangential force depending on the relative slip, $N$ – normal force, $T\_{x},T\_{y}$– longitudinal and transverse components of the tangential force, $μ\_{t}$ – traction coefficient means the adhesion used in the conditions of traction force, $μ$ – adhesion coefficient means the adhesion possible to obtain, $μ\_{d}$ – coefficient of kinetic friction, $μ\_{s}$ – coefficient of static friction.

|  |  |
| --- | --- |
| $$μ\_{t}\leq μ\leq μ\_{d}\leq μ\_{s}.$$ | (6) |

Taking into account the risk of derailment when traversing a curve, from the point of view of the Nadal Y/Q criterion, there is included, i.a. a factor in the form of a transverse creep force $T\_{y}$, which is then projected onto the track plane to obtain the total transverse force or projected onto the plane perpendicular to the track to obtain the total vertical force:

|  |  |
| --- | --- |
| $$Y=N\sin(δ)-T\_{y}\cos(δ)$$$$Q=N\cos(δ)+T\_{y}\sin(δ)$$ | (7) |

$δ$ – contact angle, $Y$ – total lateral force, $Q$ – total vertical force.

The existence of a creep force longitudinal component $T\_{x}$ reduces the maximum value that $T\_{y}$ can reach because the total friction force cannot be greater than the saturation limit. However, this does not mean that a large $T\_{x}$ value is always favourable from the derailment risk point of view (Santamaria, Vadillo, & Gomez, 2009). The course of creep forces on the outer wheel visible in Fig. 7 can be divided into stages:

1. Straight section - during free rolling with minimal micro-slip (linear part of the relative force-slip characteristic), the value of the longitudinal creep force $T\_{x}$ results from small rolling resistance and resistance in the bearings, the value of the transverse creep force $T\_{y}>T\_{x}$ results mainly from the spin micro-slip and the contact angle.
2. The initial section of the arc, when the driving/braking torque is not present - the rolling radius of the outer wheel increases, the inner one decreases, their creepages increase, and longitudinal creep forces $T\_{x}$ are generated on both wheels, which create a torque relative to the vertical axis passing through the wheelset mass centre. This moment allows the wheelset to rotate relative to the vertical axis and helps the wheelset radially position itself in an arc (at the same time, the bogie frame rotates relative to the body when the suspension resistance is overcome). Since the radial alignment is not perfect, some yaw angle remains. The transverse creep force $T\_{y}>T\_{x}$ reaches a high value due to the yaw angle $\left|ψ\right|>0$ and a large contact angle $\left|δ\right|\gg 0$ (when the wheel flange moves closer to the rail, the theoretical maximum contact angle for the adopted wheel/rail profiles is 70°). Typically, the longitudinal creep force on the rolling surface of the outer wheel has the opposite direction to the longitudinal creep force on the inner wheel, and the directions of the transverse creep forces are the same, pointing outside the arc. However, this is not a rule, especially when there are external control torques on independently rotating wheels (Chudzikiewicz, Gerlici, Sowińska, Stelmach, & Wawrzyński, 2020; Konowrocki, Kalinowski, Szolc, & Marczewski, 2021; Opala, 2016) or traction/braking torques (Liu & Wang, 2007). The yaw angle of the leading wheelset is usually much larger than the following wheelset of the first bogie, hence the transverse creep forces $T\_{y}$ are much greater on the leading wheelset. However, the longitudinal tangential forces $T\_{x}$ on the following wheelset may be even greater than on the leading wheelset. For small arc radii and traditional rigid axle wheelsets, the tangential forces on the outer and inner wheels may reach the saturation limit. In this case, the saturation level for the outer wheel $n=\left|T\right|/N/μ$ is in the range of 0.94$÷$0.96, while for the inner wheel during free running $n=$ 0.8$÷$0.94. The values of contact forces at this stage constitute a reference level for changes that will occur during acceleration/deceleration.
3. The section of the arc during the driving/braking torque acting on the wheelset axle. Based on the simulation results in the assumed driving conditions, it can be concluded that if at stage b) the driving/braking torque begins to act on the wheelset, it causes a greater change in the value of longitudinal creepage than the lateral creepage, which results in a more significant change in the longitudinal force than the lateral force $∆T\_{x}>∆T\_{y}$ (Fig. 7). Changes in tangential creep forces occur relative to the reference level described in step b). During braking, the direction of the braking torque reduces the creepage in relation to the free running conditions, and therefore the value of the longitudinal creep force $T\_{x}$ decreases, the transverse creep force $T\_{y}$ did not change its value significantly in the case of braking. During acceleration, the direction of the driving torque is opposite to that during braking, which causes an increase in the creepage and the value of the longitudinal tangential force $T\_{x}$ in relation to the free running conditions. It is possible to change the direction of the longitudinal force so that the direction is the same on both wheels. The transverse component $T\_{y}$ decreased in value during acceleration. The saturation level for the outer wheel $n=\left|T\right|/N/μ$ is in the range 0.94$÷$0.96, while for the inner wheel during acceleration $n=$ 1, and during braking $n=$ 0.3$÷$0.8.

The longitudinal acceleration of ±1 m/s2 lasting for 3.5 s causes a change in the longitudinal speed of the vehicle by approximately 12.3 km/h. During acceleration, the change is oscillatory due to the middle unit of the vehicle, which is not driven and constitutes additional inertial resistance acting through the longitudinal elastic-damping connections of the inter-car couplings. However, during braking, the speed changes smoothly. Fig. 10 shows changes in the lateral force acting in the coupling between the units. During the tests, simulations of a single wagon with parameters such as one of the traction units were also carried out. Results of similar quality were obtained, which may indicate that in the case of this vehicle model, the impact of the coupling's interactions on the contact forces is small. The change in the longitudinal speed of the vehicle in the curve affects the value of the centrifugal force and, therefore, the balance of lateral forces and the value of the derailment coefficient. However, in this case, the scope of change in centrifugal force has a smaller impact on the balance of transverse forces than the values of tangential forces in contact between wheels and rails. The presented graphs show that after the short period torque on the wheelset axis ceases, during further traversing the curve, the value of the Y/Q indicator is close to the value before the change in running speed (Fig. 8).

Fig. 7. Longitudinal and transverse creep forces in contact of the outer wheel with the rail

Fig. 8. Derailment indicator for the leading wheelset of the first wagon. OW – outer wheel, IW – inner wheel

The derailment index value is the result of interactions with the track of both the first and second wheelsets, which are connected to each other through the bogie frame and suspension, the resultant force depends on the configuration of forces on all wheels of the bogie. On the one hand, the driving/braking moment acts on the wheelsets, and on the other hand, by the third law of dynamics, it acts on the frame of the bogie to which the engine and brake actuator systems are attached. The bogie frame rotates around the transverse axis (Fig. 9), which leads to the additional loading/unloading of the wheels through the first stage suspension. This affects the change of the vertical force (Fig. 11) and the value of the derailment coefficient (Zhang & Dhanasekar, 2009). Tab. 2 shows the distribution of creep forces in the final phase of the tested circular arc section.



Fig. 9. The angle of rotation of the leading bogie frame around the transverse axis due to the traction and braking torque between the motor/brake attached to the bogie frame and the wheelset



Fig. 10. Lateral force acting in the coupling between the first and second unit

Fig. 11. Vertical force acting on the outer wheel OW of the wheelset during acceleration, braking and free running. The load on the wheel is decreased/increased by the vertical suspension as a result of the rotation of the bogie frame due to the torque between the motor/brake attached to the bogie frame and the wheelset

Tab.2. Distribution of longitudinal and lateral creep forces (N) on individual wheels of the first bogie in the final phase of the driving/braking torque. 1Z- leading wheelset, outer wheel, 2W- following wheelset, inner wheel

|  |  |  |
| --- | --- | --- |
| *Free run* | *Accelerating* | *Braking* |
| ***1Z Tx*** | ***1Z Ty*** | ***1W Tx*** | ***1W Ty*** | ***1Z Tx*** | ***1Z Ty*** | ***1W Tx*** | ***1W Ty*** | ***1Z Tx*** | ***1Z Ty*** | ***1W Tx*** | ***1W Ty*** |
| -9900 | 23600 | 10300 | 14700 | -14800 | 20400 | 3600 | 15900 | -3700 | 23260 | 14970 | 14900 |
| ***2Z Tx*** | ***2Z Ty*** | ***2W Tx*** | ***2W Ty*** | ***2Z Tx*** | ***2Z Ty*** | ***2W Tx*** | ***2W Ty*** | ***2Z Tx*** | ***2Z Ty*** | ***2W Tx*** | ***2W Ty*** |
| 19700 | 2700 | -19700 | -1000 | 10300 | 1250 | -22000 | -1500 | 19700 | 1580 | -8930 | -700 |

**4. Summary**

The use of the appropriate longitudinal accelerations of trains requires the action of specific driving and braking torques on the wheelsets, which affects the distribution of contact forces and, therefore, the safety and operating costs. In the literature, the most frequently analysed cases are the braking of heavy freight trains, while fewer analyses are devoted to the situation when a light rail vehicle traverses a curve. In this study, simulation tests of the dynamics of a three-unit train set were carried out, considering driving conditions on a curve with a small radius. Changes in the values of creep forces and the derailment index were observed depending on the direction of torque acting on the wheelset axles during short-term acceleration and braking. The effect on the vehicle is related to a greater extent to changes in the contact forces and moments of these forces acting on the individual wheels of the bogie (as well as the frame of the bogie) and is less influenced by the centrifugal force and the interaction in the coupling. Pitch rotation of the bogie frame due to the traction/braking torque results in increasing/decreasing the vertical load on the wheelsets through the first-stage suspension, which has a visible impact on the value of the derailment risk index. It was observed that the saturation level of the friction force on the leading wheelset reaches higher values during acceleration compared to the braking conditions, considering the same absolute value of the vehicle’s longitudinal acceleration. The results for the leading traction unit are qualitatively very similar to those for a single wagon with similar parameters; in this case, the structure and interactions of the central coupler have little impact on the analysed values.

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