

## DYNAMIC RESPONSE OF VEHICLE-TRACK COUPLING SYSTEM WITH AN INSULATED RAIL JOINT

Ilaria Grossoni\*<sup>1</sup>, Simon Iwnicki<sup>1</sup>, Yann Bezin<sup>1</sup>, Cencen Gong<sup>1</sup>

<sup>1</sup> Institute of Railway Research, University of Huddersfield, Huddersfield, UK  
groilla@inwind.it, {s.iwnicki, y.bezin, u1273131}@hud.ac.uk

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**Abstract.** *The dynamic behavior of vehicle and track systems is studied in the presence of an insulated rail joint through a two-dimensional vehicle-track coupling model. The track system is described as a finite length beam resting on a double layer discrete viscous-elastic foundation. The vehicle is represented through a half car body and a single bogie. These sub-systems are solved independently and coupled together through a Hertzian contact model, where the irregularity caused by the rail joint is modelled as a second order polynomial. A parametric study is carried out in order to understand the influence by the main track and vehicle parameters to the  $P_1$  and  $P_2$  peak forces. Finally, the results in terms of  $P_2$  force from the present model have been compared not only with measured values but also with both other simulated and analytical solutions and an excellent agreement between these values has been found.*

## 1 INTRODUCTION

There is currently a worldwide trend towards using continuously welded rails for minimizing the wheel-rail impact forces. Nevertheless, rail joints are still used in some areas. In particular, insulated rail joints are required for track electrical insulation to detect the train location and to isolate sections such as those near road crossings.

When a train runs over a joint, large dynamic impact forces are developed which lead to vibrations in the structures and a higher probability of component fatigue and damage. Thus, it is clear that rail joints can affect the maintenance costs, ride comfort and running security on a modern railway.

Many studies have focused on the dynamic response of the railway track under moving vehicles. Some of these [1-4] only consider the track system as a beam on an elastic foundation subjected to moving point loads. Although this approach is simple to implement, it is insufficient to fully model the dynamic behavior of the track and vehicle systems as it neglects the wheel-rail contact aspects. Some other studies [5-8] take the coupling aspects into account, but few [9-10] apply these aspects to model the dynamic behavior in presence of a rail joint.

In this paper, a two-dimensional vehicle-track coupling model is developed. The track system is described as a finite length beam supported on a two-layer discrete elastic foundation. The vehicle is represented using a half car body. These sub-systems are solved independently and coupled together through the Hertzian wheel-rail contact model [5], where the irregularity due to the rail joint is modelled as a second order polynomial. The main results from the parametric study have shown that the first impact force  $P_1$  is greatly influenced by the wheelset mass, the rail mass and the joint angle, whereas the second peak force  $P_2$  is largely affected by the wheelset mass, the rail-pad stiffness, the support stiffness and the joint angle. Finally, the results in terms of the  $P_2$  force from this model have been compared not only with measured data [9] but also with both other simulated [9] and analytical [11-12] solutions. An excellent agreement has been found in all the comparisons.

## 2 MODELLING THE VEHICLE-TRACK COUPLING SYSTEM

The vehicle-track coupling model with an insulated rail joint is shown in Figure 1.

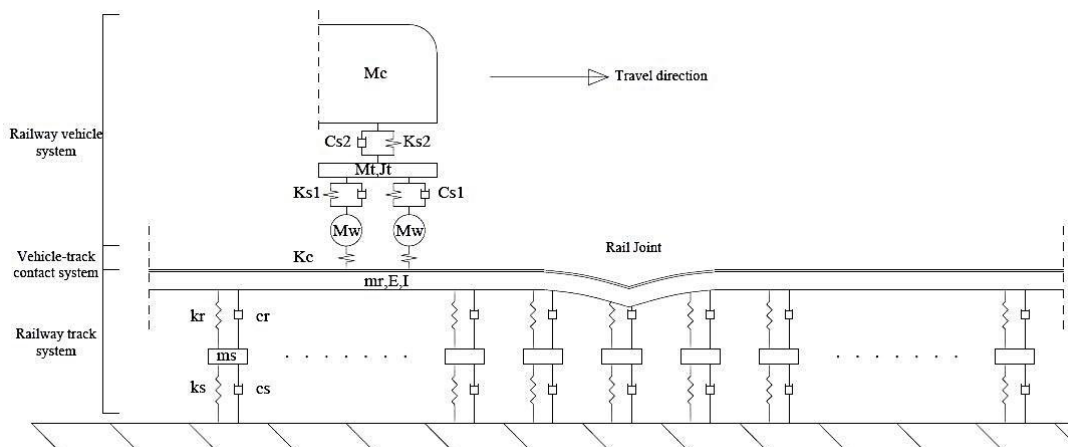


Figure 1: Vehicle-track coupling model with an insulated rail joint.

### 2.1 Fundamental assumptions

The main assumptions are listed as follows:

- 1) Only vertical dynamic forces are considered in the model. Due to the track symmetry, it is possible to use a single rail in the calculation.
- 2) The track system is modelled through a two-layer discretely supported ballast track model. A finite straight track without imperfections is considered. The number of beam elements considered is 40, because in this way it is possible to avoid the overlapping between the increasing static load during the transient time and the dynamic forces due to the rail joint.
- 3) A half car body is considered in the vehicle model. All the masses are assumed to be concentrated in the centre of gravity of the corresponding element. The two wheelsets masses and profiles are assumed to be the same.
- 4) The classical Hertzian contact model is used to couple the vehicle and track models.
- 5) The rail joint is modelled through a second order polynomial as established by a comparison between a quadratic fit function and the experimental measurements [13].
- 6) The joint fishplates contribution to the local stiffness is neglected.
- 7) It is assumed that there is one contact point at each wheel.

## 2.2 Modelling the track system

The moving force is characterized by a constant value of speed  $S$ , thus the simple formula for the uniform linear motion has been used. A finite element (FE) analysis is developed approximating the deformation within an element through using nodal values of displacement and rotation.

The beam has uniform flexural rigidity  $EI$  and mass per unit length  $\bar{m}_r$ . The sleepers are represented by their effective mass  $m_s$ , and assumed to be concentrated in the centre of gravity. The support is represented by rail-pad and ballast. The rail-pad is characterized by a spring of stiffness  $k_r$  in parallel with a damper having viscous damping coefficient  $c_r$ , while the ballast layer is similarly represented by a spring of stiffness  $k_s$  and a damper in parallel with constant  $c_s$ .

## 2.3 Modelling the vehicle system

The model used here consists of a quarter car supported by a bogie through the secondary suspension and a bogie supported by two half wheelsets through the primary suspension. All the bodies are assumed to be rigid. The car body is represented by its mass  $M_c$ , the bogie by its mass  $M_t$  and its pitch moment of inertia  $J_t$  and the wheelset by its mass  $M_w$ . Each primary suspension is modelled as a spring of stiffness  $K_{s1}$  in parallel with a damper with viscous constant  $C_{s1}$ . Similarly, the secondary suspension is characterized by a spring of stiffness  $K_{s2}$  in parallel with a dashpot having viscous constant  $C_{s2}$ .

## 3 RESULTS

In order to solve the coupling system, the track parameters used are reported in Table 1.

Variable	Value	Variable	Value
Mass per length unit (UIC60) $\bar{m}_r$	60.21 [kg/m]	Car body mass $M_c$	13000 [kg]
Moment of inertia (UIC60) $I_{yy}$	$3.04 \cdot 10^{-5}$ [m <sup>4</sup> ]	Bogie mass $M_t$	1600 [kg]
Young's modulus of rail $E$	$2.11 \cdot 10^{11}$ [N/m]	Bogie pitch moment of inertia $J_t$	1560 [kg·m <sup>2</sup> ]
G44 concrete sleeper	308 [kg]	Wheelset mass $M_w$	700 [kg]

Variable	Value	Variable	Value
mass $m_s$		Primary suspension stiffness $K_{s1}$	$1.87 \cdot 10^6$ [N/m]
Pandrol rail-pad stiffness $k_r$	$2.70 \cdot 10^8$ [N/m]	Primary suspension damping $C_{s1}$	$5 \cdot 10^5$ [N·s/m]
Pandrol rail-pad damping $c_r$	$2 \cdot 10^5$ [N·s/m]	Secondary suspension stiffness $K_{s2}$	$1.72 \cdot 10^6$ [N/m]
Typical support stiffness $k_s$	$8 \cdot 10^7$ [N/m]	Secondary suspension damping $C_{s2}$	$1.96 \cdot 10^5$ [N·s/m]
Typical support damping $c_s$	$1 \cdot 10^5$ [N·s/m]	Distance between the centre of pivot and the centre of wheelset $l_t$	1.25 [m]
Sleeper spacing $l$	0.65 [m]	Wheel rolling radius $r_0$	0.46 [m]
Number of beam elements $n$	40	Hertzian contact constant for worn wheels $G$	$4.22 \cdot 10^{-8}$ [m/N <sup>2/3</sup> ]
Maximum joint depth $D$	0.01 [m]	Newmark Beta parameter $\alpha$	0.25
Affecting length of rail joint $L$	3.65 [m]	Newmark Beta parameter $\beta$	0.50
Travelling speed $S$	160 [km/h]	Wilson Theta parameter $\theta$	1.40
Time step $\Delta t$	$5 \cdot 10^{-4}$ [s]		

Table 1: Track parameters [14], vehicle parameters [15], wheel-rail contact parameters [15], rail joint parameters [10] and integration parameters.

The response in term of wheel-rail contact force versus time is shown in Figure 2. It can be deduced that the results from the Wilson Theta integration method are higher than those from the Newmark Beta integration method. The peak difference is about 15%.

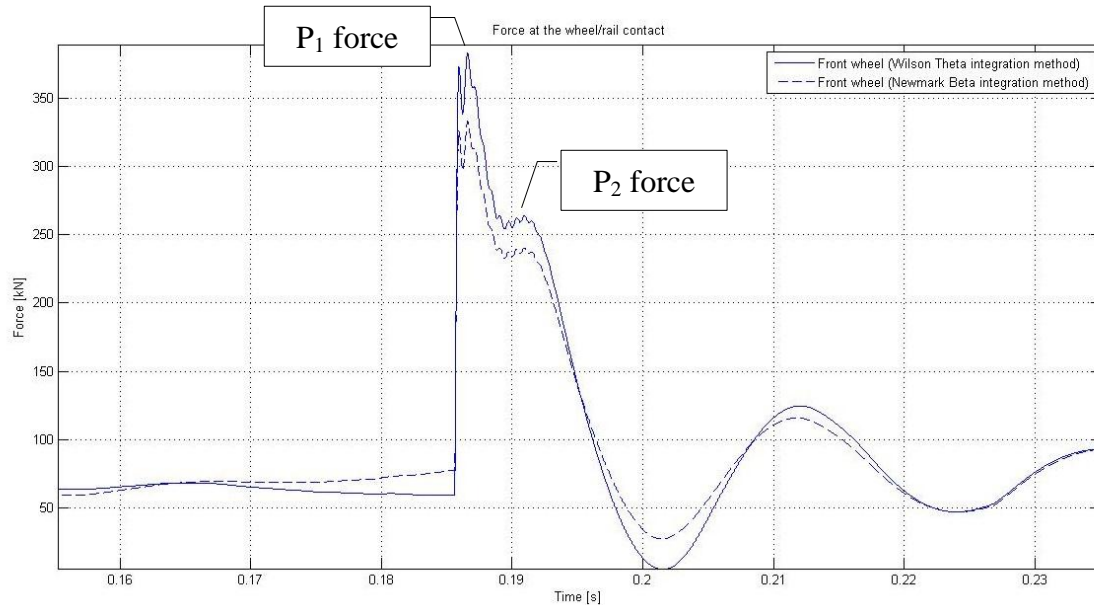


Figure 2: Wheel-rail contact force versus time.

It is also possible to recognize two impact forces ( $P_1$  and  $P_2$  forces).

The  $P_1$  force is a high frequency force (approximately 500 Hz) and it is characterized by high magnitude, which is found approximately 5 times bigger than the unsprung static load. It is associated mainly to the battering of the unsprung mass on the rail-end and is absorbed mainly by the rail and sleeper inertias. In the example shown in Figure 2, the force peak occurs 0.3 ms after crossing the joint, which is within the typical range between 0.25 ms and 0.5 m [11].

The  $P_2$  force, which occurs several milliseconds after the impact, is a medium frequency force (approximately 30-40 Hz) and its peak is lower than the  $P_1$ , around 3-3.5 times bigger than the static force. Contrary to the  $P_1$  force, the  $P_2$  force depends on the rail bending resilience and it is transmitted to the ballast, producing an acceleration of deterioration of the whole track system. This is the reason why the vehicle designers should reduce the unsprung masses as much as possible.

The  $P_1$  and  $P_2$  forces can be determined in the first approximation as in Eqs. (1-2) [11]:

$$P_1 = P_0 + 2\alpha V \sqrt{\frac{K'_H m_e}{1 + \frac{m_e}{m_u}}} \quad (1)$$

$$P_2 = P_0 + 2\alpha V \cdot \left(1 - \frac{c_t \pi}{4k_t(m_t + m_u)}\right) \cdot \sqrt{\frac{1}{1 + \frac{m_t}{m_u}} \cdot k_t m_u} \quad (2)$$

Where  $P_0$  is the static wheel load,  $2\alpha$  the total joint angle,  $V$  the travelling speed,  $K'_H$  the linearized Hertzian contact stiffness,  $P$  the generic load value,  $G$  the Hertzian contact constant,  $m_u$  the unsprung mass,  $m_e$  the effective track mass and  $m_t$ ,  $k_t$  and  $c_t$  the equivalent track system parameters defined in [11].

The UK standard GM/TT0088 states that “vehicles shall be able to run over the normal range of vertical track irregularities at normal operating speeds without generating excessive vertical loads and stresses in the rails and track” [12]. There is a limit only for the  $P_2$  force, which cannot exceed a total value of 322 kN per wheel at the maximum operative speed. This limit is made because this force is directly transmitted to the ballast, as said previously. In particular, an analytic formula has been proposed as in Eq. (3) [12]:

$$P_2 = Q + A_z \cdot V_m \cdot M \cdot C \cdot K \quad (3)$$

Where  $Q$  is the maximum static wheel load,  $V_m$  the maximum normal operating speed,  $M_v$  the effective vertical unsprung mass per wheel,  $M$ ,  $C$  and  $K$  parameters reported in [12].

### 3.1 Parametric study

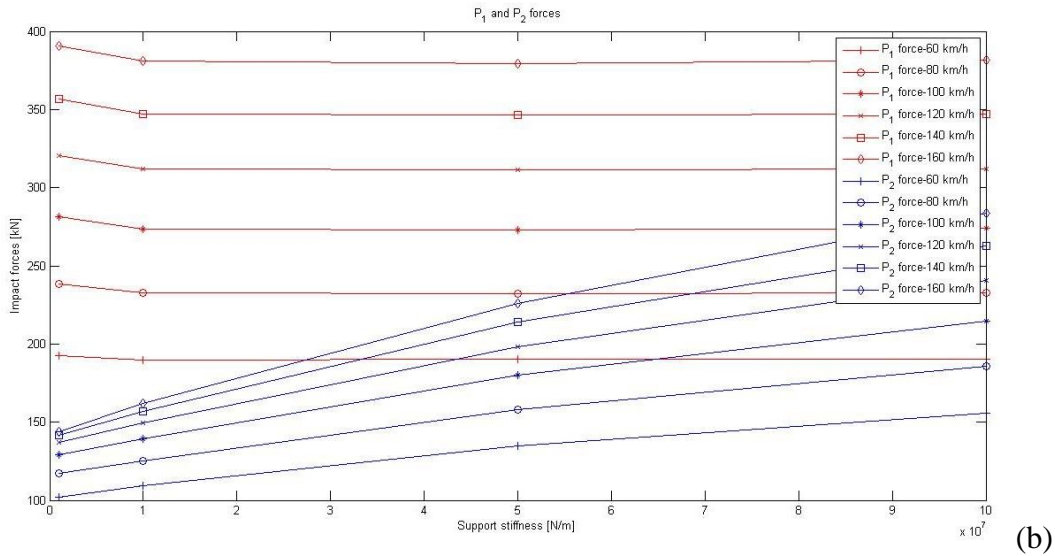
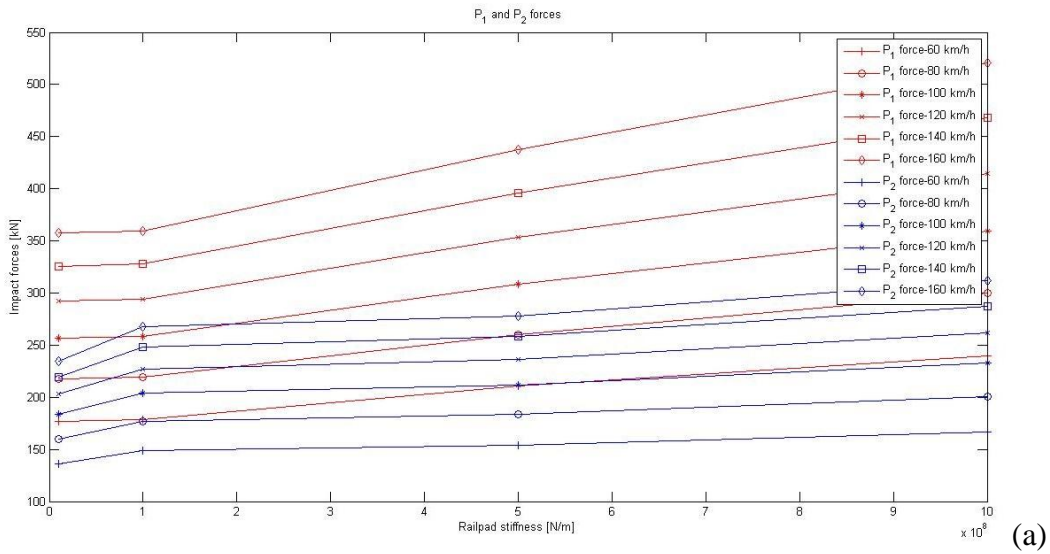
Numerical examples are shown and the effects of some parameters are investigated. The parameters of track, vehicle and joint models which have been used in the simulation are reported in Table 2.

	Track model	Vehicle model	Joint model
Parameters	Travelling speed $S$	Travelling speed $S$	Travelling speed $S$
	Rail mass per unit length $\bar{m}_r$	Wheelset mass $M_w$	Affected length $L$
	Rail-pad stiffness $k_r$	Bogie mass $M_t$	Total joint angle $2\alpha$
	Support stiffness $k_s$	Car body mass $M_c$	
	Total joint angle $2\alpha$	Total joint angle $2\alpha$	

Table 2: Parameters of simulations.

The integration method used is the Wilson Theta method because it gives higher values compared to the Newmark Beta method, as shown in Figure 2.

In the following, only the most relevant graphs are reported.



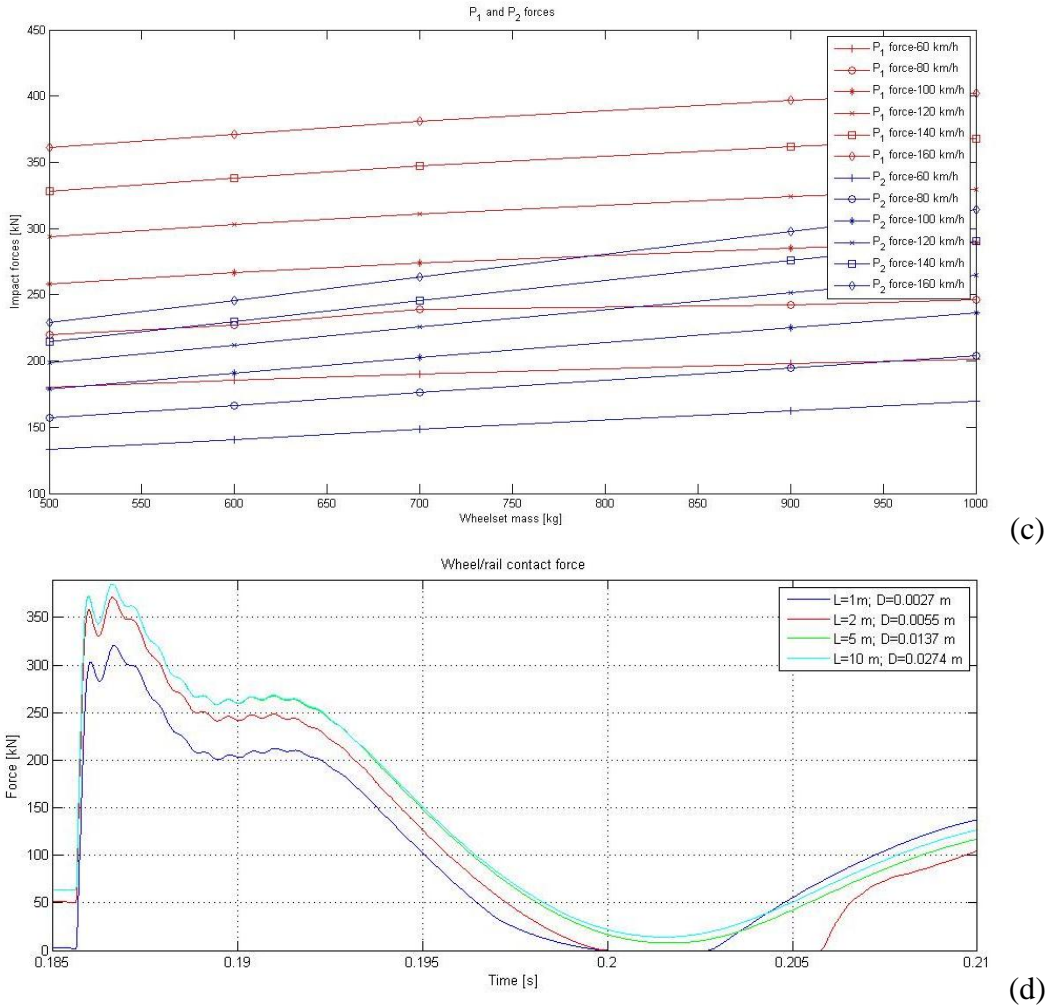


Figure 3: The variation of P<sub>1</sub> (red lines) and P<sub>2</sub> (blue lines) forces with (a) rail-pad stiffness; (b) support stiffness; (c) wheelset mass; (d) joint shape fixing the total joint angle.

From Figure 3-a, it is found that the trends of P<sub>1</sub> and P<sub>2</sub> forces with the rail-pad stiffness are different. In the case of P<sub>1</sub> force, there is a first constant trend for low stiffness values ( $10^7$ - $10^8$  N/m) and then a linearly increasing trend for higher stiffness ( $5 \cdot 10^8$ - $10^9$  N/m). The slope increases with increasing speed, from about 25% in the case of 60 km/h to about 32% in the case of 160 km/h. In case of P<sub>2</sub> force, instead, there is a relatively rapid increasing trend for low stiffness values ( $10^7$ - $10^8$  N/m) and then a relatively slower increasing trend for higher stiffness ( $5 \cdot 10^8$ - $10^9$  N/m) with a slope significantly lower than the first part.

Regarding the support stiffness (Figure 3-b), it can be found that the P<sub>1</sub> trend is relatively constant while the P<sub>2</sub> grows linearly. The average slope increases with increasing travelling speed, from about 35% at 60 km/h to about 50% at 160 km/h. This means that decreasing the ballast stiffness has little effect on the P<sub>1</sub> force, whereas it can reduce the P<sub>2</sub> force.

As shown in Figure 3-c, the impact of the wheelset mass is very large in both cases. In particular, the P<sub>1</sub> force trend is asymptotic, that is the change of wheelset mass plays a limited role in that impact force for values greater than 600-800 kg. The P<sub>2</sub> force increases proportionally with the wheelset mass, with the average slope increasing with increasing travelling speed.

Finally, from Figure 3-d it can be deduced that both the impact forces are different in case of different affected length L. Thus, the dynamic response in terms of wheel-rail contact

forces is closely related to the actual shape of the rail joint. This conclusion is against the formulae proposed by the UK researchers in Eqs. (1-2), according to which the forces are constant for constant value of total dip angle. Therefore, those formulae can give only a rough estimation of the forces.

### 3.2 Comparison

The main characteristics of two different Chinese freight vehicles, C62A and C75, are reported in Table 3.

Variable	C62A	C75
Car body mass $M_c$	19250 [kg]	22950 [kg]
Bogie mass $M_t$	565 [kg]	755 [kg]
Bogie pitch moment of inertia $J_t$	380 [ $\text{kg}\cdot\text{m}^2$ ]	780 [ $\text{kg}\cdot\text{m}^2$ ]
Wheelset mass $M_w$	600 [kg]	647.5 [kg]
Primary suspension stiffness $K_{s1}$	-	-
Primary suspension damping $C_{s1}$	-	-
Secondary suspension stiffness $K_{s2}$	$1.06\cdot 10^7$ [N/m]	$1\cdot 10^7$ [N/m]
Secondary suspension damping $C_{s2}$	$1.40\cdot 10^5$ [Ns/m]	$1\cdot 10^5$ [Ns/m]
Distance between the centre of pivot and the centre of wheelset $l_t$	0.87 [m]	0.87 [m]
Wheel rolling radius $r_0$	0.42 [m]	0.42 [m]

Table 3: Main characteristics of C62A and C75 freight vehicles [9].

In Figure 4 the comparison between the results in terms of  $P_2$  force in case of C62A and C75 freight vehicles with increasing travelling speed is shown. In particular, for each travelling speed and each vehicle type five different values of the peak force are considered: measured data [9], simulated value [9], two different analytic solutions [11-12] and the present model.

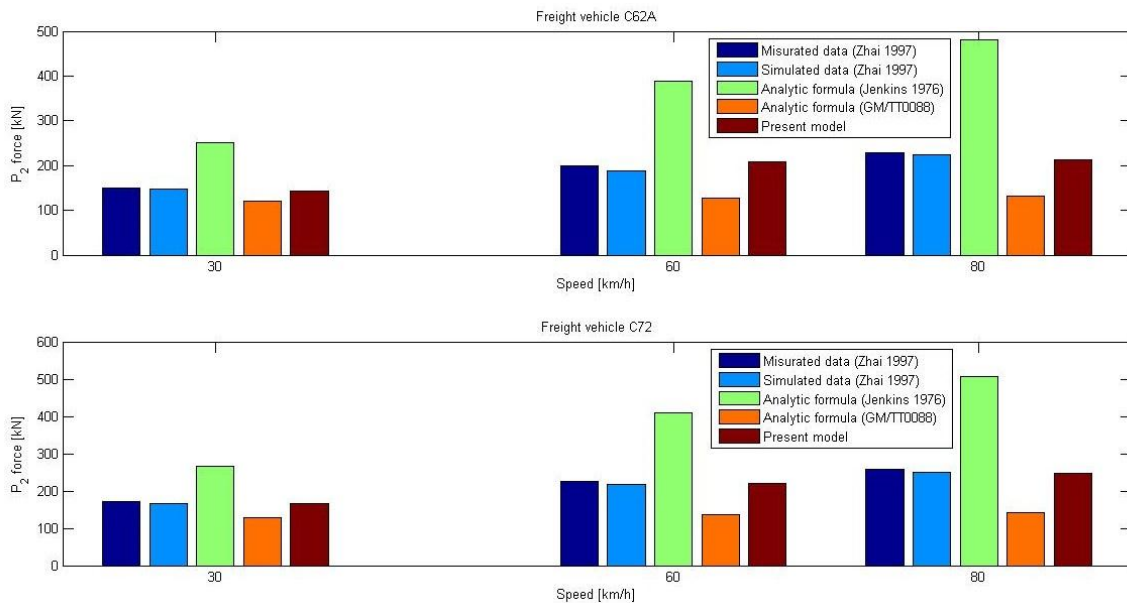


Figure 4: Comparison between the results in terms of  $P_2$  force in case of C62A and C75 freight vehicles.



It is noticeable that the results are close to each other, particularly the measured data, simulated values and the present model. The percentage differences between the present model and measured data or simulated values are reported in Table 4.

Travelling speed [km/h]	C62A		C75	
	Measured data [%]	Simulated values [%]	Measured data [%]	Simulated values [%]
30	4.5	3.2	3.2	0.8
60	3.7	10.9	2.2	1.6
80	6.5	4.4	3.7	0.6

Table 4: Percentage differences between the present model, the measured data and the simulated values.

## 4 CONCLUSIONS

The primary objective of this paper is to investigate the characteristics of vertical dynamic response due to insulated rail joint (IRJ) through developing a comprehensive finite element (FE) model of vehicle-track coupling system. The two dimensional model employed has been established by merging three elementary models, which are the track model, the vehicle model and the contact model. This strategy was proven efficient to obtain an efficient solution. Two explicit numerical integration methods have been employed in the dynamic analysis. In order to achieve a reasonable model size which is compatible to the available computing facility, several assumptions have been made in the rail joint model, track model and boundary conditions.

The wheel-rail impact mechanism can be explained through the stiffness discontinuity of the IRJ structure. The joint causes a geometric discontinuity in the running surface during the wheel passage, which therefore produces the impact in the vicinity of the gap. At impact, two peak contact forces develop. The main characteristics, such as the frequency and the magnitude, are quite different. It has been demonstrated that the lower magnitude force is the force that actually causes the track degradation because its characteristic frequencies match with the track frequencies.

Through a series of sensitivity studies of several parameters, it was shown that the dynamic response can be largely improved by optimised design parameters. The parametric simulations have shown that the first impact force  $P_1$  is greatly influenced by the wheelset mass, the rail mass and the joint angle, whereas the second peak force  $P_2$  is affected by the wheelset mass, the rail-pad stiffness, the support stiffness and the joint angle. The model has outlined that the impact forces depends on the actual shape of the rail joint. Therefore, great reductions of peak forces values can be obtained through an appropriate joint design.

Finally, the results in terms of  $P_2$  force from the present model have been compared not only with measured values but also with both simulated and analytical solutions. An excellent agreement between values has been found, with a maximum percentage difference of 10%.

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