



COMPUTATIONAL MODEL TO EVALUATE ACTIONS IN WHEEL- RAIL CONTACT INTERACTION

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Abstract. *Railway traffic is a big source of vibrations. Over the years, with higher freight capacity and speeds getting higher, excessive stresses are experienced by the subgrade. Then, stress waves propagate through the ground until buildings on the surroundings. This leads to considerable negative impacts on the neighborhood. On this context, this work is immersed in a broader scope of proposing an approach to create a model to simulate railway systems, with the main objective of predicting the vibration and stresses at the subgrade, in different situations, such as to predict the contact forces between the wheels and the rails. In this model, the dynamic characteristics of the vehicles and the contact between their wheels and the track are considered. The numerical model was solved using the software Giraffe. To simulate the traffic of a vehicle, a two-car train composition was modeled. One of the cars is loaded by an imposed torsion moment, applied on the axle of the wheels. A time-varying torsional moment was considered, so the simulation could reproduce an accelerating vehicle, the same vehicle in constant speed, as well as braking. The model, as it was proposed, represented the global mechanical behavior of the cars and it was possible to obtain the contact forces between wheels and a surface that represents the rails.*

Keywords: *Railway, Contact, Finite Element, Giraffe.*

1 INTRODUCTION

Railway transportation has been one of the most efficient mass transport system since its creation. From steam to electric engines, railways transport tons of freight and millions of people all around the globe. For years, Brazil prioritized the development of individual roadway transportation, leaving behind investments to its railroad network. Now, with a massive growing population in urban centers, increasing on exploitation of iron mines and growth in food production, it has no other way but resume the expansion of Brazil's rail network.

To serve the community more efficiently, the new lines must be located in urban centers, getting closer to residences, hospitals, schools and historic buildings, for example. Railway traffic is a huge source of noise and vibration, owing, mostly, to steel-steel contact between wheels and rails, impact excitation due to discontinuities of the wheels and the rails and also, in some circumstances, parametric excitation due to the support of the rails on sleepers in intermittent intervals (Degrande et al., 2006). With faster and heavier trains, excessive ground vibration, created due the reasons above, has become a problem. According to Suhairy, 2000, there are three principal types of waves that propagate in soils (Figure 1). Compression waves, or longitudinal waves, with their particle motion being in the direction of propagation (they are also called P or Primary waves). Shear waves, also called S or Secondary waves, are the ones with particle motion being an oscillation in a plane perpendicular to the direction of propagation. Finally, we ought to mention the Rayleigh waves, which are surface waves, with a particle motion generally elliptical in a vertical plane through the direction of propagation. Despite these, there are others, like Love waves, which are the fastest surface waves, that move the ground from side-to-side.

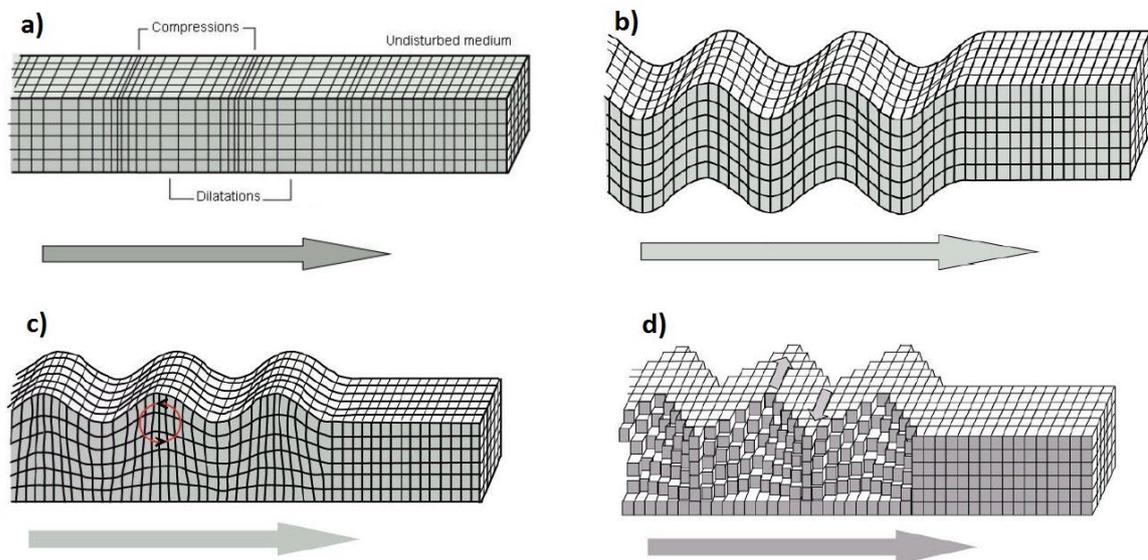


Figure 1. Different types of waves: a) P waves; b) S waves; c) Rayleigh waves; d) Love waves.

Fonte: Adapted from (Suhairy, 2000).

Ground surface vibration produced by trains is an important aspect of environmental impact of rail traffic. Due the importance of this research field, a considerable amount of models for vibration have been developed, but most of these models only take into account the vibration generated by a moving point force on an elastic half-space (Sheng et al., 1999) or “quasi-static” axle loads (Nejati et al., 2012).

Two-dimensional finite element models have been used by several authors to solve dynamic interaction problems. Although these models need important simplifications to represent a three-dimensional dynamic load. Another alternative is a 2.5D model¹, which is a resource to numeric models that account for the invariance of the track cross-section in the longitudinal direction (Degrande et al., 2006). The benefits of Finite Element models are that it has no restrictions in geometry and it can be considered the nonlinear effects (Karlström et al., 2006).

This paper aims to show a first approach on developing a finite element model of a railway vehicle and track system, using the software Giraffe, with the objective of obtaining the result of a time-varying contact force that lies on the ground. In a second moment, such result will be used as input in another model to simulate the wave propagation through the ground, which is the main motivation of this work. The proposed model includes the suspended and unsuspended mass of the cars and their primary and secondary suspensions. In a forthcoming session, further details will be given about the conception design. The wheels are placed on a horizontal flat surface, so it is possible to monitor the contact between the surface and the wheels. This preliminary study considers no track or subgrade influence.

1.1 Giraffe

Giraffe, which means “Generic Interface Readily Accessible for Finite Elements”, is a software coded using C++ language, which goal is to generate a base-interface to be used by researchers to implement their own finite element formulations (Gay Neto, 2016). It may be used on structural problems, which may include translational and rotational degrees of freedom (DOFs), and multiphysics applications, including other nature of DOFs.

In the paper Gay Neto et al., 2014, it is shown a new approach to model the contact between a circular cross section beam and a flat surface and it presents the mathematical model for the contact representation. Giraffe allows at modeling different individual elements, defined by nodes in space that can interact with each other since the user defines their connections, joints and contacts. Besides the cross-section form and its orientation, in the input code is required to introduce the characteristics of the materials of all elements. Until now, the program admits only materials that follow Hooke’s law. In a recent work (Gay Neto, 2016[b]), the mathematical models for joints, springs, dashpots and followers loads, that were considered in *Giraffe*’s formulation are presented. In Gay Neto et al., 2015, a numerical model and procedures that solve self-contact in beams under loop formation are presented.

A useful resource of Giraffe is the possibility to insert environmental data. This includes gravitational field and the mass per unit volume of a surrounding fluid, for structures experiencing fluid-structure interaction.

Giraffe is able to perform not only static but also dynamic analysis. The post-processing data consists on information about the nodes (or nodes-sets), elements and contacts, that the

¹ 2.5D refers to models that have the projection of a plane cross-section into a third dimension with no variations.

user wants to monitor. The static beam models implemented in *Giraffe* are described in Gay Neto et al., 2013 and the dynamics in Gay Neto, 2016[a]).

2 MODEL OF THE RAILWAY VEHICLE

It was used the software *Giraffe* to create and simulate a finite element model that can represent a two-car train, which may be called as an Electric Multiple Unit (EMU). EMUs require at least two carriages, a first one that carry the traction motors to move the train and a second one that has no traction or power related equipment. The proposed model consists in these two units, in the front goes the car with a traction motor and the second is connected by a rigid beam element and is pulled as a trailer car. Beam elements with rectangular and tubular cross-sections compose all parts of the model as it is shown in Figure 2. Note that the objective of the beam parts is only to serve as connections, and each beam member discretization was not performed to analyze the local structural behavior of the system. A structural more detailed and refined model may be employed in future works. Since in this work the objective is to analyze the overall system dynamic behavior, the adopted discretization showed to be adequate.

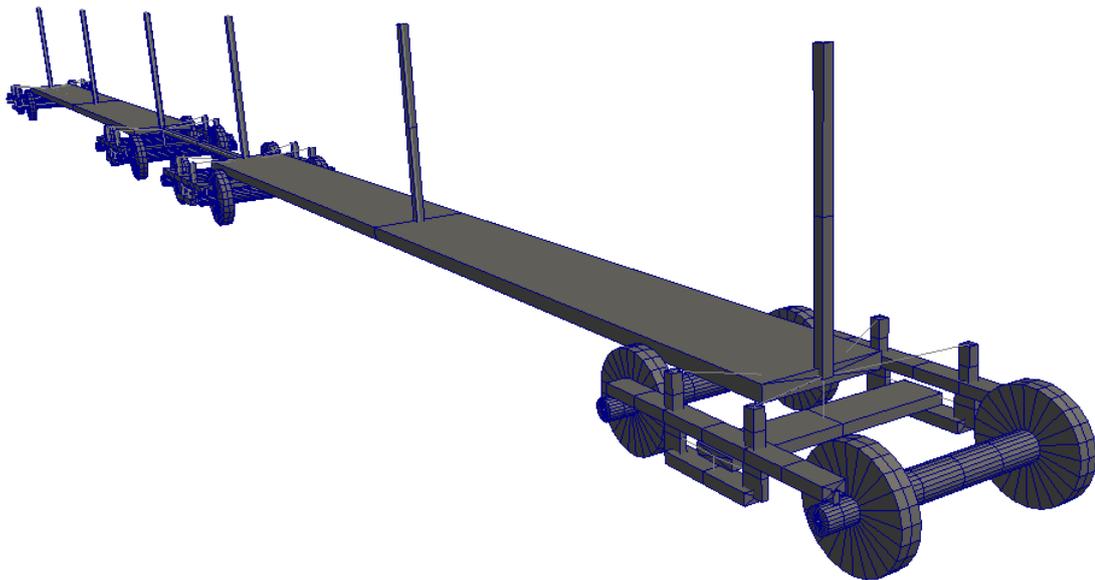


Figure 2. General render view of the model with two-car composition

The wheels and their axles were modeled as circular cross-section beams, made out of steel as a single object. The wheels and axles are connected to the bogie with a special constraint, which in *Giraffe* is called “Hinge Joint”, presented in Figure 3. It permits connecting two different elements with coincident nodes (A and B), presenting the same displacement, but the connected bodies will be able to rotate freely around an axle defined by a direction \mathbf{e}_3 from a chosen coordinate system. During the simulation, the hinge joint ensures that $\mathbf{e}_{3A} \equiv \mathbf{e}_{3B}$. In this case, the direction \mathbf{e}_3 was coincident to the axle of the wheels, see Figure 3 -b).

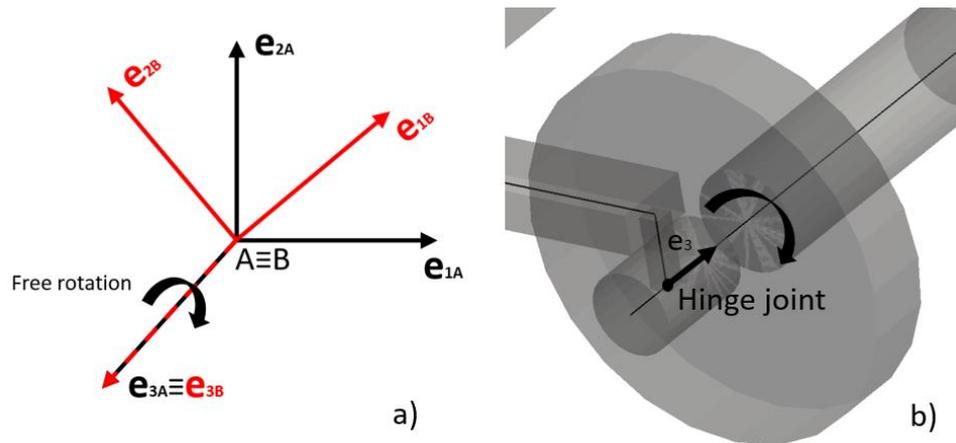


Figure 3. Hinge Joint: a) conception; b) applied to the model.

The bogie was designed, as shown in Figure 4, as beam bodies with multiple rectangular steel beam elements connected by a floating transversal beam ABC placed over the primary suspension (a1 and c1). Springs/dashpots elements represent the primary suspensions (a1, a2, a3, c1, c2 and c3). Over the beam ABC another suspension, the secondary (b, d1, d2, d3 and d4), was designed following the same principle of the primary, locking horizontal relative displacement and leaving vertical displacement free. The elements a2, a3, c2, c3, d1, d2, d3 and d4 have much higher stiffness than the elements a1, c1 and b. The horizontal elements have the role to lock the horizontal movements and maintain the beams ABC and DE in the right aligned position, while the bogies move at the longitudinal direction.

The beam DE is the connection between both bogies of a car. Over this beam, vertical bars were placed, with a mass element on their top, representing the mass at the mass center of the car, considering people's and other equipment's contributions. The mass elements were calculated such that the train have around 23 t/axle or 11.5 t/wheel, considering the mass of all other elements.

It was also defined the contacts between the wheels and a flat horizontal surface, that is presented in Figure 4 (W1, W2, W3 and W4). The coefficient of friction was input to represent the contact between two steel elements, which means wheels and rails.

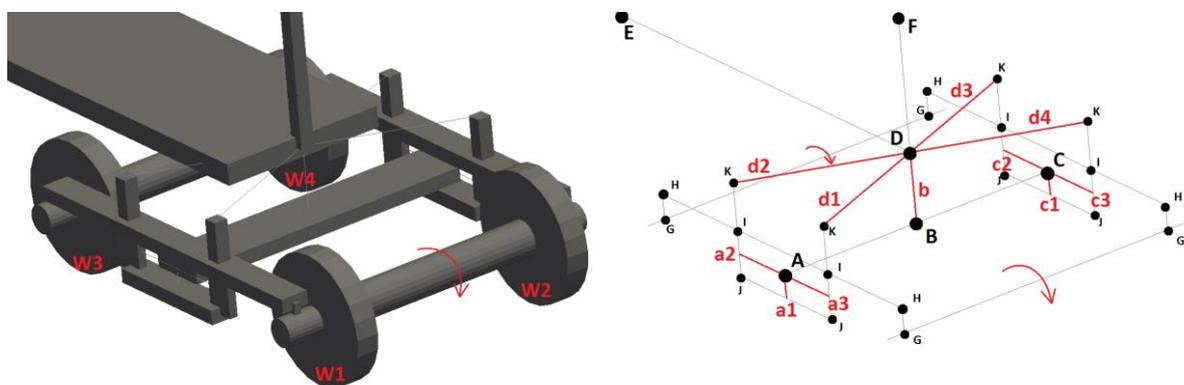


Figure 4. Concept of the modeled bogie

The data about the beams, suspensions and masses are disposed in Table 1.

Table 1. Elements of the model

Element	Identification	Characteristics		
		Section	Dimensions [m]	Length [m]
Beams	ABC	Rectangular	0.40 x 0.10	2.114
	DE	Rectangular	1.20 x 0.15	14,710
	DF	Rectangular	0.10 x 0.10	2.500
	GH	Rectangular	0.05 x 0.05	0.175
	HI and II	Rectangular	0.10 x 0.20	0.780 and 0.940
	IJ	Rectangular	0.20 x 0.06	0.329
	JJ	Rectangular	0.10 x 0.20	0.940
	IK	Rectangular	0.05 x 0.05	0.335
	GG (axles)	Circular	0.25	2.381
	Wheels	Circular	0.9144	0.164
Springs and Dashpots	Characteristics²	Stiffness [kN/m]	Damping [kNs/m]	
	a1 and c1	3274.55	63.51	
	a2, a3, c2 and c3	327455.00	0	
	b	1751.31	34.84	
	d1, d2, d3 and d4	175131.00	0	

2.1 Suspension model

In the model of the whole system, when the car starts to move, the beams ABC and DE have to accompany such movement, with no horizontal relative displacement to the whole system, while the bogies move. Due to that, the springs a2, a3, c2 and c3 were included in the suspension model, to represent this characteristic. They have greater stiffness than a1 and c1. The relation between horizontal and vertical stiffness coefficients should be as high as possible, so the nodes on extremities of the beam could not move horizontally when compared to the rest of the model. On the other hand, this relation could not be so high that

² Data taken from (Correa et al., 2005)

the vertical displacement would become too limited. The generic simple model represented in Figure 5 was modeled, so it could show the influence of the horizontal stiffness on the vertical movement of the whole suspension. On this study, the damping of the elements was disregarded, because the analysis made was only static.

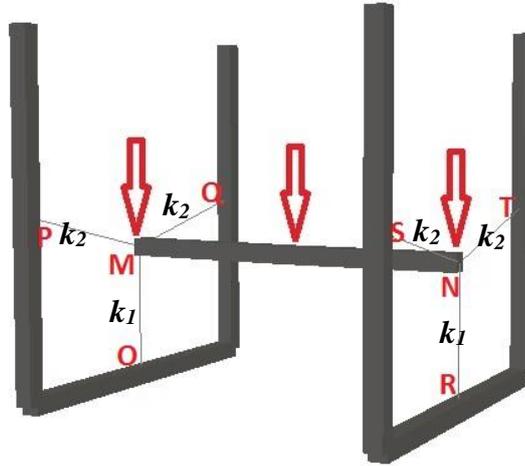


Figure 5. Simple concept of the suspension

This simple model is formed by seven bars with square cross-section with width 10 cm. The suspension concept has a geometric configuration that induces nonlinear geometric behavior. For small vertical displacements, only the initially vertical spring/dashpot (MO and NR) plays a role. When experiencing larger vertical displacements, the initially orthogonal to the vertical springs/dashpots (PM, QM and SN, TN) change their alignment, collaborating with the whole system's stiffness. The result is a typical hardening force vs. displacement curve, as depicted in Figure 6.

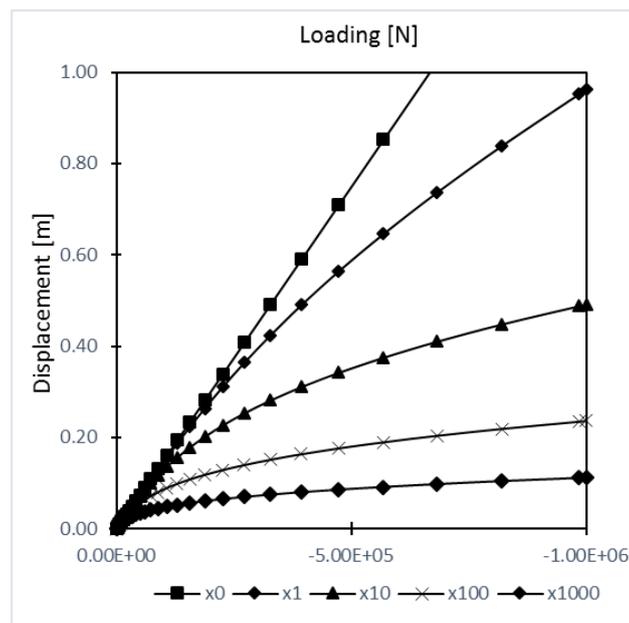


Figure 6. Vertical displacement of the beam

The graphic on Figure 6 shows the curves of the vertical displacement of the beam during the loading. With no horizontal stiffness (curve x0) the displacement has a linear behavior. As the horizontal stiffness grows, from 1 to 1000 times higher than the vertical (x1 to x1000), the nonlinearity becomes more evident. The stiffness of all spring elements is depicted in Table 2.

Table 2. Stiffness of the springs

Case	Element's Stiffness [kN/m]	
	<i>MO / NR</i>	<i>PM / QM / SN / TN</i>
x 0	1.00E+6	-
x 1	1.00E+6	1.00E+6
x 10	1.00E+6	1.00E+7
x 100	1.00E+6	1.00E+8
x 1000	1.00E+6	1.00E+9

The most important criterion to define the relation between the stiffness values, in complete model, was the limitation of the settlement.

After this study of the suspension system, we adopted in the complete vehicle model the relation x100 for the vertical/horizontal springs. This choice was done based on a criterion of providing enough horizontal rigidity for the system, to avoid excessive horizontal movements, but not creating a too stiff behavior in vertical direction. The whole system behavior is clearly nonlinear, as seen in Figure 6. One can set springs combinations to create the desired force vs. displacement relation to the suspension system.

3 RESULTS AND DISCUSSION

To avoid any oscillation of the whole structural model that represents the cars over the suspensions, it was executed first a static analysis, with the objective of establishing the initial contact condition between the system and the track (represented by a flat surface) and the suspension settlement due to the self-weight of the system modeled. The result of this simulation was taken as start-point of dynamics, considering inertial forces and damping, such as time-varying loads.

During dynamics, a torque was imposed at the four axles of the front car, causing their rotation, playing the role of a motor car. It was desired that some contacts and nodes are monitored, so the post-processing files can be used to plot data from those points of interest. The observed nodes are presented in Figure 7: the central node of the transversal floating beam (P nodes)³ the longitudinal beams that connect the bogies of a car (R nodes), the nodes

³ Must be noticed that all nodes in similar conditions to the ones in Figure 7 will be called the same.

(S) coincident to the mass elements and the ones in the middle of the axles (Q nodes). Besides the nodes, the contacts between the wheels and the generic flat surface (not represented in Figure 7) were as well monitored (W1 to W16).

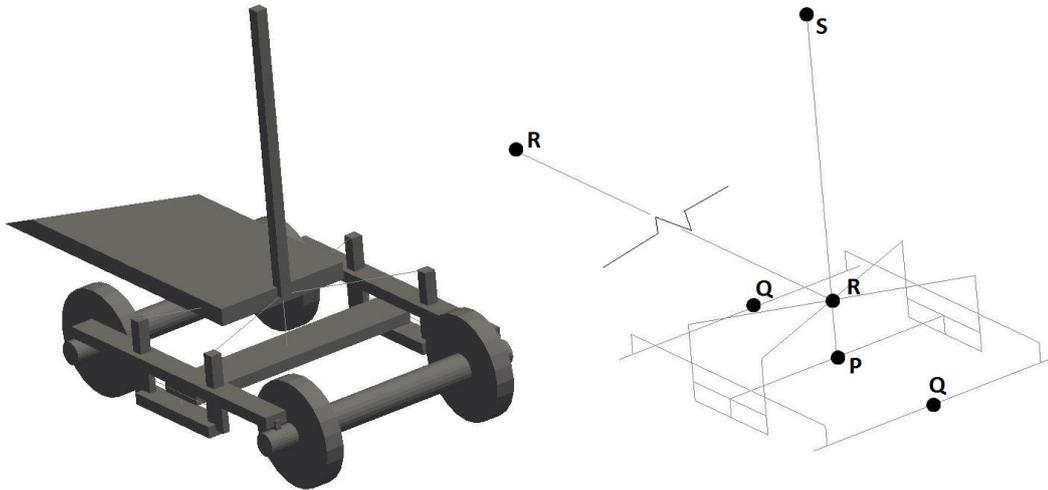


Figure 7. Monitored Nodes and Contacts

In forward sections, the monitor's results will be discussed.

3.1 Monitored nodes

The monitored nodes are presented in Figure 7. The static analysis result plot is shown in Figure 8. The primary suspension had a settlement of 3.7 cm and the total settlement of the system is 16.6 cm, which means that the secondary suspension had a settlement of 12.9 cm.

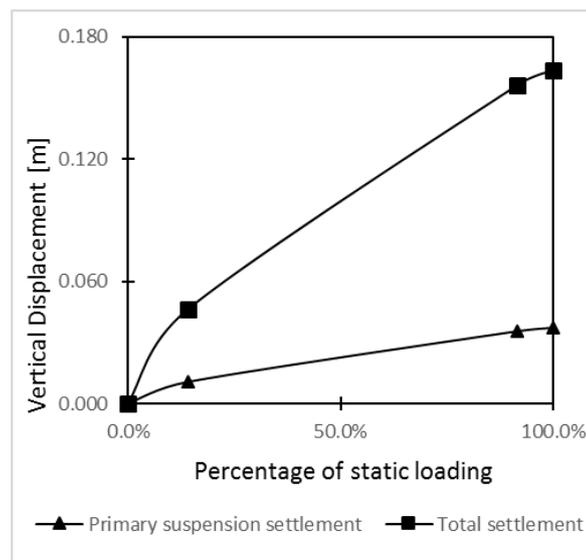


Figure 8. Vertical displacement of S nodes during statics analysis.

After the static analysis, when the system achieves an equilibrium configuration, *Giraffe* starts the dynamic analysis, when external forces can be applied during specific times. In this case, the Q nodes received a torque, each, which permits the axles of the front car to roll. This torque was applied as shown in Table 3 and Figure 9.

Table 3. Distribution of applied torque over time

Applied Torque in nodes 7, 20, 60 and 73			
TIME	M_Y [N.m]	TIME	M_Y [N.m]
0	0	24	-10000
1	10000	25	-18000
2	18000	26	-26000
3	26000	27	-34000
4	34000	28	-42000
5	42000	29	-50000
6	50000	30	-50000
7	50000	31	-50000
8	50000	32	-50000
9	50000	33	-50000
10	50000	34	-42000
11	42000	35	-34000
12	34000	36	-26000
13	26000	37	-18000
14	18000	38	-10000
15	10000	39	0
16 – 23	0	40	0

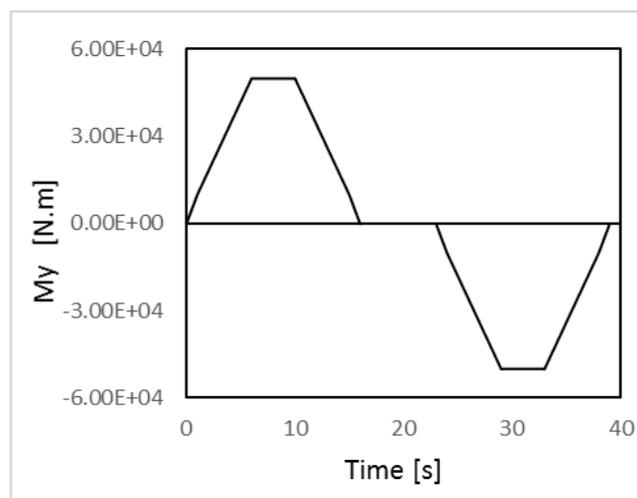


Figure 9. Distribution of applied torque over time

From time 0 s to 15 s a positive torque is applied to the center nodes of all four axles, for this reason the cars start moving, due the friction between both materials. The friction coefficient was chosen for dry and clean steel – steel contact interaction (0.74) (Grigoriev, 1997). From 16 to 23 seconds, the cars move in constant speed and later, a negative torque is applied and the cars stop completely the movement. During the simulation, as seen in Figure 10, the cars traveled 436 meters and reached 19 m/s of speed during the cruise. They achieved an average of $\pm 1,86$ m/s² during acceleration and breaking, and during the cruise 0 m/s², as expected.

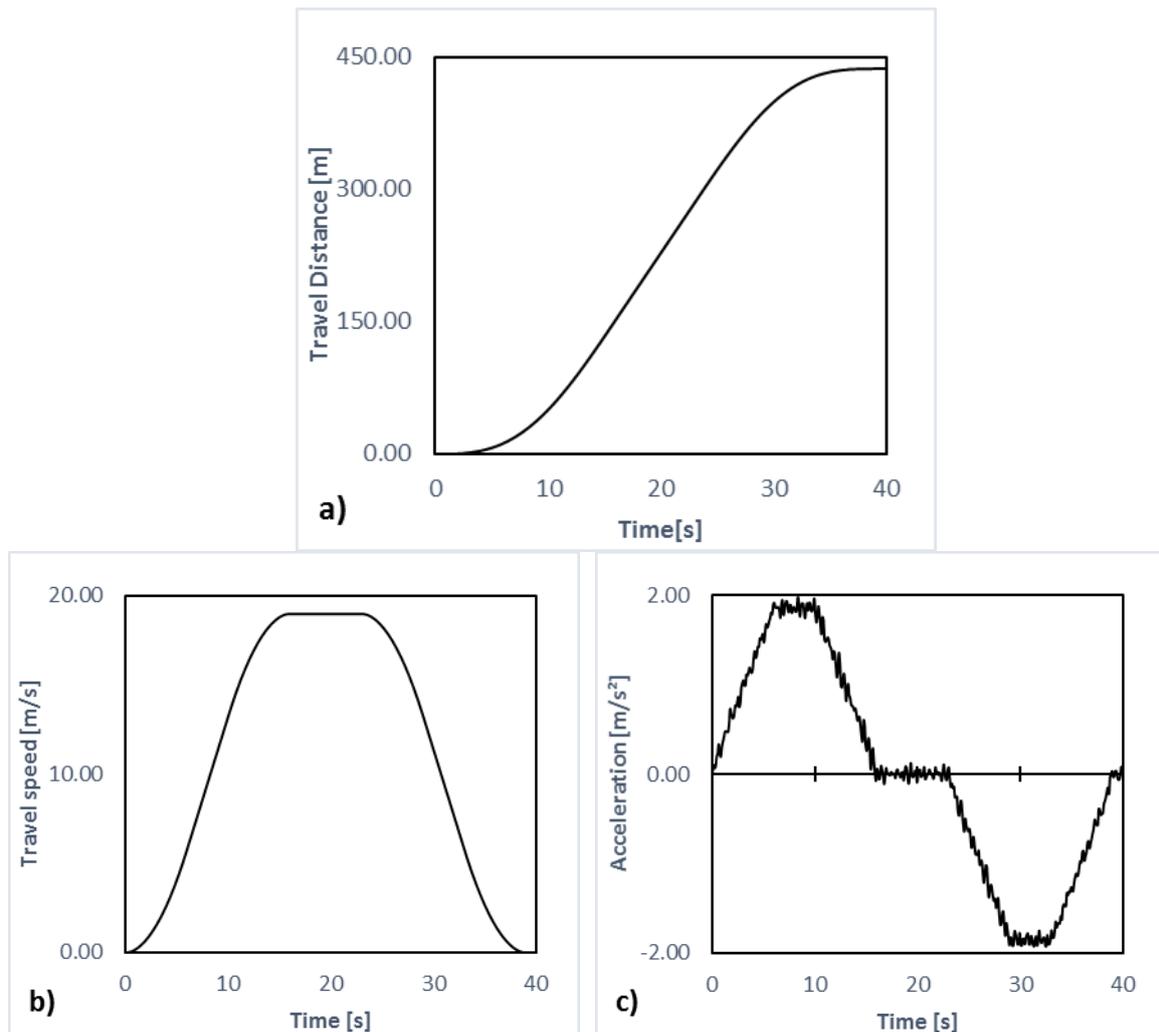


Figure 10. a) Travel distance; b) Travel speed; c) Acceleration.

Since the surface was perfectly flat, there was no disturbance during rolling. Consequently, there were no considerable vertical or transversal displacements in none of the nodes, for that reason it was analyzed only longitudinal displacement.

3.2 Monitored contacts

The contact between wheels and the flat surface had their results observed. The post-processing data consists on contact forces in three directions. The transversal forces, in Y direction, were disregarded since they were null. The motor car and the trailer car were watched separately.

When the torques were applied, varying along time, the EMU starts moving. When that happens, the wheels of both bogies exert a longitudinal contact friction force on the surface. It can be seen in Figure 11. This force grows together with the applied torque. Between 16 and 23 seconds, when there is no external moment on axles, the longitudinal contact force goes to zero. That means that the wheels of the car are just rolling but not exerting any traction.

An interesting effect is shown in Figure 11. While the car is accelerating, the rear wheels are overcharged, the front wheels discharged, and when the car is breaking, the opposite happens. It makes sense, since the suspended mass tend to stay still while the bogies starts movement and the mass tends to overload the rear wheels. The same is valid while the car is stopping movement.

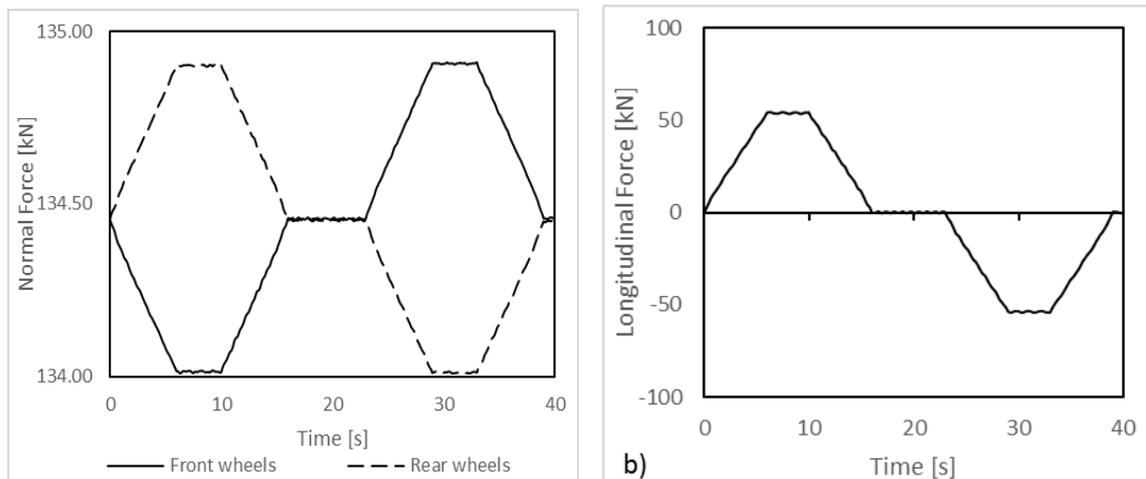


Figure 11. Contact forces of the motor car: a) Normal force; b) Longitudinal force.

For the trailer car, a similar effect happens, as it can be seen in Figure 12, but in a much smaller scale. While the longitudinal contact forces of the motor car are around 50 kN, for the trailer car it is less than 1 kN. And the variation of normal forces between rear and frontal wheels are about 0.075 kN. Such results present many oscillations, which may be attributed to the modeling of contact, done by using the penalty approach (see Gay Neto et al. (2014)).

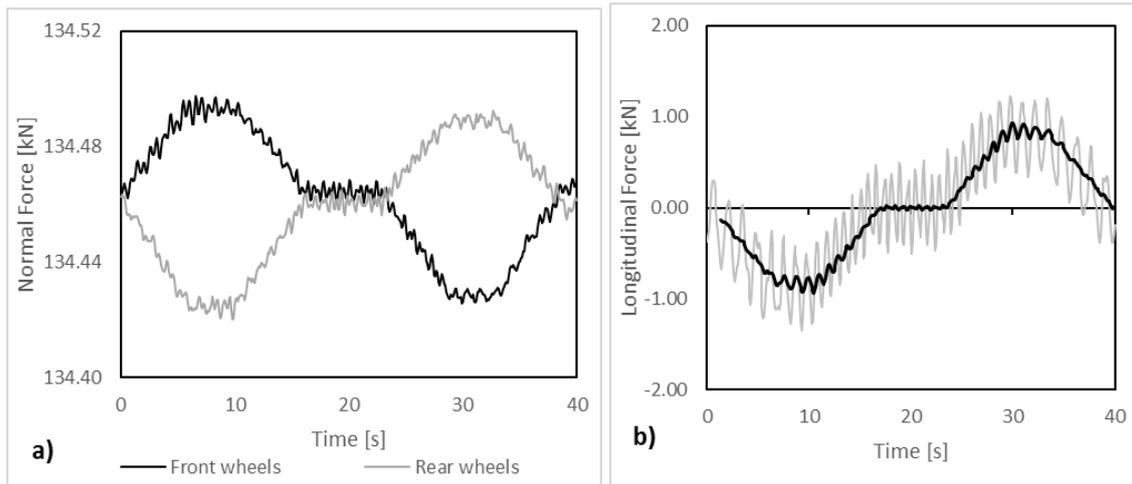


Figure 12. Contact forces of the trailer car: a) Normal force; b) Longitudinal force.

When plotted in the same scale (Figure 13), it is possible to verify that the normal and longitudinal forces of the trailer car stay practically constant when comparable to the motor car, that is, the only one exerting any traction force so the EMU can move.

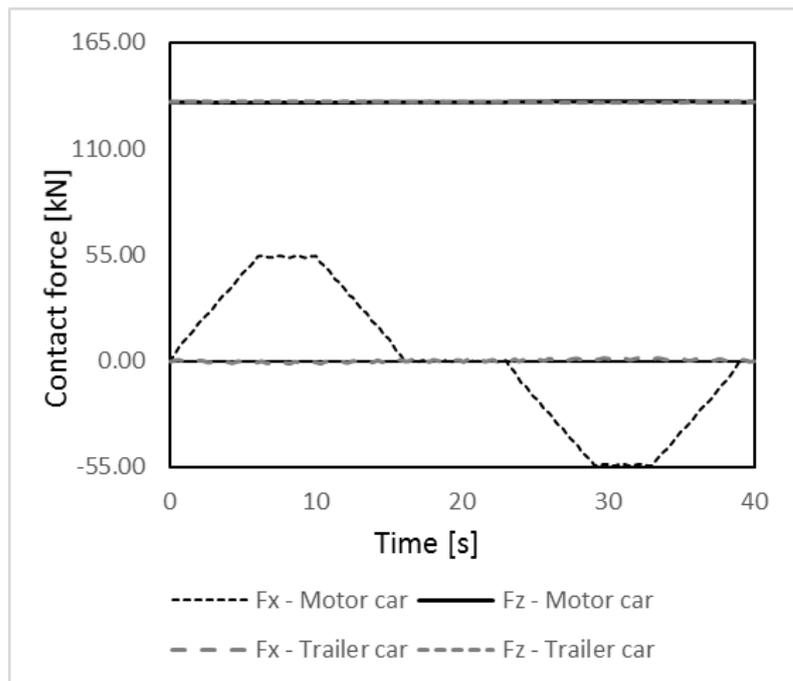


Figure 13. Contact forces of both cars.

4 CONCLUSION

This work shows that *Giraffe* may be used in future studies involving dynamic interactions between a railway vehicle and the track. It permitted also to determine the static loads that the train exert on the track. Although the software still does not consider complex geometries to determine with exactitude the contact forces between rail and wheels, which could detail the interaction between these two structures, it can present the forces in a more general approach, that can be extended to the interactions between track and ground. With the time-variation of the forces, one is able to input the excitation that the dynamic model can impose to the soil, generating vibrations.

In this study, the surface considered was completely horizontal and with no irregularities, which results in a not real situation, since the tracks present deformations. Without that, the variations of the forces during the simulation were not representative and the suspensions did not play their role, fully. Other not considered fact in railways were the discontinuities on track, like the crossings and mechanical joints. These discontinuities create impact of the wheels on track, which amplifies the ground vibrations in these points.

4.1 Future works

In order to represent even more the reality, it is proposed in future works include, in *Giraffe* simulations, irregularities and discontinuities of the track instead of a horizontal plane surface. This can be accomplished by modeling, not only the vehicle, but also the track with new elements.

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