Dynamic Assessment of Freight Rail Vehicles Passing Through a Jog Aperiodic Irregularity

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Many factors can cause accidents with rail vehicles. The main ones are related to the vehicle-load dynamics, rundown tracks and unsafe operations. In order to evaluate the railway vehicle dynamics, a multibody system was implemented to model a railway boxcar with 24 DOFs. Based on the system’s excitation, the vehicle behavior can be evaluated in several operating conditions, and the dynamic system is solved with the Runge-Kutta method for numerical integration. When comparing results from a known model to the ones from the developed program, Dynamic Analysis of Railroad Cars (DARC), a consistent correlation is found. When passing over a jog type aperiodic irregularity, the boxcar of 100 t suffers a vertical acceleration of the car body equivalent to 0.38g and thus lowers load on the wheels for an instant. Hence, it is possible to conclude that this irregularity is severe. Considering that AAR establishes a minimum value of 10% of load over the wheel for traffic conditions, the obtained results show there would not be derailment, as the minimum load obtained on the wheel is of 44% in relation to the static load value.

Keywords: Railcar, Multibody Dynamics, Rail Track Irregularity, Derailment

Introduction

More modern passenger trains currently operate in high speed as a means to reduce costs and shorten travel time. Freight trains have also increased their loading capacity and speed, with projects and operations conducted to prioritize safety. Derailment is a problem observed since the beginning of rail transport. A common derailment criterion is to measure the \( L_d/L_v \) ratio of vertical force \( L_v \) and lateral force \( L_d \) which act on the wheel before a derailment occurs. According to the Brazilian National Land Transport Agency, rail transport is characterized by its capacity of moving large volumes, with high energetic efficiency in long distances. Safety and productivity of rail operations are related to the performance of rail vehicles. Chances of derailment are associated to equipment failures, bad railroad conditions, unsafe train operations and excess of dynamic limits. Hence, it is essential to understand better the vehicle dynamic behavior. Dynamics of multibody systems can be used to solve the non-linear movement equations of rail vehicles. For instance, non-linear forces from wheel-rail contact are considered when a speed train is crossing a bridge.

On the other hand, the optimization of wedge suspensions for three-piece bogies during simulations of freight car models, using parallel multi-objective algorithms, is developed and wedge suspensions with the toe-in angle show superior dynamic performance. Rail corrugation as result of periodic wear can involve severe vibrations between railcar and track, noise and reduction of structural life. In addition, one of the most common aperiodic irregularities on tangent railways is the jog and might feature a bridge. Figure 1 and Table 1 are extracted from Shabana et al. and give the parameters of these irregularities.

Freight truck

A freight railcar is constituted by the car body and two trucks, front and rear, Figure 2(a). Each truck is formed by two wheelsets, one bolster and two side frames, Figure 2(b). The primary suspension of the truck, constituted by roller bearing and adaptor, is placed between the wheel and the side frame, while the secondary suspension, formed by springs and friction wedges, is located between the side frame and the bolster. When elastomeric material exists between the adaptor and the side frame, there is a primary suspension with stiffness in longitudinal, lateral and yaw directions. The group of springs in the secondary suspension has vertical and lateral flexibility in

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parallel to the dry friction of the wedges. Most operating freight trucks show non-linear friction damping characteristics. Relative to the secondary suspension, the friction wedge’s spring can be preloaded into a space between the vertical friction plate of the side frame and the bolster to provide damping. Different suspension models are found into commercial softwares and academic programs, which must be compared with experimental results. 

Wu et al. classify friction wedge suspension models as multibody systems, quasistatic models and force element combinations. Multibody models cover flat and curved wedge surfaces; however, they suffer to be integrated into the vehicle dynamics. Static equilibrium of wedges, bolsters and side frames is assumed for small displacements and a time step lower than 10E-4 s. Then, quasistatic models were developed in vertical, planar and three-dimensional approaches, with or without wedge mass, and with or without toe angles; e.g. wedge suspensions were modeled by Xia as variable DOF systems, changing with the stick/slip states of two-dimensional dry friction.

Combinations models are formed by springs, viscous dampers, friction and clearance elements. According to Wu et al. linear stiffness and damping combinations without friction elements are still appropriate when the nonlinearities of the friction wedge are not important to the concerned simulation results, as could be considered for this work and for practical engineering applications, because of their simplicity and computational efficiency.

**Car body**

The car body is supported by the center plate and the weight is transmitted downwards, to the road, through the bolster, secondary suspension, side frames, primary suspension, wheelsets and wheel-rail contact. The center plate is formed by a circular plate fixed on the vehicle’s body, which freely adjusts itself on the bolster’s center plate, Figure 2. The car body can sway with some constraint of the kingpin or the edges of the center plate. Considering the small spacing allowed, the car body can slide until touching the edge of the plate. When the pitching gets stronger, the car body starts to roll in relation to the bolsters and eventually touches the side bearings. The main objective of this work is to implement a simplified multibody model of a railway vehicle that allows performing dynamic simulations for any type of boxcar. The specific objective is to analyze the vehicle behavior for a jog type aperiodic irregularity on permanent track.

**A methodology of modeling for freight rail vehicles**

**Vehicle characterization**

The majority of products transported by railway are placed inside a box-shaped vehicle. The vehicle is discretized with five masses, linear springs and dampers and nonlinear effects related to the solid length of the springs and gaps. The car body is represented as a rigid body, with a concentrated mass in its CG. The characteristics and dimensions of the rail vehicle of 100 t, considered in this work, are shown in Figure 2, while the corresponding values are given in Table 2. There is no elastomeric pad in the primary suspension between the adaptor and the side frame. As result, the deformation of the primary suspension can be disregarded. The wheelsets and the side frames of the truck are rigid bodies and are considered equivalent to a concentrated mass, termed as truck by simplicity from now and located in the truck’s CG. If the weight of the car body plus the load weight are uniformly distributed, the static clearances (GAP) of the side bearings to the car body are of 6.35 mm. When the car body rolls intensely, this clearance may disappear and the load is divided between the center plate and the side bearing. The friction wedge and the lateral wear plate form a damping system. Consequently, it was necessary to use an analytical method to incorporate the damping effects onto the truck, based on the concept of viscous damping. By observing that the rails, the ballast and

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**Table 1 — Characterization of jog aperiodic irregularity**

<table>
<thead>
<tr>
<th>Function</th>
<th>Parameter</th>
<th>A (cm)</th>
<th>k (m⁻¹)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Alignment (lateral)</td>
<td>1.27 – 8.382</td>
<td>0.0197 – 0.082</td>
<td></td>
</tr>
<tr>
<td>Cross level</td>
<td>4.064 – 7.112</td>
<td>0.0656 – 0.164</td>
<td></td>
</tr>
<tr>
<td>Profile (vertical)</td>
<td>1.27 – 12.7</td>
<td>0.0262 – 0.1476</td>
<td></td>
</tr>
</tbody>
</table>

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**Fig. 1 — Jog deviation shape**

![Jog deviation shape](image-url)
the subgrade are not rigid structures, the elasticity of the permanent track is modeled with lateral and vertical stiffness.

**Model characterization**

A simplified non-linear model with 24 DOFs was implemented for the multibody discretization of the freight car, based on the IIT model\textsuperscript{21}. It simulates a boxcar of 100 t with five masses, describing the car body, the front and rear bolsters, and the front and rear wheelsets-side frames. Translational and rotational DOFs for each mass are defined according to the coordinate systems of Figure 3; disregarding the lateral and longitudinal DOFs for the front and rear bolsters, and the longitudinal DOF for the front and rear wheelsets-side frames. Each center plate is modeled with two vertical springs, disregarding the lateral motion between the car body and the center plate. The reaction of the side bearing is modeled as a non-linear spring with a dead band, because of the gap relative to the car body. The spring has no stiffness while there is the gap, after the contact the spring gets to have a high stiffness. The suspension’s springs, the torsional and bending stiffness between the bolster and the wheelset-side frame, the friction plates and the GIB gap are the elements that provide the interface between the bolster and the side frames.

![Characteristics and dimensions of a freight railcar](image)

Fig. 2 — Characteristics and dimensions of a freight railcar: (a) boxcar, (b) railroad truck
Each truck has four vertical springs, between bolster and side frames, with two springs at each side. The flexibility of the bolster is disregarded. Two non-linear springs of stiffness $k_{BOT}$ allow modeling the end of working space in intense bounce actions. Though the suspension’s springs are in the vertical axis, when the bolster moves laterally, a lateral constraint enters to the system as the bolster is restrained by gibs from moving laterally. These situations are considered in the model by four lateral springs in each truck. Table 3 presents the stiffness values for all springs used in the model, for a spring working space $s_w = 1.016E^{-1}$ m. The truck damping is considered with four dampers parallel to the four vertical springs plus four dampers parallel to the lateral springs of the truck suspension. The equivalent viscous damping coefficient $C$ used in the simulations is 35.605 N s/m, for the boxcar of 100 t in a speed of 40 km/h.

The simplification used when modeling the wheelsets and the side frames implies in adding a torsional ($k_{TBT1}=k_{TBT2}=4.294E7$ N/rad) and pitching ($k_{PBT1}=k_{PBT2}=2.680E7$ N/rad) stiffness between the bolster and the truck. GIB gap is the lateral gap between the bolster and the side frame, modeled by a non-linear spring with a high stiffness $k_{GIB}$.

The rails, ballast and subgrade are modeled with eight springs that represent the track in each direction, vertical and lateral.

**Dynamic analysis and solution method**

Custom software was written in Matlab® platform to obtain the dynamic responses of the system.
composed by 24 equations of motion. The software is termed Dynamic Analysis of Railroad Cars (DARC) and uses the ODE45, which is an available Matlab® function based on the Runge-Kutta method of fourth order for numerical integration. The simulation starts at time zero with the system in static equilibrium, and initial displacements and velocities are given. The variations of the surface of the rails are the source of excitation, which are applied to the model and the resulting accelerations are computed. All accelerations are integrated numerically in order to obtain velocities and displacements in the CG of the masses. These new velocities and displacements, together with the excitation, are the values based on which the accelerations of the next time step are calculated.

Results and Discussion

Willis and Shum developed, validated and used a non-linear mathematical model of a rail vehicle that considers the coupling between the freight element with the car body, the truck motion and the characteristics of the track. This model, which is known as IIT model, was correlated with field test data. The model IIT considers the car body and the load as two separated rigid bodies, while the DARC model assumes both bodies as a single rigid body and disregards any coupler force. In this work, results of the simplified model of the freight rail vehicle DARC were previously compared with results of the IIT model, as a means for validation.

Numerical tests for aperiodic irregularities of jog type

The existence of 3.375 level crossings in Brazilian rails was identified by CNT, and the jog can be used to represent these aperiodic irregularities. Then, DARC is used to simulate a boxcar on a tangent track with one of the most common irregularities in rails, the jog. The boxcar of 100 t is used in broad gauge tracks (1.6 m) and due to the heavier load per wheel. DARC simulations consider a speed of 40 km/h, average that freight rail vehicles use on good quality tracks in Brazil. A jog irregularity, as shown in Figure. 4a, can occur due to variations in the stiffness of the track, e.g., when a very flexible track connects with a rigid bridge. Moreover, traffic and wind loads cause damage on bridges, which can require structural health monitoring. As time goes by, the sleepers weaken and lose their solid grip over the ballast, mainly due to lack of maintenance on the track and heavy load. Figure. 4b shows the vertical acceleration of the car body, which reaches a maximum value of 0.38 g, on the moment of passing over the jog irregularity. Considering the 100 t vehicle loaded and balanced, each wheel would have a static load of 168985 N. The percentage of the minimum vertical dynamic load relative to the static load for each wheel was found by simulation as follows; front truck: wheelset 1 (left wheel 46%, right wheel 44%) and wheelset 2 (left wheel 48%, right wheel 46%), rear truck: wheelset 3 (left wheel 46%, right wheel 44%) and wheelset 4 (left wheel 48%, right wheel 48%).

There are no comparison parameters in regulations or specialized literature for the obtained results in the
study of the aperiodic irregularity. For a boxcar of 100 t, the minimum load value over the wheel (considering the worst case of the eight wheels) was 44%. As AAR\textsuperscript{24} establishes a minimum load value equal to 10% relative to the static load for normal motion condition in rails, the results show that the derailment is not possible.

Conclusions
The methodology of multibody systems allowed modeling a boxcar with its main DOFs, geometric characteristics, force and damping elements. Simplifying hypotheses were used for each mechanical element of the vehicle to reproduce its behavior and to perform dynamic simulations. In order to evaluate the translational and rotational motions of the vehicle, a model with 24 DOFs was programmed, called DARC. The software lets to analyze criteria of performance depending on the permanent track type that excites the vehicle. By passing a jog aperiodic irregularity, the boxcar of 100 t experiences a vertical acceleration of the car body equal to 0.38g and consequently loses load on wheels for a short instant, concluding that this irregularity is severe. However, the minimum load on the wheel is 44% compared to the static load value and there will not be derailment.

References