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EFFECTS OF COIL SPRING THREE DIMENSIONAL STIFFNESS ON FREIGHT WAGON LATERAL BEHAVIOR

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Abstract. *Railway bogies used in freight transportation are usually of the American design three piece type. In this kind of truck the component that effectively supports the wagon car body is called bolster and seats on two sets of coil spring packages that provide vertical, lateral and roll stiffness and damping to the car body movements. In this paper we present a set of multibody simulations of a typical freight rail vehicle supported by springs whose deflection behavior has been extracted from finite element analysis. The structural model shows that even under regular operation ranges, the spring coils tend to contact each other, artificially increasing stiffness.*

Keywords: *finite element, multibody dynamics, railway engineering, helical springs*

1. INTRODUCTION

Three piece bogies, also called three piece trucks, are the main suspension subassemblies for most Brazilian railway freight vehicles. Their current design shows just a few differences from the models that were built in the 1920's, because they are considerably cheap and reliable. The three pieces that compose them are the wheelsets, side frames (which are parallel to the railway) and the bolster, which effectively supports the car body. The bolster is connected to each side frame by a set of springs (usually 7 to 9 packages at each side of the truck, depending on truck design) and by friction wedges (Baruffaldi and Santos Jr., 2010), that are considered as the damping element. In some cases, each of these spring packs consists of a single helical spring, but most commonly there are two or three coils on each pack, with circa 2,5 mm radial spacing between wires. The spring set just described provide not only vertical stiffness, but also resists roll and lateral motions induced, i.e., by curves. On the design of high speed vehicles, the effects of radial spring stiffness on train dynamics is already known and studied and, many times, this characteristic is explored in order to obtain a certain dynamical behaviour (Michàlek and Zelenka, 2015), but this is not true for freight trains.

Our goal in this paper is to perform a initial computational investigation on how the non-axial components of spring pack stiffness can influence the response of a typical freight wagon transverse excitations. To accomplish this objective, numerical simulations were used to determine the three-dimensional stiffness of bogie's coil springs and to introduce this multi-dimensional stiffness on a full wagon model developed using multibody dynamics formalism.

2. COMPUTATIONAL PROCEDURE

To understand the three-dimensional spring stiffness effects on bolster response, two coupled computational models were proposed. The first one is a finite element analysis representation (Stewart, 2014) of a single spring pack composed of two steel concentric coils. The model was simulated using Abaqus and under the following assumptions:

- Spring material is a linear elastic steel ($E = 200\,\mathrm{GPa}$);
- Finite displacements are allowed (leading to geometric non-linearities);
- Springs can frictionlessly contact each other;

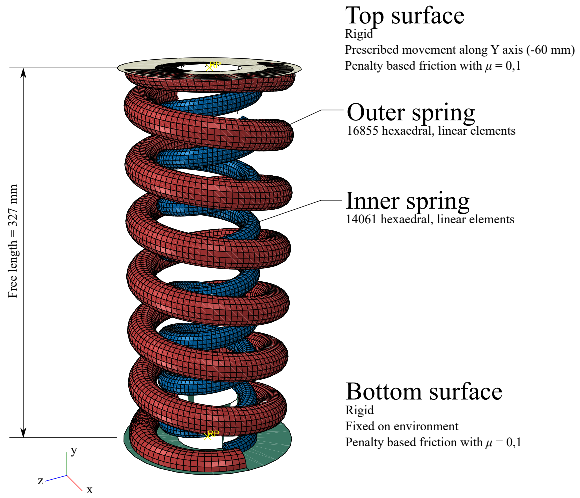

Figure 1. Finite element model of the spring pack.

- Spring ends contact rigid surfaces whose displacements are prescribed at the top and completely constrained at the bottom.

Figure 1 shows the finite element model used to perform the structural calculations. It is composed by four bodies: top surface, bottom surface, outer spring and inner spring. Both top and bottom surfaces are perfectly rigid and contact the ends of the springs. Springs are modeled with three-dimensional elements, most of them hexaedral, with a few thetraehedrons on the grinded endings, where a nice structured mesh could not be obtained. The coils are allowed to contact each other as a result of deflection induced linear buckling. All contact constraints are unilateral, which allows them to separate during simulation. Friction was considered using a penalty Coulomb model with $\mu = 0.1$.

The structural simulation was carried out using three steps: at first, a small y-negative displacement is applied on the top surface, in order to fully establish the contacts. The second step stands for the compression of the springs, when their lengths go from the original $327\,\mathrm{mm}$ to $267\,\mathrm{mm}$ – which is their expected length when under the weight of an empty wagonbox. Finally, the equivalent stiffness matrix between the top and the bottom surfaces is extracted using a variational perturbation procedure, where a small displacement is applied at each output node and the corresponding generalized reaction force is computed to keep static balance. The exact dimensions of the springs will be omitted due to comercial characterization of the product. It is enough to say that this spring pack belongs to a Brazilian heavy haul three-piece truck with constant damping. All finite element analyses were carried out using static/linear perturbation procedures on Abaqus Standard solver.

The resulting stiffness matrix is 6×6, corresponding to the three possible translations and three possible rotations of the rigid body nodes used to represent the surfaces. By denoting the generalized virtual displacements of the top node by $\delta\mathbf{q}$, the stiffness matrix of the spring pack ny $\mathbf{K}$ and the generalized reaction forces by $\mathbf{f}$, and splitting these on translational and rotational components with subscripts "T" and "R" respectivelly, one gets:

$$\begin{pmatrix} \mathbf{f}_T \\ \mathbf{f}_R \end{pmatrix} = \begin{bmatrix} \mathbf{K}_{TT} & \mathbf{K}_{TR} \\ \mathbf{K}_{RT} & \mathbf{K}_{RR} \end{bmatrix} \cdot \begin{Bmatrix} \delta\mathbf{q}_T \\ \delta\mathbf{q}_R \end{Bmatrix} \tag{1}$$

For this study, we used only the $\mathbf{K}_{TT}$ partition of the stiffness matrix on Eq. (1). The bending/twisting moment stiffnesses of the springs are expected to have small influence on truck behavior, since the bolster angular motions are supported by sets of 7 to 9 spring packs pairs – depending on truck model – these sets being separeted from each other by distances between $1000\,\mathrm{mm}$ and $1500\,\mathrm{mm}$: the binary resulting from spring *forces* is much higher than the springs reaction *moments*.

The multibody model consists of nine bodies: bolster, two sideframes, two wheelsets, and four wedges. Motions are prescribed at wheelsets reference points. The wheelsets are connected to the bolster using idealized cylindrical joints. The friction wedges between sideframes and bolster are modelled using four contact points each one with dry friction coefficient $\mu = 0.2$ and are connected to the sideframe by a spring pack. Fourteen spring packs support the bolster above the sideframes. Each of these packs have two concentric springs. In order to simulate applied wagon load, $6500\,\mathrm{kg}$ were added to the mass of the bolster.

Table 1 shows some of the parameters used to assemble the multibody model, while Fig. 2 depicts its graphical representation.

The multibody model was developed on MBSim (Friedrich *et al.*, 2015) and simulated using the time stepping integrator with step size control developed by Clauberg (2013). The simulations were carried out by imposing motions on the wheelsets in two different ways:

Table 1. Model physical properties

Property	Value
Bolster mass	$623\,\mathrm{kg}$
Bolster pitch inertia	$7000\,\mathrm{kg}\cdot\mathrm{m}^2$
Bolster yaw inertia	$99\,000\,\mathrm{kg}\cdot\mathrm{m}^2$
Bolster roll inertia	$97\,000\,\mathrm{kg}\cdot\mathrm{m}^2$
Wheelset mass	$2000\,\mathrm{kg}$
Truck wheelbase	$1.725\,\mathrm{m}$
Truck track	$1.575\,\mathrm{m}$
Friction coefficient	0.2
Structural damping	0.4%

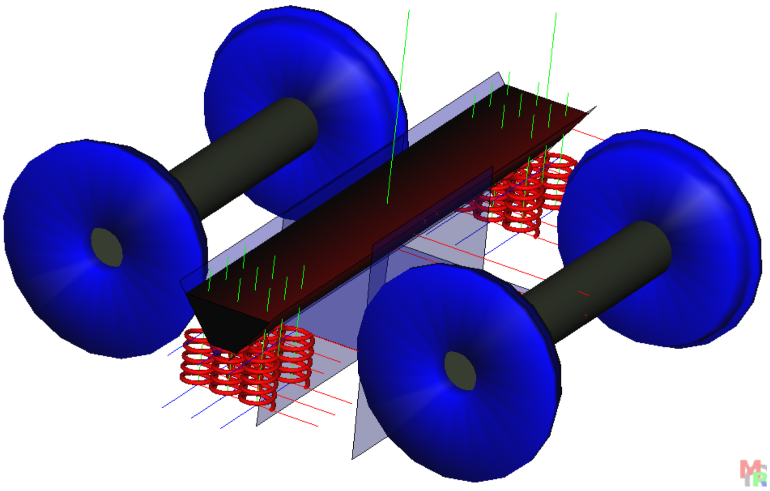

Figure 2. Multibody model

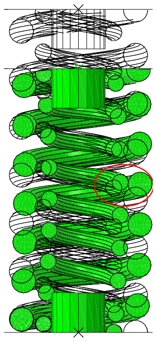

Figure 3. Deformed (green) over initial (wireframe) plot showing the contact between spring coils.

1. Longitudinal excitation with cosine function, $2\,\mathrm{Hz}$ frequency, and $10\,\mathrm{mm}$
2. Combination of the first excitation with rolling motion of the wheelsets, also with a cosine function, $2\,\mathrm{Hz}$ and total amplitude of $0.030\,\mathrm{rad}$ ($0.18°$).

3. RESULTS

3.1 Stiffness matrix extraction

The finite element analyses resulted on the following sitffness matrix (values in $\mathrm{N} \cdot \mathrm{mm}$):

$$\mathbf{K}_{TT}^{A} = \begin{bmatrix} 53.8 & 20.1 & 23.3 \\ 20.1 & 484.0 & -30.6 \\ 23.3 & -30.6 & 28.0 \end{bmatrix} \tag{2}$$

The first noticeable point in Eq. (2) is the difference between radial stiffnesses along the x and z directions. The spring pack is not a symmetric system, thus this difference was expected to some extent. The coils of the inner and outer springs, however, touch each other during simulation after only $15\,\mathrm{mm}$ of deflection. How much, then, of the matrix asymmetry comes from the unexpected contacting between coils and how much is due to different end seating conditions? To try to answer this question, we ran a second analysis, where the contact between springs is not allowed, all other boundary conditions kept the same. The resuling stiffness matrix is $\mathbf{K}_{TT}^{\star}$:

$$\mathbf{K}_{TT}^{\star} = \begin{bmatrix} 31.0 & 7.8 & 39.8 \\ 7.8 & 466.2 & -19.1 \\ 39.8 & -19.1 & 14.3 \end{bmatrix} \tag{3}$$

Clearly, both matrices in Eqs. (2) and (3) have a similar structure: axial stiffness in x is greater than in z and off-diagonal terms are also alike. All values, however, were reduced when compared to the model where contact is allowed: there is a small difference along the axial direction (around $3\,\%$), but transverse stiffenesses can face increases between $75\,\%$ and $100\,\%$ when contact between coils is considered.

The exact effects of contat model, friction coefficient and related considerations on the performance of the spring pack will be left for future studies. In this work we focus on comparing the dynamical behavior of the truck when transverse stiffnesses are or not taken into account. With that in mind, we used the first stiffness matrix, $\mathbf{K}_{TT}^{A}$ modified. That modification consists in creating a second version of the matrix, $\mathbf{K}_{TT}^{B}$ where the rows and columns related to the radial directions were permuted. The final matrix, which was implemented on the multibody code, was than calculated from:

$$\begin{aligned} \mathbf{K}_{TT} &= \frac{1}{2}\mathbf{K}_{TT}^{A} + \frac{1}{2}\mathbf{K}_{TT}^{B} = \\ &= \begin{bmatrix} 40.9 & -5.25 & 23.3 \\ -5.25 & 484 & -5.25 \\ 23.3 & -5.25 & 40.9 \end{bmatrix} \end{aligned} \tag{4}$$

3.2 Dynamic behaviour of the truck using different spring models

The results of the simulation of the longitudinal excitation are shown in Fig. 4. The continous lines represent the truck model with springs as line elements connecting sideframes and bolster, i.e., with only the $\mathbf{K}_{TT_{11}}$ term from Eq. (4). The dashed lines are from the simulation using the full stiffness matrix.

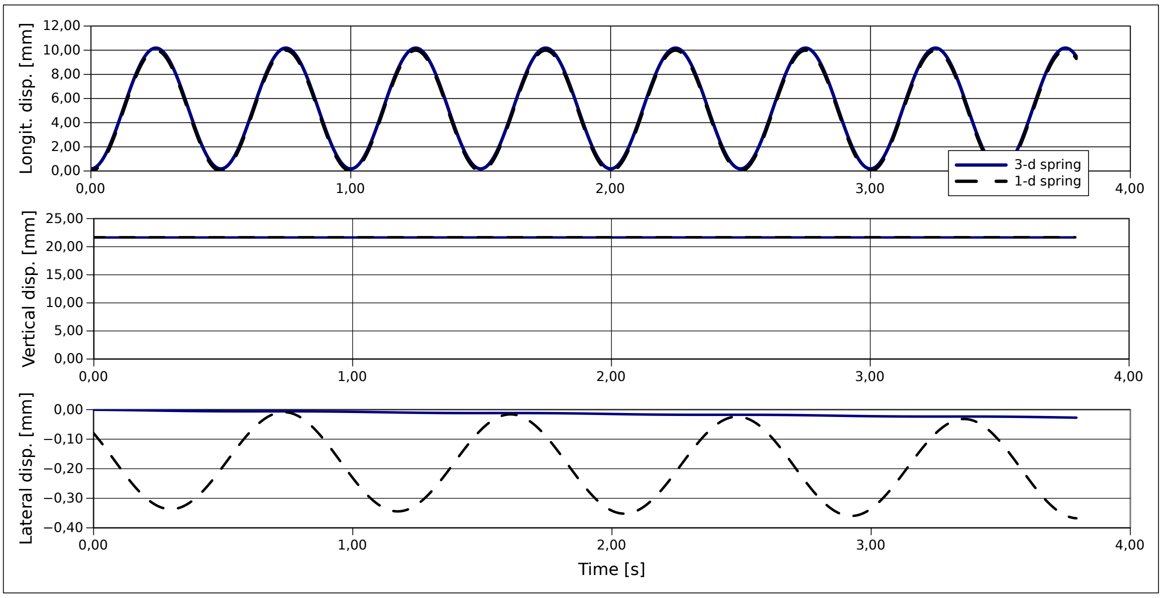

Figure 4. Absolute displacement results of the bolster for a purely longitudinal excitation on wheelsets

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Clearly, for this type of excitation, there is virtually no difference between the two approaches. This was expected from the symmetry of the system. The only noticeable difference occurs on lateral displacements, where the three dimensional stiffness model presents a vibrational behaviour. One can conclude, however, that this oscillation comes from the numeraical drift of the connecting constraints, which starts to impose a small, yet persistent, asymmetry to the system.

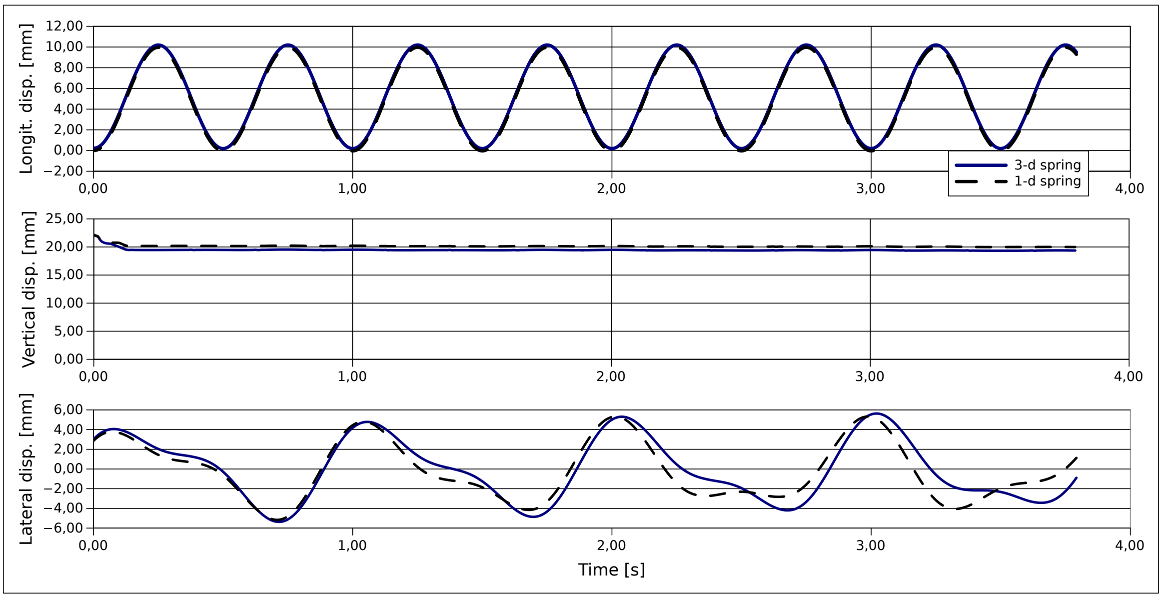

Figure 5. Absolute displacement results of the bolster for combined longitudinal and roll motions of the wheelsets.

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Figure 5 presents the same degrees of freedom plots, but for the simulations with combined roll and longitudinal excitation. Movement of the bolster along the x and y axes (respectivelly, longitudinal and vertical directions) is not significantly changed by the addition of the three-dimensional stiffness.

Lateral vibration, however, suffers some modifications when the single dimension spring model is compared to the full-dimensional one. On the one hand, both responses present similiar frequencies, with the three-dimensional stiffness spring model being of stiffer nature. This suggests that the axial and mixed – non-diagonal matrix terms – stiffnesses play some role, but also that the behaviour is ruled by the lateral components that came from the decomposition of the axial spring force.

The phase diagram displacement versus velocity of the bolster for the combined excitation is depicted in Fig. 6. The results are for a $20\,\mathrm{s}$ simulation and numerical/constraint drifts must be taken into considaration when analysing this

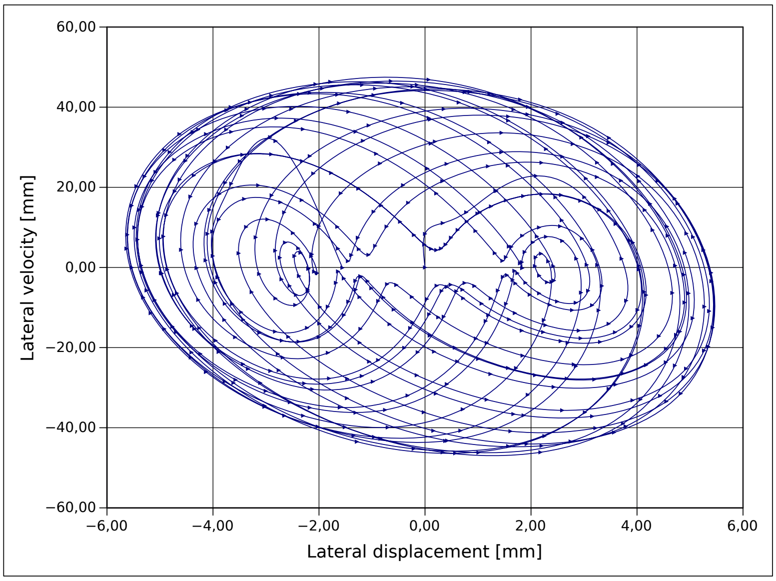

Figure 6. Phase diagram os the lateral motion under combined wheelset excitation.

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figure. Nevertheless, it seems to present two attractors symmetrically placed along the zero-displacement axis. This may indicate a system subjected to deterministic chaos, i.e., a system whose steady state response depends on the values of the initial conditions. Kaiser *et al.* (2002) had already determined that trucks damped with friction wedge could exhibit chaotic behaviour when subjected to pure vertical excitations. Our results suggest that the same may occur for rolling, but resulting on a lateral displacement chaos. Due to the lack of more extensive simulations, however, this conclusion must not be taken for granted and further investigations should be carried out to enlighten the subject.

4. CONCLUSIONS

On this work, a multibody model of a three-dimensional three-piece-truck was presented with its coil springs modeled with two different approaches: (a) the classical assumption of a linear, single-dimensional, force element, and (b) as a three-dimensional force element whose stiffness matrix was extracted from finite element analyses of a single spring package.

The finite element simulations showed that under a 60 mm deflection, which is expected to occur, the spring coils of the inner and outer spirals will establish contact. This increases effective stiffness of the spring on every direction, a phenomenom that can have consequences on suspension design. The multibody simulations carried out using the 1-d and 3-d stiffness models showed very similar responses to longitudinal excitations. In other words, when modeling vibrations that act on the same direction as the rail tracks, choosing between the two spring models is irrelevant, at least for small oscillations, where the friction elements are not active.

If in one hand the longitudinal behaviour of the trucks does not seem to be affected by the presence or not of the complete spring stiffness matrix, on the other roll motions of the wheelsets will induce bolster motions that are not the same when the spring model changes. To completely understand how the train dynamics are affected by the presence of the full stiffness, more analyses must be carried on, but our results suggest that at least for a set of parameters there are some significant differences.

Finally, we found out some oscillatory phenomena on the lateral displacement when roll motion is imposed that may suggest the presence of chaos. Again, no further conclusion can be state at this point, since more extensive simulations must be run to check a wider range of initial conditions and systemaparameters.

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6. RESPONSIBILITY NOTICE

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