Identification of railway track quality and safety through the dynamic inertial response of railway carbody and truck

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Abstract: An extended version of the inertial measuring system installed in the carbody and trucks, is used to evaluate the railway track quality and safety, observed from the vehicle dynamic performance point of view. The system measures the dynamic movements of the carbody and is extended to observe also the truck attitude and suspension torsion due to irregular track geometry. System equations for the inverse carbody and truck dynamic problem, augmented with the suspension torsion equation, are solved to directly estimate the wheels' driving forces, directly correlated with the vehicle safety. The results of a test campaign travelling on the irregular track, identify the full vehicle and trucks attitude and the suspension torsion angular movement. The safety index and the location of the most potential hazard region for track maintenance purposes were identified. Good correlation between safety index and measured track geometry is observed, being this method a promising technique.

Keywords: dynamic; safety; vehicle; railway; track; irregularity; truck; suspension.

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Biographical notes: Roberto Spinola Barbosa is a Professor at the Department of Mechanical Engineering of the Polytechnic School of the University of Sao Paulo, Brazil. His field of research is on vehicle dynamics and multibody systems.

1 Introduction

The railway track is expected to be reliable, available for use and easy to maintain. To guarantee safe traffic conditions also, the operator should keep track geometry standards at the highest quality possible, for an inspection time interval. This maintenance process is expensive due to tamping, ballast cleaning or renewal. Additionally sleeper replacement, joint repair, rail grinding or replacement and substructure treatment and other maintenance interventions are expensive. Railways also seek how to establish explicit processes for decision-making regarding various activities to be undertaken to keep the track infrastructure in satisfactory condition or within desirable/required condition/operational limits.

There are some researchers developing indirect methods for wheel–rail force measurement based on accelerometers and suspension movements and using derailment criteria for wheel set flange on railway vehicles, for track safety evaluation. Wei et al. (2014), proposed wheel set flange derailment criteria for railway vehicles and studied the influence of wheel–rail contact parameters. An indirect method for wheel–rail force measurement based on these derailment evaluation criteria is proposed. A simulation package (*Simpack*) is used to develop a passenger car model to generate wheel–rail forces and vibration signals. A feasibility test is conducted in the test line using a passenger car equipped with a set of strain gauges on the wheelset. The comparison of the force time history applied to the instrumented wheel set and that obtained using the indirect method is presented. The same group proposed an indirect method for wheel–rail force measurement to evaluate the running safety of railway vehicles (Zeng et al., 2016; Real et al., 2012). In this method, the equilibrium equations of a suspended wheelset are derived and the wheel–rail forces can then be obtained from measured suspension and inertia forces. As the wheel–rail lateral forces at two sides of the wheelset are difficult to separate, a derailment criterion by combined use of wheelset derailment coefficient and a wheel-unloading ratio is proposed. The feasibility of using this method to identify wheel–rail forces for low-floor light rail vehicles with resilient wheels is discussed. The value identified by this method is compared with that by Simpack simulation for the same low-floor vehicle, which shows a good coincidence between them in the time domain of the wheelset lateral force and the wheel–rail vertical force by measuring the accelerations on the axle box and the relative displacements of the primary suspension.

Studies on railway vehicle safety and use of low-cost *MEMs* to evaluate performance are still themes of research. Yeo (thesis, 2017) has shown that accurate estimates of the vertical track geometry can be obtained using a relatively compact and inexpensive inertial measurement unit (*IMU*) mounted on the bogie of an in-service train. Information about the rate of degradation of the track can also be gained only by verifying track geometry at sparse intervals producing profile data-sets which are repeatable to within 0.2 mm. Acquisition platform using low-cost accelerometers (González et al., 2018) applied to vehicle dynamics within less than 80 Hz data acquisition, is used for vibration acquisition and comfort studies obtaining optimum results for low-to-medium frequency operations with an error of 2.19% on road tests.

The track safety issue is still investigated by several authors. Xu and Zhai (2017) extract and recognition in the time-domain, irregularities of a track profile and a comparative analyses between the time-frequency distribution of track irregularities and the corresponding dynamic responses of a railway vehicle and track are conducted. Limit values on irregularities' amplitudes and characteristic wavelengths are obtained for dynamic indices, such as vertical and lateral accelerations of a car body and wheel-rail relative lateral displacement. Wheel-rail contact forces are greatly affected by the wavelength components of the irregularity time-series, especially for short wavelengths and peak waveforms. Jiang et al. (2017), proposed a new filtering and smoothing algorithm based on the *IMU*/odometer and landmarks integration for the railway track surveying. In order to overcome the difficulty of estimating too many error parameters with too few landmark observations, a new model with completely observable error states is established by combining error terms of the system. Based on covariance analysis, the analytical relationship between the railway track surveying accuracy requirements and equivalent gyro drifts including bias instability and random walk noise are established. The performance of a new trainset is investigated through dynamic modelling, including five wagons with single active steering wheelset bogies (Molatef et al., 2017). The simulations were carried out by means of Simpack software and the obtained results were evaluated according to EN-14363 standard. The analyses revealed that the lateral forces, derailment coefficients of the wheel and lateral accelerations of the wagons improved significantly by considering steerable mechanism.

The friction models are used for the centre bowl friction joint between the car body and bogies to study curving safety (Opala, 2018). The friction torque in the centre bowl affects the rotational resistance of the bogie in motion relative to the car body and consequently influences the interaction forces between the wheels and rails. Simulation results of safety against derailment index obtained from the vehicle-curving scenario suggest noticeable differences in predictions of the selected models of friction.

Track quality is traditionally quantified with a specialised-on-moving measuring car that measures track geometric basic parameters. Usually, the inspection car measures and records the variation of the track gage, vertical and lateral alignments and cross-level (angular variation on a track section – cant or super elevation). Additionally, the cross-level variation per metre (track twist) can be calculated depending on the data sample rate. Some systems also use the three-point middle-chord technique as a device for particular measurement (Glaus, 2006). Values recorded are confronted with standard recommended limits and harmful regions are identified for maintenance planning. These measuring techniques are focused on measuring the track geometry and local irregularities and compare values to normalise limits. Most of the track measuring systems identify only the track geometry variation within short wavelength identification. Also these systems do not deform the track during the measuring process. Therefore, the real deformed track geometry with the vehicle fully loaded is not recovered.

Additionally to the well-known geometric measuring methods, researches related to the vehicle response characterisation, when travelling over the track irregularities, are observed. Correlation metrics between the track roughness characteristic and the dynamic vehicle behaviour is the key for these methods. Several researchers (Lee et al., 2012) discuss the detection of rail track irregularities, based on the measurements of the bearing box vertical acceleration during the operation of rail vehicles. Wilson and Ketchum (2012) developed a performance-based track geometry (*PBTG*) that is an inspection method based on an accelerometer installed on a conventional track geometry inspection vehicle. The method calculates the vehicle behaviour in real-time based on the measured track geometry input.

The use of instrumented wheel-set is another method to evaluate the effect caused by the track irregularity over the vehicle behaviour in traffic. In spite of being an expensive and laborious instrument, the quantification of the wheel-rail contact force ratio is an indication of track quality (Gullers et al., 2011) and, therefore, the vehicle safety. Also the use of portable accelerometers are employed for passenger comfort measurements (Borgovini, 2012) based on the ISO-2631 and UIC-513 Standards.

Inertial measurement devices (*IMU*) are new technologies developed in the aerospace industry with widespread application in military equipment. These devices are now available in the automotive industry and are particularly used in automotive control systems. Weston et al. (2007) used rate gyros and lateral acceleration for track curvature and alignment monitoring. Xia et al. (2008) used an inverse vehicle model to estimate high-frequency wheel–rail contact forces from measurements of sensors installed in a track-recording car. One can observe in the temporal results, the difficulty of vehicle dynamics correlation with the wheel forces due to high frequency movements of the bodies with reduced weights (e.g., wheelset, side frame, etc.). Therefore at low and medium frequencies the vehicle and truck masses has predominance over the system movements and can be used for a particular application, which is the object of this research.

During its service life, a perfect track develops irregularities that cause vehicle oscillation. In the extreme case, the vehicle can lose its guidance. Defects and failure of the track superstructure and vehicle dynamic performance may be mixed and cause these undesirable derailment events. Focusing on the track geometry defects, structural elasticity, vehicle suspension characteristics and train speed, all these are potential possible contribution causes and should therefore be evaluated together to minimise hazard risk improving safe traffic.

Conversely, track geometry locations that exceed the standardised limits often do not cause obligatory poor vehicle performance. In the contrary there are good track locations under geometric limits that promote unsafe dynamic vehicle performance (Wilson and Ketchum, 2012). The larger the irregularities are, the stronger the dynamic interaction effects. This process is auto propelled and increases the track defects at each passing vehicle. Additionally, depending on the train speed, a particular track roughness wavelength excites the vehicle modal resonance that substantially magnifies the dynamic effect. Safety is a complex phenomenon and depends simultaneously on the vehicle dynamic characteristics and on the track system response and geometry. To optimise track maintenance, would be of interest to include vehicle performance on the track evaluation method. It would be of interest to also identify the problems as they arise rather than waiting for the scheduled inspection campaign.

To handle this subject a new method for track inspection is proposed to complement the traditional ones. Track irregularities excite vehicle vibrations main modes and produce translational and angular movements. Wheel/rail contact forces that support the vehicle vertical load and produce the lateral directing guiding forces cause these movements. The guiding forces are directly related to the vehicle and trucks accelerations. Additionally the vehicle longitudinal torsion stiffness relieves vertical wheel load when travelling over twisted track. Hence, the results of the vehicle and trucks dynamic behaviour can be employed to evaluate the track geometry adequacy. Complementarily to the traditional measuring method, the vehicle and trucks dynamic performance can be used to identify potential place of low safety on the track. The evaluation of these results can be used as metrics to prioritise location of maintenance on the already measured track geometry. This methodology can even more optimise maintenance intervention and improve the vehicle traffic safety.

2 Methodology

The methodology adopted to quantify track quality is to identify where it is more aggressive to the vehicle safety. The specific circumstance for this scenario is three general types of vehicle unsafe conditions. The first condition is the wheel-climb derailment at a low speed on sharp curves. Another such condition is related to vehicle main body large movements. The latter condition is relative to a specific speed and a particular type of track irregularities.

The first condition is mainly related with vehicle suspension stiffness and load distribution. The second is related with the vehicle unsprung mass dynamic movements and directing bogie/wheelset properties. The third is related to the track evenness wavelength, the vehicle natural frequencies and train speed (Barbosa, 2011). Although there are other types of unsafe conditions, including the accidental and component failure ones, those here described are only related to the vehicle body low frequency movements and small energy dissipation.

The methodology proposed and presented here is based on detection of signs of unsafe railway vehicle performance, mainly related to the second and third described types, when travelling on the track evenness. These signs are used to identify the location on the track and prioritise the pertinent track geometry correction for the most harmful irregularity to the vehicle safety.

For this purpose, the metric adopted to identify the potential harmful location is associated with the vehicle safety. An index here called Safety Index (*SI*) is proposed. This dimensionless index is directly correlated to the traditional ratio between the wheel lateral (L) and vertical (V) contact force (L/V). The wheel forces are quantified from the measurement of the carbody and trucks acceleration and attitude due to its overall dynamic behaviour. Additionally on this extended version called *SIRI*-2 (first version of this system *SIRI*, with one measuring device, is already published in Barbosa (2015) the vertical wheel relieve due to suspension torsion is also taken into account. Using four inertial measuring devices (*IMU*) with ten high-resolution transducers and a GPS signal, the safety index (*SI*) can be directly estimated from the wheel driving forces and suspension torsion. This task is performed with an inverse vehicle dynamic model, fed with data acquired from complete vehicle instrumentation, during the transit journey and an inertial navigation algorithm (*INS*) for attitude recognition.

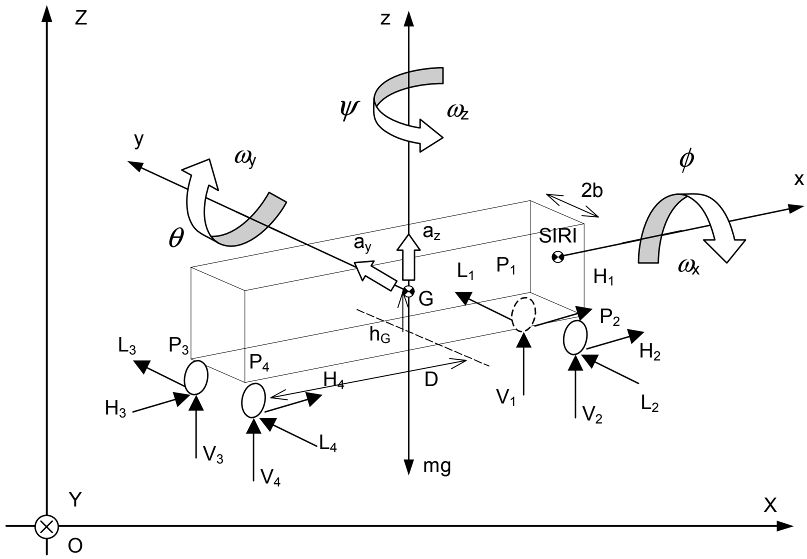
3 Vehicle dynamics

The wheel-rail contact force, due to the vehicle dynamic behaviour, is a function of the track roughness where the vehicle is travelling on. To identify the acting contact forces that produce the vehicle directing movements, it is necessary to solve an inverse dynamic problem (Barbosa, 2016a). The vehicle dynamics is governed by a set of differential equations obtained from the Newton-Euler theorems applied to the car body (considered as a rigid body) valid for a fixed reference frame *N* (*OXYZ*) presented in Figure 1. For the translational movements, the following differential equations relate accelerations and external forces in an earth fixed reference frame:

$$m\,{}^{N}\vec{a}_{G} = \sum \vec{F}^{ext} \;\Rightarrow\; m\,{}^{N}\vec{a}_{G} = \sum \vec{F}_{wheels} - m\,{}^{N}\vec{g} \tag{1}$$

This equation does not consider the drag and Coriolis effects from the Earth rotations due the irrelevant magnitude faced to the vehicle accelerations. The external forces are mainly due to wheel contact forces and gravitational effects as shown in Figure 1.

Figure 1 Vehicle attitude and forces distribution on the wheels



The equation also can be expressed in the body reference frame (*Gxyz*) using a rotational transformation matrix *T*, composed with the three *Euler* angles (roll ϕ, pitch θ, yaw ψ) as identified in Figure 1, where the accelerations are to be measured and the forces computed:

$$mT_N^B\left({}^N\vec{a}_G + {}^N\vec{g}\right) = T_N^B\sum\vec{F}_{wheels} \tag{2}$$

When the measuring system is fixed at particular point *P*, not coincident with the vehicle centre of gravity *G*, the measured acceleration must be projected according to the field acceleration equation, to be used by the *Newton* equation:

$$\vec{a}_G = \vec{a}_P + \dot{\vec{\omega}}\wedge(G-P) + \vec{\omega}\wedge[\vec{\omega}\wedge(G-P)] \tag{3}$$

For the rotational movements described in a moving reference frame attached to the vehicle, the following differential equations relates angular accelerations and velocities (roll rate $\omega_x = \dot{\varphi}$, pitch rate $\omega_y = \dot{\theta}$ and yaw rate $\omega_z = \dot{\psi}$) and external moments is used:

$$[J]_G\{\dot{\omega}\} + [\omega]\wedge[J]_G\{\omega\} = \{M_G^{ext}\} \tag{4}$$

The body external contact forces due to each wheel (H_i, L_i, V_i) are shown in Figure 1. The body external moments (M_G) due to the wheel forces are obtained from the carbody dimensions as shown in Figure 1. To work out the contact forces solving the system equation, it is necessary to know the vehicle body accelerations, as stated in equation 1. Additionally, it is also necessary to measure the angular velocity and to estimate the angular acceleration, needed to solve equation (2). Finally, the body angular attitude must be identified to solve the suspension torsion equation (6), that identifies the vertical wheel load relieve due to track twist.

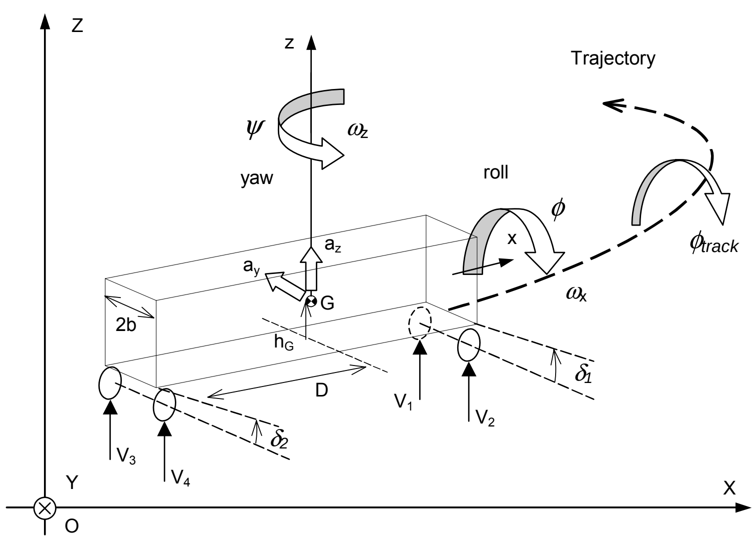
The system has six equations and 12 contact forces unknowns. Disregarding the longitudinal effects, one equation is removed and four longitudinal contact forces are ignored (no acceleration or breaking effects). Due to the system being hyperstatic, the contact lateral forces in each wheelset are summed. To solve the system with five equations and six unknowns, an additional suspension torsion equation is disclosed to access each vertical force relationship, completing the system.

Finally the contact wheel forces (L_i, V_i) is related to the measured accelerations and angular velocities, masses and inertia moments:

$$L_i = f_L(m,[J]_G,a_y,a_z,\omega_x,\omega_y,\omega_z) \text{ and } V_i = f_V(m,[J]_G,a_y,a_z,\omega_x,\omega_y,\omega_z) \quad (5)$$

The vehicle longitudinal torsion due to the track twist (ϕ_{track}) affects mainly the vertical wheel load distribution. Considering the car structure as a rigid body, the track twist deflects the vehicle suspension unloading the diagonal wheels. This effect is internal (not inertial) and depends upon the suspension stiffness, length of the vehicle, track gage and magnitude and wavelength of track twist. In this extended version of the *SIRI* system (Barbosa, 2016b), two additional instrumentation installed in each truck, can quantify the relative inclination. The *IMU* installed in each truck, allow to identify the relative roll angle (δ_i) between the trucks and vehicle body as shown in Figure 2, with the aid of the *INS* algorithm. The vertical wheel load relieve may be related directly to suspension stiffness or to a normalised angular rotation limits (*APTA*, 2017).

Figure 2 Relative vehicle/trucks roll angles



Considering the axle moving as the track on the vertical direction and only the vehicle primary suspension, the expression for the vertical load variation as a function of the track angular twist per meter (ϕ_{track}) is related to a body dimensions proportion ($D/2b$) and suspension torsional stiffness (k_ϕ) stated as:

$$\Delta V = -k_{\varphi} \frac{D}{2b} \varphi_{track} \tag{6}$$

Alternatively the vertical wheel load relieve due to suspension torsion can be quantified with the difference between the trucks absolute roll angles (δ_i) normalised by a torsional limit (δ_{Limit}) stated by the international standard (*APTA*, 2017):

$$\Delta V = (\delta_1 - \delta_2) / \delta_{Limit} \text{ where } \delta_{Limit} = \tan^{-1}(zw / 2b) \tag{7}$$

where zw = 76.2 mm (type *G* vehicle form *APTA*) and $2b$ = 1600 mm (track gage).

To identify the angles and attitude, the inertial navigation algorithm (*INS*) based on extended *Kalman* filter is used. With all this information, it is possible to solve the vehicle inverse dynamic equations to evaluate the driving contact forces and calculate the Safety Index (*SI*) on each wheel.

4 Measuring system and data treatment

The measuring system consists of four inertial measurement units, being two fixed on the vehicle and two fixed on the trucks. A *GPS* and a computer for command actions, data acquisition and storage media is also used. The inertial measurement unit, or simply *IMU*, is a micro-electro-mechanical system (*MEMS*) that measures and reports the body movement. It utilises a set of tri-orthogonal accelerometers to measures the vehicle accelerations ${}^{B}\vec{a}_G$ and angular speed device to measure the attitude variation ${}^{B}\vec{\omega}$. Additionally a tri-orthogonal magnetometer set and a precision barometer measures the orientation ${}^{B}\vec{m}$ based on the earth magnetic filled and the relative level. All the sensors measure the three-dimensional movements of the vehicle and the trucks. A *GPS* identifies the vehicle speed and position expressed in the geographic-referenced latitude and longitude. All these information are synchronised, digitalised, anti-aliasing filtered, and recorded in the on-board control computer.

To recover the complete vehicle and trucks attitude to calculate the SI index, a process based on inertial navigation algorithm (*INS*) is used to treat rough data from the sensor and identify vehicle external loads. Vehicle and trucks accelerations and angular attitudes are the main information to recover from the accelerometers, rate-gyros and magnetometers information. To this end, a strapdown inertial recovery (*SIR*) algorithm and a local level frame identification must be involved for vehicle and trucks angular attitude recognition. An integrated navigation system on terrestrian movement's methodology should combine state data, generated by the dynamic equations, with independent redundant data in a *Kalman* filter algorithm.

The vehicle translational motion expressed in a body-fix moving reference frame *B* is described by:

$$m\,({}^{B}\vec{a}_G + \vec{\omega} \wedge {}^{B}\vec{V}_G) = \sum {}^{B}\vec{F}_i^{ext} \tag{8}$$

(left superscript ^{B} over the vector states for the reference frame used). The vehicle attitude relative to an inertial reference frame *N*, is described by three Euler angles denoting vehicle roll angle ϕ, elevation angle θ and heading angle ψ as shown in Figure 1.

The absolute position of a point in the vehicle is described by the vector ${}^{N}\vec{r}$, expressed in the inertial reference frame *N*, and its time rate of change is:

$$ {}^{B}\vec{r} = T_N^B \; {}^{N}\vec{r} \text{ and } {}^{B}\dot{\vec{r}} = T_N^B \; {}^{N}\dot{\vec{r}} + \dot{T}_N^B \; {}^{N}\vec{r} \tag{9} $$

where T_N^B is the direction cosine matrix (*DCM*) formed with the three *Euler* rotation angles, which leads to the transformation matrix in terms of the three successive sequential rotations (sequence 3–2–1, according to *NASA* Standard, Baruh, 1999):

$$ T_N^B = \begin{bmatrix} c\theta\, c\psi & -c\theta\, s\psi + s\varphi\, s\theta\, c\psi & s\varphi\, s\psi + c\varphi\, s\theta\, c\psi \\ c\theta\, s\psi & c\theta\, c\psi + s\varphi\, s\theta\, s\psi & -s\varphi\, c\psi + c\varphi\, s\theta\, s\psi \\ -s\theta & s\varphi\, c\theta & c\varphi\, c\theta \end{bmatrix} \tag{10} $$

(the prefix ***s*** and ***c*** stands for sine and co-sine). The velocity vector ${}^{B}\vec{V}$ expressed in rotating body fix reference *B*, is defined in terms of position ${}^{N}\vec{r}$ expressed in the inertial fixed frame *N* as:

$$ {}^{B}\vec{V} = T_N^B \; {}^{N}\dot{\vec{r}} \text{ and its time derivative as } {}^{B}\vec{a} = \dot{T}_N^B \; {}^{N}\dot{\vec{r}} + T_N^B \; {}^{N}\ddot{\vec{r}} \tag{11} $$

where the time rate of change of the transformation matrix $\dot{T}_N^B$ and the skew symmetric rotating matrix are:

$$ \dot{T}_N^B = T_N^B \; {}^{N}\Omega_B^N \text{ and } {}^{N}\Omega_B^N = \begin{bmatrix} 0 & -\omega_z & \omega_y \\ \omega_z & 0 & -\omega_x \\ -\omega_y & \omega_x & 0 \end{bmatrix} \tag{12} $$

where ω_i are the three angular speeds expressed on the body reference frame.

The problem of attitude determination involves determining the transformation matrix that maps the on-board sensed information with model transformation to the geographic frame magnetic and gravity field components. For the body-referenced magnetic sensor to match the local geographic-referenced magnetic field, and for the body-referenced accelerometer sensor to match the local geographic-referenced acceleration then:

$$ {}^{N}\vec{m} = T_B^N \; {}^{B}\vec{m} \text{ and } {}^{N}\vec{a}_G = T_B^N \; {}^{B}\vec{a}_G \tag{13} $$

Assuming these two vectors are not parallel, a third orthogonal vector can be produced by the cross product. The matrix formed using these three vectors as columns (superscript ^{T} over the vector states for transposed vector) can be associated to:

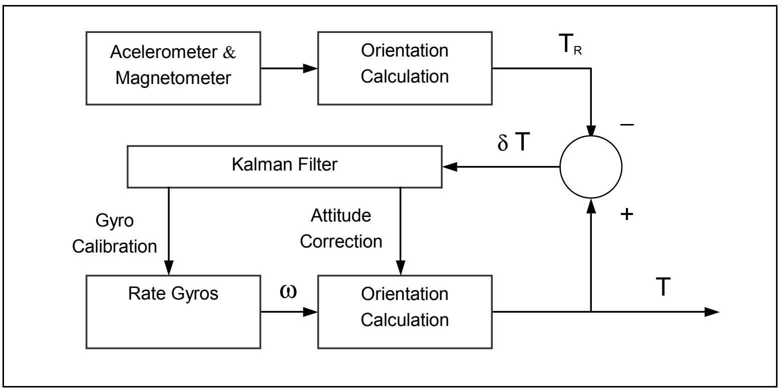
$$ \left[{}^{N}\vec{m}^{T} \quad {}^{N}\vec{a}^{T} \quad \left({}^{N}\vec{m} \wedge {}^{N}\vec{a}\right)^{T} \right] = T_B^N \left[{}^{B}\vec{m}^{T} \quad {}^{B}\vec{a}^{T} \quad \left({}^{B}\vec{m} \wedge {}^{B}\vec{a}\right)^{T} \right] \tag{14} $$

The matrix on the left-hand side is composed by known geographic-referenced information. The matrix on the right-hand side is composed by sensed information. Therefore, the unknown *DCM* orthogonal matrix can be obtained from:

$$ T_B^N = \left[{}^{B}\vec{m}^{T} \quad {}^{B}\vec{a}^{T} \quad \left({}^{B}\vec{m} \wedge {}^{B}\vec{a}\right)^{T} \right]^{T} \left[{}^{N}\vec{m}^{T} \quad {}^{N}\vec{a}^{T} \quad \left({}^{N}\vec{m} \wedge {}^{N}\vec{a}\right)^{T} \right] \text{ and } T_N^B = \left(T_B^N\right)^{T} \tag{15} $$

A better refined estimative for the *DCM* matrix to identify body attitude is obtained using a *Kalman* filter technique (Marins et al., 2001). Typical integration accumulated drifts errors, such as heading vehicle attitude, are to be corrected with multiple cross sensor information. With the accelerometers and the magnetometer a level frame is to be determined. Based on this error difference, an extended *Kalman* filter algorithm corrects and stabilises the rate-gyros orientation calculations as shown in Figure 3. Complementary *GPS* data allows estimating the vehicle speed, alignment and the curvature of the trajectory (Anderson and Bevly, 2010).

Figure 3 Block diagram



The angular description can be on the *Euler* form, or *Quaternion* form, depending on the need to solve the singular problems due to angular quantification. With the accelerations, angular rate and attitude angles, the vehicle guiding force is calculated with aid of a strapdown inertial recovery (*SIR*) algorithm that allows to determine the safety index *SI*. Data is filtered with a low-pass 3 Hz *FIR* filter for the carbody sensors and 15 Hz *FIR* filter for the truck measurements. These values are enough to identify the main carbody vibration modes and meet the typical requirements of *L/V* time duration greater than 50 milliseconds (20 Hz according to *Association of American Railroads – AAR*) for the truck oscillating movements (Barbosa, 2009).

The estimative of the lateral and vertical wheel forces due to the dynamics movements are performed from the body acceleration. These estimative are weighted with the mass-ratio between truck and car-body. Finally the contribution of the suspension torsion is inserted on the ratio.

$$\left(\frac{L_i}{V_i}\right)_{measured} = k_{cb}\cdot\left(\frac{L_i}{V_i}\right)_{carbody} + k_{tr}\cdot\left(\frac{L_i}{V_i}\right)_{truck} \quad \text{and} \quad \left(\frac{L_i}{V_i}\right)_{Total} = \left(\frac{L_i}{V_i \pm (g\cdot\Delta V)}\right)_{measured} \tag{16}$$

where the mass-ratio are: $k_{cb} = m_{cb} / m_{total}$; $k_{tr} = m_{tr} / m_{total}$ and $m_{total} = m_{cb} + m_{tr}$. For a typical empty motorcar it can be adopted $k_{cb} = 0.64$ and $k_{tr} = 0.36$.

The Safety Index is the ratio between the module of the calculated ratio *L/V* value for each wheel and the limit L/V_{Limit} (Barbosa, 2004). This value for a wheelset can be adopted from the international rules between 1.2 and 1.5 (according UIC-518 and AAR-Chapter XI respectively):

$$SI = 1 - \left(\frac{\left| L_i / V_i \right|_{Total}}{L / V_{Lim}} \right) \tag{17}$$

5 Test on the track

The evaluation test of the system was performed in the train of the series 8000, similar to the Figure 4, travelling on the line 8 of *CPTM* in the west side of Sao Paulo city. The test was performed on the track #2 in the west direction, between Station Osasco and Station Comandante Sampaio, as shown in the map of Figure 5. The test was performed outside business hours, in the dawn of 15/07/2018, with all the cars of the train empty.

Figure 4 Train series 8000 from CPTM (see online version for colours)

6 Vehicle instrumentation

The instrumentation implemented on the vehicle consists of four boxes with high precision accelerometers and *GPS*, installed on the car saloon and *IMUs* installed on the trucks, as presented in Figure 6. The truck shown in Figure 7, receive each one, an instrumentation box with *IMUs* fixed on the structure, detailed shown in Figure 8. The carbody instrumentation boxes are aligned with each truck box on car extremities, as shown in Figure 9. Additionally a *GPS* receiver is used to acquire the speed and the georeferenced position and kilometric distance (see *GPS* antenna position in Figure 6).

Figure 5 Line 8 of CPTM (see online version for colours)

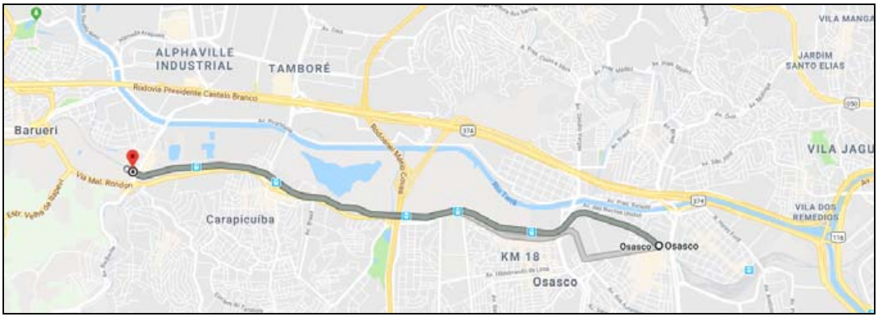


Figure 6 Location of the instruments on the car (see online version for colours)

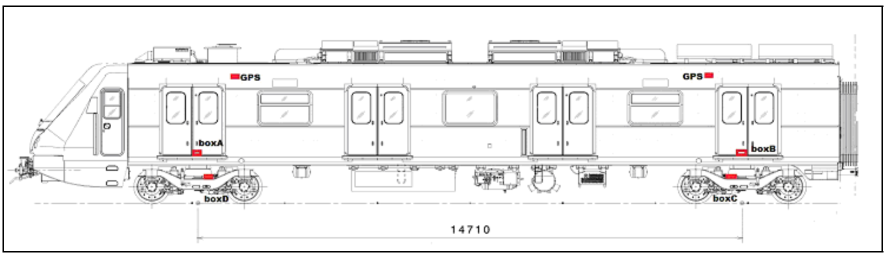


Figure 7 Truck details of vehicle of series 8000 (see online version for colours)

All data measured is time synchronised and stored in a solid-state media (micro SD-card) for post-processing analysis.

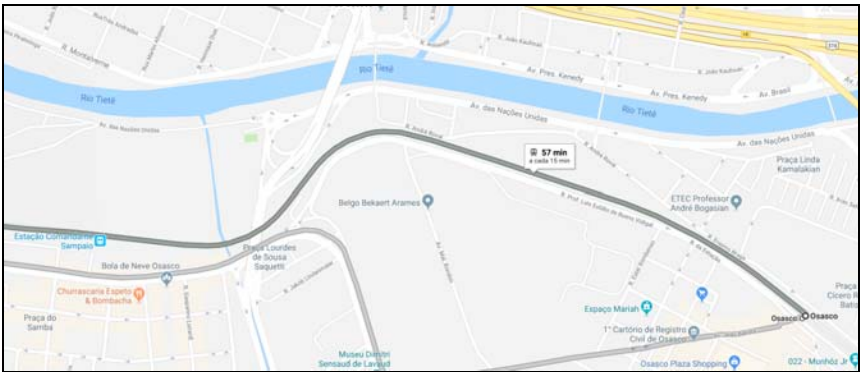
Figure 8 Instrumentation on the truck (*IMU*) (see online version for colours)

Figure 9 Instrumentation installed on the car saloon and truck (see online version for colours)

7 Results and analysis

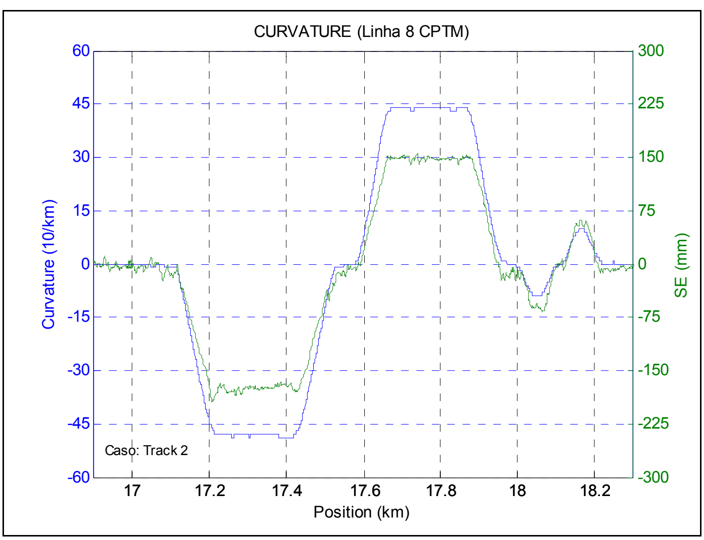
The test was performed on the track #2 of line 8 in the west direction, between Station Osasco (km 15+946) and Station Comandante Sampaio (km 18+253). The results obtained from the test was treated according to the methodology presented in item 4, with the purpose to identify the safety along the track length. A particular section that contains a reverse curve, between km 16+900 and km 18+300, as shown in Figure 10, was separated for a deeper analysis.

Figure 10 Detail of reverse curve between station Osasco and Comandante Sampaio (see online version for colours)



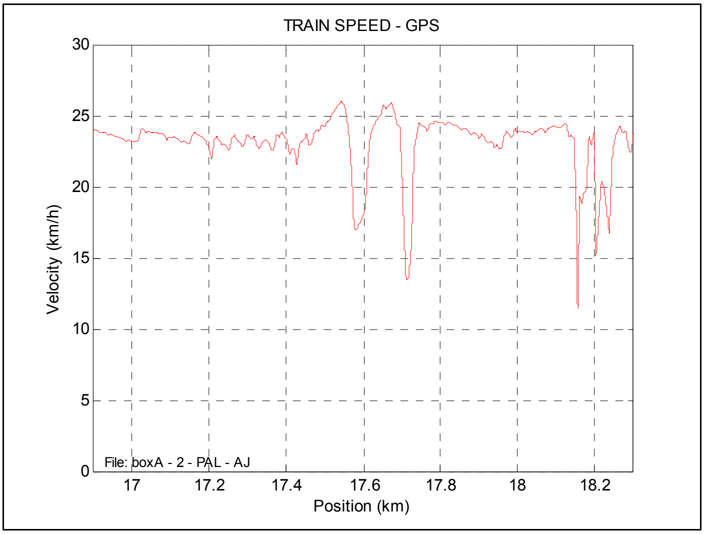
This section contains a reverse curve with 211 m and 330 of radius (superelevation of 173 mm and 153 mm) respectively, as quantified with the measuring car (*Plasser* EM-100 form CPTM) as presented in the Figure 11.

Figure 11 Measured track geometry (curvature and superelevation) (see online version for colours)



The test train passed through this section with a speed of around 25 km/h as shown in Figure 12. The vehicle dynamic behaviour including accelerations, angular speeds and vehicle attitudes were acquired and stored for post-processing analysis.

Figure 12 Train speed from *GPS* (see online version for colours)



Torsion analysis: The sensors in each truck and in the car, measure longitudinal rotation based on an *INS* algorithm (item 4) that are presented in the Figure 13 for the carbody (black line) and for each truck (front and rear truck). These values are precise and stable enough to identify the angular suspension torsion (see Figure 14).

Figure 13 Truck and car longitudinal rotation (IMU-INS) (see online version for colours)

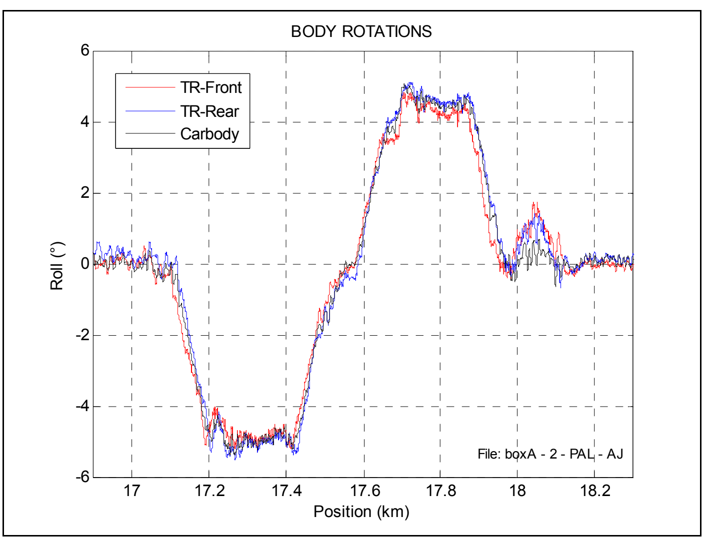
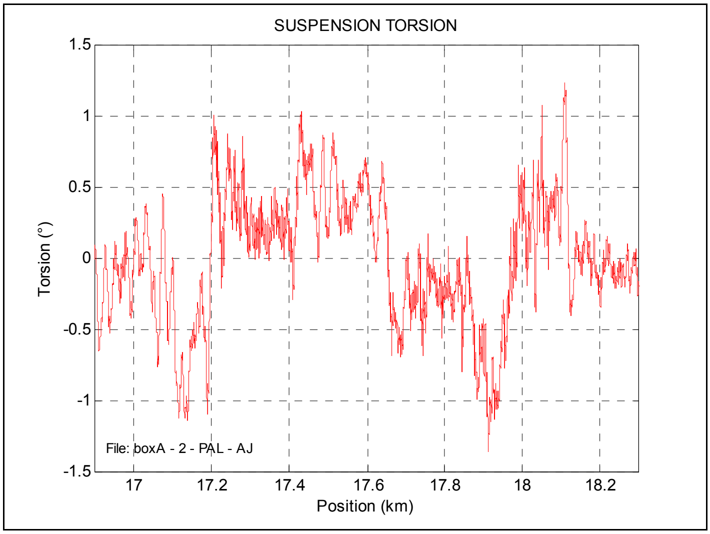


Figure 14 Suspension torsion (*IMU*) (see online version for colours)



This value is used to quantify the wheel load relieve during suspension torsion analysis as presented on item 3. Using the maximum allowed torsion presented in item 4, the calculated percentual vertical wheel load relieve is presented in Figure 15. This dynamic vehicle behaviour may be confronted in the space domain with the geometric measurements performed with the measuring car (see Figure 16, EN 13848-1, 2003). It is observed a good agreement between the on board car inertial measurements (see Figure 15) and the geometric as presented in Figure 16.

Figure 15 Vertical wheel load relieve (see online version for colours)

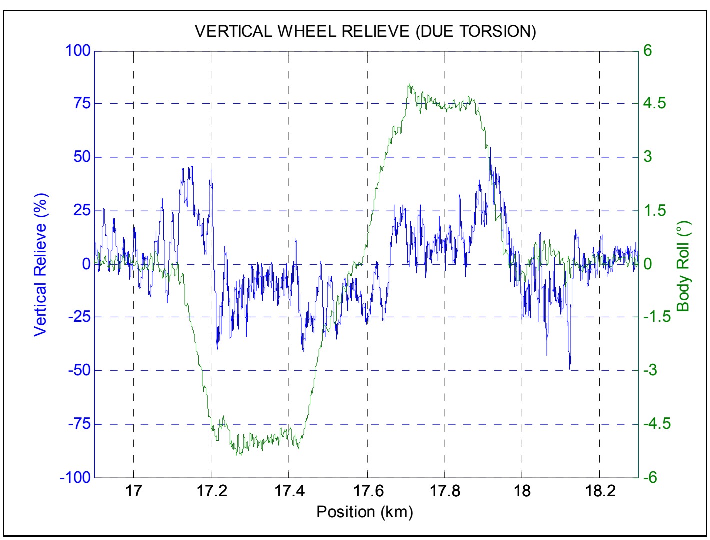


Figure 16 Track torsion and curvature (geometric measurement within 10 m base) (see online version for colours)

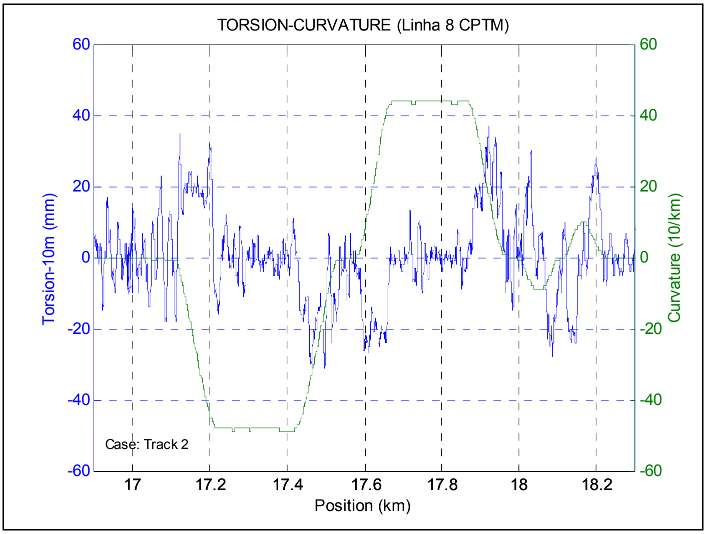
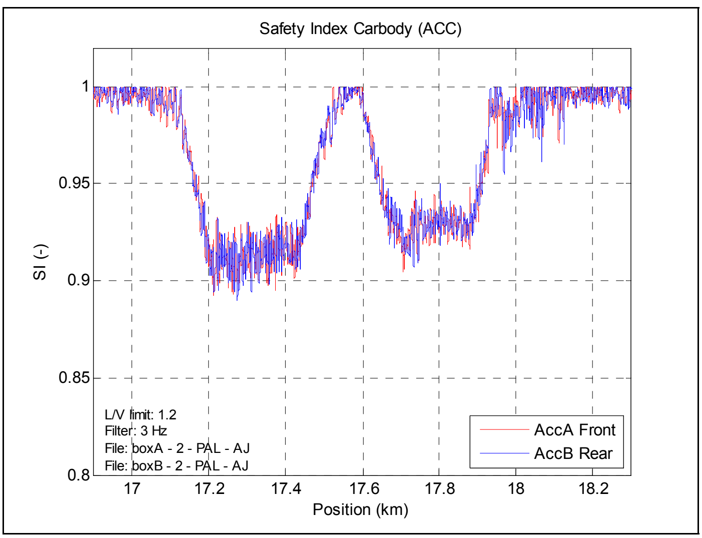


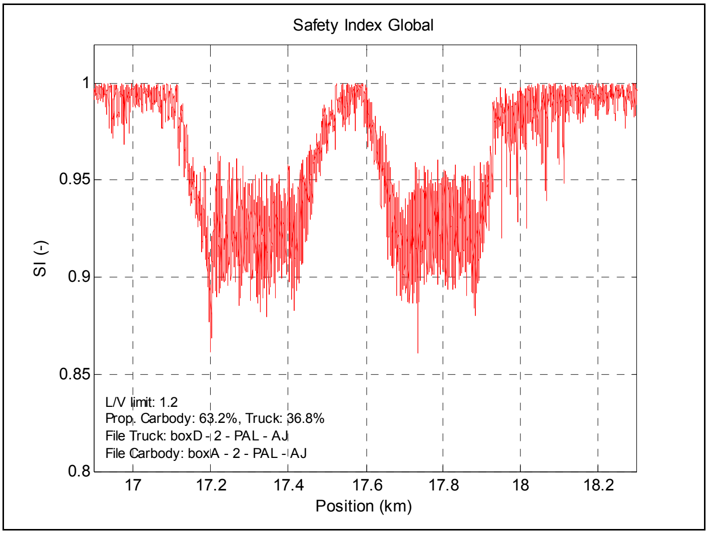
Figure 17 Carbody safety index (see online version for colours)



To compute the wheel loads, it is measure the translational acceleration and angular velocity, to identify the angular accelerations and the car body *Euler* angles as described in item 3. Values for the safety index (*SI*) can be determined at any vehicle extremity. Adopting an L/V limit of 1.2, the resulting values for the safety index of the carbody front end and rear end, are presented in Figure 17 (EN-14363, 2005).

Finally the calculated Global Safety Index (*GSI*) including the carbody and truck inertial effects and suspension torsion is presented in Figure 18, showing the worst safety condition of *GSI* = 0.86 at track location on km 17+200. The values of the GSI considering the inertial response of the carbody and trucks and suspension torsion of presented in the Figure 18, clearly identify the influence of the reverse curves and reveal the good quality of this track section.

Figure 18 Global safety index (see online version for colours)



8 Comments

The general results show good quality of the track particularly due to the low speed of the train during along this test. In this track section the maximum *L/V* are inside the track curvature. The results are related to the speed of the train during the journey. The operating speed is variable depending on the style of the driver, trainload, signalisation, climatic variations and any speed restrictions existing on the track. However in different speeds, forced movements will change its magnitude, modifying the values measured, but keeping the location identified. Even the natural movement induced by periodic irregularities changes, but location remains due to the damping factor of the suspension.

The repeatability of the system is confirmed with different runs during the test campaign. The possibility of evaluating similar vehicles in various load conditions or distinct passengers car fleets is easily performed, only by changing the installation of the measuring devices. The data measured can also be used to evaluate passenger comfort using the vertical and lateral accelerometers signals in accordance with the comfort standard (*ISO* 2631) or even the vehicle modal quantification.

Differently from the other systems that use only statistics information from few sensors, this extended version of the measuring system is *MISO* that takes into account the complete vehicle multisignal input and delivers a single output index directly associated to the safety condition.

9 Conclusions

An extended version of an inertial measuring system is presented to perform the railway track quality quantification, observed from the vehicle performance point of view. With four inertial devices, the system measures the carbody dynamic and trucks movements, including suspension torsion, during its transit along the irregular track. The values measured are used in the strapdown inertial recovery (*SIR*) algorithm with an extended *Kalman* filter, to identify the full carbody and trucks' attitude, including angular positions and accelerations. The vehicle and truck system equations for the inverse dynamic problem, augmented by suspension torsion equation, are solved to directly calculate the wheels driving forces Also the safety *L/V* contact force ratio in the high frequency region due to the truck is identified. The safety index (*SI*), directly associated with the vehicle safety, is determined based on the traditional railway *L/V* ratio. Values obtained are used to qualify track harmful locations.

Results of a test campaign travelling on the irregular track in a conventional train are presented. The full vehicle attitude and movement, including vertical wheel load relieves identification due to suspension torsion, and calculations of the safety index (*SI*) are performed. The track quality results show an overall good track safety.

The values obtained for the *SI* drop down to almost 86%, probably due to track twist in the end of the first curve of the reverse, that promote vertical wheel load relieve of the vehicle. The *GPS* signal simultaneously captures the exact georeferenced location and train speed of the most potential hazard region. The test results were compared to the measured track geometry and a good correlation was observed and the most harmful location was identified for track maintenance purposes. This quantification may complement the other existing geometric measuring tools.

Due to its simplicity and low cost, the extended system can be easily installed in any vehicle and operate with any load condition and variable travelling speed, without the traditional traffic disturbance. The system can be applied to any specific vehicle fleet, travelling in any track section, in the usual operational speed and detect the most harmful location for this specific track, to complement the geometric measuring methods. The analyses can also be focused to compute different priority criteria (passenger comfort, minimal dynamic vertical load applied to the track, instantaneous safety indicator, etc.) according to the user interests. The better classification of the most harmful track locations, allows prioritising the track intervention strategy. The complementary combination of new and traditional monitoring track inspection techniques can help to better understand asset behaviour and produce effective investment efficiency in railway track maintenance, being a promising technique. Future development aims to extend this concept to a system with a full sensor in each rigid body including the wheelsets, all of them time synchronised, and with the corresponding set of equations of motions.

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