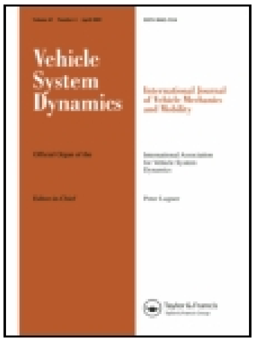

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Identification of the technical state of suspension elements in railway systems

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The running safety and passenger comfort levels in a vehicle are tightly related to the technical state of the suspension elements. The technical state of the suspension depends of the service life time as its components become old and wear out. In this paper, a study on the dynamic behaviour of a railway vehicle is established in relation to the damping elements in one of its suspension stages. An experimental measurement model is developed, obtaining a set of useful signals for the identification of the dynamic parameters of the vehicle and developing a test through the application of the operational modal analysis technique, using least-squares complex exponential method as a basis to validate the numerical model of the multi-body system. Then, the study focuses on developing numerical simulations for the identification of the technical state of the dampers by the registration of dynamic variables under commercial service conditions and on estimating the state of the suspension elements.

Keywords: IRF; LSCE; multi-body model; OMA; railway testing; stabilisation diagram

1. Introduction

The modal properties of a dynamic system are traditionally obtained using techniques known as experimental modal analysis (EMA). EMA techniques have been widely documented [1–3]; nonetheless, for many cases relative to civil and mechanical structures, it is difficult to apply the excitation with an impact hammer, shaker, etc. due to the size, shape, fragility or location of the dynamic system [4].

For systems with auto-motion (aircraft, automobile, ship, train, etc.), it is necessary to know the modal parameters under normal running conditions, considering the particular characteristics of boundary condition, distribution of forces in the frequency domain and level of response [5]; these systems possess normal operating conditions which are difficult to reproduce in a laboratory, for which it is necessary to measure the physical and modal parameters during the operation.

The operational modal analysis (OMA) technique is only based on the measurement of the response signals and uses the environmental excitations as non-measured signals [6].

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The OMA is used for the modal identification with high precision under typical operating conditions [7]. The following are some of the main advantages of the OMA:

- It is possible to test engineering systems for which the excitation through the application of artificial external forces does not allow obtaining correct measurements; there are mechanic systems (e.g. freight train, suburban train and tramway) for which the sources of excitation (creepage phenomenon in the contact patch on the wheel–rail interface) cannot be measured individually in a precise way, and as a result, an erroneous modal model is obtained [8]. The only parameters that can be measured in an exact way are the response data [9].
- *In situ* tests can be developed without interruption and in parallel with other applications. It is not necessary to move the system to a laboratory to realise the tests under controlled conditions. *In situ* tests avoid the need of building a test bed [10]. In consequence, the OMA avoids the system to be submitted to inactive time.
- It is a simple test procedure similar to the operating deflection shapes technique, which employs a reference transducer and various mobile response transducers. The OMA does not use reference transducers in the case where all the responses are measured synchronously; in the other case, one or some transducers are used depending on the number of repeated roots that are obtained. Any of the measurements can be employed as a reference, which means that that the OMA is a multiple-input multiple-output technique and allows to estimate very proximal modes with high precision [8,10].

In this study, a technique (see Figure 1(a)) has been adopted for the evaluation of the technical state of the dampers through the measurement of variables under operating conditions of the vehicle using the OMA–least-squares complex exponential (LSCE) method, meaning that with the excitations of the operating conditions which originate from uncontrolled sources. The present work presents the steps to be followed in order to identify the technical state of the suspension elements.

Using the properties upon which the natural excitation technique [11] is based, the response functions of random noise can be used to determine the impulse response function (IRF). The modal parameters of the system can be identified using standard identification methods in the time domain [10]. In this paper, the LSCE method is used.

The LSCE method is an OMA method in the time domain whose procedure is illustrated in Figure 1(b). The LSCE method determines the relation between the IRF in a multiple degrees of freedom (MDoF) system, its complex poles and residues through a complex exponential. The IRF can be derived from the inverse Fourier transform (FT) for a frequency response function through the random decrement (RD) process or other methods. An auto-regressive (AR) model is constructed by the relation between poles and residues; the AR model solution

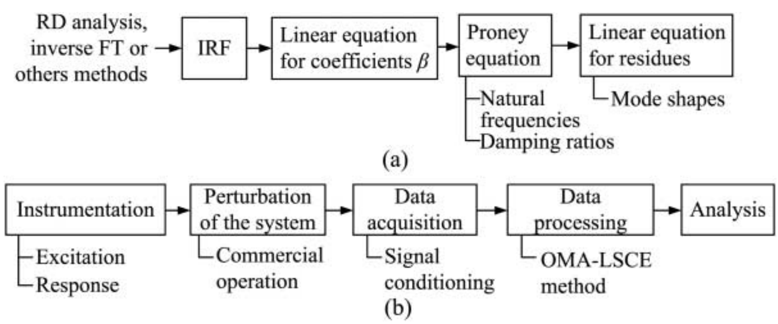

Figure 1. Flowchart of the process: (a) mathematical development and (b) the OMA–LSCE method.

allows to define a polynomial in whose roots the complex roots of the system are present. Having established the roots (equivalent to the natural frequencies Ω and the damping rates ξ), the residues can be derived through the AR model and then the modal shape ϕ_r can be obtained [2].

With the calculation of the system poles (see Appendix 2), it is possible to build the stabilisation diagram, which allows to graphically represent the poles of a system when it is excited in one point (reference) and measurements are made in another one (responses) [12]. It can be seen that the estimated poles for certain frequencies create stable vertical lines. The vertical lines are generated in the characteristic frequencies of the system; then, it is possible to affirm that the pole identifies modal parameters if [10] (i) the pole is stable at the frequency concerned and (ii) the pole frequency appears in the characteristic frequency.

The poles are codified with alphanumeric characters: stable pole (s), vibration frequency and modal vector are stable (v), vibration frequency and stifling are stable (d), only the vibration frequency is stable (f) and unstable pole (o). Once the stable poles in the characteristic frequencies have been identified, it is possible to estimate the pole parameters [10,12]: (i) frequencies of the own vibrations, Ω; (ii) the damping ratio, ξ; and (iii) the modal shape, ϕ_r.

The OMA does not require information of external perturbation forces, but assumes determined particular characteristics in the excitation force; therefore, the OMA has two requirements that imply that the excitation must be stochastic in time and space [13]: (i) the power spectrum (PS) of excitation must be a wide band and soft signal, meaning that the PS is constant and does not have poles or zeroes in the frequency range of interest (for the particular case of study, the range is defined in $\{0.5, \ldots, 3.0\}$Hz) and (ii) the excitation signal must have a uniform spatial distribution.

2. Description of the object of study

This study is applied to passenger vehicles belonging to the mass transport railway vehicle fleet of Medellín city (Colombia), which is a railway system similar to suburban trains. The original equipment manufacturers of rolling stock were *Maschinenfabrik Augsburg-Nürnberg* (MAN) for mechanical components and Siemens for electrical components. MAN has since become Adtranz company and subsequently Bombardier Transportation. The vehicles are similar in geometry and design to the ET420 train sets formerly operated by Deutsche Bahn in commuter service (e.g. the Munich S-Bahn) [4]. The railway system has 42 three-unit cars (see Figure 2(a)). Each car has two bogies that are supported over two axle–wheel sets. Each car has a suspension of two stages: primary and secondary [14].

The primary suspension stage is composed by elastic and damping elements that connect the axle–wheel to the bogie (helicoidal springs, guide leafs and vertical dampers); the secondary suspension stage connects the bogie to the car body (pneumatic spring, traction link and vertical and transversal dampers). The end cars are powered, while the centre unit is an unpowered trailer car. The end car bogies have a DC motor (see Appendix 1) located in transversal position, partially suspended over the axle and over an elastic suspension connected to the bogie frame; there is one 205 kW motor per axle.

An EMA technique [4] has been used to determine the behaviour of the vehicle under a given operating condition. The EMA technique is applied to a set of tests based on the excitation and the response measurement of the vehicle, according to the theory on the classic modal analysis. The excitation of the vehicle is achieved by manual inputs applied directly to the car body, in the location and direction necessary to induce a specific shape mode. In order to be able to generate the excitation, it is necessary to remove the damping elements of the vehicle suspension, a condition denominated $\in_0$ that allows the car body to be excited by

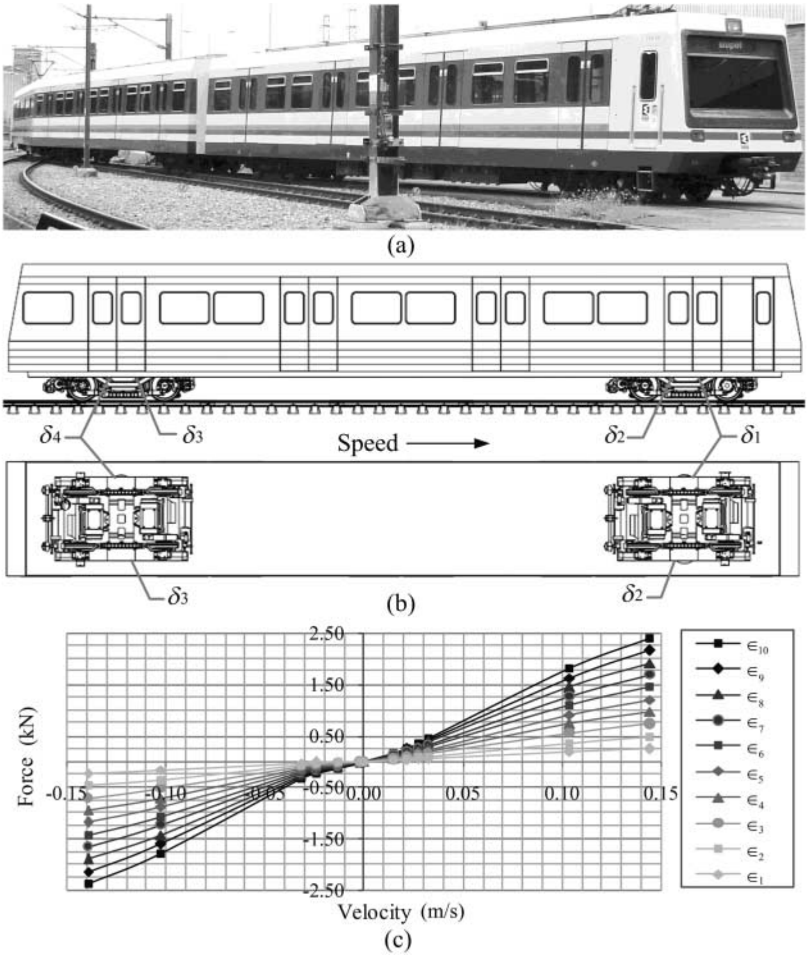

Figure 2. Railway vehicle: (a) unit of three cars; (b) location of the secondary dampers and (c) damping function of the secondary dampers.

an ordinary human force. This consists of simultaneous lifting at the corners of the car body by a group of people per corner. Force is applied to the underframe of the car body side sill or end sill; the issue is applying a rhythmic input force to bring the car to a modal shape resonance.

In order to better capture the signals in the transducers (accelerometers) [15], these have been located at the ends of the car body, collinear to the longitudinal plane of the car body. The data are recorded with a minimum sample frequency of 50 Hz.

The set of recorded signals is transformed to the frequency domain through fast FT algorithm to obtain the characteristic frequencies of the car body, finding the corresponding modal shape [16]. Table 1 presents the results obtained from the EMA technique.

Table 1. Modal parameters identified by the EMA technique, condition $\in_0$.

Modal shape, ϕ_r	Lower roll, ϕ_1	Bounce, ϕ_2	Yaw, ϕ_3	Pitch, ϕ_4
Natural frequency, Ω (Hz)	0.80	1.47	1.50	2.10

This work focuses on the study of the dynamic response of the vehicles [11], which is influenced by the technical state of the four identical vertical dampers that each car has, δ_j with $j = 1, \ldots, 4$ (see Figure 2(b)). This type of component is characterised by a set of physical laboratory tests. The testing method requires the application of cyclical displacement excitation (sinusoidal) in determined ranges of frequency and peak-to-peak amplitudes [4,14]. To define the mechanical characterisation of the damper, the railway of Medellín operator company has provided, for the test, a damper that has been classified as a component in good condition according to its maintenance criteria. Therefore, the tested damper has the following characteristics: (i) it is not new; (ii) it has no faults; (iii) it does not have an ideal mechanical characteristic; and (iv) it has the actual behaviour of a typical damper in the system.

The relation obtained from the characterisation of the element is denominated *nominal damping function*, $\in_{10}$. From $\in_{10}$, a set of nine hypothetical properties are established, $\in_i$ with $i = 1, \ldots, 9$. The hypothetical properties have a behaviour that is similar to that of $\in_{10}$, but affected by a coefficient that reduces the damping property of the element δ_j, $\in_i = \frac{i}{10} \in_{10}$. This is how the different technical states of the component are represented, causing a progressive degradation of the *damping function* (Figure 2(c)).

Three hypotheses are established in relation to the state of the suspension elements δ_j, which is defined parametrically by the progressive degradation of the *damping function*:

- hypothesis ζ_1: consists of the reduction of the damping function, varying the technical state $\in_i$ of the four secondary vertical hydraulic dampers δ_j, with $j = 1, \ldots, 4$;
- hypothesis ζ_2: consists of the reduction of the damping function by varying the technical state $\in_i$ of the two leading dampers (δ_1 and δ_2), with these being the ones located at the closest distance to the driver, and the other two dampers (δ_3 and δ_4) conserve the nominal property $\in_{10}$; and
- hypothesis ζ_3: consists of the reduction of the damping function by varying the technical state $\in_i$ of the two trailing dampers, with these being the ones located at the end opposite to the driver (δ_3 and δ_4), and the other two dampers (δ_1 and δ_2) conserve the nominal property $\in_{10}$.

3. Development of the numerical models

A model of the railway vehicle can be developed and put in motion on a typical track and instrumented in a virtual environment, which allows investigating the effects of a wide range of possible variations of the vehicle parameters [4,11]. The results obtained from a model can provide accurate predictions of the dynamic behaviour of the vehicle and the interaction with the track [17–19].

The general approach of a virtual model is to numerically integrate the ordinary differential equation that constitutes the model by using one or various integration algorithms. This approach is usually denoted simulation or numerical experimentation, which is equivalent to physical experimentation where the system is subject to given conditions and its response is registered. This approach is convenient because it allows [3] (i) different types of models and complexity degrees and (ii) evaluating the response of any type of perturbation.

But it has limitations such as [3] (i) it does not allow identifying the whole behaviour of the system, only the response under defined conditions; (ii) it requires computing resources and generates high computational costs in cases where the system is complex or the model has characteristics that make the numerical integration a difficult procedure; and (iii) it allows the variation effects of the parameters to be predicted only with a high quantity of simulations.

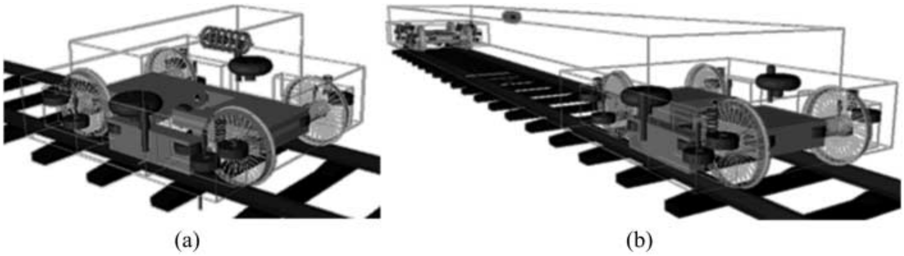

Figure 3. Graphical representation of the numerical models: (a) simple model and (b) complete model.

The numerical model used to simulate the characteristics of the system is considered to be a virtual prototype. The virtual techniques allow the model to generate information on the dynamic behaviour and the interaction of the components, which are only comparable to the physical prototypes. Simulations with numerical models are a valid source of the necessary data to apply the formulated methodology. The virtual techniques allow the model to generate a great quantity of information not only on response and dynamic behaviour, but also relative to the interaction of the components and load details [4,11,18].

The numerical model developed has 120 DoFs [14], consists of the union of two simplified models (see Figure 3(a)) and represents a complete motor car (see Figure 3(b)). It is based on the multi-body system theory using the analysis software VAMPIRE® [20]; the values of the particular parameters of the model are given in Appendix 3.

4. Test development in a virtual environment

It is necessary to point out the guidelines for the correct development of the test. The coherent design of the experiment allows a correct analysis of the data and it must be designed to evaluate the secondary suspension under normal operating and controlled conditions, without affecting the operation and the security of the system. The test is defined following these conditions: (i) the necessary load condition (AW0), meaning that the car is empty; (ii) the section track is a straight track and the track irregularities are considered [21]; (iii) the vehicle speed, $V = 80\,\mathrm{km/h}$; and (iv) a variation of the secondary suspension dampers technical state is assumed $\in_i$ (see Figure 2(c)).

The measurement points are defined according to the railway international standards [22]. Given that the interest of this work is the study of the secondary-stage vertical suspension, the signals to be recorded must be in the main bodies of the system. In this way, three signals in the vertical direction are chosen and located as follows:

- $\ddot{Z}^{*}_{qI}$, on the floor of the car body, at leader side (Figure 4(a));
- $\ddot{Z}^{+}_{qI}$, in the bogie frame near to the attack axle (Figure 4(b)); and
- $\ddot{Z}^{o}_{qI}$, in the axle box of the attack axle (Figure 4(c)).

5. Processing and results obtained with the OMA–LSCE method

The OMA–LSCE method is applied to the signals acquired during a test performed on a typical operation in a straight commercial track. The OMA–LSCE method must be applied to an excitation signal with a broad and soft band, this means that the PS must be constant and

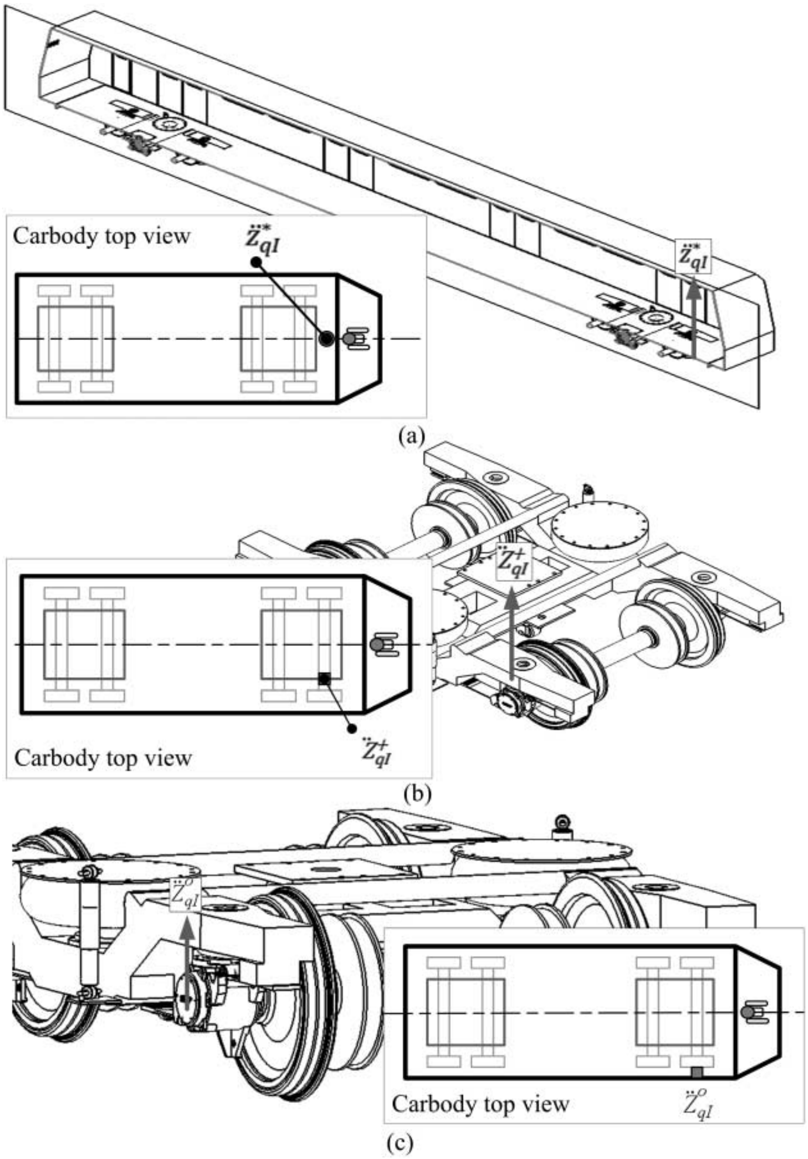

Figure 4. Location of the sensors: (a) in the car body $\ddot{Z}^{*}_{qI}$; (b) on the bogie $\ddot{Z}^{+}_{qI}$ and (c) in the axle box $\ddot{Z}^{o}_{qI}$.

without poles or zeroes in the frequency range of interest. Furthermore, the excitation force must be specially distributed in a uniform way [16]. This means that the vehicle has a stable state behaviour; therefore, the elements with nonlinear properties work in a narrow range, that is, the vehicle response tends to be highly linear.

The set of signals $\ddot{Z}^{*}_{qI}$, $\ddot{Z}^{+}_{qI}$ and $\ddot{Z}^{o}_{qI}$ is processed by the OMA–LSCE method, based on the measurement of response of the system to the excitations of the operation and from which the global modal parameters are estimated: Ω, ξ and ϕ [12]. The modal parameters are graphically represented by the stabilisation diagrams. Appendix 4 presents as an example a set of stabilisation diagrams obtained from different operating conditions of the vehicle. In the diagrams, it is possible to identify

Table 2. Modal parameters identified by the OMA–LSCE technique.

Technical state of damper, $\in_i$	ζ_1				ζ_2				ζ_3			
	ϕ_2		ϕ_4		ϕ_2		ϕ_4		ϕ_2		ϕ_4	
	Ω (Hz)	ξ (%)	Ω (Hz)	ξ (%)	Ω (Hz)	ξ (%)	Ω (Hz)	ξ (%)	Ω (Hz)	ξ (%)	Ω (Hz)	ξ (%)
$\in_1$	1.479	0.820	2.095	1.727	1.480	0.851	2.102	1.994	1.498	1.768	2.159	1.594
$\in_2$	1.480	1.001	2.112	1.882	1.480	1.001	2.112	1.882	1.502	1.987	2.160	1.785
$\in_3$	1.484	1.364	2.128	2.070	1.484	1.364	2.128	2.070	1.520	2.752	2.166	1.910
$\in_4$	1.491	1.876	2.144	2.136	1.491	1.876	2.144	2.136	1.526	2.745	2.170	2.037
$\in_5$	1.504	2.395	2.159	2.154	1.504	2.395	2.159	2.154	1.544	3.965	2.179	2.300
$\in_6$	1.515	2.757	2.167	2.190	1.515	2.757	2.167	2.190	1.559	5.410	2.184	2.596
$\in_7$	1.517	3.105	2.177	2.255	1.539	5.567	2.224	9.562	1.551	3.696	2.186	2.308
$\in_8$	1.543	3.239	2.184	2.230	1.543	3.239	2.184	2.230	1.552	3.131	2.189	2.224
$\in_9$	1.553	3.363	2.190	2.248	1.553	3.363	2.190	2.248	1.558	3.294	2.192	2.263
$\in_{10}$	1.561	3.449	2.195	2.273	1.561	3.449	2.195	2.273	1.562	3.511	2.196	2.322

- two local maximums in the range {1.0, ..., 2.5} Hz, which correspond to the two modal shapes (bounce ϕ_2 and pitch ϕ_4) and are registered in the car body by the sensor $\ddot{Z}^*_{qI}$;
- three local maximums in the range $\{4, \ldots, 10\} Hz$, which correspond to the three modal shapes recorded from the bogie by the sensor $\ddot{Z}^+_{qI}$; and
- three local maximums in the range {15, ..., 25} Hz, which correspond to the vertical irregularities of the track and are obtained from the signal registered by the sensor $\ddot{Z}^o_{qI}$.

This work focuses on the analysis of the vehicle, more specifically on the car body for which it has been possible to identify two vertical modal shapes (ϕ_2 and ϕ_4) under the load condition AW0, see Table 2. It is noted that the damping rate ξ has low values, and this is because the numerical model of the vehicle has a simple air spring modelling, which does not consider the damping effects of the change of area stiffness, the orifice damping and term relating to the surge pipe.

6. Analysis of the results

From the set of values of the modal parameters identified for the different hypothesis ζ_i, it is possible to show the existing relationship between the technical state of the damping set and

- the natural frequency, Ω, of the modal shapes ϕ_2 and ϕ_4, obtaining regressive linear models with a correlation coefficient value of $\sqrt{R^2} > 0.98$ (see Figure 5), and
- the damping rate ξ, obtaining regressive exponential models with correlation coefficient values of $\sqrt{R^2} > 0.96$ (see Figure 6).

The regressive models are considered valid given that the values $\sqrt{R^2}$ represent an association measurement of the statistical model with the obtained data [23], which have an acceptable level for the scope of this work.

7. Validation of the test

The regressive models belonging to ζ_1 and obtained by the OMA–LSCE method in a virtual environment are extrapolated to the value of the technical state of the damper with a null

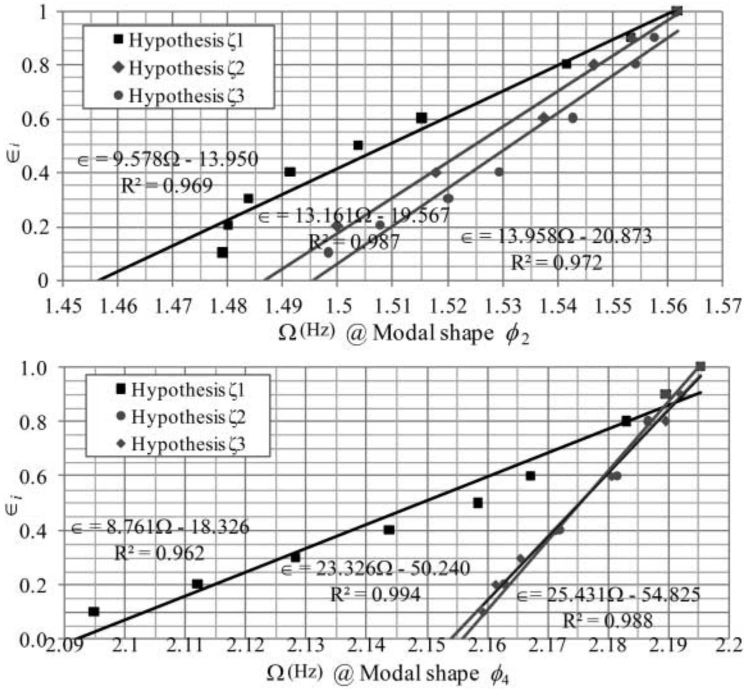

Figure 5. Development of regressive models ζ and modal parameter Ω.

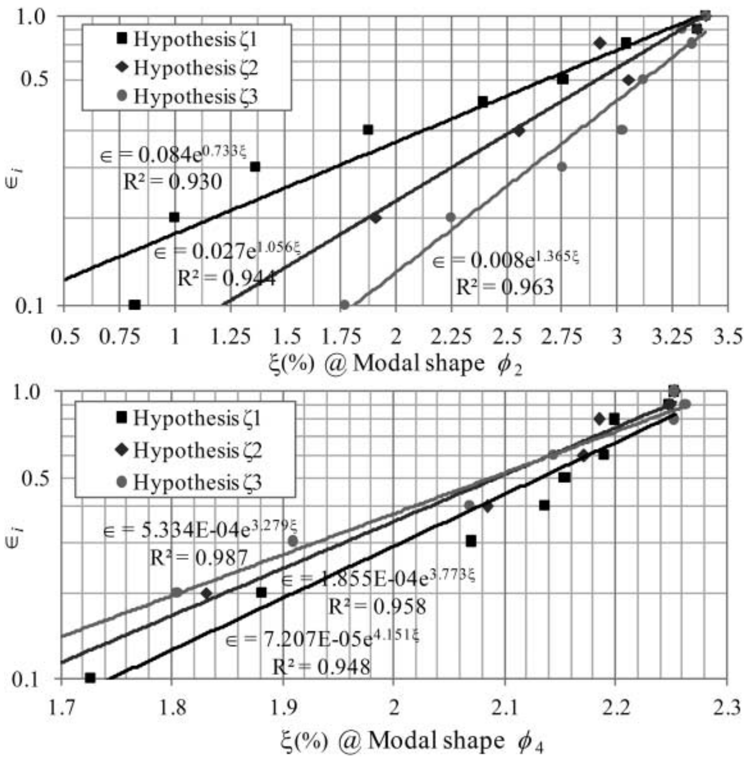

Figure 6. Development of regressive models ζ and modal parameter ξ.

damping function, $\in_0$, for the modal shape ϕ_i, meaning that

$$\text{for } \phi_2, \in(\Omega) = 9.577\Omega - 13.950 \therefore \text{ if } \in = \in_0 \longrightarrow \Omega = 1.46\,\text{Hz},$$
$$\text{for } \phi_4, \in(\Omega) = 8.761\Omega - 18.326 \therefore \text{ if } \in = \in_0 \longrightarrow \Omega = 2.09\,\text{Hz}.$$

In this way, the dynamic behaviour of the vehicle without the secondary damper is obtained, emulating the whole extraction of the damping elements of the suspension δ_j with $j = 1, \ldots, 4$, which is the same testing condition applied to the vehicle with the EMA technique.

The values reordered by the two types of analyses, EMA and OMA, are compared (see Table 2), which allows for observing that there is an estimation error of the modal parameter, $\varepsilon < 1\%$. Therefore, it is possible to consider the regressive models valid given that the ε values represent an acceptable deviation level for the scope of this work (Table 3).

8. Case study and application of the regressive models

Given the set of signals, $\ddot{Z}^*_{qI}$, $\ddot{Z}^+_{qI}$ and $\ddot{Z}^o_{qI}$, obtained from a vehicle, the data are processed through the OMA–LSCE method and the following identification of the modal parameters is obtained:

- natural frequency: $\Omega = 1.53\,\text{Hz}$ for ϕ_2, and $\Omega = 2.18\,\text{Hz}$ for ϕ_4; and
- damping rate: $\xi = 2.90\%$ for ϕ_2, and $\xi = 2.20\%$ for ϕ_4.

Starting from the identified modal parameters and based on the series of regressive models obtained for the vehicle, the probable state of the damper is identified $\in_i$, for each one of the hypothesis ζ_i. The regressive models in the identification of the modal shape ϕ_2 are as follows:

$$\text{for } \zeta_1, \in(\Omega) = 9.577\Omega - 13.950 \quad \therefore \quad \in(1.53) = 0.70,$$
$$\in(\xi) = 0.084\text{e}^{0.733\xi} \quad \therefore \quad \in(2.90) = 0.70;$$
$$\text{for } \zeta_2, \in(\Omega) = 13.161\Omega - 19.567 \quad \therefore \quad \in(1.53) = 0.57,$$
$$\in(\xi) = 0.027\text{e}^{1.056\xi} \quad \therefore \quad \in(2.90) = 0.58;$$
$$\text{for } \zeta_3, \in(\Omega) = 13.958\Omega - 20.873 \quad \therefore \quad \in(1.53) = 0.48,$$
$$\in(\xi) = 0.008\text{e}^{1.365\xi} \quad \therefore \quad \in(2.90) = 0.42;$$

in the modal shape ϕ_4 are:

$$\text{for } \zeta_1, \in(\Omega) = 8.761\Omega - 18.326 \quad \therefore \quad \in(2.18) = 0.77,$$
$$\in(\xi) = 7.207 \times 10^{-5}\text{e}^{4.151\xi} \quad \therefore \quad \in(2.20) = 0.67;$$

Table 3. Modal parameters identified by the EMA and OMA techniques, condition $\in_0$.

	Modal shape Ω(Hz)	
Modal analysis type	ϕ_2	ϕ_4
EMA	1.47	2.10
OMA	1.46	2.09
Error, ε (%)	0.91	0.04

Table 4. Evaluation of technical state $\in$ in order to obtain ζ_i.

	Modal parameter		Technical state of damper $\in$ (%)		
Modal shape, ϕ_r	Description	Value	ζ_1	ζ_2	ζ_3
ϕ_2	Ω (Hz)	1.53	0.70	0.57	0.48
	ξ (%)	2.90	0.70	0.58	0.42
ϕ_4	Ω (Hz)	2.18	0.77	0.61	0.61
	ξ (%)	2.20	0.67	0.75	0.72
Mean value, $\bar{X}_i$			0.71	0.63	0.56
Standard deviation, σ_i			0.04	0.08	0.14

$$\text{for } \zeta_2, \in(\Omega) = 23.326\Omega - 50.240 \quad \therefore \quad \in(2.18) = 0.61,$$

$$\in(\xi) = 1.855 \times 10^{-4}\mathrm{e}^{3.773\xi} \quad \therefore \quad \in(2.20) = 0.75;$$

$$\text{for } \zeta_3, \in(\Omega) = 25.4316\Omega - 54.825 \quad \therefore \quad \in(2.18) = 0.61,$$

$$\in(\xi) = 5.334 \times 10^{-4}\mathrm{e}^{3.279\xi} \quad \therefore \quad \in(2.20) = 0.72.$$

The $\in$ values from the hypothesis ζ_i are given in Table 4. The mean of the technical state is calculated, $\bar{X}_i$, as well as its corresponding standard deviation, σ_i; the later will be the criterion that defines the valid hypothesis ζ_t. ζ_t that presents the least standard deviation is the one that adapts to the obtained dynamic characteristics of the system and, therefore, the value $\bar{X}_i$ of such a hypothesis must be the technical state of the damper $\in$, meaning that $\zeta_t = \bar{X}_i(\in)\nabla \min(\sigma_i)$.

The technical state of the damper is $\in = \bar{X}(\in) \pm 2\sigma$ with 95.45% of confidence factor. For the given case study, the value of the technical state is $\in = 0.71 \in_{10} \pm 0.1$ in the four secondary-stage vertical dampers (hypothesis ζ_1) with a confidence of 95.45%. This means that the dampers are degraded 29% relative to the nominal characteristic.

9. Conclusions

The OMA–LSCE technique is a suitable technique to evaluate and identify the technical state of the dampers $\in_i$ and the operating characteristics of the components of passenger railway vehicles.

The vertical dampers of the secondary suspension stage present a direct influence on the dynamic behaviour of the modal shapes, ϕ_2 and ϕ_4; therefore, the different technical states of the damper $\in_i$ can be tested and estimated through the dynamic recording in the car body. This means that from the sensors installed in the car body that register the natural frequency Ω appropriately, it is possible to determine the variation of the damping function of these elements and to infer their degradation or failure.

The dynamic parameters, Ω and ξ, have a high degree of dependence over the secondary suspension dampers δ_j.

An experimental model of three measurement points, $\ddot{Z}^*_{qI}$, $\ddot{Z}^+_{qI}$ and $\ddot{Z}^o_{qI}$, obtained from a set of sensors can be used to identify the dynamic parameters, Ω and ξ, in a vehicle with load condition AW0 with the OMA–LSCE method.

A methodology has been proposed and applied for the evaluation of the technical state of the dampers $\in$ and the identification of the hypothesis of suspension element deterioration ζ_i through measurements of variables under operating conditions of the vehicle.

International standards for railway vehicles define the range for deficient frequencies $\omega = \{8, \ldots, 10\}$ Hz. The human body is sensible to vertical accelerations [24]; frequencies $\omega \approx 10$ Hz cause excessive oscillations on ϕ_2, generating a significant deficiency of comfort in the vertical (spinal) direction for both seated and standing passengers [25]. On comparison of the values obtained for the analysed vibration modes ϕ_r and under the different technical states of the damper $\in_i$, the natural frequency is $\Omega > 2\,\text{Hz}$. Therefore, the degradation of the damping function of the suspension elements does not lead *per se* to any violation of the railway standards.

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Appendix 1. Notation

Abbreviations, acronyms, coefficients and constants

Nomenclature

AR	Auto-regressive model
AW0	Empty car, null passenger load
DC	Direct current
DoFs	Degrees of freedom
EMA	Experimental modal analysis
FFT	Fast Fourier transform
FRF	Frequency response function
IRF	Impulse response function
LSCE	Least-squares complex exponential
MAN	*Maschinenfabrik Augsburg-Nürnberg*
MDoF	Multiple degrees of freedom
MIMO	Multiple input multiple output
ODS	Operating deflection shapes
OMA	Operational modal analysis
PS	Power spectrum
RD	Random decrement process
${}_rA_{ij}$	rth modal constant
h_k	kth IRF
$k\Delta$	Time interval
R^2	Coefficient of determination
$\sqrt{R^2}$	Coefficient of correlation
s_r	rth complex quantity root
V	Vehicle speed
$\bar{X}_i$ $i = 1, 2, 3$	ith mean
z_r	Conjugate of the roots s_r
$\ddot{Z}^*_{qI}, \ddot{Z}^+_{qI}, \ddot{Z}^o_{qI}$	Vertical acceleration at car body, leading bogie and leading axle, respectively
Ω	Natural frequency
$\alpha(\omega)$	Receptance matrix
$\beta_k, k = 0, 1, \ldots, 2N$	kth real coefficient
$\delta_j, j = 1, \ldots, 4$ jth	secondary vertical damper
ε	Error value
$\in_i, i = 1, \ldots, 10$	ith damper technical state
ϕ_r	rth mode shape
$\sigma_i, i = 1, 2, 3$	ith standard deviation
ω	Oscillation frequency
ξ	Damping ratio
$\zeta_i, i = 1, 2, 3$	ith hypothesis

Appendix 2. Conceptual basis

The inverse Laplace transform of the transfer function of an MDoF system is the IRF, h_k. This gives as a result a series of equally spaced time intervals $k\Delta(k = 0, 1, \ldots 2N)$, and then it is possible to express IRF as

$$h_k = \sum_{r=1}^{2N} {}_rA_{ij} z_r^k \quad z_r^k = e^{s_r k \Delta},$$

where

$${}_rA_{ij} = \phi_{ir}\phi_{jr}.$$

This expression is the product of the i*th* and j*th* elements in the modal shape r*th* $\{\phi\}_r$ and is named as the *modal constant*. The values in the series belong to the real numbers even if the residues and the roots s_r are complex values. It is possible to demonstrate that all imaginary parts will cancel each other because of the complex conjugates for both expressions: ${}_rA_{ij}$ and s_r. The next step is to estimate the roots and the residues for the sampled data. This solution is aided by the conjugate of the roots s_r, therefore, z_r. Mathematically, this means that z_r are the roots of a polynomial with only real coefficients [2]:

$$\beta_0 + \beta_1 z_r + \beta_2 z_r^2 + \cdots + \beta_{2N-1} z_r^{2N-1} + \beta_{2N} z_r^{2N} = 0$$

or

$$\sum_{k=0}^{2N} \beta_k h_k = \sum_{r=1}^{2N} {}_rA_{ij} \sum_{k=0}^{2N} \beta_k z_r^k.$$

This equation is known as the Prony equation. The coefficients can be estimated by the IRF values.

Appendix 3

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Table A3. Vehicle parameters.

Element	Quantity	Value	Unit
Mass			
Car body	1	24486.36	kg
Bogie frame	2	6102.44	kg
Motor	4	5551	kg
Electromagnetic brake	4	704	kg
Traction link	2	–	kg
Axle–wheel set	4	7083.52	kg
Stiffness			
Linear	1	$k_A = 1.00$	kN/mm
Nonlinear	20	Array	kN/mm
Shear	8	$k_x = 2.16$	kN/mm
		$k_y = 2.16$	kN/mm
		$k_z = 12.16$	kN/mm
Axis direction	2	0.08	kNmm/s
Air spring with auxiliary rubber stack	4	$k_z = 4.52$	kN/mm
		$k_y = 1.04$	kN/mm
Nonlinear damper	13	Array	kNmm/s
Bushing	36	$k_x = 67.32$	kN/mm
		$k_y = 36 \times 10^{-5}$	kN/mm
		$k_z = 72 \times 10^{-6}$	kN/mm
		$k_\theta = 36 \times 10^{-5}$	MNm/rad
		$k_\phi = 2 \times 10^{-6}$	MNm/rad
		$k_\psi = 1 \times 10^{-5}$	MNm/rad
DOF	120	–	–
Nonlinear conicity		Array	–

Appendix 4.

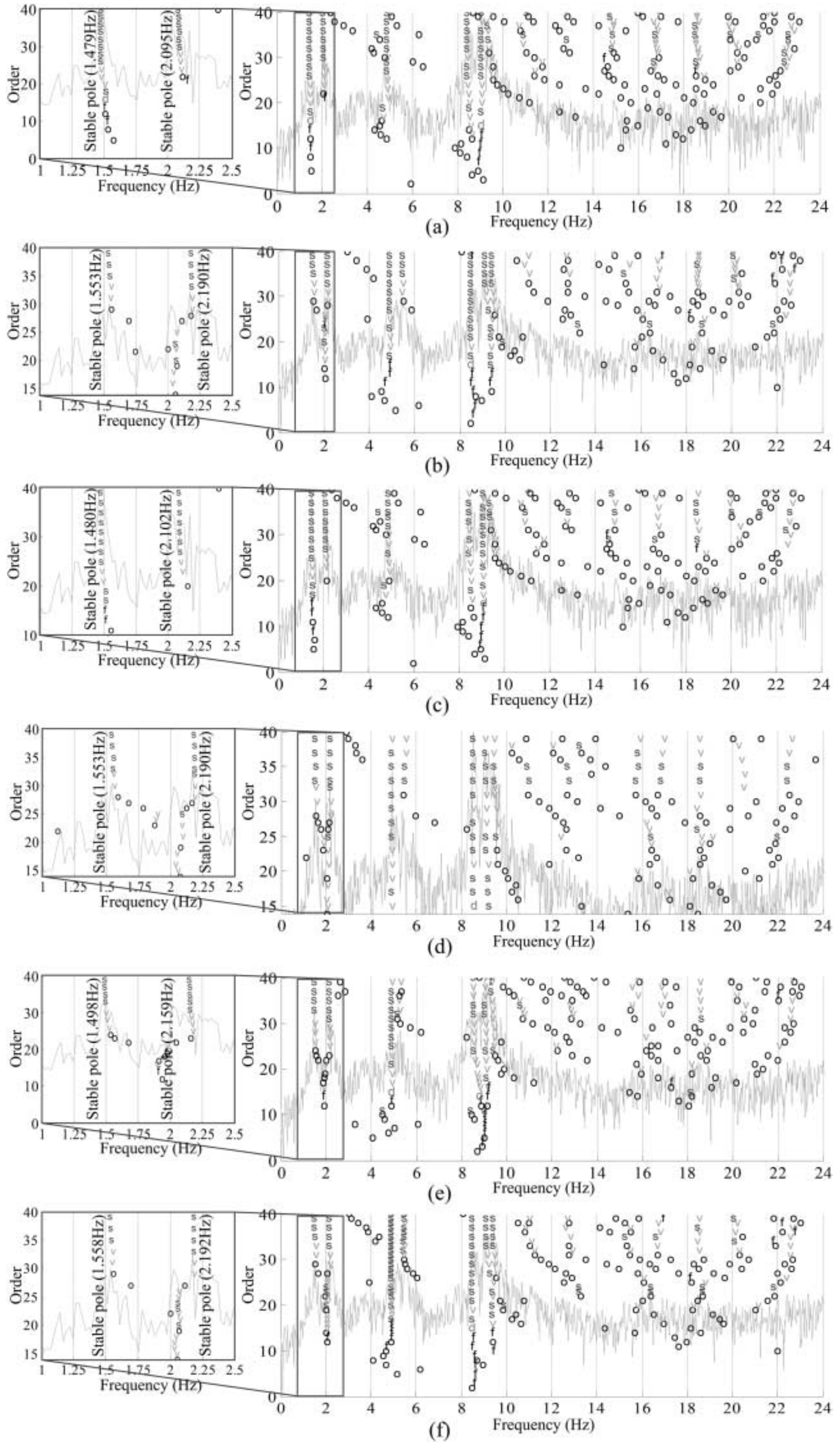

Figure 7. Stabilisation diagrams: (a) hypothesis ζ_1 with $\in_1$; (b) hypothesis ζ_1 with $\in_9$; (c) hypothesis ζ_2 with $\in_1$; (d) hypothesis ζ_2 with $\in_9$; (e) hypothesis ζ_3 with $\in_1$; and (f) hypothesis ζ_3 with $\in_9$.