Effect of Non-linear Suspension on the Recognition of the Road Disturbance

Dorra Ben Hassen[1](✉), Mariem Miladi[1], Mohamed Slim Abbes[1], S. Caglar Baslamisli[2], Fakher Chaari[1], and Mohamed Haddar[1]

[1] Mechanics, Modeling and Production Laboratory, National Engineering School of Sfax (ENIS), BP 1173, 3038 Sfax, Tunisia
Mohamed.haddar@enis.rnu.tn

[2] Department of Mechanical Engineering, Hacettepe University, 06800 Beytepe, Ankara, Turkey

Abstract. The road excitation is one of the major forces which act on the vehicle and affect the passenger's comfort so it constitutes a crucial field of interest when suspension systems are designed. Hence, identifying this type of excitation remains very benefiting since it contributes to study the dynamic behavior of the vehicle and to apply a controller law in order to ensure the passenger comfort. Direct recognition techniques (longitudinal profile analyser or laser sensors…) of the road profile are costly, thus, it is necessary to find other methods such as numeric ones to recover the road disturbance. In this paper, the inverse problem theory is employed to pick out the road profile disturbance applied to vehicle. This proposed technique, known as the Independent Component Analysis (ICA), can recreate initial excitation sources by using physically measurable signals named observed signals of the system under study. These signals are obtained numerically in this study by using the Newmark approach. Starting from these dynamic responses, the ICA algorithm is applied to a non-linear vehicle model to identify the road excitations. The performance of this technique is studied using some criteria which are the relative error and the MAC number. The obtained results show a good relevance between the original signals and the estimated ones.

Keywords: Non-linear vehicle model · Road excitation · Inverse problem

1 Introduction

The effect of the road excitation on the suspension's performance has been the center of the attention of various scientific papers. One of them is: Hunt (Hunt 1991) has scrutinized the dynamic response of a vehicle that is subjected to the random excitations. Furthermore, E. Duni (E. Duni et al. 2003) have investigated the dynamic behavior of a full vehicle model submitted to different types of road excitations through the use of finite element method. So, in order to analyze the dynamic response of a vehicle under real condition, the road profile excitation should be identified accurately in order to its effect on the ride quality and passenger's comfort (Yan 2012). Kim (Kim et al. 2002) have measured the road roughness directly by means of visual inspections. The Monte

A. Hammami et al. (Eds.): MOSCOSSEE 2021, ACM 20, pp. 65–74, 2022.
https://doi.org/10.1007/978-3-030-85584-0_7

Carlo technique (Harris et al. 2010) has been used to estimate the road profile. Fauriat (Fauriat et al. 2016) proposed the 'Augmented Kalman filter' to derive the road profile excitation. Mariem (Mariem Miladi et al. 2019) showed that ICA technique has higher efficiency than other techniques in road identification. The main aim of this paper is to use the ICA to recover the excitation of the road profile for a non-linear full vehicle model. This technique was commonly used to evaluate the excitation force in many studies (Dhief et al. 2016, B. Hassen et al. 2017, Taktak et al. 2012). Its main advantages are that it is simple to be implemented and has a feature of a real time identification process. This paper is organized as follows: the first section shows the two axle vehicle model along with its mathematical formulation. Then the applied method, the ICA, is modeled. In the third section, the obtained results are showed and finally the efficiency of the method is confirmed by means of some performance criteria.

2 Two Axle Vehicle Model

The figure (Fig. 1) depicts the full dynamic model of the car studied in Meywerk (2015).

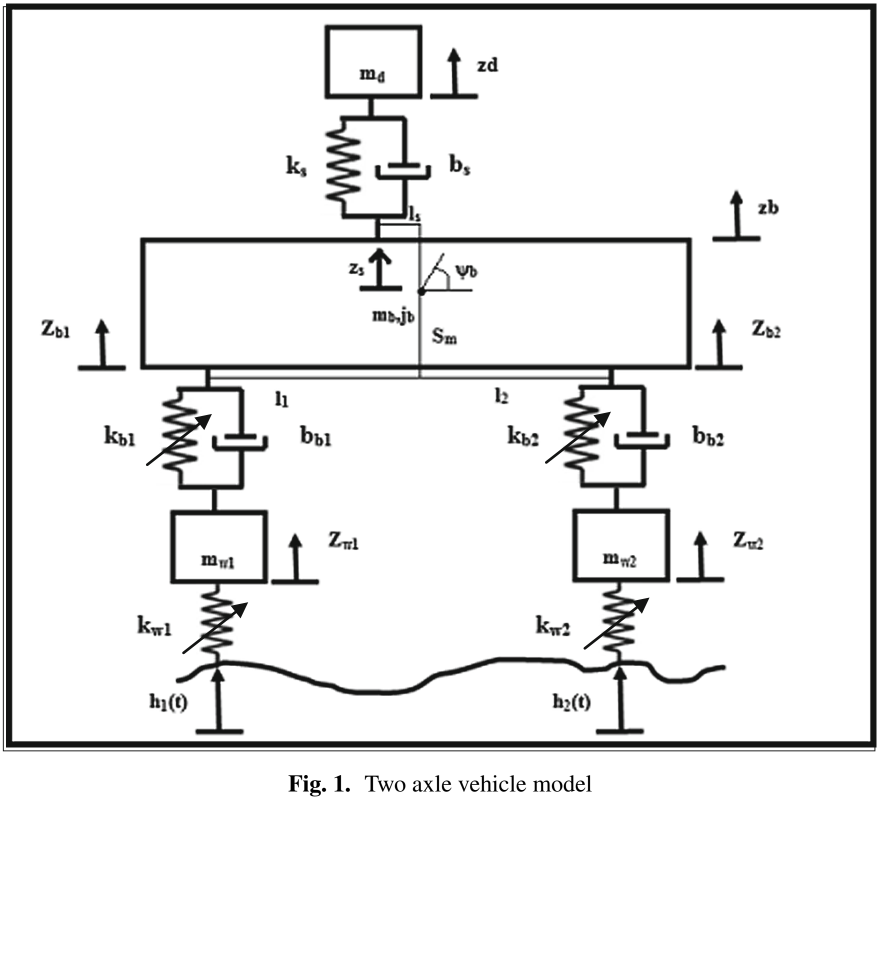


Fig. 1. Two axle vehicle model

This model consists of four masses. In order for this kind of modeling to yield a good insight of the actual vibrations of a real vehicle, the following assumptions should be included:

– The road disturbance is evenly applied to the right and the left wheel. The vehicle is considered to have symmetrical inertia features (Meywerk 2015).
– The excitations applied to the rear wheels $h_2(t)$ are taken identically to those on the front wheels $h_1(t)$, with a short delay as it is mentioned in Fig. 2.

This model is described as follows:

– m_{w1} and m_{w2} stand for the masses of the wheels. They are linked to the road by means of two non-linear springs denoted as k_{w1} and k_{w2}. The deflections of the two masses m_{w1} and m_{w2} are denoted as z_{w1} and z_{w2} respectively.
– z_{b1} and z_{b2} represent the vertical displacement of the suspension systems. These suspensions are composed by non-linear stiffness's k_{b1} and k_{b2} in parallel with the two dampers are denoted b_{b1} and b_{b2}.
– z_b is symbolized as the center of gravity's displacement
– φ_b denotes the pitch angle.
– z_d stands for the vertical displacement of the driver's seat.

Taking into account the assumptions described above, zb_1 and zb_2 are expressed as follows:

$$z_{b1} = z_b - l_1 \varphi_b \tag{1}$$

$$z_{b2} = z_b + l_2 \varphi_b \tag{2}$$

And the coordinate z_s is expressed as:

$$z_s = z_b - l_s \varphi_b \tag{3}$$

– The non-linear Spring's excitations are expressed as follows:

$$F_{b1} = k_{b1}\,\Delta l + \beta_1 k_{b1}\,\Delta l^2 + \beta_2 k_{b1}\,\Delta l^3 \tag{4}$$

and

$$F_{b2} = k_{b2}\,\Delta l_1 + \beta_1 k_{b2}\,\Delta l_1^2 + \beta_2 k_{b2}\Delta l_1^3 \tag{5}$$

Where:

- Δl stands for the deflection between z_{b1} and z_{w1} in Eq. (4) and Δl_1 is written in Eq. (5) as the deflection between z_{b2} and z_{w2}.
- $\beta_1, \ \beta_2$ represent two non-linear constants (Li et al. 2011) as $\beta_1 = 0.1 \ \ and \ \ \beta_2 = 0.4$

For the tire, it is designed as a non-linear spring k_2. The expression of the non-linear tire stiffness is taken from Li et al. (2011) as:

$$F_{w1} = k_{w1}\,\Delta l_3 + \beta_3 k_{w1}\Delta l_3^2 \tag{6}$$

And

$$F_{w2} = k_{w2}\,\Delta l_4 + \beta_3 k_{w2}\Delta l_4^2 \tag{7}$$

Where:

- Δl_3 is the difference between kw_1 and $h_1(t)$ in Eq. (6) and Δl_4 represents the difference between kw_2 and $h_2(t)$ in Eq. (7).
- β_3 is the non-linear coefficient of the tire. Its value is taken from Li et al. (2011):

$$\beta_3 = 0.01 \tag{8}$$

So the equations of motion of the half vehicle model can be written in a matrix form as: (Ref CMSM DORRA°)

$$[\mathrm{M}]\left\{\ddot{\mathrm{X}}\right\} + [\mathrm{C}]\left\{\dot{\mathrm{X}}\right\} + [\mathrm{K}]\{\mathrm{X}\} + \{\mathrm{F_{NL}}\} = \{\mathrm{F}\} \tag{9}$$

Where $\{\mathrm{X}\}$, $\left\{\dot{\mathrm{X}}\right\}$ and $\left\{\ddot{\mathrm{X}}\right\}$ are respectively displacement, velocity and acceleration vectors. [M] is the mass matrix [C] is the damping matrix and [K] depicts the stiffness matrices of the studied system. {F} is the excitation force vector due to road disturbance. $\{\mathrm{F_{NL}}\}$ is the non-linear spring force vector.

$$M = \begin{bmatrix} m_d & 0 & 0 & 0 & 0 \\ 0 & m_b & 0 & 0 & 0 \\ 0 & 0 & J_b & 0 & 0 \\ 0 & 0 & 0 & m_{w1} & 0 \\ 0 & 0 & 0 & 0 & m_{w2} \end{bmatrix} K = \begin{bmatrix} k_s & -k_s & k_s l_s & 0 & 0 \\ -k_s & k_s + k_{b_1} + k_{b_2} & -k_s l_s - k_{b_1} l_1 + k_{b_2} l_2 & -k_{b1} & -k_{b2} \\ k_s l_s & -k_s l_s - k_{b_1} l_1 + k_{b_2} l_2 & k_s l_s^2 + k_{b_1} l_1^2 + k_{b_2} l_2^2 & k_{b1} l_1 & -k_{b2} l_2 \\ 0 & -k_{b_1} & k_{b_1} l_1 & k_{b_1} + k_{w1} & 0 \\ 0 & -k_{b_2} & -k_{b_2} l_2 & 0 & k_{b_2} + k_{w2} \end{bmatrix}$$

$$C = \begin{bmatrix} b_s & -b_s & b_s l_s & 0 & 0 \\ -b_s & b_s + b_{b_1} + b_{b_2} & -b_s l_s - b_{b_1} l_1 + b_{b_2} l_2 & -b_{b1} & -b_{b2} \\ b_s l_s & -b_s l_s - b_{b_1} l_1 + b_{b_2} l_2 & b_s l_s^2 + b_{b_1} l_1^2 + b_{b_2} l_2^2 & b_{b1} l_1 & -b_{b2} l_2 \\ 0 & -b_{b_1} & b_{b_1} l_1 & b_{b_1} & 0 \\ 0 & -b_{b_2} & -b_{b_2} l_2 & 0 & b_{b_2} \end{bmatrix}$$

$$F = \begin{bmatrix} 0 \\ 0 \\ 0 \\ k_{w1} h_1 \\ k_{w2} h_2 \end{bmatrix}$$

The implicit Newmark's technique coupled with Newton Raphson Method has been used to resolve the non-linear dynamic equations governing the motion of the system.

A residue is computed. At each iteration k this residue has the following expression:

$$R_{i+1}^{k} = \left[\overline{K}\right]\{X\}_{i+1}^{k} + F_{NL}^{k} - \left\{\overline{F}\right\}_{i+1} \tag{10}$$

With:

$$\left[\overline{K}\right] = [K] + a_0[M] + a_1[C] \tag{11}$$

And:

$$\begin{aligned}\left\{\overline{F}\right\}_{i+1} &= \{F\} + [M]\left(a_0\{X_i\} + a_2\left\{\dot{X}_i\right\} + a_3\left\{\ddot{X}_i\right\}\right) \\ &+ [C]\left(a_1\{X_i\} + a_4\left\{\dot{X}_i\right\} + a_5\left\{\ddot{X}_i\right\}\right)\end{aligned} \tag{12}$$

Where a_i ($i = 0..5$) are the Newmark's constants.

If the residue is not acceptable i.e. $R > \varepsilon$, a correction should be made for the displacement vector as following:

$$\{\Delta X\} = \left(\left.\frac{\partial R}{\partial X}\right|_{i+1}^{k}\right)^{-1}\left(-R_{i+1}^{k}\right) \tag{13}$$

So that the displacement will be:

$$\{X\}_{i+1}^{k+1} = \{X\}_{i+1}^{k} + \{\Delta X\} \tag{14}$$

The values of the model parameters are presented in the following Table 1:

Table 1. Parameters of the studied vehicle model

Parameters	Variable value	Variable unit
Mass of the chassis	$mb = 960$	[Kg]
Mass of the tires	$m_{w1} = m_{w2} = 36$	[Kg]
Suspension stiffness	$k_{b1} = k_{b2} = 16000$	[N/m]
Tire stiffness	$K_{w1} = k_{w2} = 10^5$	[N/m]
Suspension damping	$bb_1 = bb_2 = 100$	[Ns/m]
Driver's mass	$m_d = 90$	[Kg]
Moment of inertia	$Jb = 500$	[Kg/m^2]
Driver seat's rigidity	$k_s = 2000$	[N/m]
Driver seat's damping	$b_s = 10$	[Ns/m]
l_1	$l_1 = 1.8$	[m]
l_2	$l_2 = 0.8$	[m]

For the road profile excitation, a random road profile has been included in the first wheel. For the second the same excitation with a short delay is taken (Fig. 2). This profile is modeled according to ISO 8608 (ISO 8608) (Table 2) which classifies the profiles to different classes based on the power spectral density (PSD) (Yan 2012).

Table 2. Road profile classification

Road Class	Degree of roughness $Gd(n_0)$ (10^{-6} m^3)		
	Lower limit	Geometric mean	Upper limit
Road A	–	16	32
Road B	32	64	128
Road C	128	256	512
Road D	512	1024	2048
Road E	2048	4096	8192

The Integral White Noise method is employed to create the road roughness. It considers that the road roughness is the issue of a filtered white noise defined by Eq. 15:

$$\dot{q}(t) = 2\pi n_0 \, w_1(t)\sqrt{G_d(n_0)v} \tag{15}$$

Where: $w_1(t)$ stands for the Gaussian white noise with a variance equal to 1, q(t) denotes the road roughness while v represents the vehicle velocity. The applied road in this paper is a profile of type A as mentioned in Fig. 2.

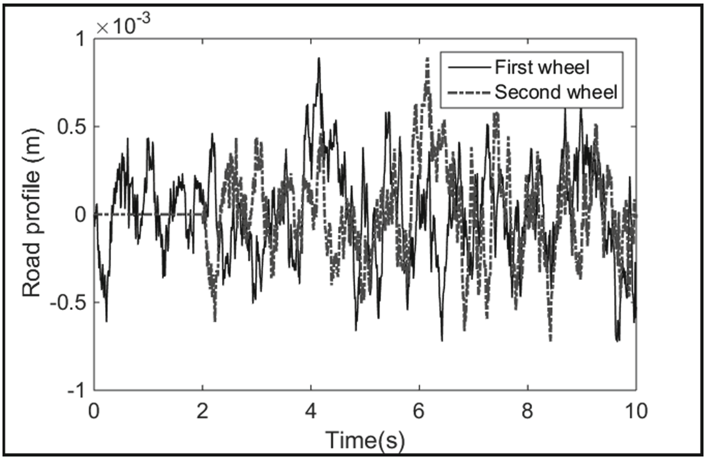


Fig. 2. Road profile

3 Identification Technique: ICA

This algorithm aims to decompose a random signal X in statistically independent components (Dhief et al. 2016). This random signal is expressed by the following equation (Hassen et al. 2017)

$$X(t) = [A]\{S\} \tag{16}$$

Where:
A: The mixing matrix.
S: The source signals' vector.
The ICA has to estimate A and S based only on the cognition of the vector X. This estimation requires some assumptions and some pretreatments.

So a matrix W must be searched where the estimated signal $\overline{y}(t)$ are defined by the following equation

$$\overline{y}(t) = [W]\{S\} \tag{17}$$

To find the matrix W the independence criterion are used in the sense of the maximization of non gaussianity defined by the kurtosis (Eq. 18)

$$\{Y\} = [W]^{H}\{X\} \tag{18}$$

Where $(.)^{H}$ denotes the conjugate-transpose operator To find the matrix W the independence criterion are used in the sense of the maximization of non gaussianity defined by the kurtosis (Eq. 19) defined by Zarzoso and Comen (ISO 8608) article apac6344 as the normalized fourth-order marginal cumulate defined by the following equation in order to guarantee a non-Gaussianity distribution.

$$K(ka) = \frac{E\left\{|y|^{4}\right\} - 2E^{2}\left\{\left|y^{2}\right|\right\} - \left|E\left\{y^{2}\right\}\right|^{2}}{E^{2}\left\{\left|y^{2}\right|\right\}} \tag{19}$$

Where E is is the orthogonal matrix of eigenvectors of $E\left\{X\,X^{T}\right\}$.

For more details about this method, the reader can refer to these references (Hassen et al. 2017; Comon 1994).

4 Numerical Results

The dynamic responses which are the deflections of the two suspensions system (Fig. 3) are used as the observed signals for the ICA algorithm. Based on the knowledge of these signals (Noted X1 which is equal to $z_b - z_{w1}$ and X2 which is equal to $z_b - z_{w2}$), the ICA aims to identify the road profile excitations.

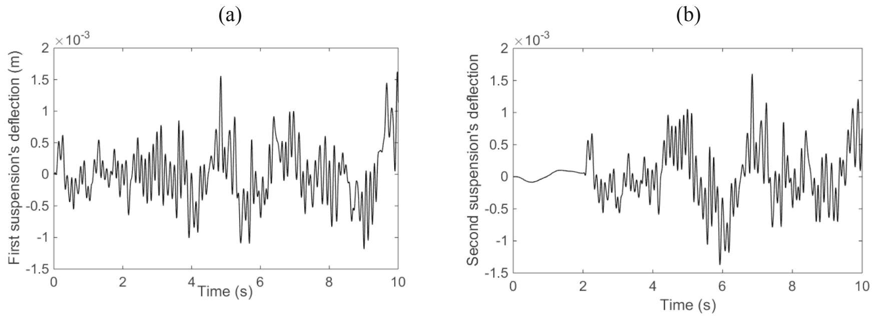


Fig. 3. Observed signals (a) X1 (b) X2

After applying the ICA, The results of the road profile estimation are presented by the following Fig. 4:

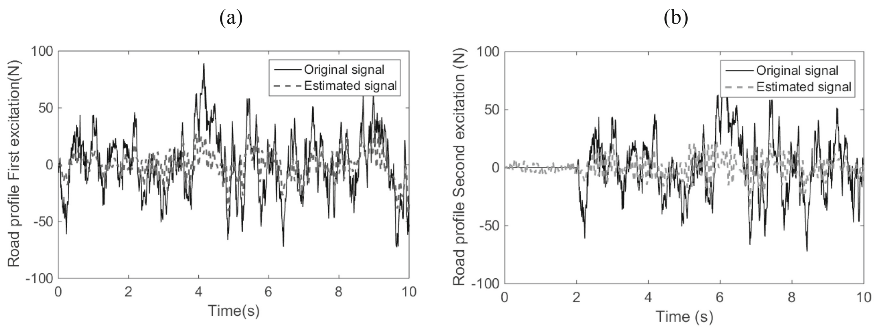


Fig. 4. Estimation of the road profile excitation (a) excitation1 (b) excitation2

Based on these results, it can be said that the ICA can identify the original signals. The small delay and perturbation are due to the effect of the non-linearity. To study the efficiency of this method, the Mac number and the relative error are computed according to the following equations. If the MAC number has a value close to zero, then the obtained results are not compliant and if it has a value close to 1, then the results are compliant. Table 3 resumes the obtained results:

$$E_r = 100 * \frac{y_i - \overline{y}_i}{y_i} \tag{20}$$

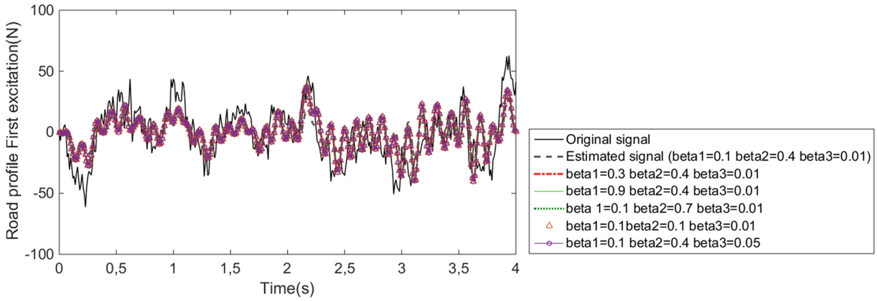
$$\mathrm{MAC_i} = \frac{(y_i^{\mathrm{T}}\overline{y}_i)^2}{(y_i^{\mathrm{T}}\overline{y}_i)(\overline{y}_i^{\mathrm{T}}\overline{y}_i)} \tag{21}$$

Where yi is the original signal and $\overline{y}_i$ is the estimated one.

Table 3. Performance criteria

	MAC	Relative error E_r (%)
Profile 1	0.65	4.712
Profile 2	0.71	3.85

It is seen that the Mac value is near to one for the two studied signals and this shows that the estimated profile is conformed to the original one. Moreover, the values of the relative error confirm these results.



5 Conclusion

The proposed method ICA gives a good estimation of the road profile excitation even with the use of the non-linear parameters. The strength of the ICA technique is that it is applicable with no need to specific road instruments and it is inexpensive.

This method based on the inverse problem, can be used over thousands of kilometers as a real time estimation which is rapid enough.

This identification process will help us to choose the adequate controller law in future work in order to ameliorate the passenger comfort.

References

Hunt, H.E.M.: Stochastic modelling of traffic-induced ground vibration. J. Sound Vib. **144**, 53–70 (1991)

Duni, E. et al.: Numerical simulation of full vehicle dynamic behaviour based on the interaction between ABAQUS/Standard and explicit codes. In: Abaqus Users' Conference, June Munich (2003)

Yan, Z.: Road condition predicting with Kalman. Filter for magneto-rheological. Damper in suspension system, Thesis, Department of Applied Signal Processing, Blekinge Institute of Technology, July 2012

Kim, H.J., Yang, H.S., Park, Y.P.: Improving the vehicle performance with active suspension using road-sensing algorithm. Comput. Struct. **80**, 1569–1577 (2002)

Harris, N.K., González, A., OBrien, E.J., McGetrick, P.: Characterisation of pavement profile heights using accelerometer readings and a combinatorial optimisation technique. J. Sound. Vibr. **329**, 497–508 (2010)

Fauriat, W., Mattrand, C., Gayton, N., Beakou, A., Cembrzynski, T.: Estimation of road profile variability from measured vehicle responses. Vehicle Syst. Dyn. **54**(5), 585–605 (2016)

Chaabane, M.M., et al.: Road profile identification using estimation techniques: comparison between independent component analysis and Kalman filter. J. Theor. Appl. Mech. **57** (2019)

Dhief, R., Taktak, M., Tounsi, D., Akrout, A., Haddar, M.: Application of the independent components analysis in the reconstruction of acoustic sources in duct systems. Arabian J. Sci. Eng., 1–10 (2016)

Hassen, D.B., Miladi, M., Abbes, M.S., Baslamisli, S.C., Chaari, F., Haddar, M.: Application of the operational modal analysis using the independent component analysis for a quarter car vehicle model. In: Fakhfakh, T., et al. (eds.) Advances in Acoustics and Vibration. Applied Condition Monitoring, vol. 5, pp. 125–133. Springer, Cham (2017). https://doi.org/10.1007/978-3-319-41459-1_12

Taktak, M., Tounsi, D., Akrout, A., Abbès, M.S., Haddar, M.: One stage spur gear transmission crankcase diagnosis.using the independent components method. Int. J. Vehicle Noise Vibr. **8**(4), 387–400 (2012)

Meywerk, M.: Vehicle Dynamics. John Wiley & Sons (2015)

Li, S., Lu, Y., Li, H.: Effects of .parameters on dynamics of a nonlinear vehicle-road coupled system. JCP **6**(12), 2656–2661 (2011)

ISO, 8608.: Mechanical vibration. Road surface profiles. Reporting of measured data. ISO 8608:1995 (1995)

Hassen, D.B., et al.: Road profile estimation using the dynamic responses of the full vehicle model. Appl. Acoustics (2017)

Comon, P.: Independent component analysis, a new concept? Signal Process. **36**, 287–314 (1994)