OPERATIONAL MODAL ANALYSIS OF PASSENGER CARS: EFFECT OF THE CORRELATION BETWEEN FRONT AND REAR INPUTS

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ABSTRACT

Operational Modal Analysis (OMA) allows the structural identification of systems in working conditions, moving from output data only. External forces remain not measured. Basically some fundamental assumptions have to be satisfied: (i) the not known loads acting on the system need to have the form of white noise sequences, (ii) in the case of multi-point excitation the external inputs are required to be strictly uncorrelated. In the field of vehicle dynamics, OMA can be utilized to assess the performance of different suspension systems equipping the same type of vehicle. The output data recorded at the different sensor locations during vehicle road tests are affected by a certain correlation among the road forces acting on the wheels, mainly that loading the front and the rear axle, even if inputs on wheels belonging to the left and the right side could be also in some way correlated. In this paper, the effect of the existing correlation between the road inputs, acting on the front and the rear axles of a numerically simulated passenger car, on the estimates of modal parameters is investigated. The estimates are obtained by using a standard curve fitting technique, as the LMS PolyMAX for Operational Modal Analysis, on the time data evaluated in several virtual sensor locations.

1. INTRODUCTION

Operational or Output-only Modal Analysis [1, 2] allows the experimental identification of structural dynamics models during operation. Strength of the technique is, indeed, that frequently the system identification has to rely only on output-only data.Moreover, since all the real systems are to a certain extent non-linear and often also subjected to non-linear constraining conditions, the modal models obtained under real loadings give a picture of the system once linearised around the more interesting and representative working points. In the case of control systems, the identification of the real in-operation behaviour is essential in the areas of the optimal design and verification of the controller performance.
All these characters are needed for the experimental assessment of suspension systems equipping a passenger car. In particular, if the vehicle ride performance has to be assessed, the heave, roll and pitch rigid body motions have to be investigated. This can be done mainly by performing (i) driving tests in controlled conditions, i.e. proving ground tests on specific road tracks and (ii) laboratory tests in which the input excitation is emulated by a four poster road simulator [3].

A four poster is a test rig basically consisting of four actuators, one for each wheel, able to reproduce, on a car that actually stands still on it at null speed, a running condition on a certain road profile. Advantages of using multi-shaker rigs are that measurements are more repeatable than in the case of road tests, with extremely low level of external noises, and that it is possible to choose several different deterministic or stochastic excitation signals, allowing tests not feasible on road. Although pure or swept sine waveforms or white or pink random sequences do not represent any realistic road condition, they can be effectively utilised to perform input-output modal analysis (EMA), paying particular attention to the assessment of the non-linearities that affect the whole system [4, 5]. Moreover the possibility of reproducing on a shaker rig the loads coming from real road tracks driven at certain speeds, allows mainly to perform durability tests for the fatigue assessment of suspension components [6]. The main drawbacks are, of course, availability and overall cost of a lab equipped with those shaker rigs.

With regards to road tests, it has to be stressed that since the external loads acting on the vehicle commonly remain not known (they could be measured only by using expensive transducer-instrumented wheels) EMA cannot be performed.

For all the reasons reported, in a previous effort of the same research group an OMA approach to the performance assessment of vehicle suspension systems has been suggested [7]. In particular, main findings were that due to the nature of road input excitation, the better excited modes are actually the lower frequency ones and, hence, the so-called rigid body modes, related to the aforementioned rigid body motions. But attention has to be paid to the effects of the existing correlation between the road inputs, mostly that between forces acting on the front and the rear axles.

Goal of the research here presented is to understand if the presence of correlated inputs still allows reliable operational modal parameter estimation, performed by using a standard operational curve fitting algorithm. To this aim the OMA of a passenger car has been numerically simulated. Firstly, the mathematical model of the vertical dynamics of a vehicle, when excited (i) by uncorrelated brownian noise sequences at the wheels, as during a laboratory test and (ii) by correlated road inputs has been developed. By simulating those two different dynamic conditions, changing in the latter case even parameters from which the solution depends on, as the amount of damping in the system and the car velocity, the output time history data have been collected in four sensor locations. Secondly the collected throughputs have been OMA processed by using the standard LMS operational PolyMAX (pLSCF) algorithm, available in the Test.Lab Software Suite.

We show that although the inputs are correlated, it is still possible to find accurate modal parameters in the case the amount of damping in the system is quite low (heave mode damping ratio $\zeta \leq 0.1$). Since in the case of suspension systems for passenger cars high damping levels are expected, the correlation between the inputs introduces distortions in the magnitude and the phase of the output Power Spectra that do not allow the curve fitting estimator to synthesize good quality functions.

A special time data pre-processing aimed at eliminating the effects of the temporal correlation between the inputs, before performing standard operational modal parameter estimation, or a novel, special curve fitting formulation for vehicle OMA are possible solution to be investigated.
2. MATHEMATICAL MODEL

2.1 Road input excitation

The Power Spectral Density (PSD) approximation proposed by Sussman (1974), holding for
long wavelengths, is utilised as stochastic description of road unevenness:

\[ S_d(n) = \frac{C}{n^2 + n_0^2} \quad \text{with} \quad 0 \leq n \leq \infty \]  \hspace{1cm} (1)

where \( C \) and \( n_0 \) are two constants and \( n \) is the wavenumber [8, 9]. This approximating function trend can be obtained by processing a white noise input sequence \( w(t) \) with an applied first order shaping filter:

\[ H_{dw}(s) = \frac{D(s)}{W(s)} = \frac{1}{n_0^2 + 1} \]  \hspace{1cm} (2)

where \( D(s) \) is the Laplace Transform of the displacement \( d(t) \) and \( v \) the forward velocity of the vehicle. In order to obtain the transfer function \( H_{dw}(s) \), the following Eq. 3 has to hold in the time domain:

\[ n_0vw(t) = d(t) + n_0vd(t) \]  \hspace{1cm} (3)

The relationship holding between input and output PSDs for a SISO system can be then used to find the analytical expression of the road profile PSD in the frequency domain:

\[ S_d(f) = |H_{dw}(f)|^2 S_w(f) = \frac{2\sigma_w^2 f_0^2}{f_s f^2 + f_0^2} \quad \text{with} \quad f_0 = \frac{n_0v}{2\pi} \quad \text{for} \quad 0 \leq f \leq f_s/2 \]  \hspace{1cm} (4)

where

\[ S_w(f) = 2\sigma_w^2 / f_s \]  \hspace{1cm} (5)

is the half PSD of a band-limited white noise with variance \( \sigma_w^2 \), sampled at frequency \( f_s \).

A comparison between analytical and numerical PSDs is proposed in Fig. 1, obtained by considering the displacement induced by the road profile when smooth asphalt is simulated.

2.2 Vehicle dynamics

An half car model, as shown in Fig. 2, is used to describe the vertical dynamics of the vehicle [10]. The 4 degree of freedom system includes the heave \( z_g \) and the pitch \( \theta_g \) rigid body motions of the sprung mass and the vertical movements \( z_f \) and \( z_r \) of the unsprung masses,

Figure 1: Comparison between analytical and simulated PSDs in the case of smooth asphalt \((n_0=0.2 \ \text{rad/m}, \ \sigma_w=0.05 \ \text{m}, \ v=60 \ \text{km/h})\).
even referred as rattle displacements. The equations of motions for this system can be written as:

\[
\begin{align*}
    m_g \ddot{z}_g &= f_s + f_{rs} \\
    j_s \dot{\theta}_g &= f_s d_r - f_f d_f \\
    m_f \ddot{z}_f &= f_f - f_s \\
    m_r \ddot{z}_r &= f_r - f_{rs}
\end{align*}
\]

where \( f_s \) and \( f_{rs} \) are the forces acting on the sprung mass, while \( f_f \) and \( f_r \) are the forces exerted on the unprung masses:

\[
\begin{align*}
    f_s &= -k_f (z_{fs} - \dot{z}_f) - c_f (\dot{z}_{fs} - \dot{z}_f) \\
    f_{rs} &= -k_r (z_{rs} - \dot{z}_r) - c_r (\dot{z}_{rs} - \dot{z}_r) \\
    f_f &= -k_f (z_f - d_f) \\
    f_r &= -k_r (z_r - d_r)
\end{align*}
\]

and \( z_{fs} \) and \( z_{rs} \) are the vertical movements of the sprung mass at front and rear axles respectively:

\[
\begin{align*}
    z_{fs} &= z_g - \theta_g l_f \\
    z_{rs} &= z_g + \theta_g l_r.
\end{align*}
\]

### 2.3 Simulations

The dynamics of a passenger car during a road test is simulated using the Matlab Control System Toolbox to obtain virtual accelerometer measurements at several locations. The equations of motion are implemented in state space form:

\[
\dot{x} = Ax + Bu
\]

\[
y = Cx + Du
\]

where the state vector \( x = \begin{bmatrix} z_g & \theta_g & z_f & z_r & d_f & d_r & \dot{z}_g & \dot{\theta}_g & \dot{z}_f & \dot{z}_r \end{bmatrix}^T \) and the matrices \( A, B, C \) and \( D \) define the relationship between the input vector \( u \) and the output vector \( y \).

The sprung mass \( m_g \) of the simulated vehicle is set to 1000 kg, while the lateral inertia \( j_y \) is considered equal to 1620 kgm². The remaining needed parameters are summarized in Tab. 1.
<table>
<thead>
<tr>
<th></th>
<th>Front</th>
<th>Rear</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wheelbase (m)</td>
<td>1.2</td>
<td>1.5</td>
</tr>
<tr>
<td>Spring Rate (kN/m)</td>
<td>31.5</td>
<td>28</td>
</tr>
<tr>
<td>Damping Rate (kNs/m)</td>
<td>2.95</td>
<td>2.62</td>
</tr>
<tr>
<td>Unsprung Mass (kg)</td>
<td>50</td>
<td>50</td>
</tr>
<tr>
<td>Tire Stiffness (kN/m)</td>
<td>280</td>
<td>280</td>
</tr>
</tbody>
</table>

Table 1. Vehicle parameters.

Figure 3. Offprint of the road displacements on front and rear axles.

To account for the existing correlation between front and rear axle, the road input considered acting on the rear axle is obtained by shifting samples of the input vector acting on the front one by a time delay \( \tau = (l_f + l_r)/v \) (Fig. 3).

Theoretical values for eigenfrequencies \( f_d \) and damping ratios \( \zeta \) for each mode are reported in Tab. 2. These figures are used as reference values to be compared with those obtained by OMA processing the simulated signals in each considered virtual sensor location. In particular, as anticipated, in order to find the modes of the vehicle system, the frequency band from 0 to 15 Hz is investigated, by processing the throughputs by means of the LMS Test.Lab Software Suite [11].

3. OMA OF SIMULATED SENSOR DATA

3.1 Theoretical background

Operational Modal Analysis theory moves basically from the well-known relationship between Power Spectrum and Frequency Response Function matrices

\[
S_{yy}(\omega) = H(\omega)S_{xx}(\omega)H^H(\omega)
\]  

where \( S_{yy}(\omega) \) is the output Power Spectrum matrix, \( H(\omega) \) the Frequency Response Function matrix and \( S_{xx}(\omega) \) is the input Power Spectrum matrix [12]. Only output data are measured and the input signals have to satisfy the condition of being uncorrelated white noise sequences, hence \( S_{xx}(\omega) \) ends to be independent on frequency. The modal decomposition of the output Power Spectrum matrix is

\[
S_{yy}(\omega) = \sum_{i=1}^{n} \frac{v_i g_i^*}{j\omega - \lambda_i} + \frac{v_i^* g_i}{j\omega - \lambda_i^*} + \frac{v_i g_i^*}{-j\omega - \lambda_i} + \frac{v_i^* g_i}{-j\omega - \lambda_i^*}
\]  

where \( v_i \) are the modal vectors, \( g_i \) are the operational reference factors and \( \lambda_i \) are the system poles. In order to estimate the Power Spectra the Weighted Correlogram Method can be
Correlation functions have firstly to be calculated in the time domain

\[ R_i = \frac{1}{N} \sum_{k=0}^{N-1} y_{k+i} y_k^T \]  

and Power Spectra are then evaluated by taking the Fourier transform of the got correlation functions

\[ S_{yy}(\omega) = \sum_{k=-L}^{L} w_k R_k e^{-j\omega k\Delta t}. \]  

An exponential window can be used for reducing the effect of leakage and, moreover, to increase the frequency resolution, the number of time lags \( L \) has to be lower than the number of the elements of \( R_i \).

The polyreference Least Square Complex Frequency domain (pLSCF), also known as LMS PolyMAX, is one of the most used frequency domain modal parameter estimators [14, 15]. Frequency domain techniques provide better stabilization diagrams and are able to take into account the effects of the eigenmodes outside the band of interest, by using the so-called upper and lower residuals. The estimation process can be divided into three steps: (i) polyreference Least Square Frequency Domain (pLSCF), (ii) stabilization diagram, (iii) Least Square Frequency Domain (LSFD). First, the spectra are replaced with rational functions by using the pLSCF:

\[ S_{yy}(\omega) = \sum_{r=1}^{p} \frac{\beta_r}{\alpha_r} \]  

where \( \beta_r \) and \( \alpha_r \) are the numerator and denominator matrices of coefficients, \( p \) is the maximum polynomial order and \( z = e^{j\omega \Delta t} \). The system poles and the operational reference factors are respectively the eigenvalues and eigenvectors of the matrix \( \alpha_r \).

In the stabilization diagram [16, 17] the physical poles can be selected, then the LSFD allows to calculate the modal vectors and the upper and the lower residuals

\[ S_{yy}(\omega) = \sum_{j=1}^{n} \frac{v_j g_j}{j\omega - \lambda_j} + \frac{v_j^* g_j^*}{j\omega + \lambda_j} + \frac{L R}{j\omega} + j\omega UR. \]  

### 3.2 Virtual sensor location lay-out and test scenarios

The vehicle geometry is implemented in LMS Test.Lab in order to process the time data throughputs of virtual accelerometers (Fig. 4). The transducer locations include the vertical accelerations of the sprung mass at the center of gravity (channel 0), front (channel 1) and
rear (channel 2) axles and the vertical movements of front (channel 3) and rear (channel 4) unsprung masses.

As anticipated, two different types of test scenarios are simulated: The former is a four-poster test, the latter a road test. To simulate the four-poster test scenario, two uncorrelated brownian noises of 10 mm RMS are applied to the vehicle wheels, while in the case of the road test, the two shifted road profile inputs corresponding to smooth asphalt displacements (see Sec. 2.1) are employed.

4. DISCUSSION OF RESULTS

4.1 Simulated four-poster test

In Fig. 5 acceleration time histories evaluated at the four virtual sensor locations are highlighted. As one can clearly see, the values attained by the unsprung masses are higher than those of the sprung one, as expected. Therefore to obtain good quality mode shapes, channel 1 and channel 2 are used as references for Cross Spectrum calculation. A well-known, pow-

![Figure 5: Channel throughputs: Green and red for the unprung masses, cyan, purple and blue for the sprung mass.](image)

Figure 5: Channel throughputs: Green and red for the unprung masses, cyan, purple and blue for the sprung mass.

erful tool for evaluating the quality of estimated mode sets is the Modal Assurance Criterion (MAC). In Fig. 6, the results obtained in the case of simulated four-poster test are represented. In Tab. 2, the percentage error between theoretical values of eigenfrequencies and damping ratios and Test.Lab estimates is reported.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Theoretical $f_d$ (Hz)</th>
<th>$\zeta^f$</th>
<th>Estimated $f_d$ (Hz)</th>
<th>$\zeta^e$</th>
<th>$\epsilon_f$ (%)</th>
<th>$\epsilon_{\zeta}$ (%)</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.1200</td>
<td>0.3096</td>
<td>1.1160</td>
<td>0.2594</td>
<td>0.3602</td>
<td>16.2301</td>
<td>heave</td>
</tr>
<tr>
<td>2</td>
<td>1.2062</td>
<td>0.3401</td>
<td>1.1910</td>
<td>0.2847</td>
<td>1.2625</td>
<td>16.2892</td>
<td>pitch</td>
</tr>
<tr>
<td>3</td>
<td>11.3184</td>
<td>0.3878</td>
<td>11.4530</td>
<td>0.3242</td>
<td>0.1170</td>
<td>6.7356</td>
<td>vertical stroke of $m_r$</td>
</tr>
<tr>
<td>4</td>
<td>11.4396</td>
<td>0.3476</td>
<td>11.5250</td>
<td>0.3582</td>
<td>1.8251</td>
<td>7.6504</td>
<td>vertical stroke of $m_f$</td>
</tr>
</tbody>
</table>

Table 2: Simulated four-poster test results: Comparison between theoretical and estimated modal parameter values.

The percentage error on the damping ratio estimates is, in particular, lower than 20%, while that on the values of eigenfrequencies is very small. Since the estimation of modal parameters from four-poster test data has well performed, then these results are used as references for the simulated road test.
4.2 Simulated road test

In this case the effect of the correlation existing between the front and the rear axles is investigated, by considering different values of the theoretical damping ratio $\zeta$ of the first mode, the heave: 0.01, 0.02, 0.03, 0.05, 0.07, 0.10, 0.20, 0.30.

Figure 7: Simulated road test results: Percentage error between theoretical and estimated eigenfrequency values for different damping ratios and velocities.

Figure 8: Simulated road test results: Percentage error between theoretical and estimated modal damping values for different damping ratios and velocities.
More over, since the forward velocity of the vehicle has a fundamental influence on the gain of input signals, two velocity values are considered: 60 km/h and 90 km/h. In Figs. 7 and 8, the percentage errors obtained in the cases of the different considered damping ratios and vehicle speed values are summarized. Stabilization diagrams and autoMAC matrices in the two representative cases of (i) low damping ($\zeta = 0.01$ for the first mode) and (ii) high damping ($\zeta = 0.30$ for the first mode), and for the two considered vehicle speeds, are collected in Figs. 9 and 10. Yet, for both the low and high damping cases, the MAC matrices, between the mode sets obtained in the two vehicle velocity cases, are reported in Figs. 11 and 12, and Tables 3 and 4. An overall comparison of synthesized and measured Cross Spectra is finally depicted in Fig. 13.

![Figure 9](image-url)

**Figure 9:** Simulated road test results for $\zeta = 0.01$ on the first mode: (a) Stabilization diagram and (b) autoMAC matrix for $v=60$ km/h, (c) stabilization diagram and (d) autoMAC matrix for $v=90$ km/h.

The correlation between front and rear inputs gives rise to a typical behaviour of measured Cross Spectra, introducing a saw tooth trend in the phases and some humps in the magnitudes (see e.g. Figs. 13 (c) and (d)), that become visible and even more important for increasing amounts of damping, causing additional difficulty when performing the pole selection. The percentage error of modal estimates, in the high damping case, becomes consequently higher than in the low damping one (Figs. 7 and 8).

In the case of low damping, indeed, although correlated input signals are exciting the system, a standard curve fitting algorithm is still able to accurately estimate the modal parameters (Figs. 7 and 8). A low percentage error between synthesized and measured Power Spectra can be more over noticed (Figs. 13 (a) and (b)), as a satisfactory MAC comparison between modal vectors found in the cases of 60 and 90 km/h (Fig. 11 and Tab. 3), pointing out the similarity between the two obtained mode sets.

On the other hand, in the high damping case, the pole selection in the stabilization diagram becomes tougher (Fig. 10) and worse autoMAC matrices are obtained for each of the considered car velocities, showing, in particular, high correlation between all the modes of the vehicle. An high percentage error between synthesized and measured Cross Spectra is also found out (Figs. 13 (c) and (d)).
Figure 10: Simulated road test results for $\zeta = 0.30$ on the first mode: (a) Stabilization diagram and (b) autoMAC matrix for $v=60 \text{ km/h}$, (c) stabilization diagram and (d) autoMAC matrix for $v=90 \text{ km/h}$.

Figure 11: Simulated road test results: MAC matrix between two different modal vectors sets ($v=60 \text{ km/h}$ and $v=90 \text{ km/h}$) for $\zeta = 0.01$.

<table>
<thead>
<tr>
<th>$v = 60 \text{ km/h}$</th>
<th>1.154 Hz</th>
<th>1.251 Hz</th>
<th>12.463 Hz</th>
<th>12.586 Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.152 Hz</td>
<td>99.97</td>
<td>3.270</td>
<td>0.190</td>
<td>0.43</td>
</tr>
<tr>
<td>1.252 Hz</td>
<td>3.58</td>
<td>99.72</td>
<td>0.34</td>
<td>0.16</td>
</tr>
<tr>
<td>12.483 Hz</td>
<td>0.06</td>
<td>0.51</td>
<td>92.28</td>
<td>20.35</td>
</tr>
<tr>
<td>11.576 Hz</td>
<td>0.45</td>
<td>0.14</td>
<td>5.43</td>
<td>88.64</td>
</tr>
</tbody>
</table>

Table 3: Simulated road test results: Values of the MAC matrix between two different modal vector sets ($v=60 \text{ km/h}$ and $v=90 \text{ km/h}$) for $\zeta = 0.01$.

A novel curve fitting formulation for vehicle OMA, able to take into account the effects of time-shifted excitation is then required. Of course, to develop specific time data preprocessing techniques, aimed at decorrelating the output responses before performing the modal parameter estimation, is even a not easy route that could be taken.
Figure 12: Simulated road test results: MAC matrix between two different modal vector sets \(v=60 \text{ km/h} \) and \(v=90 \text{ km/h} \) for \(\zeta = 0.30\).

<table>
<thead>
<tr>
<th>(v = 60 \text{ km/h})</th>
<th>0.697 Hz</th>
<th>1.230 Hz</th>
<th>10.126 Hz</th>
<th>11.732 Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>(v = 90 \text{ km/h})</td>
<td>0.822 Hz</td>
<td>91.42</td>
<td>50.76</td>
<td>5.48</td>
</tr>
<tr>
<td></td>
<td>1.273 Hz</td>
<td>62.94</td>
<td>88.64</td>
<td>12.52</td>
</tr>
<tr>
<td></td>
<td>10.580 Hz</td>
<td>18.68</td>
<td>22.09</td>
<td>67.76</td>
</tr>
<tr>
<td></td>
<td>12.268 Hz</td>
<td>32.19</td>
<td>31.03</td>
<td>66.34</td>
</tr>
</tbody>
</table>

Table 4: Simulated road test results: Values of the MAC matrix between two different modal vectors sets \(v=60 \text{ km/h} \) and \(v=90 \text{ km/h} \) for \(\zeta = 0.30\).

5. CONCLUSIONS

Operational Modal Analysis of vehicles can be used as a powerful tool for the performance assessment of car suspensions. Road tests have the advantages of being less expensive than laboratory tests and, moreover, road input excitation is suitable for exciting the lower “rigid body” vehicle modes, since the lower is the frequency, the higher is the input incoming energy. The main drawbacks consist in the existing correlation between the input forces acting on the front and the rear axles and in the high damping introduced by suspension shock absorbers.

Goal of the research presented in this paper is to understand if by using a standard operational curve fitting algorithm is still possible nonetheless to perform a reliable modal parameter estimation. To this aim, the OMA of a passenger car road test output acceleration signals has been simulated. At first the theoretical model of the vertical dynamics of a vehicle excited by correlated road inputs has been developed and by simulating several running conditions, time history data have been collected in four sensor locations. Secondly the collected throughputs have been OMA processed by using the standard LMS operational PolyMAX (pLSCF) algorithm. Specifically, different values of the amount of damping in the system and of the car velocity have been considered.

We found out that although the inputs are correlated, it is still possible to find accurate modal parameters in the case the amount of damping in the system is quite low (heave mode damping ratio \(\zeta \leq 0.1\)). Since in the case of suspension systems for passenger cars high damping levels are expected, the correlation between the inputs introduces distortions in the magnitude and the phase of the output Power Spectra, that do not allow the curve fitting estimator to synthesize good quality functions. By using the standard formulation of the pLSCF, indeed, extracting system modal parameters becomes really tougher, and, although increasing the vehicle speed is expected to get the quality of processing better, it introduces
Figure 13: Comparison between the synthesized and measured Cross Spectra: (a) $\zeta=0.01$ and $v=60$ km/h, (b) $\zeta=0.01$ and $v=90$ km/h, (c) $\zeta=0.30$ and $v=60$ km/h, (d) $\zeta=0.30$ and $v=90$ km/h.

issues in the estimation of the unsprung mass modes. In this case, indeed, it is only possible to extract the rigid body (lower frequency) modes, higher excited by the road profile.

A specific curve fitting OMA formulation is then required, able to take into account the effects of correlation between time-delayed inputs, in the case of the highly damped vehicle systems. Of course, the development of dedicated time data pre-processing techniques, aimed at eliminating those undesired effects is also a not easy route that could be taken.

REFERENCES


