The Method of Estimating Kinematic Road Excitation with Use of Real Suspension Responses and Model

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Abstract

The paper deals with the problem of vertical kinematic excitations in road vehicle dynamics simulation, with the main focus on reconstruction of random excitations using measured dynamic responses of a car suspension. The possibility of causing excitations adequately in terms of chosen conditions of exploitation and in reliable way is crucial to properly asses ride comfort, ride safety as well as rattle space and fatigue strength of suspension elements. The paper presents a method of generating equivalent kinematic excitation allowing for reconstruction of suspension dynamic responses in simulation. The method uses unsprung mass accelerations acquired during test rides and a model of vertical suspension dynamics. The method uses estimated displacements of unsprung mass as a preliminary approximation of kinematic excitation and tracking control system with a PID controller, which causes corrections of kinematic excitations transforming it to the form that allows for faithful reconstruction of unsprung mass accelerations and, in turn, kinematic excitations. The paper presents the basic structure of kinematic excitations' reconstruction system as well as a method of tuning PID controller's coefficients so that the error in estimation is minimized. Research and verification of results were done using a sine chirp signal and constant frequency sine waves. The similarity of estimated road profiles is high with error no larger than 8% of the original signal's amplitude.

Keywords: suspension, kinematic excitation, remote parameter control, simulation, PID controller

1. Introduction

A moving vehicle is suspect to two types of excitations – dynamic and kinematic ones. The most common dynamic excitations are the forces connected with acceleration and deceleration of a vehicle. These however happen much more infrequently in comparison to the ones caused by road unevenness which are called kinematic excitations. These excitations are consequence of road irregularities' heights and velocity of their changes which are proportional to vehicle speed. It is impossible to create a perfectly smooth surface, on which the vehicles would travel. Every road has a profile of irregularities' heights – a geometric structure of the pavement [1]. In mathematical terms it can be described as a function in which the height of the profile is dependent on longitudinal and lateral coordinates along a plane that represents ideal road surface. In simulation the profile is often simplified to just a single longitudinal line directly under vehicle tyre – as this is the direction the vehicle is traveling and the lateral profile is often assumed to be the same on the whole width of road-tyre contact path or even the whole road. The effects such profile has on a vehicle depends however also on other factors, mainly the vehicle's speed and tyre's filtering properties. Resulting function of the heights of the profile is time-dependent, not distance-dependent, as the road profile is. It is also subject to the tyre filtration, which acts as a low-pass filter. It smoothens sharp edges, which is known as tyre enveloping [2]. The knowledge of kinematic excitations is necessary if the goal of the simulation analysis is to evaluate ride comfort, safety or a durability.

The problems in estimation of these excitations comes from the fact that they are random in nature and there is no simple way to define all the factors and their influence on vehicle responses on a certain road. Given the long time for which the researchers have studied this topic (dating as far back as 1910s [3], for more information see [4]), there are many proposed methods of estimating kinematic excitations. Typical road surfaces such as different kinds of paved roads have been investigated thoroughly many times and the excitations' results acquired on them were used to create statistical databases. This in turn allows researchers to estimate expected levels of kinematic excitations on those types of surfaces. This lead to the creation of International Roughness Index (IRI in 1986 [5]), classifying roads based on total suspension deflection over distance travelled, and later of ISO 8608 standard (in 1995 [6]), which classifies roads based on power spectral density (PSD) of the road irregularities' heights encountered on them. Both of those have their limitations, tough. IRI, being a 1-dimensional index, can describe vastly different roads as the same, based purely on the cumulative suspension deflection, as described in [7]. The PSD classification also does not work in every situation, as most real roads do not belong to one class only, but several at once in different frequency ranges (Figure 1) - [8], [9]. It should be noted, that normally PSD of road irregularities' heights is a function of spatial frequency, but can be easily recalculated to temporal frequency if a constant speed assumption is made. If that is the case, then the temporal frequency ω is equal to spatial frequency Ω times velocity v [10].

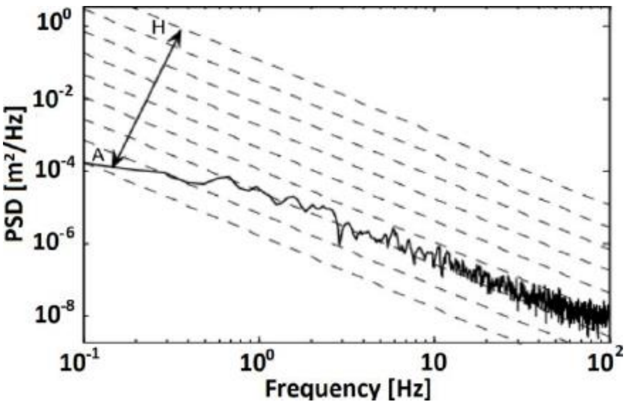

Figure 1. An example of a single road's PSD in multiple classes.
Dashed lines indicate borders between road classes from ISO 8608 standard [9]

The fact that roads of similar statistical characteristics can vary so much means that simulations need to be made on a case-by-case basis. The methods that can be used to obtain kinematic excitations on a specific road vary, but can be generally divided into 2 categories:

I. Profile measurements, which includes different forms of geodetic measurements that give the heights of irregularities very precisely. To derive kinematic excitations from these measurements one needs to know the transfer functions between profile and kinematic excitation. This can be done assuming that the road-tyre pair is a linear time-invariant system. This means the correlation between the system input and output can be written as in formula (1)

$$H(s) = \frac{Y(s)}{X(s)} \tag{1}$$

where $H(s)$ is a tyre transfer function, $Y(s)$ is a Laplace transform of kinematic excitation and $X(s)$ is a Laplace transform of the road profile. This assumption however does not apply to every situation and as such has limited use.

II. Kinematic excitation estimation, which can be further divided into [2]:
   a. direct measurement, which produces road profile data for further processing. It is slow to conduct and expensive and relies heavily on the parameters of a profilometer. The name "direct" comes from the fact, that the sensors used record the profile directly, with no need for further processing,
   b. response-based methods, which register vehicle responses, which then need to be processed to kinematic excitations. These are the least costly and the fastest, but they their accuracy heavily depends on the accuracy of the model used as well as the sensors,
   c. non-contact measurements that produce road profile. They are fast, easy to conduct and quite accurate, but expensive and prone to errors due to environmental factors. They combine the aspects from both previous groups – requiring vehicle models to work and correct errors stemming from vehicle movement, but measuring the displacements directly in reference to the road surface.

The response-based methods are not new – the first tries date as back as 1950's when response-type road roughness measuring systems, RTRRMS for short, started gaining popularity. They consisted of a vehicle and a towed trailer with measuring equipment, that registered the responses of a trailer, such as accelerations. To use this data to reconstruct road profile, a vehicle model needed to be put together. Based on the test that were conducted in the 60's and 70's, quarter-car model was chosen as sufficient for kinematic-excitation estimation and "*the Golden Car"* model was created – which was aimed to simulate a typical passenger car using American roads at the time [11]. It was later used for example for defining IRI. The response-based method, while not very accurate for road-profile estimation, is actually quite well suited for kinematic excitation estimation, as the hardest part to simulate in a vehicle model are tire dynamics. These are responsible for changing the road profile signal into kinematic excitation via the tire filtration properties. The suspension on the other hand acts in a much more predictable way, especially if we do not exceed the range of its linear work. This approximation is permissible if simulation is set to recreate most asphalt roads of modest quality – [12].

One way to use the measured responses to replicate kinematic excitations is a method called Remote Parameter Control, or RPC. The basic concept is as follows: firstly, the real vehicle is driving on a chosen road and unsprung mass accelerations are being measured. Then, the same vehicle is placed in the lab on four hydraulic actuators and the accelerations from the tests are turned into displacements by a computer, which then activates the actuators. The accelerations in the test are measured and compared with the original ones – if there are differences spotted, the computer lowers or strengthens the kinematic excitation signal for that wheel accordingly. After a few iterations acceleration signals from the lab become very similar to those gathered in real life tests, forming the so-called "equivalent road", equal to kinematic excitation [13].

The models used in all those methods however are far from perfect, so new methods of estimating kinematic excitations are being constantly developed, one of which will be presented in this paper. The innovations come in the form of possibility to easily switch between the linear and non-linear models for kinematic excitation estimation, as the whole vehicle model is treated like a black box, which needs only to have kinematic excitation as an input and unsprung mass acceleration as an output. Secondly, the method uses the PID controller, which makes it more flexible, as one can tune its coefficient to make the method work for different types of vehicles.

2. Proposed "Virtual RPC" method

The proposed algorithm (shown in Figure 2) works as follows: first, one needs to obtain an unsprung mass acceleration $\ddot{z}_{m\_T}(t)$ signal from real life test (hence "T" in the underscore), which will act as a base signal, that the algorithm will try to reproduce. However to run a simulation, the kinematic excitation $h(t)$ signal is needed. By using double integration on unsprung mass acceleration $\ddot{z}_{m\_T}(t)$, one will calculate the wheel displacement $z_{m\_T\_E}(t)$, which is assumed in this paper to be roughly the same as the kinematic excitation $h(t)$. This of course is not correct - that is the first estimation (hence the "E" in the underscore to mark that), however this estimation will serve as an input to the simulation, which will be corrected in the next steps. The reasons for which $z_{m\_T\_E}(t)$ and $h(t)$ are different are two-fold – first, they come from the faulty assumption that kinematic excitation $h(t)$ translates directly to the wheel displacements $z_{m\_T}(t)$, secondly they come from the imperfections of recorded acceleration signal $\ddot{z}_{m\_T}(t)$ which is used to calculate the estimated wheel deflection. Those imperfections include noise, trends appearing in the signal or inclusion of results of forces other than those caused by road unevenness. All this contributes to the fact that if a simulation was to be run with the input being double-integrated unsprung mass accelerations, the resulting responses would not match those registered in the tests. To calculate kinematic excitation from the measured accelerations that yields results closer to the responses from the test rides, the authors of this article proposed an algorithm that deals with these problems. It is noteworthy, that the first group of problems connected with the inherent difference between $h(t)$ and $z_{m\_T\_E}(t)$ is more pronounced, so the signal imperfection errors will be not be analyzed further, as they will be removed alongside the bigger error coming from the former source.

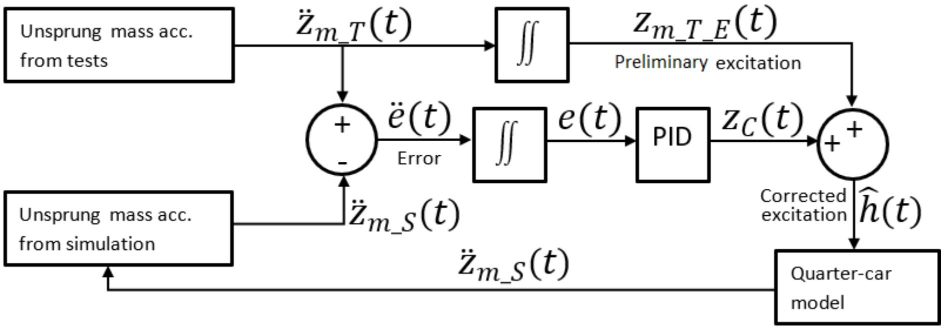

Figure 2. Diagram showing the principle of kinematic excitation estimation using "Virtual RPC" method

The algorithm works as follows: first, the unsprung mass acceleration $\ddot{z}_{m\_T}(t)$ measured in the test is integrated twice and that signal ($z_{m\_T\_E}(t)$) is used as an input to the quarter-car model.

Table 1. Parameters of quarter-car model. "M" refers to sprung mass/suspension parameters, while "m" refers to unsprung mass/tire parameters.

	m [kg]	M [kg]	k_m [N/m]	k_M [N/m]	c_m [Ns/m]	c_M [Ns/m]
Value	50	400	138200	19300	220	2500

The springs and dampers in the quarter-car model were linear and were described using coefficients k_m, c_m, k_M and c_M. Based on that, the responses of the model are calculated, including a new unsprung mass acceleration from the simulation – $\ddot{z}_{m\_S}(t)$. That acceleration $\ddot{z}_{m\_S}(t)$ is then compared with the one from the tests $\ddot{z}_{m\_T}(t)$ and the resulting error $\ddot{e}(t)$ is then also integrated twice, ran through a PID controller and that correction of wheel displacement $z_C(t)$ is added to the wheel displacement (preliminary excitation) $z_{m\_T\_E}(t)$ signal calculated from the acceleration $\ddot{z}_{m\_T}(t)$. The correction is added, because the error is defined as the difference between acceleration from tests $\ddot{z}_{m\_T}(t)$ and from the simulation $\ddot{z}_{m\_S}(t)$. This means that if a simulation signal is smaller than the reference one, the error is positive and it will be added to the corrected signal and vice versa. That corrected signal $\hat{h}(t)$ then enters the quarter-car model as a new excitation $\hat{h}(t)$ and is used for the next iteration. As one can tell, the correction occurs an iteration after the original error was calculated, so it is obvious that the value of reference acceleration might be different in the next step, so it might seem redundant to apply an error correction that comes from the iteration before. This however is dealt with by using a simulation with a small time-step, so that the values of acceleration do not change too quickly for the correction to follow. The exact value for the time-step was chosen empirically to be 0.00001 s, keeping in mind that the changes to which the suspension is able to react do not exceed the frequency of 30 Hz, which translates to a period of 0.03 s repeating, which gives 3333 calculation points every period in the worst case scenario.

The tasks that the authors of the article were met with were as follows:

- to build a mathematical model and its Simulink implementation of presented estimation method,
- to choose PID controller's coefficient so that the error in excitation prediction is minimized,
- to create an array of test signals, which will be used to verify the degree of similarity of estimated profile.

3. Model and its parameters

The vehicle model was implemented in MATLAB/Simulink. The quarter car model was chosen being the simplest, which decreased the number of variables researchers had to take into account analysing the results with its parameters akin to the typical passenger car from C segment [12].

The PID controller's parameters were at first all set to 0, reflecting the situation with no correction. Then, a number of tests was conducted, in which consecutively k_p, k_i and k_d gain values were changed and the extent of allowed values (those that did not cause the simulation to crash) was established, creating three intervals – 1 to 91 for k_p, 0 to 100 for k_i and 0 to 0.2 for k_d. Having done that, the researchers picked 10 evenly distributed numbers within those intervals, with the lowest values being 0 and the highest being the borders of each consecutive interval. The authors ran then the simulation, listing the cumulative error that occurred when the algorithm was trying to reproduce a random signal of 0.01 m amplitude. The k_i coefficient had the biggest influence on the error, increasing it with when its value grew, that is why it was set to 0. The other two displayed similar levels of influence, with k_d having much bigger impact on the stability of the system – that is why the highest value that could be tested was 0.2.The results were then saved in 3-D errors matrix and the lowest value of that matrix was found. The coefficients' values for that lowest error are presented in Table 2.

Table 2. Empirically found optimal values for PID controller's coefficients

	k_p	k_i	k_d
Value	25	0	0.194

4. Verification of reconstruction procedure of kinematic excitation

After the model and its parameters were set, the verification process could begin. Firstly, there was a need to determine exactly what excitations should the authors try to estimate. The authors chose to focus on the determined excitations (sine wave – to be exact) as the results are easy to interpret, both when it comes to amplitudes as well as phase shifts. The important factors in the case of sine waves were their amplitudes and frequencies. The amplitude chosen was 0.003 m as this is also the amplitude of base displacements during EUSAMA test [14]. The frequencies on the other hand were chosen so that they covered the full range of meaningful excitations. Very low frequencies (below 0.5 Hz) do not affect responses of the vehicle in a significant way, as those excitations do not cause big enough accelerations. At the same time, they can be detrimental for this estimation method, as they can cause the appearance of constant values, which lead to the creation of linear trends in estimated excitation signal. On the other side of the spectrum, high frequencies of over 25 Hz also do not contribute to the excitations, as they are filtered out in real life by pneumatic tyre. At the same time, in the simulation environment with tracking control their inclusion leads to destabilization of the estimation. That is why the chosen frequencies for the sine waves were discrete values 1 Hz, 5 Hz, 12 Hz and 25 Hz and also the linear chirp signal that changes frequency from 0.1 Hz to 30 Hz.

The verification method was as follows – firstly, the quarter car model was subjected to a chosen excitation and its unsprung mass acceleration signal was registered. Then, in another Simulink model, the unsprung mass acceleration signal was loaded as an input and based on it the algorithm proposed by the authors reconstructed the kinematic excitation, which caused that acceleration. That estimated kinematic excitation was then compared with the original signal, that the first Simulink model used as an input.

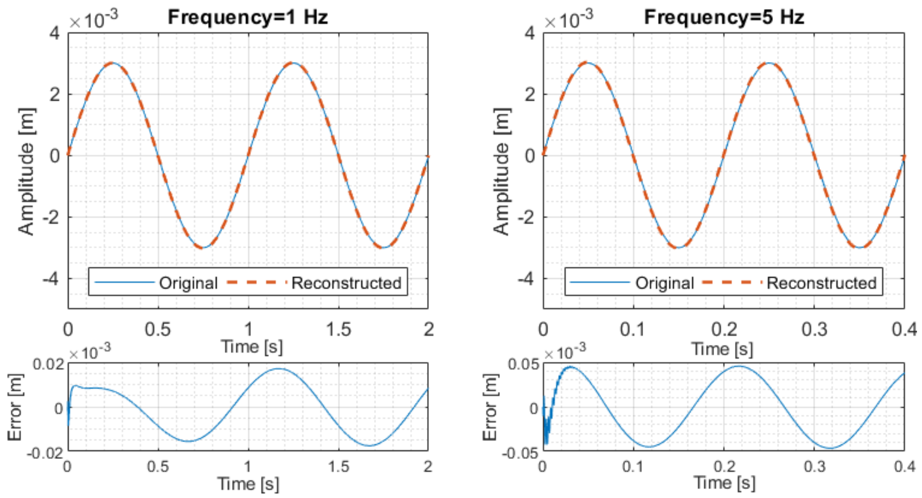

Figure 3. Original and reconstructed signals and errors for sine waves of constant frequencies 1 Hz and 5 Hz

The results for constant frequency sine waves are presented in Figure 3 and Figure 4. As was expected, the method is really effective for slow changing signal, with the maximum error between the original and reconstructed signal being $2 \cdot 10^{-5}$ m, which is 0.7% of the original signal. The error gets bigger with increasing frequency, until it reaches it maximum value for the 25 Hz sine wave – $2.3 \cdot 10^{-4}$ m, which translates to 7.7% of the original signal. The increase in error is almost linear between cases studied.

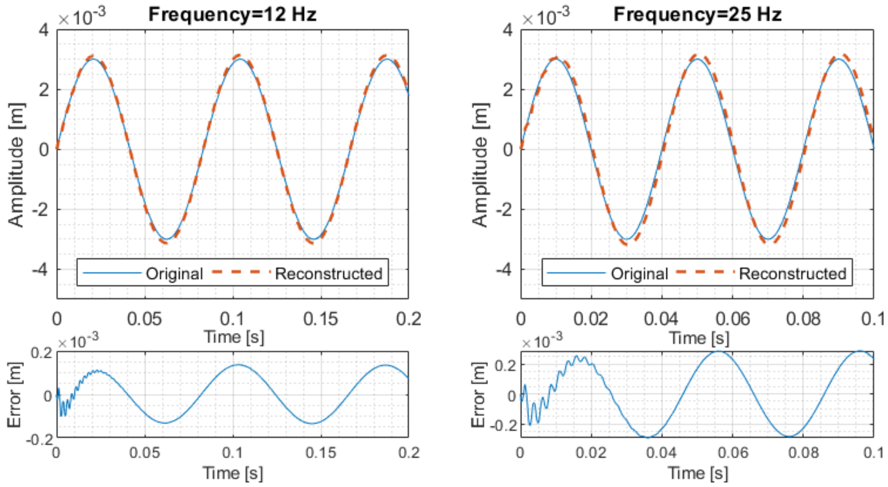

Figure 4. Original and reconstructed signals and errors for sine waves of constant frequencies 12 Hz and 25 Hz

The linear chirp signal is a sine wave that changes its frequency linearly over time. The amplitude was 3 mm, and the interval of frequencies was 0.1 Hz to 30 Hz changing over the span of 30 seconds. As one can see, these go beyond the limits described in the previous paragraph, to test whether or not the method would deal with more extreme cases than those anticipated.

The trend in the whole frequency range is depicted in Figure 5. The frequency changes from 0.1 Hz to 30 Hz in the span of 30 s, so the time is roughly equivalent to the frequency at that time. The relative error and phase shift were calculated by estimating transfer function between estimated and original kinematic excitation signals. The top chart shows the relative error that is slightly lower than the one calculated for constant frequency sine waves. This is because for constant sine waves, the error was calculated as the difference between time signals – and because there is a phase shift, the original signal was decreasing in value before the estimated signal could reach its peak, so that the momentary difference between the two was greater than the difference between peak values, as is the case here. The maximum error was calculated to be 6.1% for 21.5 Hz. The phase shift is very minimal for low frequencies up to about 15 Hz, where it starts to increase in absolute value linearly, reaching almost 6° for 30 Hz.

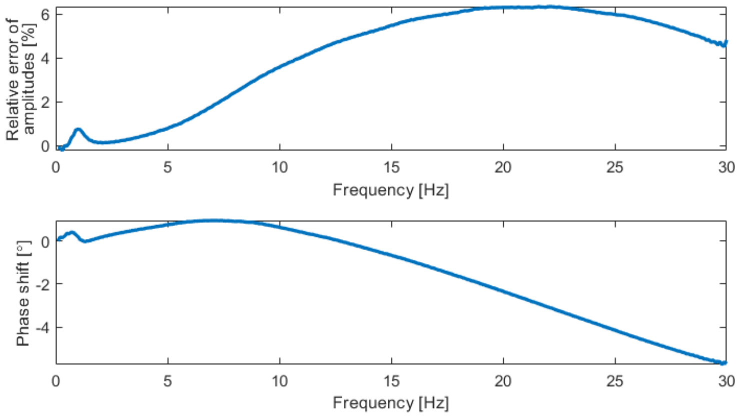

Figure 5. Relative error of estimated signal to the original one and the phase shift between them

5. Conclusions and further studies

The authors of the article set out to describe a new method of estimating kinematic excitations in the simulation based on the unsprung mass accelerations. In the process, they came to several conclusions concerning topics related to the stated goal.

First off, the proposed method can generate results containing quite big errors for frequencies outside of the described scope. This however does not diminish its usefulness, as in the real exploitation only a specific range of frequencies has significant influence on the vehicle's responses. The range of frequencies chosen as significant was between 0.5 Hz and 25 Hz. Taking this into consideration as well as the length of the tyre-pavement contact patch those temporal frequency values translate to the unevenness lengths of 20 cm and 120 m respectively (this is considering that maximum velocity a vehicle can achieve on a very few roads like German Autobahns is 60 m/s).

Considering those limitations, the method proves to be working very well in estimating the determined kinematic excitations that were tested. For low frequencies the error is very minor – not even 1%. It grows to 7.7% for 25 Hz or even 11.3% for 30 Hz, these however are high frequencies on the border or even outside of the scope of frequencies that are important in comfort, safety and fatigue analysis. For the 12 Hz sine wave, which corresponds roughly to the natural frequency of most passenger vehicles' unsprung masses, the error is 5.7%, which is satisfactory.

The fact that the results are satisfactory does not mean no further work is planned to improve the method. The method will be tested by estimating kinematic excitation signals that correspond to roads of classes defined in ISO 8608 standard to check if the results for sine waves and chirp signal correspond to random signals and more complicated tire models and what magnitude the errors would be. The influence of the noise in the signal will be examined and possible solutions to arising problems will be drawn.

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