Vibration study of a vehicle suspension assembly with the finite element method

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Abstract. The main steps of the present work represent a methodology of analysing various vibration effects over suspension mechanical parts of a vehicle. A McPherson type suspension from an existing vehicle was created using CAD software. Using the CAD model as input, a finite element model of the suspension assembly was developed. Abaqus finite element analysis software was used to pre-process, solve, and post-process the results. Geometric nonlinearities are included in the model. Severe sources of nonlinearities such us friction and contact are also included in the model. The McPherson spring is modelled as linear spring. The analysis include several steps: preload, modal analysis, the reduction of the model to 200 generalized coordinates, a deterministic external excitation , a random excitation that comes from different types of roads. The vibration data used as an input for the simulation were previously obtained by experimental means. Mathematical expressions used for the simulation were also presented in the paper.

1. Introduction
The suspension system of a vehicle is highly important for the comfort of the vehicle passengers. The way the suspension reacts to the road shape is reflected into the vibrations that can be felt inside a car.

The present paper is structured in eight major parts and approaches vibration aspects regarding the suspension from both theoretical and simulation point of view. There were considered two types of vibration sources that act upon the spring and damper assembly, first inducing vertical vibrations from the road through the wheel, and then vibration caused by the engine. Several other researches were done on vibrations acting on the suspension approaching various aspects. For example, in paper [1] there was analyzed and then performed an optimization for a suspension system generated with ADAMS CAR software. Paper [3] presents a mathematical model for the noise and vibration coming from suspension systems. The paper introduces the basic concepts of vibration theory and importance to vehicle design. It considers the role of the designer in vibration control and demonstrates methods for the control of vibration to help reduction of noise and harshness. Another vibration approach is paper [4], that presents the noise and vibration transmitted to the interior of the car. Paper [5] is a technical report that documents the results of extensive vehicle-based measurements on Sports Utility Vehicle, for the purpose of vibration source characterization. A sophisticated sensor and data acquisition system was fitted to a vehicle, supported by simultaneous video and CAN data recording. The paper presents a more experimental approach of road-vehicle vibration study.
Article [6] provides a discussion on the studies comprising active experiments conducted on selected structural elements of vehicles. The vibration excitation was achieved by applying an engine working on idle gear. The studies were conducted on vibration propagation for different constant rotational velocity. The changes of the signals were observed in time, frequency and simultaneously in time frequency domains. In [7] there are presented outcomes of a research project concerned with the development of a method for synthesizing, under controlled conditions in the laboratory.

The present paper is based on aspects that can be found in the papers mentioned before and studies the behavior of a city vehicle suspension assembly under two different external excitations: engine and road. The analysis includes several steps:

a) Preload. A static analysis where all the external forces are applied to the model which simulate the vehicle at its initial position. About that position, the model will oscillate.

b) Modal analysis. The model was linearized from the preloaded position and 200 frequencies were extracted. The model was reduced to 200 generalized coordinates.

c) A deterministic external excitation that is transmitted by the engine was applied. The model was tested for different engine rotational speeds.

d) A random excitation that comes from the wheel interaction with the road was applied. The model was tested for different types of roads.

Note that step c) and d) are perturbation steps and are independent on each other. Each are restarts from step b) modal analysis.

The software used in this analysis is Dassault Simulia Abaqus 2017.

2. Theoretical background

A general finite element model is build which include geometric and boundary non-linearities (contact and friction). This model is used to run first step which include bolt preload and gravity. After this general step, a mode base dynamic system has been modelled. The model use mode shapes from a preceding modal analysis and is used in our case to perform sinusoidal and random forced vibration analyses. The analysis is linear and the response is studied in frequency domain. Few theoretical aspects about modal dynamics are showed below.

The finite element dynamic linear model has the following equilibrium equation [9, 11]:

\[ \delta u^N (M^{NM} \ddot{u}^M + C^{NM} \dot{u}^M + K^{NM} u^M) = \delta u^N F^N \]  

(1)

where \( M^{NM} \) is the mass matrix, \( C^{NM} \) is the damping matrix, \( K^{NM} \) is the stiffness matrix, \( F^N \) are the external loads, \( u^N \) is the value of degree of freedom \( N \) of the finite element model (displacement in our case), and \( \delta u^N \) is an arbitrary virtual variation.

For eigenvalue extraction, the following undamped system is solved:

\[ (M^{NM} \omega^2 - K^{NM}) \phi^N_{\alpha} = 0 \]

(2)

Note that the repeated subscripts and superscripts are assumed to be summed over the appropriate range. The modes are orthogonal across mass and stiffness matrices. The eigenmodes are normalized so the largest value of \( \phi^N_{\alpha} \) is 1.

Using eigenmodes from (2), the equation (1) projects into a set of uncoupled modal response equations,

\[ m_\alpha \ddot{q}_\alpha + c_\alpha \dot{q}_\alpha + k_\alpha q_\alpha = f_\alpha \]

(3)

where \( f_\alpha = \phi^N_{\alpha} F^N \) is the generalized load for mode \( \alpha \) and \( q_\alpha \) is the “generalized coordinate” (the modal amplitude) for mode \( \alpha \).

Steady-state excitation is of the form
\[ f_\alpha = A e^{i2\pi ft} \]  

(4)

and creates response of similar form that we write as

\[ q_\alpha = H A e^{i2\pi ft} \]  

(5)

where \( H(f) \) is the complex frequency response function.

In the case of sinusoidal forced vibration, excitation is given by equation (5). For random vibration, the excitation is characterized by cross-spectral density matrix \( S_{NM}^f(f) \) which links all degree of freedom N and M. Projecting the matrix onto the modes, the cross-spectral density matrix for generalized coordinates is obtained:

\[ S_{\alpha\beta}^f(f) = \phi_\alpha^N S_{NM}^f(f) \phi_\beta^M \]  

(6)

The complex frequency response function then defines the response of the generalized coordinates as

\[ S_{\alpha\beta}^q(f) = H_\alpha S_{\alpha\beta}^f(f) H_\beta^* \]  

(7)

where \( H_\alpha^* \) is the complex conjugate of \( H_\alpha \).

Finally, the response of the physical variables is recovered from the modal responses as

\[ S_{NM}^q(f) = \phi_\alpha^N S_{\alpha\beta}^q(f) \phi_\beta^M \]  

(8)

so that the power spectral density of degree of freedom \( u_N \) is

\[ S_{NN}^q(f) = \phi_\alpha^N S_{\alpha\beta}^q(f) \phi_\beta^N \]  

(9)

3. Road profile

Road can be modelled as random excitation. Since we are going to run the road excitation analysis in spectral (frequency) domain, we need Power Spectral Density (PSD) of the road profile function of frequency. The road roughness PSD distribution can be approximated by [12]:

\[ \Phi(\omega) = \frac{2\alpha V^2 \sigma^2}{\omega^2 + \alpha^2 V^2} \]  

(10)

Where \( \sigma^2 \) is the road roughness variance, \( V \) the vehicle speed, \( \alpha \) depends on the type of road surface, and \( \omega \) is frequency in rad/s. Variance and \( \alpha \) are adopted from Table 1 in function of road class.

| Road class       | \( \sigma \times 10^{-3} m \) | \( \Phi(\Omega_0) \times 10^{-6} m^4 \), \( \Omega_0 = 1 \) | \( \alpha \) (rad/m) |
|------------------|---------------------------------|-------------------------------------------------|---------------------|
| A (very good)    | 2                               | 1                                               | 0.127               |
| B (good)         | 4                               | 4                                               | 0.127               |
| C (average)      | 8                               | 16                                              | 0.127               |
| D (poor)         | 16                              | 64                                              | 0.127               |
| E (very poor)    | 32                              | 256                                             | 0.127               |
For a vehicle speed of 17.7 km/h the excitation PSD displacement is obtained and shown in Figure 1a). Since the input in Abaqus require acceleration PSD for a random process, we obtained that by double derivation of displacement. The obtained curves are shown in Figure 1b) and are used in the road excitation analysis from paragraph 5.4.

4. Finite element model

The strut suspension main assembly was 3D modelled in order to use its geometry for the finite element analysis. Its most important elements can be seen in Figure 2. The representation in the left is the left side assembly of the front suspension seen from behind. The other is the side view of the same assembly.

The two components of the model are the suspension damper and the knuckle. The suspension damper was modelled by spring – dashpot connector with a stiffness of 25 N/mm and damping coefficient 0.0959. On its top mount it acts a mass of 328 kg that represents the static vertical load on one side of the front suspension in the case of a car with two passengers, each of them weighing 75 kg.

Both the damper support and the knuckle were modelled for the finite element analysis using tetrahedral elements [10]. The link between these two parts is done by two M12 bolt – nut assemblies. These were modelled by using hexahedral elements. The bolt axial preload is 40kN.

The tire that links the vehicle to the road surface was modelled by spring – dashpot connector elements with stiffness of 200 N/mm and damping coefficient 0.001562.

| Component         | Material          | Density | Elasticity Modulus | Yield  |
|-------------------|-------------------|---------|--------------------|--------|
| Suspension damper support | Hard Steel       | 7.8 g/cmc | 210000 MPa        | 1300 MPa |
| Knuckle           | Forged Steel      | 7.8 g/cmc | 210000 MPa        | 420 MPa |
5. Analysis

5.1. Preload

The preload step is a general nonlinear analysis. This step includes preload of the M12 bolts and gravity load. The model includes geometric and boundary nonlinearities. Geometric nonlinearities are modeled using Large Displacement Finite Element theory and boundary nonlinearities are due to contact and friction.

Figure 2. Quarter car suspension Finite Element Model.

Figure 3. Preload.

a) displacement contour plot; b) VonMises contour plot: knuckle; c) VonMises contour plot: spring-damper support
Figure 2 shows the displacement and stress contour plots after the preload step. Bolts are tightened at 40kN axial load. The stress obtained in this step is mainly compressive and is due to bolt preload.

The next step in the analysis is modal analysis. Modal analysis is a linear perturbation step. When considering nonlinear geometry, Abaqus starts the perturbation step from the previous general step. In our case the modal analysis starts from the displacement state in Figure 3a taking into consideration the preload stress (Figures 3b and 3c).

5.2. Modal analysis
A modal analysis is run first up to 200 modes. Table 3 shows first 7 modes.

| Mode No. | Frequency [Hz] | Frequency [rpm] | Observation                      |
|----------|----------------|-----------------|----------------------------------|
| 1        | 1.3            | 78              | Spring-damper axial motion       |
| 2        | 3.25           | 195             | Tire lateral motion              |
| 3        | 3.3            | 198             | Tire fore-aft motion             |
| 4        | 15             | 900             | Tire vertical motion             |
| 5        | 27             | 1620            | Twist around fore-aft axis       |
| 6        | 28             | 1680            | Twist around lateral axis        |
| 7        | 521            | 31260           | High frequency mode              |

For this analysis, first 6 modes are of interest since both road and engine excitation are within this frequency range. Above 7th mode, the frequency is too high for our analysis purpose. However, we extracted 200 modes to reduce the model in the subsequent modal dynamic analysis steps to 200 generalized coordinates.

The following analyses are:
- Engine excitation having the frequency range: 600-6000 rpm. Modes 4, 5 and 6 are in the excitation range of the engine
- Road excitation having the frequency range: 0.5-15 Hz. Modes 1-4 are in the road excitation range

5.3. Engine excitation analysis
This step is a modal dynamic steady state analysis and uses the modes from the frequency step (modal analysis) to build the reduced model. The analysis is done in frequency domain.

A constant acceleration excitation of 4g is considered in all three directions, spanned over a range of frequencies (angular velocity of the engine). We simulate an engine malfunction which generates sinusoidal excitations transmitted through vehicle body to the suspension.

Figure 4a shows the direction of the excitation and boundary condition of the quarter-car suspension. Figure 4b shows the excitation spanned over the engine angular velocity rage: 600-6000 rpm.

Figures 5a and 5b show the acceleration response at wheel center and spring base. The main resonant frequencies are 1620 and 1680 rpm which correspond to modes 5 and 6 of the suspension system. The resonant frequencies are caused by lateral and fore-aft excitation. It is observed a maximum acceleration of 15 g acceleration at wheel center and 10 g at spring base. There is also observed a smaller peak at 900 rpm (4th mode) in the case of vertical excitation. Note that mode 5 (1620 rpm) is excited by lateral excitation and mode 6 (1680 rpm) by fore-aft excitation.
Figure 4. Excitation.

a) Excitation and Boundary Conditions; b) Engine Excitation vs. angular velocity.

Figure 5. Acceleration response.

a) Wheel center; b) Spring base

Figure 6. Displacement Response.

a) Wheel center; b) Spring base
Figures 6(a) and 6(b) show the displacement response at wheel the center and at spring base. Still resonant frequencies at modes 5 and 6 for lateral and fore-aft excitation and mode 4 for vertical excitation.

![Figure 6. Displacement response.](image)

**Figure 6. Displacement response.**

- a) Tire spring-damper
- b) Suspension Spring-Damper

Figures 7(a) and 7(b) show the forces at tire and suspension spring-damper. A maximum force of 1000 N is obtained at tire.

![Figure 7. Force response.](image)

**Figure 7. Force response.**

- a) Tire spring-damper
- b) Suspension Spring-Damper

Figures 8 show the displacement and Von Mises stress contour for Lateral Excitation at 1620 rpm.

![Figure 8. Lateral Excitation at 1620 rpm](image)

**Figure 8. Lateral Excitation at 1620 rpm**

- a) displacement contour plot
- b) VonMises contour plot: knuckle
- c) VonMises contour plot: spring-damper support

There can be observed the values of stress over yield in the knuckle. However, the values are localized in few spots and may not create major damage if the engine malfunction is sporadic. If the malfunction is persistent and the engine angular velocity is kept constant at 1620 rpm, permanent deformation or even crack might appear in the knuckle. Similar stress values are obtained in the fore-aft excitation.

5.4. Road excitation analysis

This step is a modal dynamic steady state analysis and uses the modes from the frequency step (modal analysis) to build the reduced model. The analysis is done in frequency domain. In this step, we consider a vehicle which travels at constant speed of 17.7 km/h on flat road.
Five types of roads are considered: very good, good, average, poor, very poor. The road is modeled as random excitation and it is described in paragraph 3.

![Model showing the road excitation and boundary conditions](image)

**Figure 9.** Road excitation and Boundary Conditions

Model showing the road excitation and boundary conditions is in Figure 9.

![Acceleration response](image)

**Figure 10.** Acceleration response: Car body (sprung) mass

![Displacement response](image)

**Figure 11.** Displacement response: Car body (sprung) mass

We monitor the mass point that represents the car body mass called sprung mass. Figures 10 and 11 show the acceleration and displacement response at sprung mass. The suspension resonates at two frequencies 1.3 and 3.25 Hz, modes 1 and 2 respectively. Mode 1 is the spring-damper first mode and mode 2 is tire mode. Maximum acceleration is obtained at 3.25 Hz and its control depends only on the tire damping, any modification in the spring-damper system will not affect the acceleration response at this frequency. Maximum acceleration is 0.9g for the very poor road and 0.05g for the very good road. Regarding the displacement, it’s maximum is found at 1.3 Hz and that can be controlled by varying the spring-damper parameters. Maximum displacement is 51mm for the very poor road and 3mm for the very good road.
Figures 12 and 13 show the relative displacement and force in the damper. As expected the highest values are at damper first frequency of 1.3 Hz. The maximum relative displacement is 43 mm for the very poor road and 3 mm for the very good road. The maximum force is 1090N for the very poor road and 68N for the very good road. There can be observed large differences in values between very good roads and very poor roads. As expected, the comfort increases significantly with increase of road quality.

Next, we are going to show the effect of the road quality on the knuckle stress. Since we have a random excitation, the stress contour plot is in Root Mean Square (RMP) values and it is presented in the stress components: 11 (xx), 22, (yy) and 33 (zz). There is shown only normal stresses for the very poor road, average road and very good road for comparison purpose.

Figure 12. Relative displacement response: Damper

Figure 13. Force response: Damper

Figure 14. RMS Stress Sxx.
   a) very poor road; b) average road; very good road
Figures 14-16 show knuckle RMS normal stresses on x, y and z directions for a very poor, average, and very good roads. As the vehicle travels with constant 17.7 km/h, we observe that on average and very good roads, the stress values are below yield. For very poor road the stress is above yield on
several spots and the knuckle can encounter permanent deformation or even crack. So even on low vehicle speed, the low-quality roads can cause serious damage on the suspension knuckle.

6. Conclusions
The results are part of a study regarding the effect of the random vibration generated by the road shape, combined with sinusoidal vibration generated by the engine. A goal was to compare the effect of engine vibration with the road vibration and to study the displacement, stress and strain for suspension system parts for various working conditions.

The authors performed several physical tests on the vehicle which the suspension is analyzed. The physical tests were used as inputs on the actual analysis. A realistic model which includes several types of nonlinearities was developed. The model was correlated with the actual tests performed on the vehicle. A nonlinear finite element model was developed and used it at the initial state for the following linear perturbations analyses. Both engine and road excitations were used in the analysis.

The paper shows the methodology of analyzing a suspension mechanism using Abaqus software. It starts from generating the virtual model and ends with interpreting results and effects of both random and sinusoidal vibration over stress, strain and displacements for various mechanical parts. Simulations were done with different input data for the two sources of excitation and a comparison between their effects over the passenger comfort was performed.

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