

Identification of the Natural Frequencies of a MacPherson Suspension Using FFT, FRF, and Spectral Analysis [†]

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Abstract

This paper presents the results of identification of the natural frequencies of a MacPherson suspension using fast Fourier transform (FFT) analysis, frequency response function (FRF) analysis, and spectral analysis. Experimental investigations are conducted to determine the accelerations of a quarter-car MacPherson suspension system of an Audi A3 passenger car at a tire inflation pressure of 0.22 MPa. The investigated suspension control arm was equipped with one original equipment manufacturer (OEM) bushing and one polyurethane bushing. The paper also presents the results obtained using a set of MATLAB scripts developed for processing the vibration measurement data acquired from the components of the vehicle's quarter-car suspension system, with each script corresponding to a specific analysis approach. The developed scripts implement several methods for analyzing the recorded acceleration signals, including FFT, FRF, and spectral analysis. The results obtained by applying these three methods to the experimental data are compared and discussed.

Keywords: experimental study; OEM bushing; polyurethane bushing; natural frequencies; MacPherson suspension; MATLAB; FFT; FRF; spectral analysis

1. Introduction

The vehicle suspension system plays a major role in determining ride comfort, handling stability, and driving safety. Its performance is directly related to the transmission of dynamic excitations from the road surface to the wheel, suspension components, and vehicle body. For this reason, the investigation of vertical vibrations and the frequency response of suspension systems remains an important scientific problem with clear practical relevance, especially when an optimal compromise between comfort and handling is sought. In the literature, this issue is commonly addressed using quarter-car models, which provide a sufficiently accurate interpretation of the main dynamic processes in the system within a relatively simplified framework [1].

Frequency-domain analysis is particularly suitable for the investigation of vehicle suspensions because it enables both a clear identification of the characteristic resonance regions and an assessment of the influence of individual design and operating parameters on the vibration response. In quarter-car models, two main frequency ranges are usually distinguished: a low-frequency range associated with the oscillations of the sprung



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mass and, consequently, ride comfort, and a higher-frequency range associated with the unsprung mass, the tire–road contact, and the so-called wheel-hop mode [2]. Therefore, determining the natural frequencies of the suspension is essential both for analyzing its dynamic behavior and for comparatively evaluating different design solutions.

The frequency response of a suspension system is strongly affected by the mechanical properties of the elastic elements and vibration isolators used in the structure. Their stiffness, damping capacity, and frequency-dependent behavior largely determine the manner in which vibrations are transmitted from the wheel to the load-bearing suspension elements and subsequently to the vehicle body. Recent studies have shown that the actual behavior of rubber-elastic isolators cannot be described with sufficient accuracy solely by constant linear parameters, since their stiffness is functionally related to frequency, amplitude, and preload [3–5]. This necessitates an experimental approach when assessing their influence on vehicle vibrations.

The MacPherson suspension system is of particular interest because of its widespread application in passenger cars. It is characterized by a relatively simple design, compactness, and good functional performance. In this suspension layout, the transmission of vibrations to the vehicle body depends not only on the characteristics of the spring and the damper, but also on the properties of the rubber-elastic connecting elements. Published studies on MacPherson suspension systems have shown that changes in the characteristics of these elements may lead to significant variations in the amplitude-frequency response, in the propagation of vibrations through the individual components, and in the natural frequencies of the system [6–8].

In addition to experimental measurements, the method used for processing the recorded signals also plays an essential role in the correct interpretation of dynamic processes. The vibration acceleration signals measured at characteristic points of the suspension can be analyzed using the Fast Fourier Transform (FFT), frequency response function (FRF), and spectral analysis, with each of these approaches providing different types of information about the system. The combined use of several methods allows the natural frequencies to be identified more reliably and enables a more comprehensive assessment of the influence of various design factors on suspension behavior [9].

Despite the considerable number of publications devoted to the modeling and analysis of vehicle suspension systems, the number of experimental studies in which the influence of bushings with different mechanical properties on the natural frequencies of a quarter-car MacPherson suspension is evaluated through the simultaneous application of several frequency-domain methods to real measured signals remains relatively limited. The aim of this study is therefore to experimentally assess the influence of vibration isolators with different mechanical properties on the natural frequencies of a quarter-car MacPherson suspension system, with two original equipment manufacturer (OEM) bushings and one polyurethane replacement bushing. For this purpose, vibration measurements were conducted at characteristic points of the suspension, and the recorded signals were processed using a MATLAB-based script. The data were analyzed by means of the Fast Fourier Transform (FFT), frequency response functions (FRFs), and spectral analysis, which enabled a comparative evaluation of the three approaches in identifying the characteristic frequencies of the investigated system.

2. Materials and Methods

2.1. Experimental Equipment

2.1.1. External Excitation

For the experimental investigation of the natural frequencies of the quarter-car suspension system, a shock absorber tester was used as the source of external excitation.

The experiments were performed using a Beissbarth SA 640 suspension tester (Beissbarth GmbH, Munich, Germany) operating according to the EUSAMA principle [9–15]. During the test, the so-called sweep curve is obtained, expressing the relationship between the wheel-to-plate contact force and the excitation frequency [8,9].

The test procedure is as follows. First, the static wheel load acting on the vibrating plate is measured. The plate then generates vibrations with a frequency of up to approximately 25 Hz and a constant amplitude of 6 mm [9,10]. Subsequently, the excitation is terminated, i.e., the electric motor is switched off, while the test bench continues to record the wheel load and the excitation frequency as functions of time until the oscillations decay [10,11]. This gradual passage through the frequency range constitutes the so-called sweep [10,11].

2.1.2. Measurement System

Vibration measurements were conducted at characteristic points of the suspension using a measurement system developed by the authors in LabVIEW 2021 SP1 and MATLAB R2022b [16].

Figure 1 shows the suspension tester, the vehicle, and the measurement system.

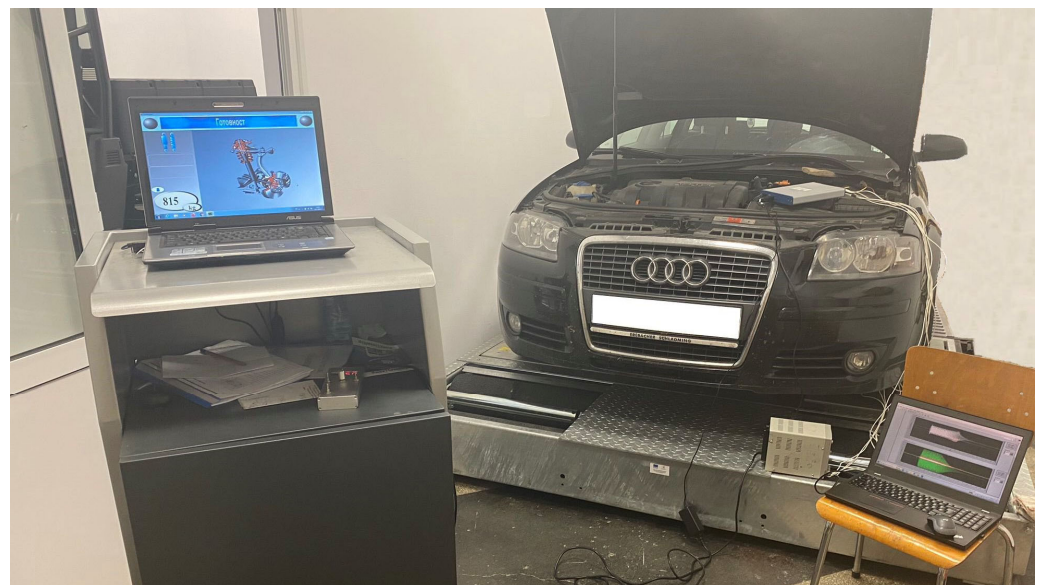


Figure 1. Experimental equipment [16].

2.2. Determination of the Natural Frequencies

To determine the characteristic frequencies of the investigated suspension system, the recorded vibration signals were processed using the Fast Fourier Transform (FFT), the frequency response function (FRF), and spectral analysis. These methods were applied to identify resonance regions, natural frequencies, and the frequency-dependent dynamic response of the suspension system.

The Fast Fourier Transform (FFT) is an efficient algorithm for signal spectral analysis that transforms the time-domain representation of the measured vibration excitation into a frequency spectrum. This provides information on the amplitudes of the individual frequency components contained in the signal, thereby enabling the identification of resonance regions and the determination of the characteristic frequencies of the dynamic system. In studies of automotive suspension systems, the FFT method is widely used for processing vibration acceleration signals, since it allows a reliable evaluation of the frequency response and a comparative analysis of different experimental cases.

The frequency response function (FRF) is one of the principal tools used in the analysis of linear dynamic systems, as it characterizes the relationship between the input and output

signals in the frequency domain. It provides information about the amplitude and phase characteristics of the system as a function of the excitation frequency. Consequently, this method is particularly well suited for identifying resonance modes, determining natural frequencies, and evaluating the dynamic behavior of automotive suspension systems subjected to external excitation.

Spectral analysis is a signal processing approach used to examine the frequency content of a signal through the analysis of spectral density or amplitude spectrum. The method makes it possible to identify the dominant frequency components in the measured signal and to evaluate how the energy is distributed over different frequency ranges. In studies of automotive suspension systems, spectral analysis complements the results obtained from FFT and FRF by providing more detailed information on the frequency structure of the vibration process and supporting a more accurate determination of the characteristic frequencies of the system.

3. Results

3.1. Determination of the Natural Frequencies

3.1.1. Fast Fourier Transform

A MATLAB script was developed to process the acceleration data obtained from the three sensors. The script loads the data file containing the measured acceleration signals, performs FFT analysis for `acc_plate`, `acc_arm`, and `acc_body`, and plots the signals in the time domain. It also generates the corresponding one-sided amplitude spectra, identifies the dominant frequency components. This processing procedure allows a clear assessment of the frequency content of the measured signals and facilitates the comparison of the dynamic response at the selected measurement points. The acceleration signals recorded by the three sensors are shown in Figure 2.

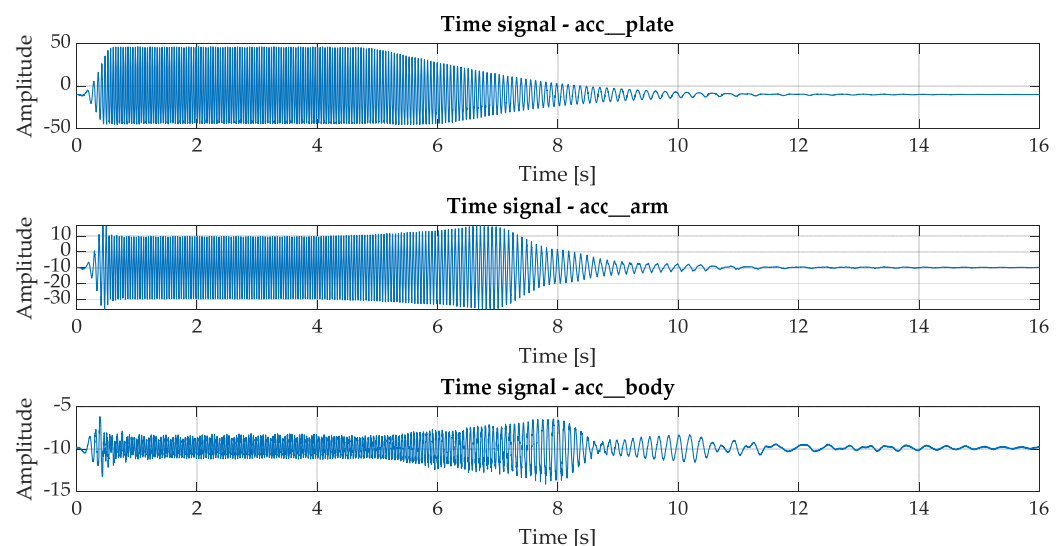


Figure 2. Time signals.

Figure 3 presents the results of the FFT analysis.

The analysis of signal “`acc_body`”, corresponding to the vehicle body response, revealed a dominant low-frequency component at approximately 3.17 Hz, which may be associated with the natural frequency of the sprung mass. A second pronounced peak was observed at approximately 5.62 Hz, indicating the presence of an additional vibration mode in the body response.

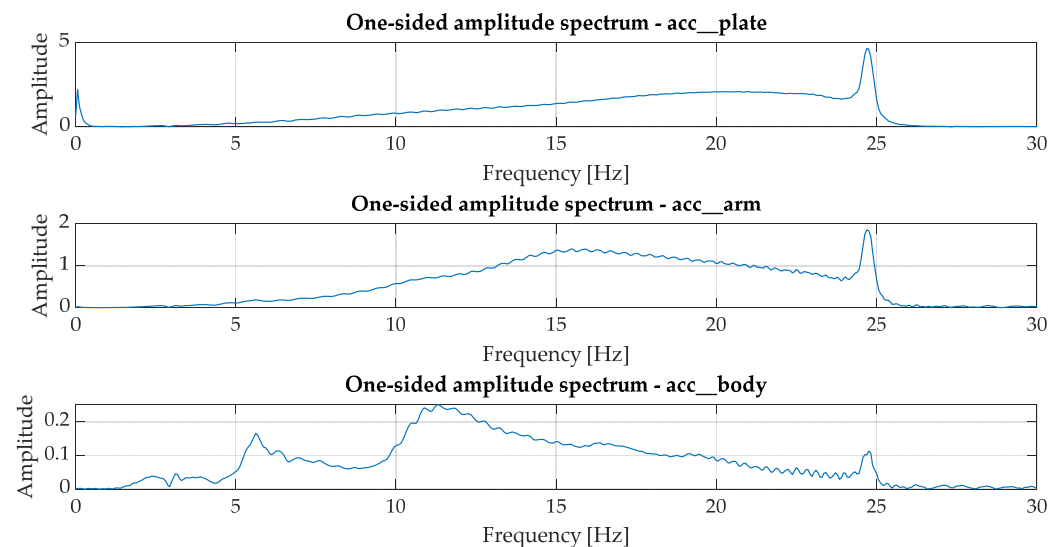


Figure 3. One-sided amplitude spectrum.

3.1.2. Frequency Response Function

A MATLAB script was developed for processing the measured signals from the three channels corresponding to the test plate, suspension arm, and vehicle body. The signals were first centered by removing their mean values and then band-pass filtered in the range 0.5–30 Hz. The frequency response functions between the plate signal and the responses of the suspension arm and the body were subsequently determined, allowing the characteristic frequencies of the system to be identified. The coherence function was used to assess the quality of the obtained results, while the damping ratio for the dominant body mode was estimated by the half-power bandwidth method.

The ordinate axis represents $|H(f)|$, defined as the ratio of the output acceleration amplitude to the input acceleration amplitude, as shown in Figure 4.

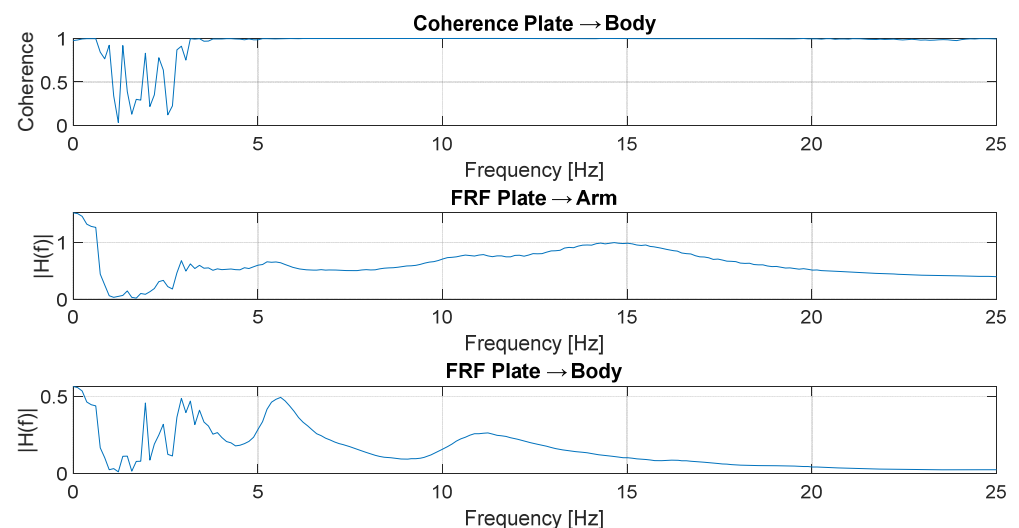


Figure 4. Coherence Plate → Body and magnitude of the frequency response function (FRF) for the Plate → Arm and Plate → Body transfer paths.

The combined interpretation of the FRF and coherence results allows the main vibration modes of the investigated quarter-car suspension system to be identified with sufficient confidence. The coherence function for the Plate → Body path is close to 1 over almost the entire frequency range above approximately 3 Hz, which indicates a strong linear relationship between the excitation from the test bench plate and the measured body response.

This confirms that the corresponding FRF peaks are physically meaningful and that the results in this range may be considered reliable. In contrast, the coherence reduction below approximately 3 Hz suggests that this low-frequency region is more sensitive to noise, insufficient excitation energy, or nonlinear effects and should therefore be interpreted with caution. The FRF results show a pronounced resonance peak for the Plate → Body path at approximately 5.5–6 Hz, which may be associated with the body mode of the suspension system. A second, broader maximum is observed around 11 Hz, indicating an additional vibration mode contributing to the body response. For the Plate → Arm path, the dominant peak is located in the range of approximately 14–15 Hz, which can be associated with the unsprung-mass mode or wheel-hop behavior of the wheel–suspension assembly. The significantly lower FRF magnitude for the body compared with the suspension arm over most of the analyzed frequency range indicates that the suspension system effectively attenuates the transmitted vibration before it reaches the vehicle body. Overall, the results suggest that the tested suspension exhibits stable and predominantly linear dynamic behavior, with no clear indication of severe clearance, strong nonlinearity, or irregular vibration transmission.

3.1.3. Spectral Analysis

The spectrogram shown in Figure 5 reveals distinct regions of increased vibration energy as the excitation sweeps through the frequency range, indicating resonance phenomena in the investigated suspension system.

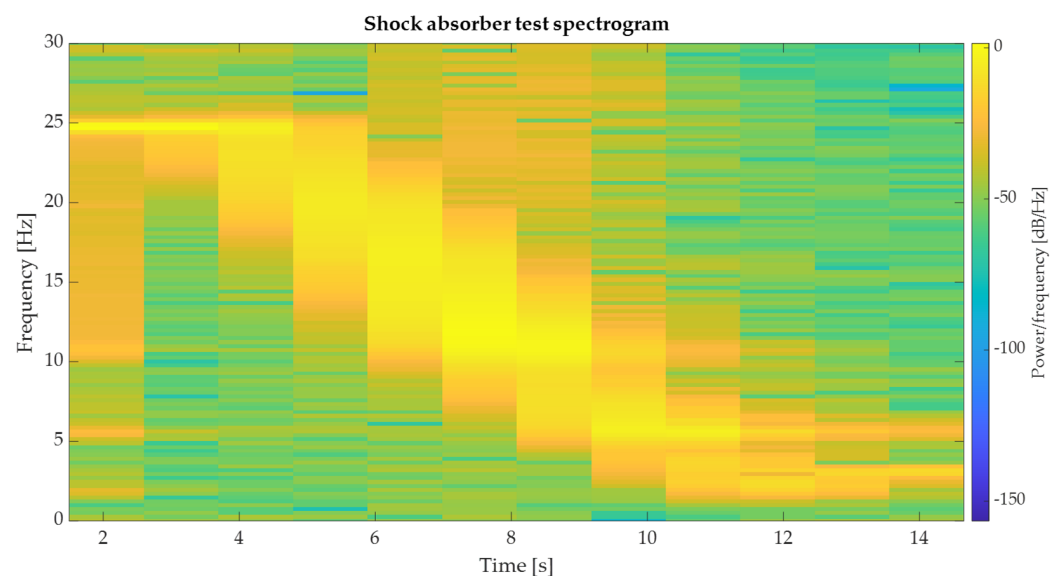


Figure 5. Shock absorber test spectrogram.

The strongest energy concentration is observed at approximately 5–6 Hz, which can be considered the dominant natural frequency of the body-related mode. In addition, a secondary energy concentration is visible around 10–11 Hz, indicating the presence of another characteristic vibration mode. Therefore, the spectrogram confirms that the investigated system exhibits at least one clearly dominant low-frequency natural mode and a second, less pronounced mode at higher frequency. These observations are in good agreement with the results obtained from the FRF and FFT analyses.

4. Discussion

The FFT, FRF, and spectrogram analyses provide a consistent interpretation of the dynamic response of the investigated quarter-car MacPherson suspension system. All three methods indicate a dominant body-related resonance in the range of approximately

5–6 Hz, a secondary mode around 10–11 Hz, and a higher-frequency suspension-related mode near 14–15 Hz. The agreement between the FRF peaks and the spectrogram energy concentrations suggests that these frequency components are physically meaningful characteristics of the measured response rather than signal-processing artifacts. However, the resonance at 5–6 Hz should not be interpreted as the classical global body bounce mode of the entire vehicle, which is typically expected at lower frequencies. Instead, it is more likely associated with a local or coupled body-related mode of the quarter-car suspension–body system. This interpretation is consistent with the experimental configuration, in which excitation was applied locally by the suspension tester and the response was measured at a point on the body close to the suspension attachment. In addition, one of the original equipment manufacturer (OEM) rubber bushings in the investigated suspension was replaced by a polyurethane bushing, which may have increased the local stiffness of the suspension connection and contributed to the shift in the identified body-related resonance toward higher frequencies. The coherence values, which remain close to unity above approximately 3 Hz, further confirm the reliability of the identified resonance peaks within the relevant frequency range.

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