Time and frequency analysis of the vehicle suspension dynamics

Rosen Miletiev, Ivaylo Simeonov, Emil Iontchev, Rumen Yordanov

***Abstract*—**The current paper describes MEMS inertial data acquisition system which is installed on the vibration stand to analyze the static vehicle suspension dynamics. The analysis is accomplished in the time and frequency domain using STFT. Also the STFT parameters are analyzed to obtain the optimal time and frequency resolution of the spectrum response. The time attenuation and total frequency range of the oscillations are established.

***Keywords*—** MEMS inertial sensor, suspension dynamics.

I. Introduction

The MEMS accelerometers may perfectly address active safety systems in the automotive domain. Control of car roll-over, vehicle stability for skidding and antilock braking, parking brake energy, activation of wheel pressure monitoring, suspension adaptation to car and road condition. MEMS sensors allow the implementation of a lot of different functions, as free-fall detection, precise tracking [1], kinetic measurements [2], tilt measurements [3], vehicle vibration monitoring [4], antitheft and many others.

The main reasons of the suspension failures are recognized as deformation multiplications of the flexible suspension elements and liquid leaks from the dampers while the main suspension failure symptoms are directed to the increased noise, increased oscillation amplitudes of the vehicle body and their slow attenuation.

Several methods exist for the damper diagnostics, as the study of the unstrained oscillations of the vehicle body after an artificial roughness transition and comparison of the recorded oscillation curve with the standard one (Siems & Klein KG, M-Tronic BIG RED), study of the maximum amplitude and average resonance frequency of the vehicle wheels (Boge, Sachs) or measurement of the friction between the wheel and the test stand (Ravaglioli Beissbarth).

The drawbacks of these methods are obliged to the dependency of the obtain results from the status of the all suspension parts not only from the dampers and springs. Also the definition of the standard attenuation curves is very difficult or sometimes impossible task.

The evaluation of the suspension status is accomplished on the basis of the expensive diagnostic equipment for the dampers and springs while the other suspension elements status are frequently determined by subjective methods like shaking, shifting, pressing, etc.

There are two popular suspension testing systems that utilize the sine swept tire shaker. They each characterize the damper condition in slightly different manners [5-8]. Regardless of some shortcomings (damper fluid temperature [9] and tire pressure dependence [10]), if care is taken these methods may be effective.

The proposed instrumentation for calculation of the suspension dynamic response is made with MEMS technology that is less expensive and more accurate then the technology used in the noted suspension testers. Another feature is that the system is portable and easy to use. Also the elaborated MATLAB routine for data analysis allows developing a quick and time efficient test to assess the performance of the suspension. The application of the numerical integration allows calculating the speed and the displacement of the suspension elements as the low frequency components of the inertial data are rejected from the signal. This procedure is accomplished by a band-pass filter, which is constructed on the basis of the signal spectrum.

II. Experimental study

A common approach for Suspension Parameter Testing (SPT) systems is to take the vehicle in static positions, and apply forces in specific ways to record the deflection of the component to be characterized, i.e. the tire, the springs, or the various linkages. These quasi-static tests are great for suspension diagnosis or further developing the models that have helped advance the automotive design process.

The testing procedure is accomplished by the execution of the strength F with a constant magnitude and an alternate direction to the longitudinal axis of the wheel (Figure 1).

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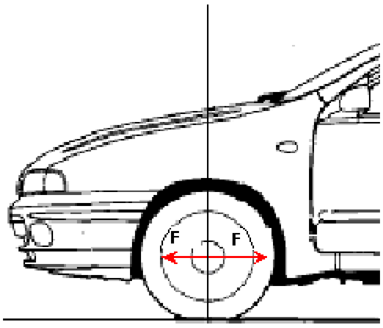


Fig.1. Test formulation

The experimental car is lifted on the vibration stand type BOGE-AFIT ShockTester, which excites the platform to 16 Hertz, and then shuts off. The ensuing platform vibrations decay at a rate that infers the performance of damper. The data acquisition system based on inertial MEMS sensor is located on the vibration stand and measure the vehicle suspension response. The vibration amplitude and frequency depend from the suspension design and condition.

The test car is Fiat Bravo with installed Macpherson strut front suspension and its position on the vibration stand is shown at Figure 2.

The vibration stand generates a vertical force G_Z which is decomposed to one force G to the direction of the steering knuckle and another force G_X which is directed to the horizontal axis of the vibration stand. The amplitude of the horizontal force and suspension carrier depends from the slope of the steering knuckle axis γ and the vibration stand inclination angle γ^1 according to the equation:

$$G_X = G_Z \sin(\gamma+\gamma^1). \qquad (1)$$

The applied vertical force G_Z of the vibration stand is equal approximately to 2500N and the geometry angles γ and γ^1 are equal respectively to 2° and 3°. Therefore the horizontal force G_X is equal approximately to 200N. This force causes the X axis vibrations of the data acquisition system and the vehicle suspension elements.

The data acquisition system is based on the MEMS accelerometer sensor LIS3LV02DQ – a three axes digital output linear accelerometer, produced by *ST*. This sensor includes a sensingelement and an IC interface able to take the information from the sensing element and to provide the measured acceleration signals to the external world through an I^2C/SPI serial interface. The MEMS sensor has a user selectable full scale of ±2g, ±6g and it is capable of measuring acceleration over a bandwidth of 640 Hz for all axes [11]. The inertial MEMS sensor reads the data with a sampling frequency of 40Hz to satisfy the Nyquist criteria.

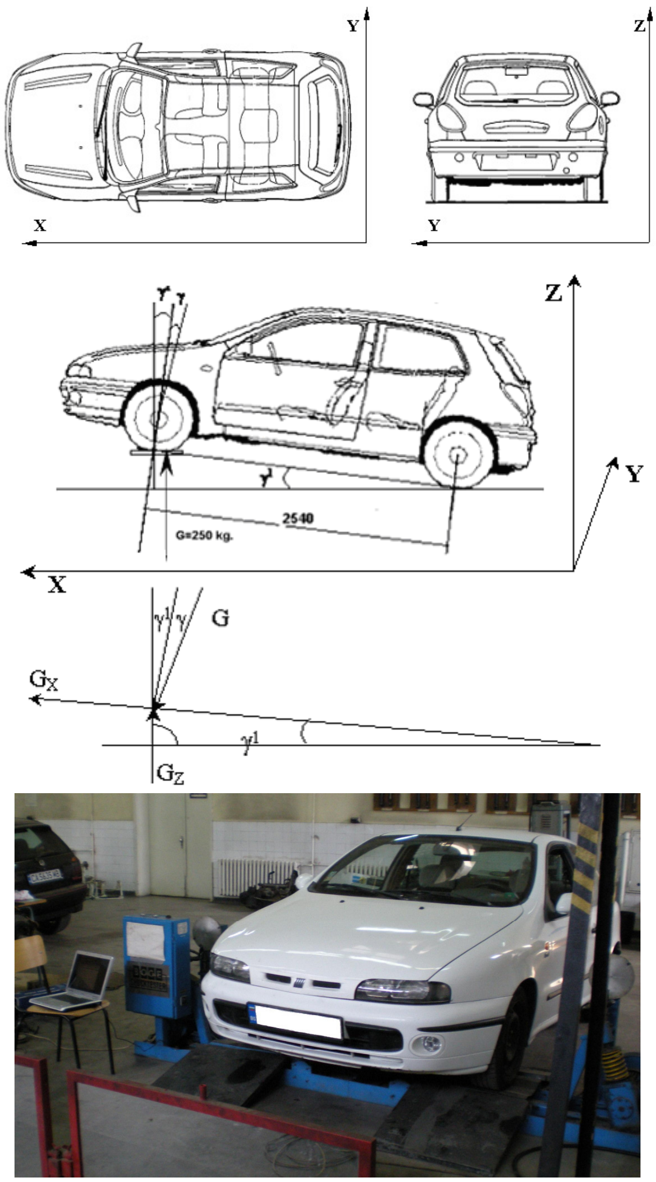


Fig. 2. Vehicle situation on the vibration stand

III. Data Analysis

The data analysis of the suspension dynamics had to determine the following characteristic:

- Attenuation time;
- Attenuation line slope (attenuation speed);
- Main dominant frequencies and magnitudes;
- Dependence of the determined frequencies of the suspension elements status;
- Calculation of wheel speed and displacement;
- Integral evaluation of the suspension.

During the experiments every vibration attempt is accomplished three times (Figure 3). The time domain analysis clearly shows the three main processes of the vibration cycle which may be defined as (Figure 4):

1. vibration stand acceleration process
2. stationary vibration process
3. oscillation and attenuation process

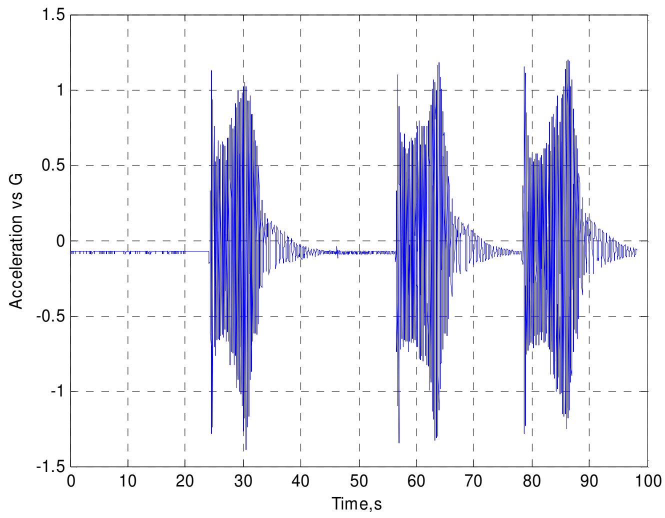


Fig. 3. Recorded vibration diagram

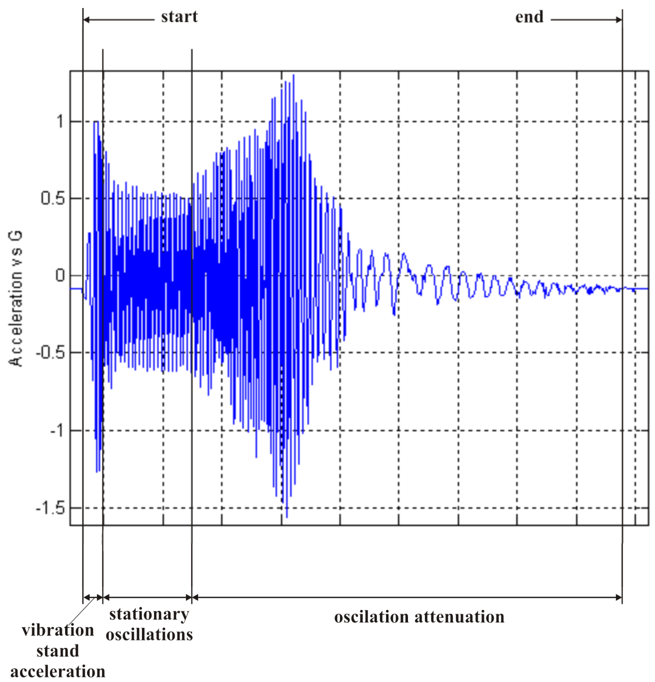


Fig.4. Vibration stages in the time domain

Vibration process can be analyzed with time-frequency analysis. The most popular analysis are:

a) Short Fast Fourier Transform (SFFT) [12,13]
b) Wigner-Villes Transform (WVD) [13]
c) Wavelet Transform [14]
d) Linear decimation [15]
e) Frequency Response Functions (FRF) [16]
f) Power spectrum density (PSD) [17]
g) Auto-Regressive Model [18].

The frequency analysis is realized on the basis of a Short Fast Fourier Transform. The window has a rectangular shape with different size (from N=32 to N=128 samples) and 16 overlapped samples. The time – frequency distribution is shown at Figure 4.

The results show that if the window size is very small (up to 32 points – first picture, Figure 5) the frequency resolution is too small and the both spectrum peaks (resonance frequency of the left and right damper) appear as one. If the frequency resolution is increased too much i.e. the window size is increased from 64 to 128 points (second and third picture, Figure 5) than the suspension resonance peaks are too small to be separated and remain only the spectrum peak at the stand shaking frequency (16Hz). Therefore, the optimal window size is set to 64 points and the both resonance frequencies of the left and right damper are clearly visible (Figure 6) and attenuation lines are well defined (Figure 7).

The line attenuation slope at Figure 7 may be calculated according to the equation:

$$k = \frac{\Delta w}{\Delta f}, \qquad (2)$$

where $\Delta w = w_2 - w_1$ - the difference between the window numbers of two neighbor peaks;

$\Delta f = f_2 - f_1$ - the frequency difference between the neighbor peaks

The window sliding ($\Delta w = 1$) is equivalent to the time shifting $\Delta T = OV.T = OV.N.\Delta t$, where OV – window overlapping coefficient. Therefore the window difference is equal to:

$$\Delta w = w_2 - w_1 = \frac{t_2 - t_1}{OV.N\Delta t},$$

and the line slope may be expressed as:

$$k = \frac{(t_2 - t_1)F_d}{OV.N\Delta f}. \qquad (3)$$

The frequency analysis not only defines the main dominant frequencies and the attenuation line slope, but it is capable to specify these parameters for the both dampers and springs according to Figure 7. The analysis shows that the both dampers have the same characteristics (otherwise the two line slopes will distinguished with different line slopes) while the left and right suspension elements have slightly different resonant frequencies.

The data analysis shows the following conclusions:

1. The calculated attenuation coefficient is equal to 0.33Hz/s according to the equation (3) and the data represented at Figure 7;
2. The attenuation defines two main frequencies which difference remains constant all the time and it's equal to $\Delta f=2f_d/N=1.25$Hz;
3. The maximum amplitude of the spectrum peaks appears at the frequency f_0=8Hz;
4. The spectrum peaks appear in the frequency range from 6 to 10Hz.
5. The frequency range of the oscillations is equal to 16Hz (from 2Hz to 18Hz).

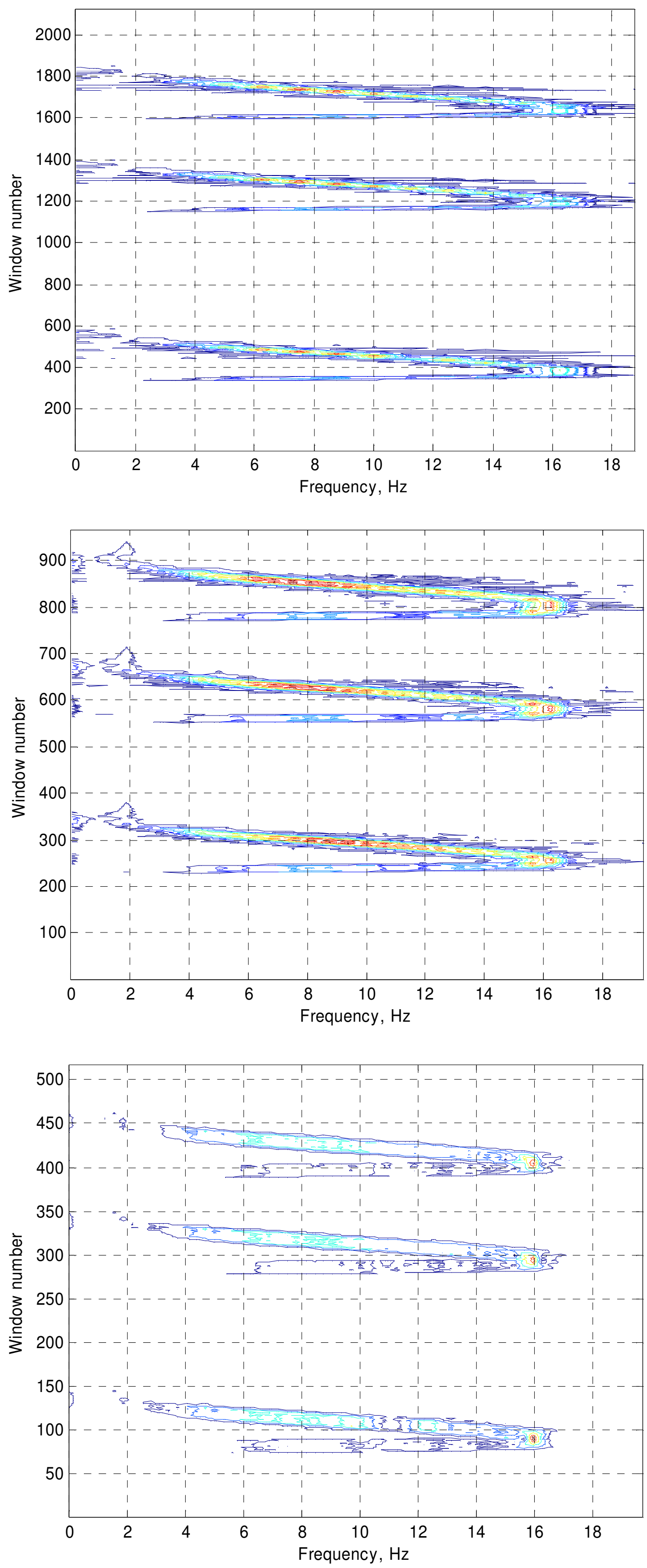


Fig. 5. Time-frequency distribution of the oscillations using different window size N=32, 64 and 128 respenctively

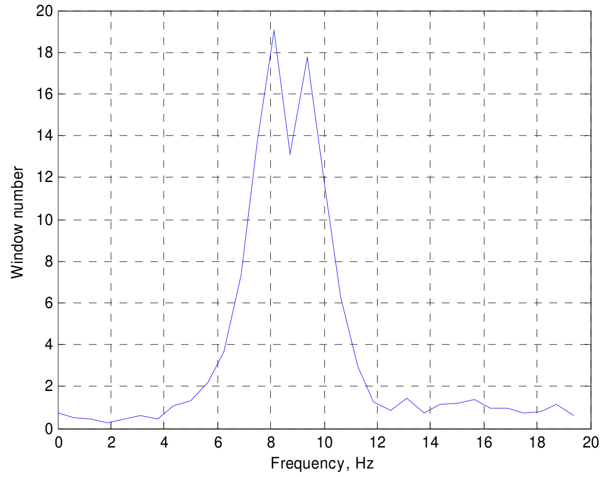


Fig. 6. Resonance frequencies of left and right dampers

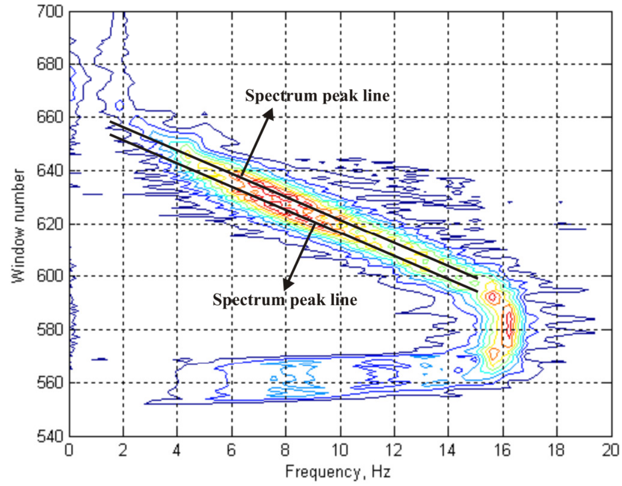


Fig. 7. Definition of the spectrum peak lines

The speed and displacement of the tested dampers are calculated according to the double integration of the acceleration as the speed and the acceleration may be presented as first and second derivatives of the displacement respectively:

$$\begin{aligned} v &= \dot{d} \\ a &= \ddot{d} \end{aligned}. \tag{4}$$

The displacement could be calculated according to equation for the continuous signal:

$$d(t) = d_0 + v_0 t + \int_0^t dt \int_0^\tau a(\tau) d\tau , \tag{5}$$

where:

d_0 - initial displacement

v_0 - initial speed

$d(t)$ - current displacement.

As the used accelerometer has the digital output, the speed could be calculated by numerical integration of the digital acceleration data:

$$v(i)=\int_0^t a(\tau)d\tau \cong \sum_0^n \left(\frac{a(i)-a(i-1)}{2}\right)\Delta t \ , \tag{6}$$

where:
n – number of points;
$\Delta t=\frac{1}{F_d}$ - time interval between (i) and (i-1) sample;
F_d - sampling frequency.

Therefore, the speed and the displacement could be established by numerical integration of the inertial data using trapezoidal rule:

$$\begin{aligned} v_c(i)&=v_c(i-1)+\frac{a(i)-a(i-1)}{2}\Delta t \\ d_c(i)&=d_c(i-1)+\frac{v_c(i)-v_c(i-1)}{2}\Delta t \end{aligned} \tag{7}$$

Before start the measurement process and the numerical integration procedure it is necessary to obtain the sampling frequency to satisfy the Nyquist. It is well known that a harmonic vibration of frequency ω and displacement amplitude d_{max}, results in a maximum acceleration a_{max} calculated as follows [19]:

$$a_{\max}=d_{\max}.\omega^2 \tag{8}$$

Therefore, if the expected maximum acceleration is equal to ±1.5g and the displacement amplitude is equal to 20mm, the harmonic vibration frequency is equal to:

$$f_{\max}=\frac{\sqrt{\frac{a_{\max}}{d_{\max}}}}{2\pi}=6.1Hz \tag{9}$$

The sampling frequency has to be equal to $F_d \geq 2f_{max}$. The inertial MEMS sensor reads the data with a minimum sampling frequency of 40Hz, therefore the Nyquist criteria is satisfied.

The low frequency components of the inertial data are source of the integration errors while the suspension speed and displacement are calculated. According to the data frequency range, the passband filter had to be designed to reject the bias offset and low frequency components. The following filter properties are defined according to spectrum response shown at Figure 7:

- filter passband - 1÷19Hz
- stopband cut-off frequencies – f_1=0.5Hz; f_2=19.5Hz
- maximum passband ripple – 0.1dB
- Minimum stopband attenuation – 40dB
- Approximation – IIR, Butterworth

On this bases MATLAB buttord() function generates 21-th order IIR digital filter. The amplitude and phase response of the constructed filter are shown at Figure 8.

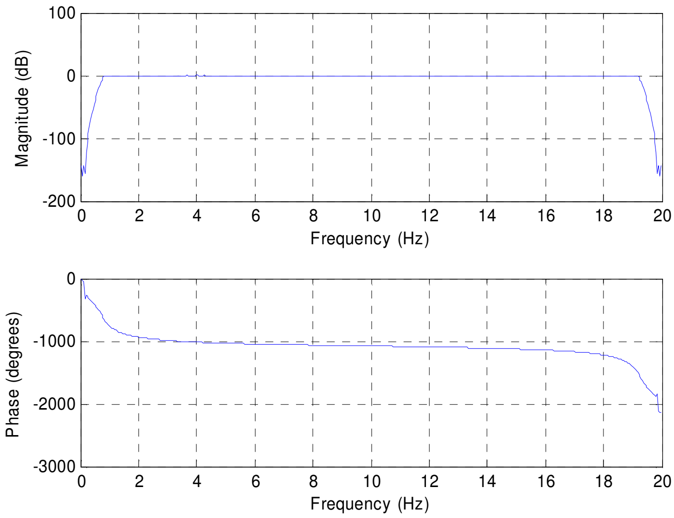


Fig.8. Spectrum response of the band-pass filter

The calculated filter coefficients are given at Table 1.

Table 1. Filter coefficients

n	a	b
1	1	0,451
2	0	0
3	-8,414	-4,512
4	0	0
5	31,965	20,302
6	0	0
7	-72,198	-54,139
8	0	0
9	107,344	94,743
10	0	0
11	-109,760	-113,692
12	0	0
13	78,156	94,743
14	0	0
15	-38,263	-54,139
16	0	0
17	12,325	20,302
18	0	0
19	-2,358	-4,512
20	0	0
21	0,204	0,451

The acceleration, calculated speed and displacement according equation (7) after band-pass filter are shown at Figure 9 respectively. It is well seen that the

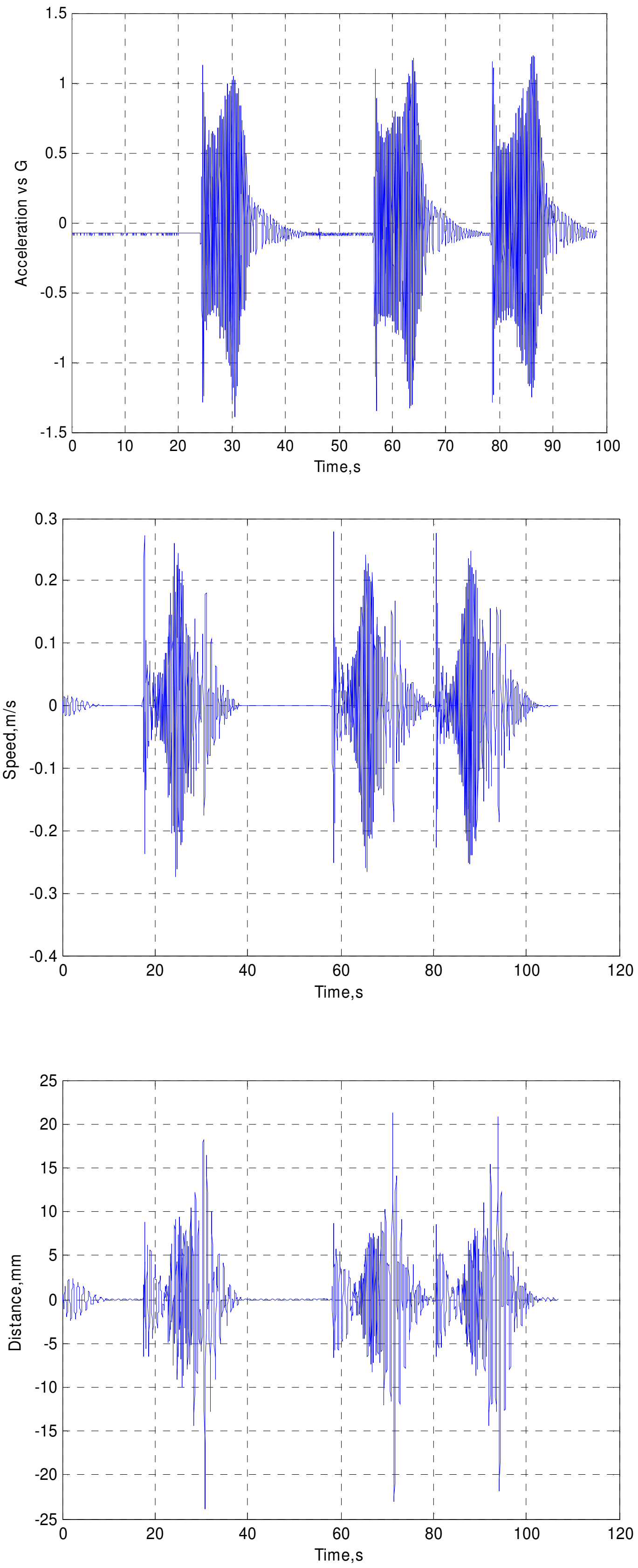


Fig.9. Measured stand acceleration and calculated speed and displacement after band-pass filter respectively

The last investigation of the suspension dynamics studied the dependence of the determined frequencies of the suspension elements status. The control parameter is defined as the clearance between the lower control arm of the suspension and the lower ball joint of the "candle". The distance between the selected suspension parts is changed from 0.0mm to 1.2mm with 0.4mm step.

During the experiments every vibration attempt is also accomplished three times and the numerical results are shown at Table 2 and the 2D STFT results – at Figure 11.

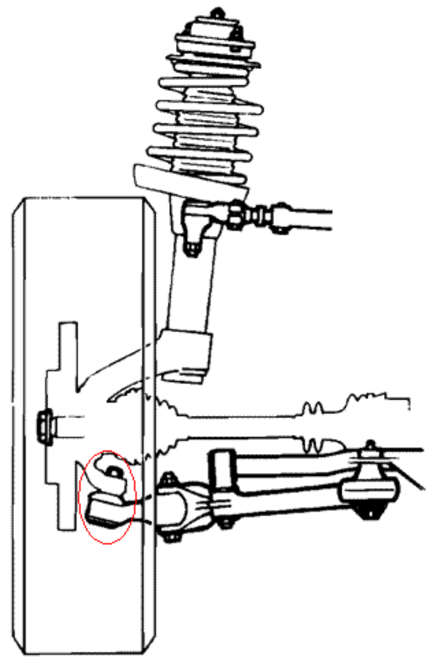
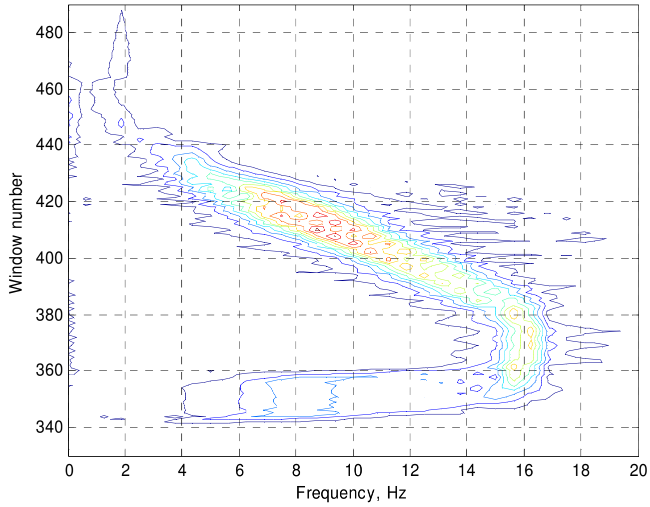


Fig.10. Studied clearance between suspension parts

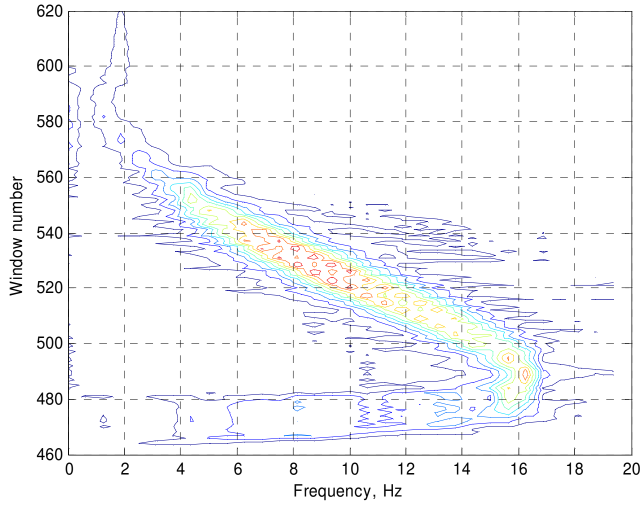
Table 2. Spectrum magnitude dependence

Clearance x,mm	Maximum amplitude of the spectrum peaks		
	Test 1	Test 2	Test 3
0.0	20.0	21.0	21.0
0.4	19.5	19.0	18.0
0.8	20.0	20.0	19.5
1.2	20.2	20.0	19.1

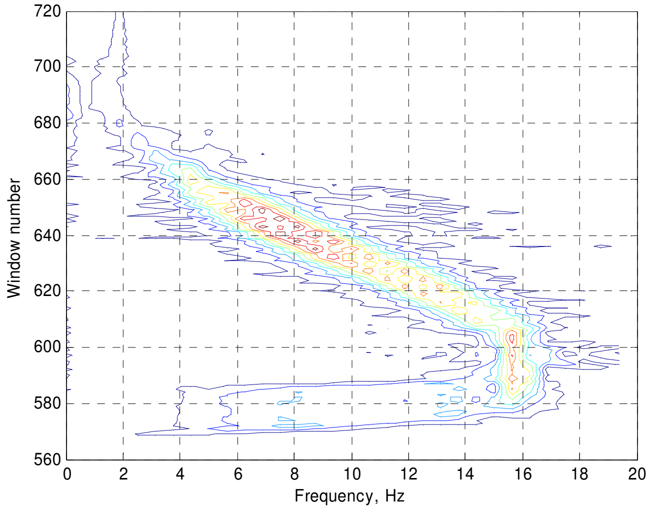
The results show that the ball joint clearance affects the spectrum response slightly and the variation of the spectrum peaks magnitude is obliged to variation in the vibration stand excitation process. This circumstance allows testing and diagnostic of the suspension status regardless of the status of the suspension elements. The suspension elements status could be obtained only by installation of the individual inertial sensors on the studied elements as is shown at our previous paper [4].



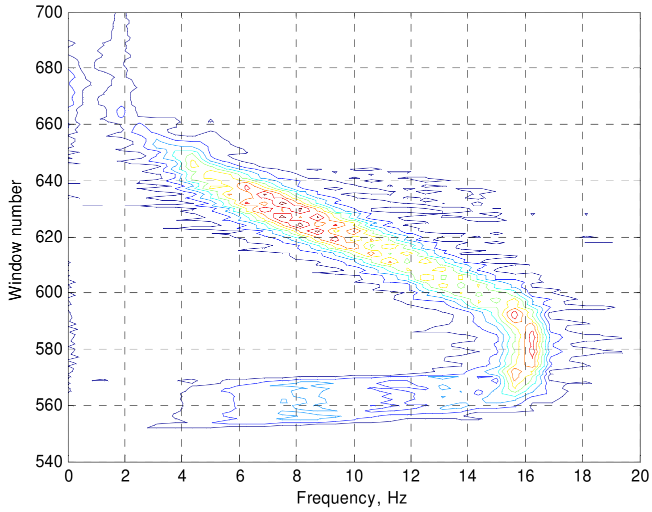
x=0.0mm



x=0.4mm



x=0.8mm



x=1.2mm

Fig.11. Spectrum response of the stand vibration depending on the clearance between the lower control arm of the suspension and the lower ball joint of the "candle"

The future work will discuss other spectrum analysis methods such as wavelet functions to obtain the dynamic characteristics because Fourier transform of a signal does not contain any local information which is the major drawback of the Fourier transform. The choice of the window size is also a critical problem in the Fourier transform because the time and frequency resolutions are tied and the better time resolution conducts to the poor frequency resolution and vice versa.

IV. Conclusion

The paper discusses the measurement and data analysis of the suspension vibrations based on micromechanical inertial sensors (MEMS) to measure the dynamic response and status of the vehicle suspension elements.

The inertial sensor system allows real-time calculation of:

- Total attenuation time;
- Resonant frequency according to the maximum amplitude of the spectrum peaks;
- The total frequency range of the oscillations;
- The calculated attenuation coefficient which represents the system dynamic response;
- Dependence of the determined frequencies of the suspension elements status;
- The suspension speed and displacement of the wheel according of the numerical integration of the inertial data.

In the same time the proposed system is less expensive and more accurate due to the low cost and high sensitive MEMS inertial sensor (1mg resolution). The combined time and frequency domain analysis allows fast and user friendly calculation of the suspension dynamic response, detection of suspension problems and prevention of the suspension element failures.

ACKNOWLEDGMENT

This paper was prepared and supported by the National Fund under contract number No.DTK02/2-2009.

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