### 7. The influence of the technical condition of suspension elements and vehicle operating parameters on the dominant frequency components of the vibration

The evaluation of the technical condition of suspension elements and operating parameters of the motor vehicle based on time function of vibration signals or global estimators can be considered as preliminary overall assessment. The results presented in previous chapter confirm relations in founded case study and distribution of vibration in vehicle. The interpretation and conclusion based only on those results can be difficult. It enables comparison of total energy of vibration propagation in vehicle construction.

The human vibration perception has to be analysed in multi-criteria terms. It relies on amplitude and dynamics exposure to vibration. The evaluation of influence of technical condition of suspension elements and vehicle operating parameters on the vibration in terms of safety and comfort, should include analysis of the changes in dynamics of the phenomena. Thus the vibration signals have to be transformed to the frequency domain, because it enables analysis of dynamics. The most useful for that purpose is the Fourier transform.

#### 7.1. Fourier transform

A signal is represented in the domain of frequency by application of the discrete Fourier transform. In the sphere of signal processing, it is mainly used to transform the f(t) function, being continuous in the domain of time, into the  $F(\omega)$  function, continuous in the domain of frequency. The discrete Fourier transform is based on an assumption that every signal may be obtained by adding sinusoid properties with appropriate phases and amplitudes. Therefore, a result of the discrete Fourier transform may be interpreted as a set of properties of the signal being examined in the function of frequency of component sinusoids. The Fourier transform based on Fourier series as periodic functions written as the sum of simple waves mathematically represented by sines and cosines. Due to the properties of sine and cosine, it is possible to recover the amplitude of each wave in a Fourier series using an integral. Thus the Fourier series, with period T, is an infinite sum of sinusoidal functions, each with a frequency that is an integer multiple of 1/T. The Fourier series also includes a constant, and hence can be written as:

$$g(t) = a_0 + \sum_{m=1}^{\infty} a_m \cos\left(\frac{2\pi mt}{T}\right) + \sum_{n=1}^{\infty} b_n \sin\left(\frac{2\pi nt}{T}\right),$$
(7.1)

where: t - time, T - period of fundamental function,  $a_0 - \text{constant}$  value,  $a_m$ ,  $b_n - \text{constants}$  coefficients of the Fourier series.

In many cases it is desirable to use complex exponential function (Euler's formula) to write Fourier series in terms of the basic waves  $e^{-i2\pi\omega t}$ , where Euler's equation can be derived by expanding the left side in a Taylor series (with variable theta or omega). Then expand the right side using the Taylor series expansions for cosine and sine and the results are identical:

 $e^{it} = \cos t + i\sin t. \tag{7.2}$ 

Thus the Fourier transform can be express as formula:

$$F(\omega) = \int_{-\infty}^{\infty} f(t) \cdot e^{-i2\pi\omega t} dt.$$
(7.3)

Fig. 7.1 illustrates identification of Fourier series from fundamental function and transformation from time domain to frequency domain.



Fig. 7.1. Illustration of Fourier Transformation from time-domain function to frequency domain function

The continuous Fourier transform converts a time-domain function of infinite duration into a continuous spectrum composed of an infinite number of sinusoids. In engineering we deal with signals that are discretely sampled and of finite duration or periodic. For such data, only a finite number of sinusoids is needed and the Discrete Fourier Transform (DFT) is appropriate. The DFT of N sampled data points  $f_i$  are defined by:

$$X_k = \sum_{j=0}^{N-1} f_j \cdot e^{-i2\pi jk/N}.$$
(7.4)

The result of the DFT of an N-point input time series is an N-point frequency spectrum, with Fourier frequencies k ranging from (N/2 - 1), through the 0-frequency and up to the highest Fourier frequency N/2. The Discrete Fourier Transforms can be implemented with the Fast Fourier Transform (FFT) algorithm. FFTs are most efficient if the number of samples, N is a power of 2. The advantage of the FFT over the DFT is that the operational complexity decreases from  $O(N^2)$  for a DFT to  $O(N\log_2(N))$  for the FFT, so the speed of the algorithm is much higher.

It is important for a DFT or FFT to represent a function accurately, the original function must be sampled at a sufficiently high rate. The appropriate rate for a uniformly sampled time series is determined by the Nyquist-Shannon Theorem or Sampling Theorem. This theorem states that any continuous baseband signal may be identically reconstructed if the signal is bandwidth limited and the sampling frequency is at least twice the bandwidth of the signal. That critical sampling rate is known as the Nyquist rate, and it is a property of the time-domain signal based on its frequency content.

The chapter address results of analysis of influence of technical condition of suspension elements and vehicle operating parameters on the dominant frequency components of the vibration. For the purpose of identification of dominant frequency components of vibration dynamics it is sufficient to apply Fast Fourier Transformation (FFT).

In order to analyse frequency components of vibration signals tests on vehicles excited to vibrate by means of an inductor controlled with a frequency converter were conducted. It enabled the system to be excited to vibrate within the chosen band of stabilised frequencies and, once the testing station had been switched off, made it possible to analyse the natural vibrations being damped. It is calculated for a car moving at a most popular speed about of 80 km/h, secondary roads, assuming an average wavelength of existing roads, the most energy of the vibration signal is contained in the band up to 20-30 Hz. The active research experiments were conducted on real passenger car. The vehicle was excited to vibration by special kinematic excitation machine. The range of the frequency of the forced was set as dynamic linear increase up to ca. 21 Hz, excitation with constant frequency (ca. 21 Hz) for 5 seconds and excitation frequency decrease down to 0 Hz for 30 second period. This set up allows analysis bands of sprung and unsprung masses resonances.

# 7.2. Dynamics of vibration in vehicle structure for different technical conditions of suspension elements and operating parameters of the car, identification of dominant frequency components

For the purpose of analysis of the dynamics phenomena the transformation of the signal to the frequency domain has to be conducted. In the course of the results analysis the spectrums of the vibration were determined. The comparison of the vibration dynamics registered during research of the vehicle with build in shock absorbers with 100 % and 50 % of liquid volume and under research with different pressure level in the tires are presented in Figs. 7.2-7.5.

From the most general perspective of vibration phenomena that one may consider, what matters most is the natural vibration frequency bands for both sprung and unsprung masses, arranged in a vertical direction. For the proper identification of the dominant frequency component, which can determine dynamics of the vibration, analysis of the chosen frequency bands was conducted. Some of the exemplary results of the analysis are shown in Figs. 7.6-7.9.



Fig. 7.2. Comparison in the vibration spectrums of exciter and suspension for the vehicles with build in shock absorbers with 100 % (blue) and 50 % (green) of liquid volume















Fig. 7.6. Comparison of the spectrums of vibration under driver feet for the vehicles (2-18 Hz frequency band – resonances of sprung and unsprung masses)











**Fig. 7.8.** Comparison of the spectrums of vibration under driver feet for the vehicles (105-110 Hz frequency band – frequency of 5th harmonics of constant period of excitation)



**Fig. 7.9.** Comparison of the spectrums of vibration under driver feet for the vehicles (148-155 Hz frequency band – frequency of 7th harmonics of constant period of excitation)

#### 7.3. Frequency based estimators

Frequency distribution of the signal enables analysis of the predominant dynamic components in the system. It also makes it possible to seek resonant frequencies and input frequencies. The detailed frequency analysis conducted consisted in a preliminary selection of characteristic frequency bands (with predominant values of the signal spectrum amplitudes) for individual sensor mounting points. The following frequency bands were selected for the analysis of the vehicle floor pan vibrations:

- from 2 [Hz] to 4.5 [Hz]
- from 5 [Hz] to 7 [Hz]
- from 12 [Hz] to 17 [Hz]
- from 21 [Hz] to 22 [Hz]
- from 33 [Hz] to 38 [Hz]
- from 62 [Hz] to 66 [Hz]
- from 85 [Hz] to 88 [Hz]
- from 105 [Hz] to 110 [Hz]
- from 150 [Hz] to 155 [Hz]
- from 190 [Hz] to 196 [Hz]

Fig. 7.10 illustrates the signal frequency bands assumed for the analysis of the vehicle floor panel vibrations.

The next stage of the analysis involved developing a collation of maximum values of the signal spectrum amplitudes in the chosen frequency bands.

#### 7. THE INFLUENCE OF THE TECHNICAL CONDITION OF SUSPENSION ELEMENTS AND VEHICLE OPERATING PARAMETERS



rig. 7.10. Danus analyses for the venicle noor paner vibration signals recorded

## **7.3.1.** Influence of the damping of shock absorber on dominant frequency components of vibration in vehicle structure on the floor pan

The results of analysis of vibration for the chosen shock absorber technical condition parameters examined were collected in the database form. Figs. 7.11-7.12 graphically present chosen analysis results obtained.





(shock absorber filled with working medium in 100 %)

### **7.3.2.** Influence of the suspension spring properties on dominant frequency components of vibration in vehicle structure on the floor pan

The results of analysis with build in new and used (worn-out) suspension spring vehicle's vibration were collected in the database form. The examined cases represent spring properties presented in Fig. 6.5, as the force vs. deflection characteristics of coil springs. Figs. 7.13-7.14 present chosen analysis results obtained.



Fig. 7.13. Distribution of maximum amplitudes of vibration signal spectra obtained for a floor panel of passenger car for the selected characteristic frequencies (the vehicles with build in new suspension spring)





### **7.3.3.** Influence of the pressure in tires on dominant frequency components of vibration in vehicle structure on the floor pan

The analysis results of operation parameter's influence, as tire pressure, on vehicle vibration were collected in the database form. The examined cases represent tire pressure changes from 600 to 2600 hPa, with 200 hPa steps. Fig. 7.15 present chosen analysis results obtained.





Fig. 7.15. Distribution of dominant frequencies of vibration of driver's feet resting spot for the pressure in tires

### **7.3.4.** Influence of the load of the passenger car on dominant frequency components of vibration in vehicle structure on the floor pan

The influence of operation parameter, as extra load, on dynamics of vehicle vibration were collected in the database. The examined cases represent extra loads of 75, 150, 225, 300 kg, which can be correlated with 1-4 persons sitting in a car. Figs. 7.16-7.17 present chosen analysis results obtained.









#### 7.4. Result analysis and discussion

This chapter addresses analyses results of frequency based vibration measures recorded on a vehicle floor panel and suspension elements in a function of technical condition of the suspension elements installed in automotive vehicles and its operation parameters. For the identification of the dominant frequency component, which can determined dynamics of the vibration, analysis of the chosen frequency bands have been conducted. The distribution of vibration in frequency bands: 2-18 Hz, 20-25 Hz, 105-110 Hz and 148-155 Hz show high sensitivity to changes of damping and stiffness (volume of liquid in shock absorber and tire pressure). The vibration are much higher for the vehicle with shock absorber filled with working medium in 50 % and it increase due to level of pressure in tire.

The signal frequency bands assumed for the analysis of the vehicle floor panel vibrations enable involved developing a collation of maximum values of the signal spectrum amplitudes in the chosen frequency bands.

The highest sensitivity to vibrations was determined to be characteristic of a symptom of the shock absorber's technical condition, namely the maximum value of the vibration amplitude within the band of 12-17 Hz. It corresponds to a band of resonant frequency of unsprung masses. In the results of passenger car tests discussed in the chapter, this frequency equals ca. 12.5 Hz. A drop in the amount of shock absorbing fluid causes a decrease of the vibration damping efficiency and an amplitude increase in the resonant band of unsprung masses. It is a very hazardous phenomenon, since it has a direct effect on the vehicle safety, and in extreme cases, it may even lead to a loss of road adhesion of a wheel. An interesting phenomenon was observed for a frequency of ca. 65 Hz which may be perceived as the 5th harmonic of resonance of unsprung masses. In this special case, one could observe a reverse dependence assuming the form of a decline in the value of maximum amplitudes as the amount of shock absorbing fluid decreased. Additionally, basing on the comparative analysis of the vibration distribution in the front and back of the floor panel, it was determined that the predominant frequency components, carrying the largest amounts of vibration energy, in locations where vibrations penetrated bodies of the sitting passengers were to be found within the band of 62-66 Hz when testing shock absorbers filled with the working medium in 100 %. One may consequently assume that correct technical condition of a shock absorber, damping efficiency aside, exerts an impact on shifting the dynamics of the system vibrations outside the band of hazardous frequencies, i.e. the resonance of unsprung masses.

The influence of the suspension spring properties on dominant frequency components of vibration in vehicle structure on the floor pan shows less sensitivity. The values of frequency based estimators are smaller for dynamics of vibration of vehicle with build in used (worn-out) suspension spring. The slight increase of the values can be observed in frequency components, as ca. 12 Hz and decrease in estimator of 21.5 Hz (correspond to the constant excitation).

The influence of operation parameters of the vehicle on the vibration dynamics were tested on two cases, tire pressure and extra load. The changes of the vibration dynamics due to the tire pressure level are observed for ca. 12 and 21.5 Hz frequency based estimators. This result are similar to influence of suspension spring properties, so it can be assumed that most sensitivity of suspension stiffness to vibrations dynamics is frequency component corresponded to the constant excitation (i.e. driving with constant speed on similar roughness road type). The distribution of maximum amplitudes of vibration signal spectra for the selected characteristic frequencies on different extra load show some influence. It is difficult to describe any clear conclusion for this case but one can be assumed that dominant frequency component is correlated to the constant excitation frequency.

Basing on empirical studies, resonant phenomena at higher frequencies, even exceeding 5 [Hz], have been identified, namely those which may cause considerable discomfort. In terms of unsprung masses, free vibration frequencies assume values within a range from several to more than a dozen hertz (i.e. 8-18 [Hz]). While an automotive vehicle is moving, free vibrations of sprung and unsprung masses occur simultaneously and overlap.