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IMPLEMENTATION OF MULTIDIMENSIONAL IDENTIFICATION OF SIGNAL CHARACTERISTICS IN THE ANALYSIS OF VIBRATION PROPERTIES OF AN AUTOMOTIVE VEHICLE'S FLOOR PANEL

IMPLEMENTACJA WIELOWYMIAROWEJ IDENTYFIKACJI CHARAKTERYSTYCZNYCH CECH SYGNAŁU W ANALIZIE WŁASNOŚCI DRGANIOWYCH PANELU PODŁOGOWEGO POJAZDU SAMOCHODOWEGO*

The article provides a proposal of software application of a method and an algorithm developed to identify signal characteristics in the analysis of vibration properties of an automotive vehicle's floor panel. Due to the complexity resulting from nonlinear and random nature of vibration phenomena in automotive vehicles, the analysis in question is multidimensional. The property table being established consists of numerous measures and estimators, both dimensional and dimensionless ones, in the domains of amplitudes, time, frequency and time-frequency. The foregoing enables observation and separation of signal components in multiple domains, but it also makes it possible to define signal measures depending on stationary and non-stationary characteristics as well as accurate time positioning of resonant frequencies. Multicriterial approach to identification of vibration enables determining the table of vibration properties measures of floor panel. The table is numerical form of characteristics properties of the vibration signal.

Keywords: *vibration signal processing, wavelet transform, FFT.*

W artykule przedstawiono programową aplikację opracowanej metody i algorytmu matematycznego identyfikacji charakterystycznych cech sygnału w analizie własności drganiowych panelu podłogowego pojazdu samochodowego. Z uwagi na złożoność, wynikającą z nieliniowości i losowości, zjawisk drganiowych w pojazdach samochodowych analiza ma charakter wielowymiarowy. Wyznaczana tabela właściwości składa się z wielu miar i estymatorów wymiarowych i bezwymiarowych w dziedzinach amplitud, czasu, częstotliwości i czasowo-częstotliwości. Pozwala to na obserwację i separację składowych sygnału w wielu dziedzinach. Umożliwia definiowanie miar sygnału w zależności od cech stacjonarności i niestacjonarności oraz precyzyjną lokalizację czasową częstotliwości rezonansowych. Wielokryterialne podejście do identyfikacji drgań umożliwia wyznaczenie zbioru właściwości drganiowych panelu podłogowego, który jest numerycznym odzwierciedleniem charakterystycznych cech sygnału drgań.

Słowa kluczowe*: analiza sygnałów drganiowych, transformata falkowa, FFT*

1. Introduction

The vehicle vibration are results from many kind of dynamic interactions. The proper identification of the vibration is very difficult research and scientific problem. It requires good knowledge fundament and correct measurement tools and signal processing. An automotive vehicle, being a complex mechanical system, includes a set of specific free vibrations frequencies depending on the direction of the oscillatory wave propagation. From the most general perspective of vibration phenomena that one may consider, what matters most is the free vibration frequency bands for both sprung and unsprung masses, arranged in a vertical direction. Various publications mention different ranges for these resonant bands. The free vibration frequency of an automotive vehicle's sprung masses is assumed to be contained within the range from 1 to 2.5 [Hz]. Such dynamics of vibration phenomena does not essentially exert any negative effects on passengers, since it corresponds to man's natural frequency of making steps. Vibrations of the frequency below 1 [Hz] cause effects similar to seasickness in people, whereas those of the frequency exceeding 2.5 [Hz] bring prompt weariness and pain. The first resonant frequency for a man in a sitting position comes to ca. 4–6 [Hz] depending on individual body build features [14]. Input functions with the frequency of 3–4 [Hz] trigger strong vibrations in the abdominal cavity organs. The amplitude maximisation of the effects caused by these vibrations occurs at the frequency of 5–8 [Hz]. Close to these frequencies are those causing resonance in a human chest (i.e. 7–8 [Hz]). Organs of the head resonate in the band of 20–30 [Hz], whereas eyeballs at 60–90 [Hz]. However, it is the nervous as well as the cardiovascular system that are the most sensitive to the whole organism vibrations. The responses of these systems and their respective organs manifest themselves in their functions being disturbed, in poor physical and mental state, and even in certain forms of damage on higher amplitudes of effects and long exposure times. Some interesting investigation on influence of chosen driving parameters on vibration comfort according to Human-Vehicle-Road (HVR) model and vibration exposure metric described in the ISO 2631 have been presented in [18]. In a wide variety of transport environments the vibration transmitted through seats is associated with discomfort [14]. Seats can either reduce vibration discomfort or increase vibration discomfort [29]. The paper [29] presents results of the study on determine how factors, as age, gender, physical characteristics, backrest contact, and magnitude of vibration affect seat transmissibility. The paper presents analysis of the vibration registered on vehicle floor panel in location when it penetrate to the human organism via feet. Based on empirical studies, resonant

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(*) Tekst artykułu w polskiej wersji językowej dostępny w elektronicznym wydaniu kwartalnika na stronie ***www.ein.org.pl*** 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. Designers of automotive vehicles in mass production strive to limit the vibrations of sprung masses, trying to maintain sufficient rigidity of the suspension system at the same time, so that suitable steerability is ensured [1, 3, 4, 6, 7, 11, 16, 19, 22, 30]. Consequently, material properties and metallurgical technologies applied in the automotive industry are gradually growing in importance [2, 12, 13, 17, 23, 31] with the many analysis on influence of some parameters on physical and chemical properties [9, 10, 15].

As the results of observing and acquisition of vibration phenomena are received signals of displacement, velocity or acceleration of vibration. A vibration signal is a carrier of information on the state, the changes or the process to which the given physical or technical system is subject [24, 33]. Vibroacoustic signals are characterised by the largest information carrying capacity and they enable observation of changes occurring in a broad frequency band.

Numerous measuring problems may be considered on a general level of a signal, perceiving the signal as an entirety in the course of observation. They may be examined in the domains of amplitudes, time and frequency [8]. As far as random vibration phenomena are concerned, the signals recorded are of non-stationary nature which requires that the signal distribution is observed in the domains of time and frequency simultaneously. However there are some methods, for example as conjugate-pair decomposition (CPD) for signal decomposition, dynamics characterization, and nonlinearity identification in the time domain only [26]. The paper [25] presents novel time–frequency signal processing methodology based on Hilbert–Huang transform (HHT) and a new conjugate-pair decomposition (CPD) dedicated for characterization of nonlinear normal modes and parametric identification of nonlinear multiple-degree-of-freedom dynamical systems.

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 *y(t)* function, being continuous in the domain of time, into the *Y(f)* 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 [20]. The fast Fourier transform (FFT) is more frequently applied in practice, since it is a computational algorithm of the discrete Fourier transform as well as of an inverse transform, making use of the sine function symmetry.

In the field of technical diagnostics, time implementations of physical quantities may be perceived as a sum of two components: the determined and the random one. It is assumed that the determined component carries information on the wear of the given device being examined, whereas the random one is a measure of noises and interferences. The only data relevant from the technical diagnostics' perspective are those contained in the determined component, and the data must necessarily be separated [21, 27, 28, 30, 32]. One of the mathematical instruments enabling separation of non-stationary signal components is a wavelet transformation which consists in distinguishing a part of the *f(t)* signal being similar to a present template, i.e. the part which corresponds to the determined component. The template role is performed by basic wavelet $\psi(t)$. The wavelet functions as a transformation kernel. A single wavelet is used in the given transformation, however, due to modification of scale coefficient a and modification coefficient b, it forms what is referred to as a *wavelet family*. A continuous wavelet transform in the domain of time and frequency is defined as follows:

$$\tilde{s}_{\Psi}(a,b)=\frac{1}{\sqrt{|a|}}\int_{-\infty}^{+\infty}s(t)\Psi\left(\frac{t-b}{a}\right)dt \tag{1}$$

gdzie:

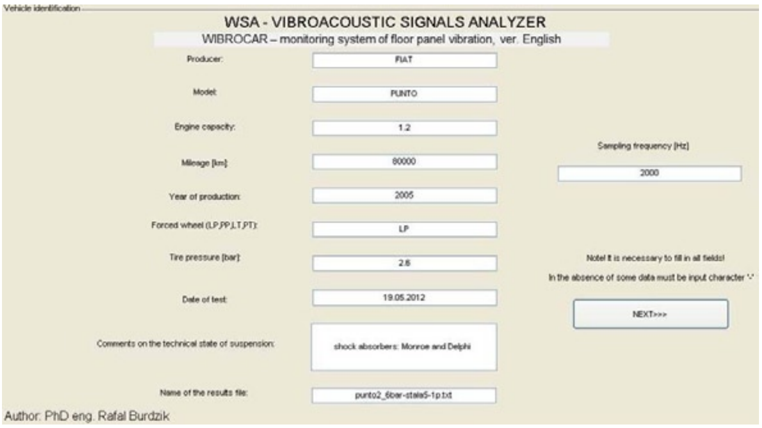
a – scale coefficient,
b – modification coefficient,
s(t) – value of the signal examined in the function of time,
$\tilde{s}_{\Psi}(a,b)$ – wavelet coefficient dependent on a and b,
ψ – wavelet function,
$\Psi\left((t-b)/a\right)$ – transformation kernel.

The value of wavelet coefficient $\tilde{s}_{\Psi}(a,b)$ established by means of the above formula is generally understood as a measure of similarity between the signal examined and the chosen wavelet [20].

Furthermore, due to dimensional estimates' sensitivity to the stationary nature of operating conditions, in the process of identification of signal characteristics, besides dimensional estimates, one applies quotients of these measures being dimensionless amplitude discriminants. They are obtained by dividing moments of various ranks by one another.

3. Method of multidimensional identification of vibration signal characteristics of an automotive vehicle's floor panel – WSA WIBROCAR

For the sake of identification of signal characteristics in the analysis of vibration properties of an automotive vehicle's floor panel, a complex mathematical algorithm was developed to be subsequently implemented in the MatLab environment, and a user interface was created named WIBROCAR. The programme developed was given the name of WSA, and it was then extended with several modules dedicated to analysis, monitoring and diagnostics of selected vehicle systems and structural assemblies. Procedure of testing is starting by vehicle data and research parameters entry (Fig. 1).



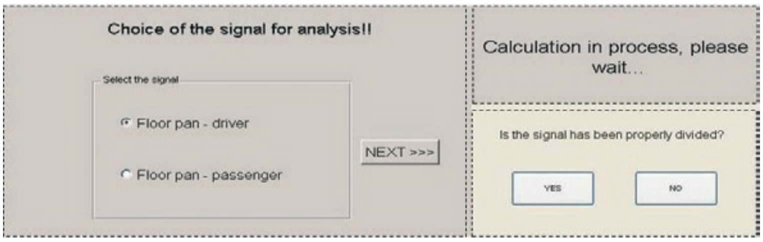
Producer: FIAT
Model: PUNTO
Engine capacity: 1.2
Mileage [km]: 80000
Year of production: 2005
Forced wheel (LP,PP,LT,PT): LP
Tire pressure [bar]: 2.6
Date of test: 19.05.2012
Comments on the technical state of suspension: shock absorbers: Monroe and Delphi
Name of the results file: punto2_6bar-stala5-1p.txt
Sampling frequency [Hz]: 2000

Note! It is necessary to fill in all fields!
In the absence of some data must be input character '-'

Fig. 1. First window of WSA program

The implementation of the WSA program was assumed the utilitarian of the software. For this purpose it is very important to communicate to the user with clear orders and information reports. The work in the WSA should be close to intuitive. Some examples of the communication windows have been depicted in Figure 2.

Due to the complexity resulting from nonlinear and random nature of vibration phenomena in automotive vehicles, the analysis in ques-


Choice of the signal for analysis!!

Select the signal

● Floor pan - driver

○ Floor pan - passenger

Calculation in process, please wait...

Is the signal has been properly divided?

Fig. 2. User – WSA program communication windows

tion is multidimensional. The property table being established consists of numerous measures and estimators, both dimensional and dimensionless ones, in the domains of amplitudes, time, frequency and time-frequency. In order to accurately identify signal characteristics, one needs appropriate analytical methods depending on the stationary and non-stationary nature of the signal. An automatic algorithm was developed for positioning of stationary and non-stationary signal cycles. For this purpose identification of next cycles of forced machine working there were next phases of vibration inductor working identification measure formulated. The markers of next cycles of forced machine working measures based on STFT (Short Time Fourier Transform) transformation were used. The main reason of choosing this transformation was short realization time. There was 21-22 Hz frequency band cut out from STFT spectrum for analysis. Based on time function of cut off frequency band identifying algorithm of end of stand run up and start of stand coasting time coordinates was created. Elaborated algorithm is based on comparing next value of analysed frequency band (“analysis of edge”) around set parameters. Locating of end of stand run up and start of stand coasting enables to divide signal on three time windows. First window for fragment of signal growing according to constant frequency increase of the forced system. Second window for signal with constant frequency and the third one for coasting stand – decrease of signal amplitudes according to constant frequency decrease of the forced system. This method and algorithm has been depicted in Figure below.

An example of such a division has been provided in Fig. 4. It is the very first step towards identification of signal characteristics using dedicated methods in the analysis of stationary and non-stationary signals.

For the purposes of analysis of the stationary signal part, an algorithm based on FFT was developed. The signal characteristics are then identified by amplitude based correlation of successive signal harmonics which have been accurately separated from non-stationary signal components. Results of this algorithm have been depicted in Figure 5. Preliminary tests of a car’s floor panel proved various sensitivities to deviation of vibration damping parameters of successive harmonics from a constant input function.

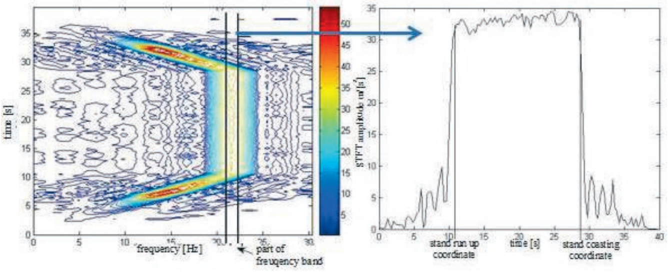


Fig. 3. Calculation and analysis of time function of STFT coefficients for identification of stationary and non-stationary parts of the signal

In order to analyse predominant components of resonant frequencies of sprung and unsprung masses, a transformation algorithm was developed for the non-stationary signals recorded during a rundown of the vibration forcing station and once it was completely shut down. Finally, for the purposes of identification of the signal characteristics, a vehicle free vibration suppression window was chosen, where the vibrations of a system subject to free suppression were recorded. It enabled the system’s free vibration frequency bands to be accurately observed and defined. The window used to analyse and define the range of resonant frequency bands for sprung and unsprung masses has been provided in Fig. 6. The wavelet based time and frequency distribution of a signal enables accurate definition of resonant windows.

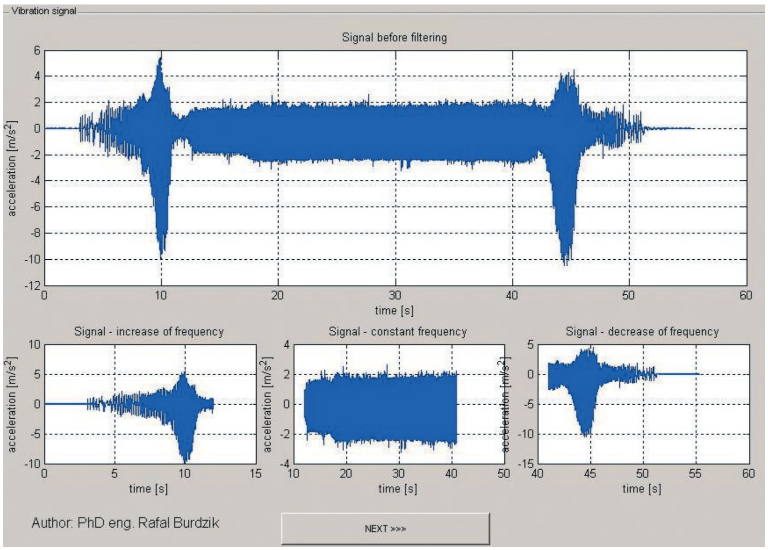


Fig. 4. Vibration of the floor panel - automatic algorithm for positioning of stationary and non-stationary signal cycles

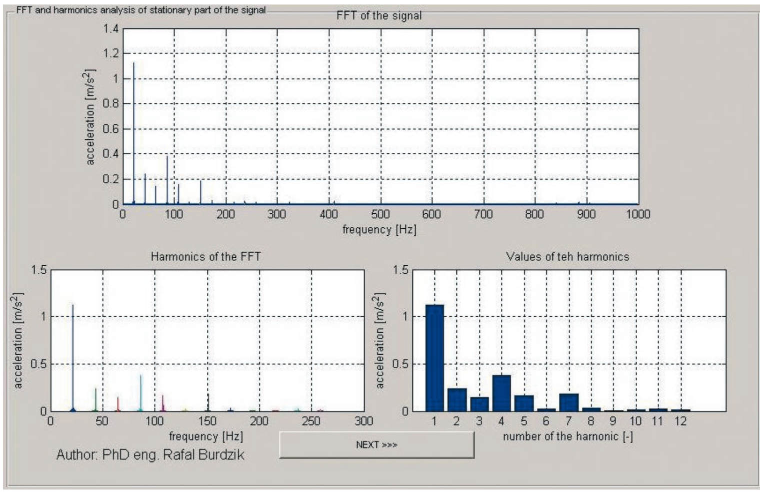
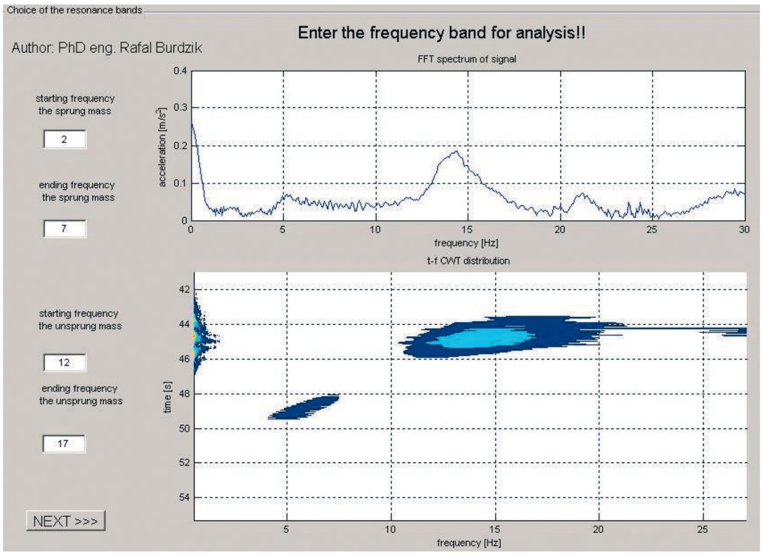


Fig. 5. Results of the FFT analysis for the stationary signal portion

75-elemnent matrices of measures of signal characteristics were used as a multi-parameter measure of signal characteristics for an automotive vehicle’s floor panel. They were established as estimators based on averaged time and frequency courses of resonant windows for sprung and unsprung masses (Fig. 7).



Choice of the resonance bands

Author: PhD eng. Rafal Burdzik

Enter the frequency band for analysis!!

starting frequency the sprung mass: 2

ending frequency the sprung mass: 7

starting frequency the unsprung mass: 12

ending frequency the unsprung mass: 17

Fig. 6. Identification of resonance frequency bands – non-stationary signal portion

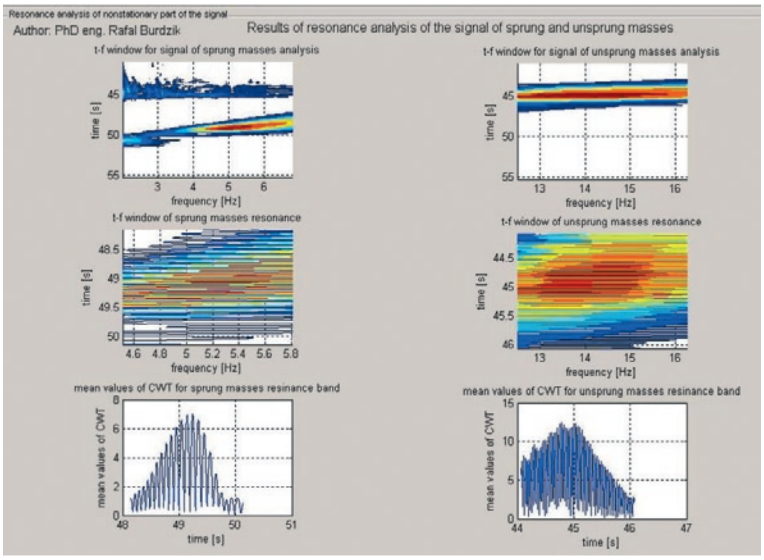


Fig. 7. Time and frequency resonance windows and averaged courses of resonance for sprung and unsprung masses

4. Table of properties of floor panel vibration

The method of multidimensional identification of vibration signal characteristics, described in previous chapter, allows to determine table of properties of an automotive vehicle's floor panel. The complicated vibration phenomena and random character of excitation forces acting on car vehicle determine to use many estimators to define vibration occurring in the car. The described method enables determining measures of signal distribution in time, frequency and time-frequency in terms of stationary and non-stationary parts of the signal.

The tables below contain a collation of the chosen estimators of vibration characteristics of an automotive vehicle's floor panel featuring built-in shock absorbers filled with working medium in 50%. These measures form 75-element table of measures of signal characteristics. From the time realization of acceleration of vibration registered during slowing of excitation, when the mechanical system goes by resonance frequencies bands of sprung and unsprung masses of the vehicle the 16 global estimators have been determined (tab. 1).

Based on the preliminary experimental research it was specified that stationary part of the vibration signal, during excitation force with constant frequency, is sensitive on changes of technical condition of car suspension. Thus for the vibration properties table were added estimators calculated on spectrum of vibration as 12th next harmonics values. The values of those estimators for the same case study (shock absorbers filled with working medium in 50%) have been presented in Table 2.

Some extra "control" estimators of identification of resonance occurring in time and frequency domains for sprung and unsprung masses of vehicle have been added to the table (tab. 3). The values can change for different technical parameters of the suspension system (masses, stiffness).

For the precise time-frequency characteristics of the resonance windows, according to the methodology described in chapter 3, the estimators of CWT (Continuous Wavelet Transform) have been determined. Time and value of the exposure on resonance vibration have been determined separately for sprung and unsprung masses. The tables below contain a collation of the chosen estimators of vibration determined from resonance distribution of CWT. Those estimators have been added to the table of properties of floor panel vibration.

Based on the previous research some extra estimators have been proposed to the table of properties of floor panel vibration. The relative (total) estimators of CWT distribution between resonances of sprung and unsprung masses have been presented in Table 6. Those are the measurements of representation of the relation of vibration characteristics of sprung and unsprung masses. Those estimators have been defined as below.

C_w – half of the sum of maximum values of amplitude of CWT of unsprung masses resonances (unsprung resonance P2P – scope range measurement):]

Table 2. Spectrum of the vibration estimators (stationary signal)

FFT estimators					
1th harm.	2nd harm.	3rd harm.	4th harm.	5th harm.	6th harm.
1,121	0,242	0,142	0,378	0,159	0,019
7th harm.	8th harm.	9th harm.	10th harm.	11th harm.	12th harm.
0,186	0,034	0,007	0,017	0,027	0,016

Table 1. Global estimators of time realization of vibration

Global estimators (amplitude, time) – resonance window					
max	skewness	kurtosis	play factor	root amplitude	standard deviation
2,951	-2,533	14,072	-30,490	0,004	1,157
shape factor	P2P	peak factor	impulsivity factor	RMS	momentum 1
-10,206	5,663	4,229	-43,166	1,339	0,000
correlaction	variance	covariance	median		
1,000	1,339	1,339	0,002		

Table 3. Estimators of resonances location

Estimators of value and location of the resonances					
sprung masses			unsprung masses		
max value	time	frequency	max value	time	frequency
7,511	49,142	5,078	13,909	45,072	13,542

Table 4. Collation of estimators of sprung masses resonance distribution of CWT

Estimators of resonance distribution of CWT – sprung masses window					
max	skewness	kurtosis	play factor	root amplitude	standard deviation
6,995	0,800	2,437	1,642	1,483	1,900
shape factor	P2P	peak factor	impulsivity factor	RMS	momentum 1
1,479	3,457	0,960	1,420	3,601	0,000
correlaction	variance	covariance	median	integral of average CWT	mean/max
1,000	3,610	3,610	1,703	4,883	0,698

Table 5. Collation of estimators of unsprung masses resonance distribution of CWT

Estimators of resonance distribution of CWT – unsprung masses window					
max	skewness	kurtosis	play factor	root amplitude	standard deviation
12,512	0,246	1,938	0,726	7,591	3,357
shape factor	P2P	peak factor	impulsivity factor	RMS	momentum 1
2,040	6,160	0,548	1,118	11,239	0,000
correlaction	variance	covariance	median	integral of average CWT	mean/max
1,000	11,267	11,267	5,283	11,048	0,883

Table 6. Relative dimensionless estimators of the relation of CWT vibration characteristics of sprung and unsprung masses

Dimensionless relative estimators (CWT)				
C_w	L	E_{sr}	E_{max}	E_w
6,352	0,726	7,946	19,507	4,910

$$C_w = \frac{Wz_{\max} + Wz_{\min}}{2} \tag{2}$$

where:

$Wz_{\max}$ – maximum value of the average of CWT distribution for the unsprung masses resonance window,

$Wz_{\min}$ – minimum value of the average of CWT distribution for the unsprung masses resonance window.

L – play factor of average of CWT distribution for the unsprung masses resonance window:

L – play factor of average of CWT distribution for the unsprung masses resonance window:

$$L = \frac{\overline{w}}{(\frac{1}{n}\sum |w_i|^{\frac{1}{2}})^2} \tag{3}$$

where:

w_i –average of CWT distribution for the unsprung masses resonance window,

n – number of samples of CWT distribution average values.

E_{sr} – sum of the average of CWT distribution for the sprung and unsprung masses resonance windows:

$$E_{sr} = Wz_{sr} + Wn_{sr} \tag{4}$$

where:

Wz_{sr} – mean value of CWT distribution for the unsprung masses resonance window,

Wn_{sr} – mean value of CWT distribution for the sprung masses resonance window.

E_{max} – sum of maximum values of the average of CWT distribution for the sprung and unsprung masses resonance windows:

$$E_{\max} = Wz_{\max} + Wn_{\max} \tag{5}$$

where:

$Wz_{\max}$ – maximum value of average of CWT distribution for the unsprung masses resonance window,

$Wn_{\max}$ – maximum value of average of CWT distribution for the sprung masses resonance window.

E_w – concentration coefficient of the average of CWT distribution for the resonance windows:

$$E_w = \frac{E_{max}}{\frac{E_{sr}}{2}} \qquad (6)$$

For the conclusion it can be stated that the table of properties of floor panel vibration is collected from estimators determined from time realization of the vibration, spectrum and time-frequency distribution of the vibration. Exemplary structure of those table have been presented in Table 7. It represents the vibration estimators calculated on the results of the research of the real object, as passenger car with shock absorbers filled with 50% of fluid volume. The colour of the next values represents the estimators presented in tables 1–6.

Table 7. Table of properties of floor panel vibration

2,951	0,002	5,078	0,000	0,548
-2,533	1,121	13,909	1,000	1,118
14,072	0,242	45,072	3,610	11,239
-30,490	0,142	13,542	3,610	0,000
0,004	0,378	6,995	1,703	1,000
1,157	0,159	0,800	4,883	11,267
-10,206	0,019	2,437	0,698	11,267
5,663	0,186	1,642	12,512	5,283
4,229	0,034	1,483	0,246	11,048
-43,166	0,007	1,900	1,938	0,883
1,339	0,017	1,479	0,726	6,352
0,000	0,027	3,457	7,591	0,726
1,000	0,016	0,960	3,357	7,946
1,339	7,511	1,420	2,040	19,507
1,339	49,142	3,601	6,160	4,910

The proper conclusion based on the such large data collection is very difficult. Thus the paper [5,7] presents some application of neural networks as classifier or input module for the control system of vibration absorbing elements in vehicle structure. The scheme of the conception of those system have been presented in the Figure below.

5. Conclusion

Analysis and evaluation of the vibration phenomena in car vehicles are very difficult and it requires using of proper methods and mathematics algorithms. The number of physics and chemical phenomena occurring during working of many systems of vehicles which are affecting on propagation of energy in different forms [9, 10, 15].

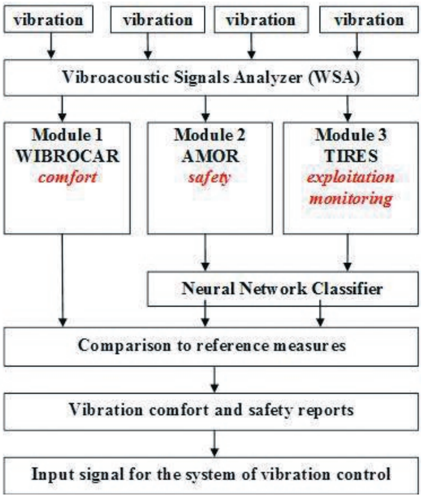


Fig. 8. Scheme of the modular conception of the monitoring and control system of vibration comfort and safety of the passenger car

Thus research on this kind of phenomena has to be conduct and the results and developed methods should be analysed for different parameters of mechanical systems working. The paper presents method verified for different exploitation parameters of the vehicle.

The method proposed and described in the article for multidimensional identification of signal characteristics in the analysis of vibration properties of an automotive vehicle's floor panel enables observation and separation of signal components in various domains. It also makes it possible to define signal measures depending on stationary and non-stationary characteristics as well as accurate time positioning of resonant frequencies. Further conclusions and assessments may rely on selected measures having the properties of state symptoms or may be achieved by means of neural algorithms to function as input databases for a neural network. The measures applied in the table of signal characteristics determine a range of properties such a dynamics, amplification, scattering, concentration, attenuation, stability etc.

The described software implementation of those method has the utilitarian character. WSA program is provided in friendly user interface. The results as table of properties of floor panel vibration could be adopted as mapping input signal to system of monitoring and control of vibration.

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