TESTING METHODS FOR VEHICLE SHOCK ABSORBERS

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Abstract

The suspension is the component decisive about safety of drive of cars, in which dumping elements have essential meaning, which the most often are telescopic hydraulic shock-absorbers. Therefore in exploitation process so important investigations of their technical condition are in track which opinion of their usefulness should be made not only to more far exploitation, but also the identification of typical damages. The common used method in practical ways of diagnosing of shock-absorber in article were introduced. There are two methods of investigations: built from and shock-absorber built-in in vehicle suspension. In first method - in support about indicatory graph, the characteristic of suppression be appointive. They are the basis to diagnostic opinion of technical condition of studied damper. In second method the results of investigations have approximate character. From those methods the most often used station control of vehicles in Poland - method post EUSAMA and BOGE in article were described, which used in investigations the harmonic excitatory of vibrations stimulant to vibrations unsprung and sprung masses of vehicle. The car is so folded, vibrate mechanical stimulated by excitator to vibrations arrangement. The vibro-acoustic methods of analyses' of signals in article were introduced, which can be practical in investigations of diagnostic shock-absorber. They are then the method of multidimensional analysis of signal - the Short-Time Fourier Transforms (the STFT), Wigner - Ville'a and Wavelet.

Keywords: *transport, vibro-acoustic methods, testing methods, shock absorbers*

1. Introduction

Road traffic safety depends on many factors, including technical condition of vehicle suspensions. Normal operation of a vehicle unavoidably leads to wear and tear of its knuckle joints and spherical couplings as well as changes in characteristics of its springs and damping elements. Diagnosing shakes in connections between dampers, rods or bearings is fairly simple. Sufficient information about them can be obtained through testing suspensions on excitation test rigs (commonly called jerkers). In order to make such shakes visible, it is necessary to obtain a high level of excitation because suspension must bear pre-tension forces exerted by the springs.

Spring wear can be seen through photo-elastic tests on suspensions and when measuring static spring deflection. A serious testing problem arises when quick assessment of technical condition of in-mounted dampers is required.

Apart from shock absorbers, there are tyres, springs, spherical couplings and metal-rubber sleeves to play a role in damping the vibrations. But the role of absorbers is often decisive as they have strong damping effect combined with unsymmetrical characteristics.

When in cars, they must ensure good tyre adheson to the road surface with varying types of the surface or varying travel speeds and during braking.

Defective absorbers may cause:

- excessive acceleration amplitudes in car body vibration,
- excessive car rolls, both lateral and longitudinal (important for comfortable travel and car travel stability),
- excessive braking distance,
- excessive dynamic forces will result in fatigue destruction of car components and road surfaces.

Modern car suspensions are complex material systems. Damping elements and springs contained therein have non-linear characteristics undergoing changes due to wear and tear. This feature leads to difficulties with proper diagnosing or differentiating between specific defects.

A car has many resonance frequencies, but sprung mass resonance (1-3 Hz) and unsprung mass resonance (8-18 Hz) are the most important. Comfortable travel is strongly related to the first frequency band, and safe travel to the second frequency band.

Generally, there are two types of testing methods for shock absorbers:

a) using a test rig,
b) using no test rig,

and the test rig methods can be divided into:

- those for absorbers mounted in the car,
- those for absorbers removed from the car.

2. Testing methods for shock absorbers mounted in the car

Testing methods for shock absorbers mounted in the car can be divided as follows:

1) forced vibration type,
2) free vibration type,
3) using test rig plates.

Out of them, the third type is practically not used any more due to its poor accuracy.
The two methods are based on forcing vertical vibrations in the test wheel till above the resonance frequency. To perform the test, the wheel is placed on the vibration inductor plate. Upon exceeding resonance frequency, the exciting force is switched off. Vibration frequency will soon go down in result of damping by the absorber, suspension elements and tyres. Technical condition of the absorber is indicated by amplitude occurring with the suspension resonance frequency.

Damping efficiency can be determined by means of vibration analysis:

- in function of wheel pressure on the plate (EUSAMA method),
- in function of time (Boge method).

3. EUSAMA method

Eusama method, developed by The European Association of Shock Absorber Manufacturers, attempts to assess adhesive force (in %) of the wheel on the ground. Damping efficiency of the absorber is shown by Eusama factor as follows:

$$WE = \frac{W_{\min}}{W_{st}} \cdot 100\%, \tag{1}$$

where:

W_{min} - minimum measured dynamic tyre-to-support contact force
W_{st} - static tyre-to-support contact force (static weight)

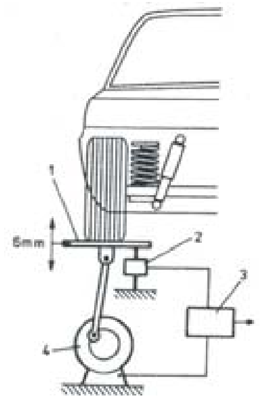

Fig. 1. Schema of research test stand for examine shock absorbers with EUSAMA method.:1-stand platform, 2-tensometric measure system, 3-analysing system, 4-electric engine

Static adhesive force (Wst) is measured with the wheel resting on a motionless plate of the inductor. In such case the value of WE factor is 100%. Upon starting the excitation system, the plate will receive vibrations with aplitude 4-8 mm and frequency about 25 Hz. Minimal dynamic pressure force Wmin of the wheel on the plate is measured upon stopping the system when the frequency drops to about 16 Hz.

If the wheel comes off the plate with this frequency, the value of WE factor is considered 0%. This philosophy of measurement is clear and logical. The method described above does not require a database.

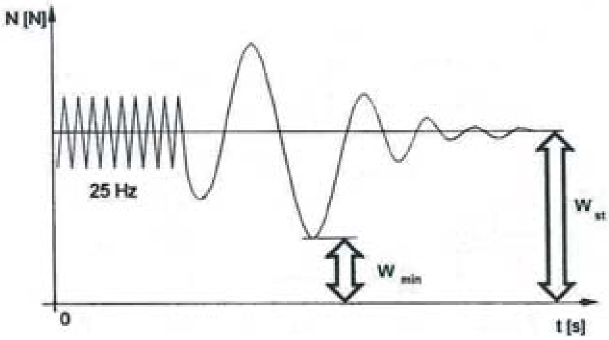

Fig. 2 Graphic interpretation of EUSAMA coefficient

A disadvantage of the method is that it is very sensitive to tyre rigidity and static load. Tyre rigidity depends on the contained intensity of pressure. Low pressure increases and high pressure decreases EUSAMA factor. On the contrary, high static load increases and low static load decreases the factor.

The assessment criteria are as follows:

a) WE = 0-20%, bad technical condition of the absorbers (insufficient damping rate),
b) WE = 21-40%, acceptable damping rate, the absorber requires inspection (upon removal) on indicator test rig,
c) WE = 41-60%, good damping rate,
d) WE > 60%, very good damping rate.

20-30% differences in EUSAMA value between left and right vehicle sides indicate that the absorber is defective. However, the tests raise doubts when used for rear axle suspension in light cars with front-wheel drive. The problem seems to be solved by Messrs. Hoffman who propose to classify the cars into four groups according to their natural masses:

1) Group I - vehicles of higher class, natural mass over 1400 kg, acceptable adhesion difference: DRP = 25%
2) Group II - vehicles of medium class, natural mass 900-1399 kg, DRP = 25%
3) Group III - compact vehicles, natural mass below 899 kg, DRP = 20%.

The data obtained through EUSAMA method are approximate in character. Using databanks, it is possible to get information about reference values (e.g.based on inspection of new vehicles) or to observe the trends in changes occurring during operation. Accepting a stable resonance frequency (16 Hz) causes limitation to the method. The wheels can have the same WE values, e.g. 60%, but their resonance frequencies need not necessarily be the same. Such differences have an influence on the travel safety. Therefore a modified testing method was developed.

4. Testing method for shock absorbers on a plate test rig with varying frequency of excitation vibrations

Testing method for shock absorbers based on varying resonance frequency is called EUSAMA plus. This method consists of two measurement stages:

a) preliminary phase,

b) taking measurement of damping coefficient.

Preliminary phase is devoted to warming up the absorber fluid for the purpose of obtaining its proper viscosity. This process takes about 10 seconds and runs with low vibration frequency. Intensity of pressure in tyres is measured. Contact between the plate and the tyre is checked. If air pressure deviation exceeds 0.05 MPa of nominal pressure, the measurement process is stopped automatically (this is signalled by appropriate instruction).

Upon completion of the warm-up phase, the measurement process begins to proceed. Plate vibration frequency is reduced by 1 Hz from 30 Hz to 8 Hz using optoelectronic converters. The converters are to stabilize each frequency throughout the measurement process. Careful consideration is given to the range from 13 Hz to 18 Hz (unsprung mass resonance). Pressure force on the plate is measured for each frequency range when the absorbers are in compression and in rebound. Thereupon the ratio of mases can be determined for each vehicle type:

$$c = \frac{m_r}{m_n} \ , \tag{2}$$

where:

m_r - sprung mass
m_n - unsprung mass

Then a relationship between damping coefficient and sprung mass-unsprung mass ratio is plotted to describe condition of the suspension when the absorber is in compression and in rebound (irrespective of the vehicle type).

Basing on the obtained diagram, it is possible to assess whether the damping coefficient value is within:

- green zone (over 80%) - good condition of the absorber,
- yellow zone (50-80%) - satisfactory,
- red zone (below 50%) - unsatisfactory.

Such testing system is used by Beissbarth Micro-SAT 6600. Precise changes in the plate vibration frequency and stable elasticity of the tyre (air pressure) combined with a warm-up phase enable accurate measurements.

Another modification was introduced to Eusama method by Hunter Engineering Company. What they used additionally was a phase angle, i.e. a phase shift angle between sinusoidal signals of the plate dislocation and the wheel pressure on the plate (Fig. 3). The test rig did not need to be rebuilt, but only the test line had to be expanded by a device to measure dislocation of the plate.

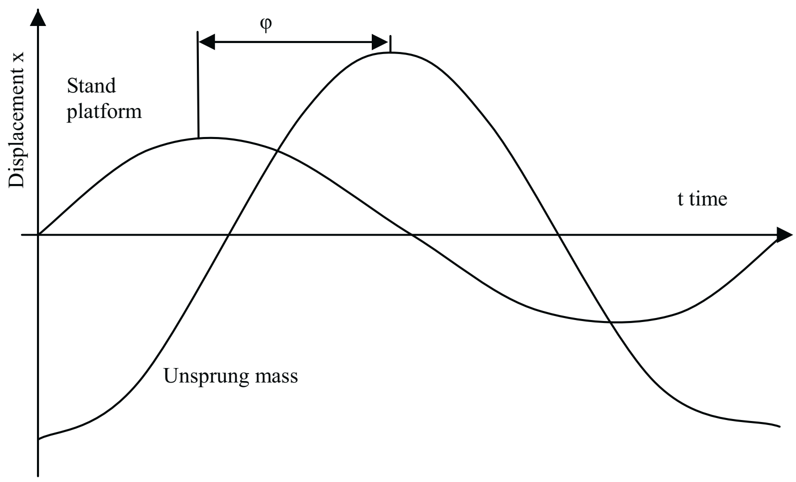

Fig. 3. Phase difference of platform and unsprung mass

Unsprung mass dislocation is proportional to temporary pressure exerted on the plate. Therefore its peak value comes with the phase angle equal to zero. Amplitude of the unsprung mass dislocation gets reduced with the phase angle increase, even if the pressure value remains unchanged.

Within the band of sprung mass resonance frequency, when there is no damping in the suspension, the phase angle between maximal amplitude of unsprung mass dislocation and maximal amplitude of sprung mass dislocation will be 180^{o}. When damping rate increases, both dislocation amplitude and phase angle decrease. Within unsprung mass resonance band, when there is no damping in the suspension, the phase angle is 0^{o} while the dislocation comes to its peak value. Pressure between the unsprung mass (wheel) and the plate is as low as possible. When damping in the suspension increases, dislocation rate of the wheel gets reduced while the phase angle increases.

Phase angle is closely connected with damping rate.

If the car suspension has adequate damping rate, i.e. adequate shock absorber, the minimal phase angle between sprung mass resonance frequency and unsprung mass resonance frequency will be above 90^{o}.

5. Boge method of forced vibration type in function of time

The method used by Messrs.Boge consists of a plate test rig to stimulate vibrations, and a crankshaft unit to set the plate in motion.

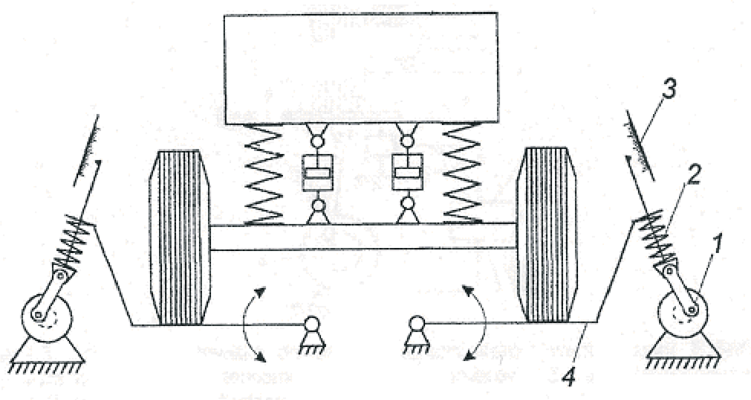

Fig. 4. Schema of Boge test stand:1-electric engine, 2-spring, 3-recording device, 4-stand platform

Testing procedure for shock absorbers is as follows: The car is made to go with its wheels on the plates which are immediately forced to vibrate. Each wheel is tested separately. As soon as the exciting force is switched off, both the suspension and the excitation unit are subject to resonance frequency. Vibrations of one wheel are recorded on half part of a round disk. Example result of Boge examine shown Fig. 5.

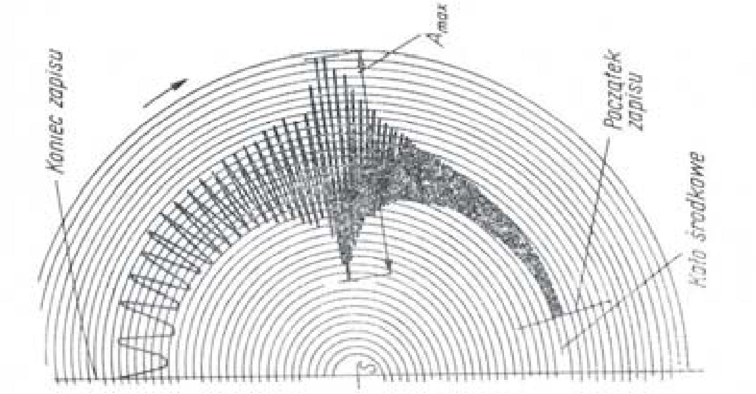

Fig. 5. Example result of Boge examine.

Condition of the absorber is measured by double amplitude of resonance vibrations (Amax). Value of the amplitude is compared with reference amplitude for shock absorbers specific vehicle types.

Damping decrement measurement is a reasonable method for linear systems. As there are non-linear elastoplastic elements in modern car suspensions, the results should be considered as purely approximate.

6. Testing methods of free vibration type

The method is based on the analysis of car chassis motion after it has been excited to vibrate. There are many excitation methods, including:
- drop,
- fall from an inclined plane,
- rebound of the compressed body.

Among those methods, the third one is nowadays most widely used. Testing procedure for a shock absorber mounted in the vehicle is as follows: The car is made to go with its test wheel on the rig mobile bracket connected through lever unit to a lifting gear (mechanical or pneumatic). As soon as the gear is unlocked by the control system, the bracket will drop together with the car.

As it is dropping, the wheel will hit on the plate causing vibrations in unsprung and sprung masses. Amplitude of vibration dislocations in function of time is recorded in the form of diagrams.

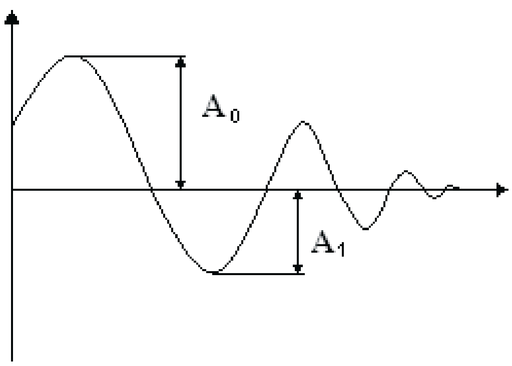

Fig. 6. Result of examine by „drop" metod

Basing on such diagrams, mean damping rates „k" (second-to-first vibration amplitude ratio) were determined.

$$k = \frac{A_1}{A_0}, \tag{3}$$

where:

A_1 - second observed amplitude
A_0 - first observed amplitude

Initial amplitudes should be equal. Acceptable difference between them should not exceed 7%. Testing methods of free vibration type enable detection of shock absorber defects which have significant influence on vibration amplitude i.e. insufficient filling in the absorber, damage to check valve spring, worn plates in overflow valve or throttle valve.

However, defective spring or overflow valve seizing have only very slight effect on the absorber characteristics, being therefore hard to detect. The method assumes that the system is linear.

7. Testing method on indicator test rigs

Kinematic diagram of a test rig for shock absorbers removed from vehicles and view of belt transmission is shown in Fig. 7

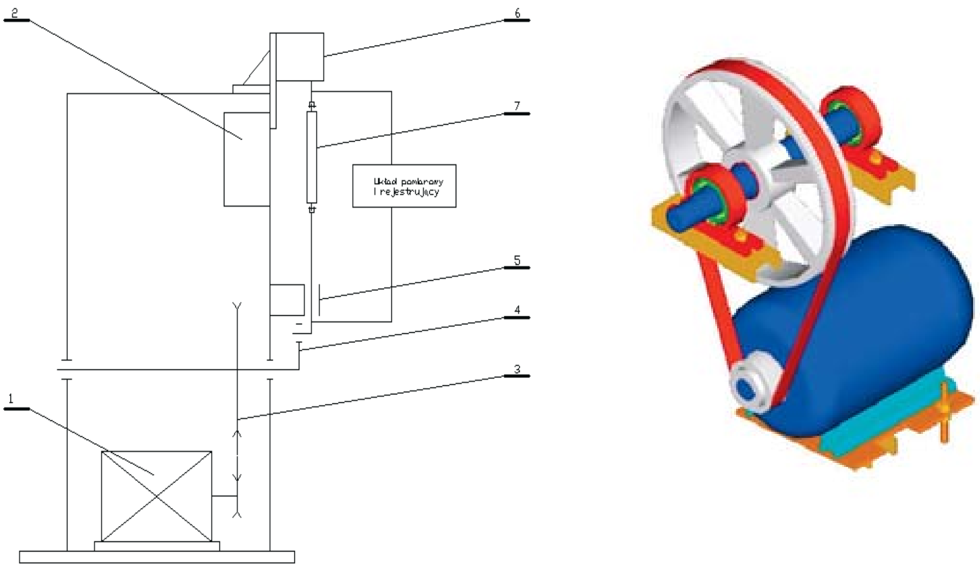

Fig. 7. Indicator test stand: kinematics scheme and view of belt transmission:1-electric motor, 2-frequency modulator, 3-belt transmission, 4-eccentrik system, 5-slider ways, 6-force sensor, 7-schock absorber

The rig is driven by a two-speed, alternating-current electric motor mated with a frequency converter (inverter). The motor drive is transmitted to an eccentric crankshaft unit with varying length of the arm where lower holder of the absorber will be connected. Slider mechanism is used to transmit motion in lower and of the absorber from the crankshaft unit, whereas a force converter is used to connect its upper and with the rig casing.

The rig is provided with double measuring system (mechanical and electronic) to enable recording indicator diagrams of the absorber damping force in function of piston travel at deflection and rebound. Excitation system used with the rig is a typical crankshaft unit (eccentric) where changes in the crank arm dislocation in function of crank angle are described by the following relationship (upon differentiation):

$$\dot{z} = r \cdot \omega(\sin\alpha + \frac{\lambda}{2}\sin 2\alpha), \tag{4}$$

where:

$\lambda = \frac{r}{l}$ crank throw to connecting-rod length ratio,

ω - crank angular velocity

$$\omega = \frac{\pi n}{30} \quad rad/s, \tag{5}$$

n - crankshaft rotational speed

Tests performed for various crankshaft rotational speeds enable determination of shock absorber speed characteristics i.e. damping force in function of piston rod motion.

Maximal amplitude of damping force is widely used now as a diagnostic parameter in indicator tests for shock absorbers.

When assessing their technical condition, the amplitude is compared with a reference damping force of a new shock absorber. Such method of assessment is deceptive because the reference model is established through calculating an average result of tests performed on many new shock absorbers (having a wide range of the force).

Therefore a comparison of one test absorber is inevitably erroneous. Moreover, assessment by amplitude will not provide information about what type of damage or defect has occurred.

In light of the above, a new diagnostic parameter is proposed, namely damping power of the absorber. Damping power can be obtained through integrating indicator diagrams based on the tests. Usefulness of this method was confirmed by experimental tests on new shock absorbers and those with programmed defects.

Test shock absorbers were taken from Fiat Seicento 900 and Skoda Fabia 1.4 cars. For the purposes of the tests original unremovable shock absorbers were rebuilt into removable types. Tests were performed at stable temperature under laboratory conditions. Changes of damping force in piston travel function were recorded by closed diagrams. Forces can be measured either with stable piston travel and varying piston motion velocity or with varying piston travel and stable velocity.

Velocity diagrams of force changes in function of piston motion were obtained by differentiation of dislocations. Damping power was determined basing on one cycle of absorber performance, through indicator diagram integration.
Compressive power and rebound power should be tested separately due to asymmetrical characteristics. For diagnostic purposes the following programmed defects were made:

- loss of fluid in result of poor absorber sealing,
- deterioration of fluid,
- damages to valves for rebound and compressive motions,
- damages to piston seal.

Examples of shock absorber characteristics showing fluid losses are presented in the Fig. 8-9.

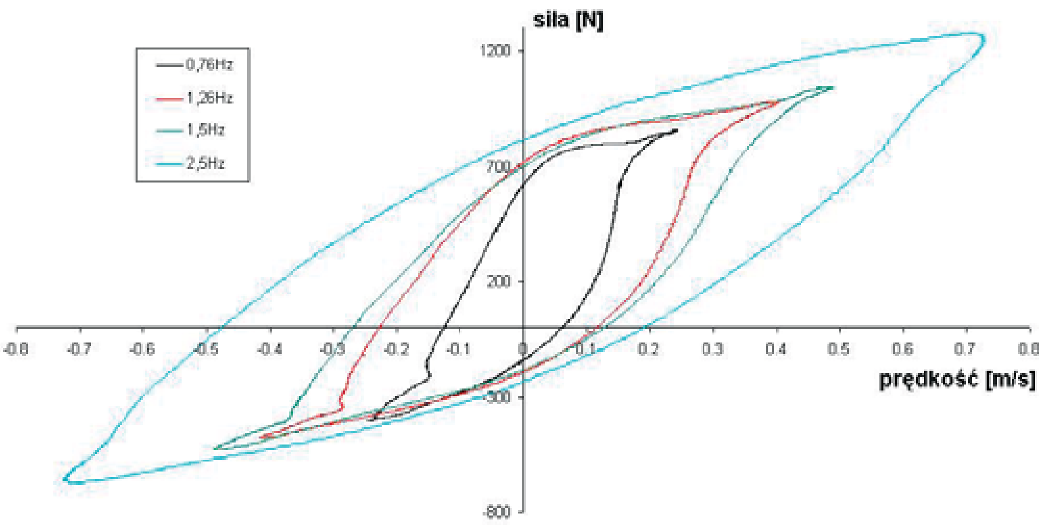

Fig. 8. Characteristic of new shock absorber

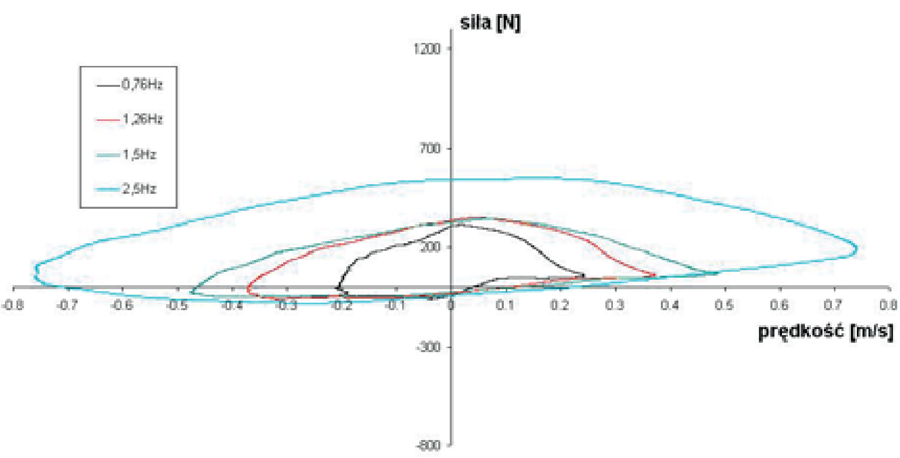

Fig. 9. Characteristic of shock absorber with 75% fluid volume

Analyses of damping powers with various excitation frequencies resulted in good identification of fluid losses in the absorbers. The proposed diagnostic parameter can be used for diagnosing car absorbers.

Damping power should not exceed certain specific limits. Such limits can be determined upon testing a large group of shock absorbers of one type and of one manufacturer.
The range of limits should be based on statistical analysis of the above tests.

8. Vibration analysis methods used to measure technical condition of suspension components

Results of tests for shock absorber condition performed on test rigs can only be treated as approximate. The assessment scale has four grades but it practically tells only whether the absorber is good or bad. Moreover, it does not enable identification of damages/defects.

All methods described above involve vibrations stimulated on both sprung and unsprung masses. So the tests refer to a material system generating a vibro-acoustic signal which will be, upon converting, a non-stationary random process. This process should be tested by mathematical analysis.

Assessment methods used for diagnostics should provide a large amount of information to enable assessment of usual wear and tear together with identification of damages/defects.
In light of such approach, it is important to note that:

a) vibro-acoustic processes provide sufficient information and are highly sensitive to changes occurring in the test car,
b) vibro-acoustic processes enable measurements on the actual car at work.

Random processes, in which random events can be functions of frequency and time, are of great importance for the analysis of non-stationary processes.

8.1. Short Time Fourier Transform

An effective method of testing signals in time-frequency field was proposed by Gabor. The method is now widely accepted. It consists of frequency analysis of the signal fragments, one by one, using the so-called window function.

During the analysis, the window is shifted in time field by an interval equal to the window width. Frequency analysis is performed for the fragments independently, enabling a relation between spectral periodical components and time together with determination of moments they occur.
Gabor proposed a gaussoidal form to present window function:

$$g(t) = e^{-\frac{1}{2}t^2}, \tag{6}$$

and formulated a principle of constructing a family of analytic functions arising from the window function through shifts in the field of time „b" and that of frequency „ω":

$$w_{\omega,b}(t) = g(t-b)e^{-i\omega t}. \tag{7}$$

Upon time-frequency analysis of signal f(t), the following coefficients are received:

$$S(\omega,b) = \int_{-\infty}^{+\infty} x(t)\cdot w_{\omega,b}(t)dt = \int_{-\infty}^{+\infty} x(t)\cdot g(t-b)e^{-i\omega t}dt. \tag{8}$$

Relations described above can be transformed into the following formula for analytical function, suitable for practical application:

$$W_{\omega,b} = e^{-\frac{1}{2}\left(\frac{t-b}{d}\right)}\cdot e^{-i\omega t}, \tag{9}$$

where:

ω - analysing frequency,
b - window shift,

g(t - b)=const - constant width of the subsequently analysed window.

Of the three coefficients contained in the formula, „b” revers to window shift (time parameter), „ω” revers to analysis rate for the window (frequency parameter) and „d” is invariable throughout the family of analytical functions (window width parameter).

The method proposed by Gabor, called Short Time Fourier Transform (STFT) or Fourier Window Transform, is based on stable width of the analytical window. Thus both the precision in time field and frequency range of the analysis are fairly limited.

If performed in the best possible manner, the analysis (with ic accuracy limited only by Heisensberg uncertainty principle) occurs in lower range of frequencies i.e. for analytical functions Wω,b where period 1/ω is equal to gaussoid window width „d”.

All higher frequencies of the signal are detected with the same accuracy, being much lower than it could be obtained in accordance with Heisensberg principle.

8.2. Wavelet transform

Another method of time-frequency analysis was proposed by Morlet and Grossman in 1984. Like in STFT, they formulated a basic function, principles of constructing the family of analytical functions, and a formula to describe the result in field „t-f” (time-frequency) in the form of two variables („t” and „f”).

Morlet proposed a non-periodical basic function:

$$\psi(t) = e^{-\frac{t^2}{2}} \cdot \cos(5t), \tag{10}$$

and called it „mother wavelet” (in French ondelette-mere) because it is used to construct the family of analytical wavelets [30,37,53] ψa,b :

$$\psi_{a,b}(t) = \frac{1}{\sqrt{a}} \cdot \psi(\frac{t-b}{a}), \tag{11}$$

a > 0 is the so-called contraction-expansion coefficient, responsible for frequency and time of the analysis.

Upon introducing an additional scale parameter „a” to the location window, Gabor's transform included Wavelet Transform (WT). Analytical function $\psi\left(\frac{t-b}{a}\right)$ called „main wavelet”, contains coefficient „a”, causing a change in duration of the wavelet, and coefficient „b” causing a change in wavelet location on the time axis. The equation represents band-transmit filtration of the signals using filters with various transmit bands. Wavelet transform (WT) is a two-dimensional function where „a” is a scale parameter (of frequency) and „b” is a translation parameter (of time shift) $a, b \in R, \ \ a \neq 0$

Coefficients representing the signal within t-f field are described by the following relationship:

$$WT(a,b) = (x(t) * \psi_{a,b}) = \int_{-\infty}^{+\infty} x(t) \cdot \psi_{a,b}(t)dt = \frac{1}{\sqrt{a}} \int_{-\infty}^{\infty} x(t)\psi(\frac{t-b}{a})dt. \tag{12}$$

This relationship, like (3) proposed by Gabor, is equivalent to convolution of the analytical signal x(t) with analytical wavelet ψa,b (t) or, in case of Gabor analysis (STFT), with analytical function $w_{\omega,b}$ (t).

In contrast to window functions, used in STFT with more or less approximate gaussoid function, function ψ (t) has different features, being an even function with local oscillation whereas outside connected interval being equal to zero. Run of the function ψ (t) resembles little waves in appearance with local oscillations fast fading when away from the centre.

Principle of constructing the family of analytical functions is different in the wavelet analysis when compared with STFT. In STFT analysis, the window width is stable, and window oscillation

increases with an increase in frequency. In wavelet analysis, the number of wave oscillations is stable but changes in frequency are accompanied by proportional change in the wave duration.
Analysis of non-stationary spectral properties of the signal requires that windows should be used. The windows will automatically become narrower when analysis of high frequencies is performed, or wider in case of low frequency analysis.

8.3. Wigner-Ville Distribution (WVD)

Wigner-Ville mutual spectrum of two signals x(t) and y(t) is defined as follows:

$$WVD_{xy}(t,f) = \int x(t+\frac{\tau}{2})y^*(t-\frac{\tau}{2})\,e^{-j2\pi f\tau}\,d\tau, \tag{13}$$

however, if x(t) = y(t), the above transformation can have the form:

$$WVD_{xx}(t,\Theta) = \int x(t-\frac{\tau}{2})x^*(t+\frac{\tau}{2})e^{-j2\pi\theta t}\,d\tau\,, \tag{14}$$

where:

WVD (θ, τ) - Wigner-Ville pseudo-transformation
x*(t) - combined signal coupled with x(t),
τ - shift in time domain,
θ - shift in frequency domain.

This transform enables transformation of the signal plotted against time into a time-frequency spectrum. Matlab program has an implemented discrete version of this transformation to enable processing of the digitized signal. As the algorithm performing this transform uses double Fourier transformation, it is necessary to sample the continuous signal with at least twice higher frequency than Nyquist criterion, in order to avoid aliasing.

As it is important to reduce the effects connected with spill of the spectrum, which would blur interpretation of the results, tapering functions are used to perform WVD filtration. In our tests we used Choi-William tapering function as follows:

$$\phi(\theta,\tau) = \exp(-\theta^2\tau^2/\sigma^2), \tag{15}$$

where:

σ - parameter proportional to amplitude of spectral spill.
The transformation can be shown as:

$$WVD(t,\Theta) = \int x(t-\frac{\tau}{2})x^*(t+\frac{\tau}{2})\,e^{-j2\pi\Theta t}\,e^{-\left(\frac{\theta\tau}{\sigma}\right)^2}\,d\tau. \tag{16}$$

An essential problem connected with the above analyses is how to present the results. There are many methods, but out of them the following two seem to be worthy of attention:
- contour line diagrams,
- profile diagrams.

Contour line diagrams are of two-dimensional type in time-frequency coordinate system in which power density is shown in the form of lines.

Profile diagrams are pseudo-spatial drawings showing spectral concentration of power in function of time and frequency. However, the „slice-like” method of visualisation may cause skipping significant results. Moreover, some of the results are hidden by preceding profiles. Making a separate diagram of power changes in function of time for each frequency seems to be a reasonable solution to this problem. Spectral structure of non-stationary signals can be determined through double Fourier transform of non-stationary correlation functions which have been determined through expectation values.

Literature

[1] Bocheński, C., Lozia, Z., Mikołajczuk, J., *An attempt to objectively assess the testing methods for shock absorbers mounted in vehicles as recommended by EUSAMA association*, Tenth Conference on Diagnostics for Work Machines and Vehicles, Bydgoszcz-Borówno 1999.

[2] Gardulski, J., *A method for assessing technical state of car suspensions using no test rigs*, Published by Institute of Service Technology, Radom 2003.

[3] Gardulski, J., *Comparison of analyses methods - continuous wavelet transform and wigner-ville'a transform in vehicle shock absorber diagnostic* KONES Vol. 14, No. 4, Warsaw 2007.

[4] Gardulski, J., Warczek, J., *Diagnostics for shock absorbers mounted in vehicles based on analysis of nonlinear effects*, Silesian Technical University Scientific Journals, No.48, ser.transport, Gliwice 2003.

[5] Gardulski, J., Warczek, J., *Technical state of a fluid telescopic shock absorber as identified on the basis of its characteristics analysis*, 30-th Jubilee Polish Symposium on Diagnostics for Machines, Węgierska Górka 2003.

[6] Gardulski, J., Warczek, J., *Damping power as a diagnostic factor for shock absorbers in cars*, DIAGNOSTYKA, Vol.29, Warszawa 2003.

[7] Lozia, Z., Mikołajczuk, J.: *Assessment of usefulness of the diagnostic test on vehicle-mounted shock absorber condition using kinematic extortion for wheel-ground contact*, Automotive Archives 2, 1997.

[8] Ringus, G., *A neural network based diagnostic test system for armored vehicle shock absorbers. Expert Systems With Aplications*, Vol. 11. No. 2, pp. 237-244, Great Britain, 1996