DIAGNOSTYKA, 2017, Vol. 18, No. 1

ISSN 1641-6414
e-ISSN 2449-5220

DIAGNOSTIC OF SHOCK ABSORBERS DURING ROAD TEST WITH THE USE OF VIBRATION FFT AND CROSS-SPECTRUM ANALYSIS

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Abstract

In this article an attempt to diagnose damage to the shock absorbers of a passenger car during road operation with the use of vibration response measurement has been described. Accelerometers were mounted on the body – sprung mass. Based on the recorded signals, FFT and Cross-Spectrum graphs were prepared. The test was performed for several variants shock absorber damage.

The single-number index of vibration amplitude increase, in the range of resonance frequency, was calculated, based on spectrum analysis. The results are shown as graphs and tables, for different damage types. The realization of measurement and adopted method of damage diagnosis are described.

Keywords: Shock absorbers, diagnostic, vibration, fft, cross-spectrum

DIAGNOSTYKA AMORTYZATORÓW PODCZAS PRÓBY DROGOWEJ Z WYKORZYSTANIEM WIDMA DRGAŃ FFT ORAZ WIDM KRZYŻOWYCH

Streszczenie

W pracy przedstawiono i opisano próbę diagnozowania uszkodzenia amortyzatorów w samochodzie osobowym przy użyciu odpowiedzi z czujników drgań podczas jazdy. Czujniki zamontowane były na nadwoziu – na masie resorowanej. Na podstawie zebranych sygnałów z czujników drgań wyznaczono widma drgań FFT odpowiedzi oraz widma krzyżowe. Badania wykonano dla kilku różnych wariantów uszkodzeń amortyzacji.

Na podstawie analizy widm przyjęto jednoliczbowy wskaźnik obrazujący wielkości wzrostu amplitudy w obszarze częstotliwości rezonansowej. Wyniki przedstawiono w formie wykresów i tabel dla poszczególnych rodzajów uszkodzeń amortyzatorów. Opisano przebieg wykonanych pomiarów oraz przyjętą metodę diagnozowania uszkodzenia amortyzatorów.

Słowa kluczowe: amortyzatory, diagnostyka, drgania, FFT, widmo-krzyżowe

INTRODUCTION

Due to the great increase in the quantity of vehicles on the world's roads and despite the fact of continuous expansion of infrastructure, road traffic is denser and denser. Active and passive safety systems are ever more significant for this reason. Active safety systems are at least as important as passive ones, because it is better to prevent accidents altogether, rather than to minimalize their consequences. One of these active safety factors is diagnostics of the most important car subassemblies that affect such parameters as braking effectiveness or traction. There are many electronic systems that improve those parameters. For example, ABS does not allow the wheels to be blocked during braking, allowing the driver to steer even during high deceleration. ESP prevents getting into an uncontrolled skid. But these systems will not help much if mechanical elements of the suspension fail.

One of the most important suspension elements that directly influences active safety are shock absorbers. Damping of vibration is responsible for ensuring wheel grip to the road during driving. Insufficient damping fosters disconnection of the wheel from surface which can lengthen the brake distance or can even be the reason for losing control of the vehicle.

Damping efficacy testing is usually only performed during vehicle servicing. Testing at diagnostic stations gives sufficient damping estimation results, but is usually performed with the use of the EUSAMA[1] stationary method. During recent years, many methods risen to estimate condition of vehicle dampers. But all these test methods need to be carried out on special stands. There is still the problem of rapid drop of airtightness on the damper during for example long travel; most drivers will not be able to notice this problem. That is the reason why a system which would be able to detect damper damage during normal road operation is desirable. It would give the driver information that there is a danger and that the

[1] EUSAMA – stationary test method of shock absorbers used in diagnostic stations

speed of car should be decreased, as well as that extreme caution should be exercised.

This paper concerns the results of research performed on a passenger car that was carried out to detect damping loss of damaged shock absorbers with the use of sprung mass vibration measurements.

1. TESTING METHOD

To detect any loss of damping in a suspension system, measurement of vibration is necessary. Where possible, the object is excited with a known or measured force and then the response is measured at defined points. The frequency and shape of all modes in the range of interest are calculated with the use of formula (1) FRF - magnitude and phase graphs. The damping factor is calculated with the use of half power method (4). This kind of analysis is called experimental modal analysis. Unfortunately, this method is not always suitable. During normal road operation only response measurement is possible without the use of complicated and expensive systems.

1.1 FRF - Transfer function

The transfer function, in other words frequency response, is defined as a complex ratio of the output signal spectrum to the input signal spectrum as a function of the frequency [2]:

$$H_{xy}(f) = \frac{F_y(f)}{F_x(f)}, \qquad (1)$$

where $H_{xy}(f)$ is the transfer function from point x to point y,
$F_y(f)$ – Fourier spectrum on output of the signal measured at point y,
$F_x(f)$ – Fourier spectrum on input of the signal measured at point x.

1.2 Cross-spectrum

The cross-spectrum can be obtained from the individual Fourier spectra $F_x(f)$ and $F_y(f)$ as follows [2]:

$$F_{xy}(f) = F_x^*(f) \cdot F_y(t), \qquad (2)$$

where $F_{xy}(f)$ is a typical cross-spectrum component and $F_x{*}(f)$ is the complex conjugate of $F_x(f)$.

1.3. Half power method

Each mode is characterized by three parameters: frequency, shape and damping. The frequency is estimated from the FRF characteristic directly after the measurement. The natural frequency is read as the maximum (peak) value of the FRF characteristic [9]:

$$|\alpha(\omega)_{max}| = \omega_r, \qquad (3)$$

For both values ω_a and ω_b localized on each side of the local maximum the amplitude is equal to half of the power, which equals: $\frac{\alpha_{max}}{\sqrt{2}}$ (fig.1). The damping ratio ξ may be calculated from the following formula [9]:

$$\xi = \frac{\omega_b - \omega_a}{2\omega_r}, \qquad (4)$$

An example of FRF graph and half power method of damping estimation are shown in fig.1

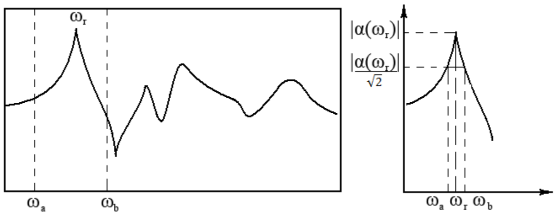

Fig. 1. FRF graph - half power method of damping estimation [9]

2. TESTED OBJECT

The measurements were done on the internal road of the Institute. The track is made of an asphalt and the structure is comparable to typical local road. There are some little irregularities typical for local roads also. The measurements were done at the range of speed 20 – 50 km/h on first three gears. Test track is shown in satellite photo – fig. 2. The drive can be compared to city traveling at increased traffic density.

Fig. 2 Test track (orange color)

Four points near the absorber fixings were chosen to measure 3 axial vibration. Additionally, Z axis vibration was measured at the center of the bulkhead. Location of the accelerometers with simplified a dynamic model (without taking into consideration wheel springiness) are presented in fig. 3. In fig. 4 the transducers measuring vibration at points 1 and 4 are shown.

Vibration was measured on a Nissan Sunny. Before the measurements, the suspension was fully serviced. All loose subassemblies were replaced; new dampers were installed. After servicing, a normal stationary damper diagnostic was performed. The EUSAMA values for the new shock absorbers were over 80 % for all wheels. To reduce the quantity of transducers, measurement points were only on the body. Measurements with the use of points on unsprung mass elements (such as the swingarm) would be much easier because the FRF characteristic would give the damping ratio directly. But the aim of the test was to minimalize the quantity of accelerometers and to use only responses similar to Operational Modal Analysis.

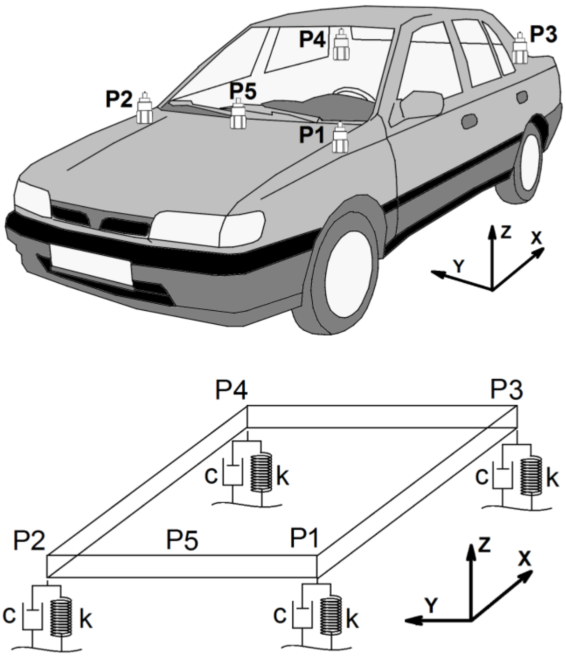

Fig. 3 Transducers localization (upper) and simplified dynamic model (lower)

3. RESULTS OF THE TEST

Vibration measurements were performed for five scenarios:

- all dampers undamaged
- one damper damaged (rear right)
- two dampers damaged (both rear)
- two diagonal dampers damaged (rear right and front left)
- all four dampers damaged

It was assumed that for initial measurements fully damaged absorbers should be tested. The problem with partial damage is that the damping is not a linear function of oil loss. Shock absorbers were twin-tube Magnum dedicated to Nissan Sunny.

The "damage" was done by drilling a hole in the damper to remove the oil. The EUSAMA values for the damaged shock absorbers were: 0; 4; 14 and 17 % (rear left; rear right; front left; front right). For all conditions the measurement was performed on the same road, 5 minutes of driving. The mass of the car was always the same. During each measurement only the driver and the measuring system operator were in the car.

a)

b)

Fig. 4. Location of the accelerometer near the shock absorber - a) front left, b) rear right

Acceleration was recorded without any integration, HP[2] or any other filter applied. The only exception was an anti-aliasing filter, which is indispensable. The signal was recorded with sampling at 1024 Hz (400 Hz frequency band). The natural frequency of the sprung mass in a car is very low (approx. a few Hz) and so a wide frequency range of the measurement system is not necessary. The low frequency characteristic of the transducers is more important. The laboratory accelerometers (PCB 356A02) chosen for this research have a flat amplitude vs frequency characteristic (0.1 dB deviation) from 0.5 Hz to several kHz.

After recording, the signals were post processed. For each 5-minute drive the linear average FFT spectra were calculated with a frequency span of 40 Hz and a delta F 100 mHz. The graphs are presented in fig 5. The maximum values are listed in table 1. As can be seen in the graphs, the maximum values were for the Z direction. It can be observed for that direction that there was resonance at approx. 2 Hz that shifted to approx. 1.7 Hz when the dampers

[2] HP - High Pass filter

were damaged. It can also be observed that for first condition the resonance was strongly dampened.

Table 1. FFT max vibration values of the first mode

Direction	Point	All dampers good		Left rear broken		Both rear broken		Front right and rear left broken		All broken	
		f [Hz]	a [m/s²]	f [Hz]	a [m/s²]	f [Hz]	a [m/s²]	f [Hz]	a [m/s²]	f [Hz]	a [m/s²]
X	P1	2.10	0.07	1.80	0.08	1.70	0.13	1.70	0.10	1.70	0.20
	P2	2.10	0.07	1.80	0.08	1.70	0.14	1.70	0.10	1.70	0.22
	P3	2.10	0.08	1.80	0.10	1.70	0.19	1.70	0.14	1.70	0.30
	P4	2.10	0.10	1.80	0.12	1.70	0.24	1.70	0.16	1.70	0.38
Y	P1	1.40	0.07	1.60	0.08	1.60	0.08	1.70	0.08	1.40	0.21
	P2	1.40	0.07	1.60	0.09	1.60	0.07	1.70	0.13	1.40	0.17
	P3	1.40	0.08	1.60	0.09	1.60	0.10	1.70	0.16	1.40	0.14
	P4	1.40	0.09	1.60	0.09	1.60	0.11	1.70	0.14	1.40	0.16
Z	P1	2.20	0.19	1.80	0.20	1.70	0.17	1.70	0.37	1.40/1.70	0.80/0.42
	P2	2.20	0.18	1.80	0.17	1.70	0.20	1.70	0.23	1.40/1.70	0.78/0.40
	P3	2.20	0.23	1.80	0.25	1.70	0.73	1.70	0.34	1.40/1.70	0.52/1.04
	P4	2.20	0.24	1.80	0.32	1.70	0.70	1.70	0.45	1.40/1.70	0.51/1.03
	P5	2.20	0.17	1.80	0.15	1.70	0.13	1.70	0.26	1.40/1.70	0.75/0.33

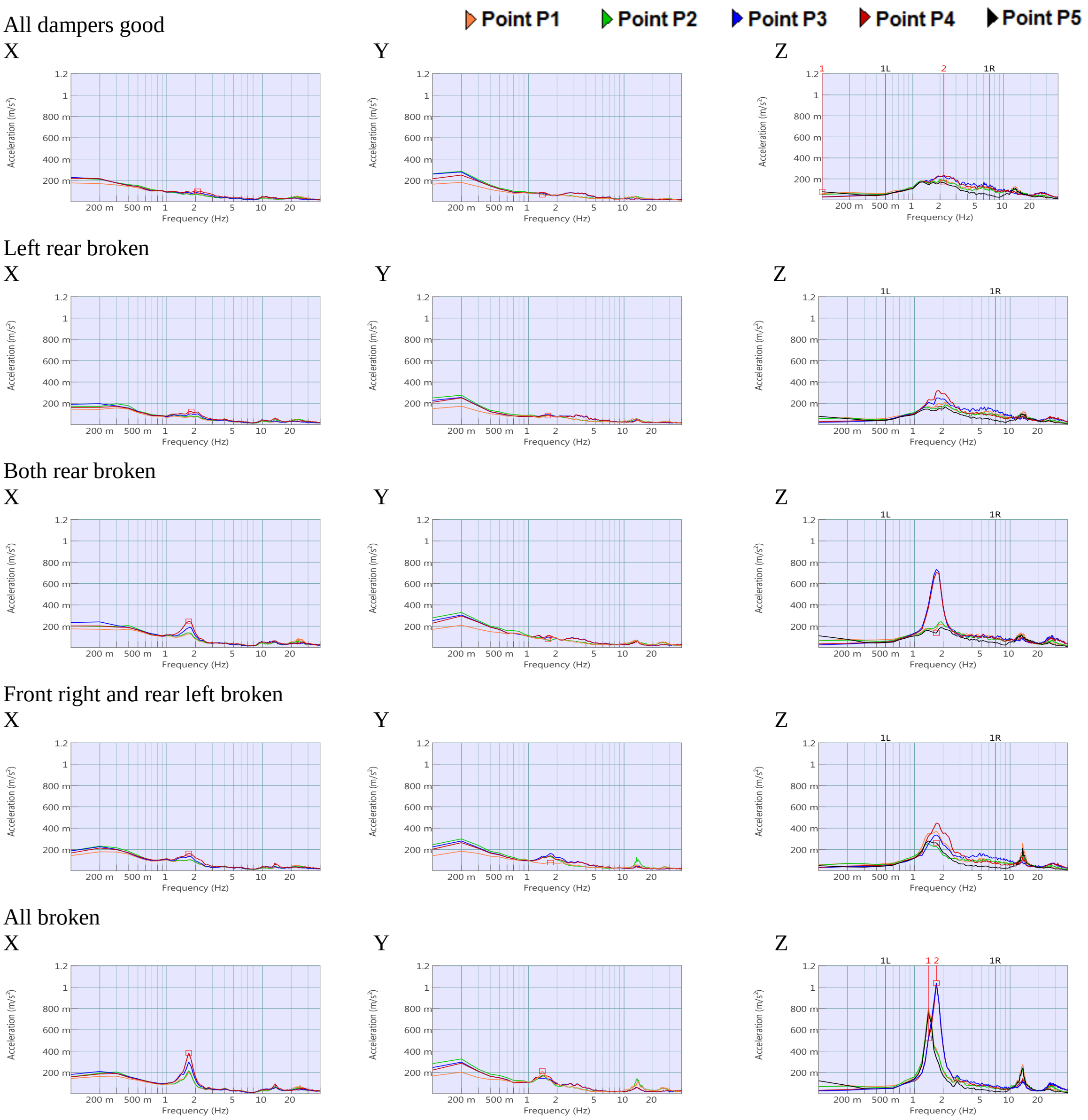

Fig. 5. FFT spectrums of 3 axial vibration for all conditions

Table 2. Change of dominating mode amplitude for direction Z (for frequency of mode)

Point	**Vibration amplitude change [%]**			
	One rear broken	**Two rear broken**	**One front and one rear broken**	**All broken**
P1	5.3	-10.5	94.7	321.1
P2	-5.6	11.1	27.8	333.3
P3	8.7	217.4	47.8	352.2
P4	33.3	191.7	87.5	329.2
P5	-11.8	-23.5	52.9	341.2

With the following conditions and with more dampers broken, damping was lower, which resulted in an increase of the amplitude of the narrow resonance peak. On graph for all dampers broken and direction Z, two high peaks may be observed at 1.4 Hz and at 1.7 Hz. The first mode was for the front suspension and second was for the rear. The rear frequency was higher because of the location of the center of mass. On the other graphs those two modes are visible, but the one related to the rear suspension is dominant.

For the most difficult to detect condition, when only one absorber was damaged, the highest difference of maximum FFT vibration (33.3%) was near the damaged damper (table 2 - P4). If more absorbers are damaged, the results are even better. This would be a success but vibration is strongly related to driving character, type of road, vehicle speed, total mass, roughness and many other factors. That is why a new calculation was used. To get rid of the effect that vibration values depend on the type of road and driving paramters, a new estimation was created: **Max value/band power.**

When the damper is broken, the damping of the system goes down and the amplitude of the resonance increases. So the aim was to find a function that would show the rise of the value in the resonance independent of the increase in total vibration (power band).
In FRF this is realized by the half power method. In this case, it was difficult to use that approach, because some approximation of curves was needed (involving too much error).

The results for that estimation are listed in table 3. The results are a little better for one damper broken. For other conditions this function decreases the results. But the main aim was to detect damage independently from driving conditions and not to improve the measurements themselves.

Cross spectra were prepared. A cross spectrum is function other than an FRF that gives phase information. The cross spectra were calculated for direction Z and for the following points:

- Both front points (point 1 and point 2),
- Both rear points (point 3 and point 4),
- One front point and one rear front (diagonal – point 1 and point 4),
- One front point and one rear front (diagonal – point 2 and point 3).

The cross spectrum graphs are presented in fig. 6 and values are listed in table 4. Two values were taken: the maximum value in the range of 0 – 40 Hz and band power of 0.5 – 7 Hz. Cross spectrum maximum values gave better results than the FFT. But the measurements were performed on the same road and with a very similar type of driving. Change of road, speed, weight and other factors would change the FFT values and the cross spectra even more. Under normal conditions a vehicle travels on many types of roads so the diagnostic must not be sensitive to this.

The biggest problem was for one damper broken, because damping loss and vibration increase were not significant. The highest increase was observed for spectra when one of the signals was from the accelerometer near the broken damper (point 4) - for cross spectra 3&4 and 1&4 the increase was approximately 50%. For 1&2 results were similar to those for all good dampers. For 2&3 results were a little lower (-16% approx.) than for all dampers good. The max value/power band estimator improved those results to 21%, 50%, 66% and -2%.

For both rear dumpers broken, as is reasonable, the highest increase was for cross spectrum 3&4 and the lowest for 1&2.

For condition when the rear right and front left shock absorbers were destroyed, the lowest increase (47%) was for cross spectra from points near the good dampers (2&3). The highest was for 1&4 damaged (327%). For two front and two rear points, results were near the average of the above (154% and 188%).

For all dampers broken results were very high (minimum 820% for 2&3 and maximum almost 2000% for 3&4). For all options, besides one damper broken, the max value/band power factor gave lower results. Nevertheless, they were still high.

According to the phase of the signals for the rear points, both were in phase and flat up to 4 Hz, which means that point 3 and 4 moved with the same sign at first resonance. For the front points the phase changed by approx. 180 degree at 0.4 Hz, but for the main resonance (1.4 - 2 Hz) there was no phase change either. On the graphs for the diagonal points there was some phase change because of the difference between the front and rear modes which was caused by the location of the center of mass being nearer the front axis.

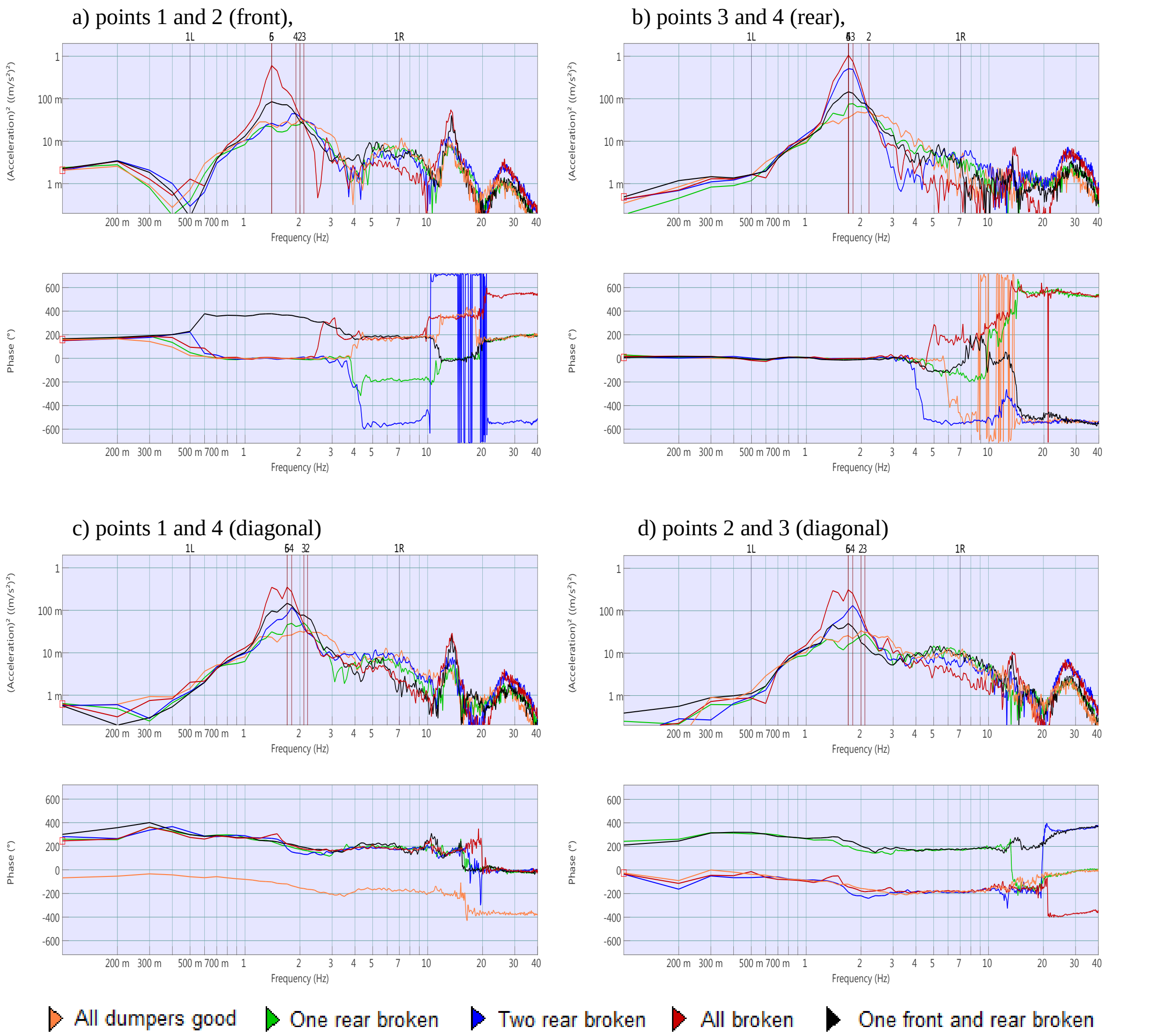

Fig. 6 Cross spectra magnitude (upper graph) and phase (lower graph)

Table 3. FFT amplitude divided by band power for direction Z

Condition	P1		P2		P3		P4		P5	
	max value / band power	Increase of max value / band power [%]	max value / band power	Increase of max value / band power [%]	max value / band power	Increase of max value / band power [%]	max value / band power	Increase of max value / band power [%]	max value / band power	Increase of max value / band power [%]
All dampers good	0.214	-	0.220	-	0.216	-	0.234	-	0.244	-
One rear broken	0.238	11.3	0.214	-2.8	0.240	11.0	0.323	37.9	0.251	3.1
Two rear broken	0.195	-8.9	0.237	7.6	0.492	127.2	0.490	109.3	0.208	-14.6
One front and one rear broken	0.356	66.1	0.278	26.2	0.314	45.0	0.375	60.2	0.367	50.4
All broken	0.596	178.5	0.596	170.7	0.578	167.2	0.594	153.9	0.650	166.6

Table 4. Cross spectra magnitude values

Cross spectrum	Condition	Freq Hz	max value (m/s²)²	**increase of max value**	band power (m/s²)²	max value / band power	**increase of max value / band power**
1 X 2	All dampers good	2.0	0.0335	-	0.4720	**0.071**	-
	One rear broken	2.1	0.0327	**-2.388%**	0.3810	**0.086**	**20.926%**
	Two rear broken	1.9	0.0446	**33.134%**	0.4040	**0.110**	**55.543%**
	One front and one rear broken	1.4	0.0851	**154.030%**	0.6180	**0.138**	**94.016%**
	All broken	1.4	0.6040	**1702.985%**	1.4850	**0.407**	**473.070%**
3 X 4	All dampers good	2.2	0.0505	-	0.6850	**0.074**	-
	One rear broken	1.8	0.0763	**51.089%**	0.6920	**0.110**	**49.561%**
	Two rear broken	1.7	0.5120	**913.861%**	1.8580	**0.276**	**273.786%**
	One front and one rear broken	1.7	0.1453	**187.723%**	0.9330	**0.156**	**111.243%**
	All broken	1.7	1.0600	**1999.010%**	2.8300	**0.375**	**408.064%**
1 X 4	All dampers good	2.2	0.0343	-	0.5880	**0.058**	-
	One rear broken	2.1	0.0506	**47.522%**	0.5240	**0.097**	**65.540%**
	Two rear broken	1.8	0.1173	**241.983%**	0.7500	**0.156**	**168.114%**
	One front and one rear broken	1.7	0.1463	**326.531%**	0.9450	**0.155**	**165.397%**
	All broken	1.7	0.3510	**923.324%**	1.5190	**0.231**	**296.125%**
2 X 3	All dampers good	2.0	0.0339	-	0.5990	**0.057**	-
	One rear broken	2.1	0.0285	**-15.929%**	0.5160	**0.055**	**-2.406%**
	Two rear broken	1.8	0.1331	**292.625%**	0.7480	**0.178**	**214.415%**
	One front and one rear broken	1.7	0.0500	**47.493%**	0.5770	**0.087**	**53.116%**
	All broken	1.7	0.3120	**820.354%**	1.4400	**0.217**	**282.842%**

4. UNCERTAINTY AND REPEATABILITY

Because research was done on one test track with one road profile the excitation of the system was strongly narrowed. On different road profiles such as gravel or on highway with higher speeds conditions would be much different. The research on different surfaces and different speeds are planned with the next research stage. The results of those measurements will allow wider uncertainty and repeatability analysis according to variable factors such as: surface, velocity, driving character and vehicle load.

For those measurements an estimated extended uncertainty (95% confidence level with coverage factor of 2) according to measurement system is 0,74 m/s2 (25% of values measured). In this uncertainty estimation was taken into consideration spread of the results for the track that measurements was done on.

5. CONCLUSIONS

Vibration measurements on a car body can help in the detection of failure of one or more dampers. With the measurement of the response only (without the excitation force), it is possible to find not only the fact that there is a problem with a shock absorber, but even which particular shock absorber is faulty. Other problems such as the breaking of a swingarm, spring or other subassembly of the suspension should also be detectable via this method.

The proposed method depends on many variable parameters such as: tire stiffness (winter/summer), the damping of tires material, suspension rubber elements stiffness, total load applied to the vehicle, etc. Additionally there is an influence of factors related to the absorbers directly: pressure in gas shock absorbers, oil condition related to the temperature and wear. Those factors influence significantly precision of the test method.

Cross-spectrum analysis gave better results than the simple FFT vibration analysis, but both are sensitive to changes in driving conditions. The maximum cross spectrum divided by band power in the range of 0.5 – 7 Hz improved results for one damper damaged (which was the most difficult condition to detect).

The reference suspension is not needed for evaluation of progressive dampers degradation. The long time averaged vibration spectrums of vehicle with undamaged shock absorbers are taken as the reference.

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Received 2016-05-23
Accepted 2016-11-21
Available online 2017-03-23

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