MAINTENANCE ON DEMAND FOR VEHICLE SUSPENSION SYSTEM

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Summary

This paper presents a method for evaluation health condition of car suspension. Research covered verification between 3D model from Virtual.Lab and mathematical models created in the Matlab/Simulink environment. All the analysis is performed for the full car model.

Created models are used for verification of suspension test in cars: EUSAMA and Minimum Phase Shift (MPS). Subsequently they are simulated in order to present behavior of degrading suspension with gamma distribution.

Research was performed under Maintenance on Demand project with cooperation with LMS International, DHL, Volvo.

Keywords: Structural Health Monitoring, vehicle suspension, mathematical car model.

MONITOROWANIE STANU ZAWIESZENIA W POJAZDACH

Streszczenie

W artykule przedstawiono wyniki prac nad wyznaczaniem stanu amortyzatorów w zawieszeniu samochodów osobowych. Badania obejmowały weryfikację modeli 3D w programie Virtual.Lab z modelami matematycznymi wykonanymi w środowisku Matlab/Simulink. W pracy przedstawione będą wyniki analiz uproszczonego modelu samochod.

Opracowane modele posłużyły w dalszej części do testowania metod używanych do sprawdzania stanu zawieszenie w pojazdach: EUSAMA i Minimum Phase Shift (MPS). Następnie prezentacji ich zachowania dla degradowanych amortyzatorów przy użyciu dystrybucji gamma.

Prace realizowane były w ramach projektu Maintenance on Demand przez m. in. LMS International, DHL, Volvo.

Słowa kluczowe: Monitorowanie stanu konstrukcji, zawieszenie pojazdu, matematyczny model pojazdu.

1. INTRODUCTION

Machine condition monitoring and fault diagnostic is a technical activity in which selected physical parameters are observed in order to determine machinery integrity. Once it is diagnosed, this information can be used for maintenance activities. The ultimate goal in case of maintenance is optimum use of resources [1].

Machine failure can be determined when machine exceeds operational limit which is designed for safe operation. As machine gradually reaches the end of its designed lifetime, the frequency of failures increases. It is connected with degradation of the parts, metal fatigue, wear mechanisms between moving parts, corrosion. Such failures are called "wearout".

Maintenance strategies can be divided into types:
- run-to-failure – also called breakdown maintenance or corrective maintenance is a strategy in form of repair work or replacement if and only if machinery, object has failed. This type of maintenance can be used if the equipment is redundant, spare parts are cheap or the repair/replacement is quick [1],
- scheduled maintenance – also called planned maintenance is any scheduled service to ensure that machinery is working within operational limits [2],
- condition-based maintenance – is a strategy requiring assessing the actual condition of the machinery thus machinery could be corrected at the right time [2].

2. FULL CAR MODEL

2.1 Mathematical model

Mathematical model was created in Matlab/Simulink software. As illustrated in fig. 1 model has 7 degree-of-freedom: main body (chassis – can rotate along the pitch and roll poles, and translate along z axis) is connected to four unsprung masses (wheel hubs) located at each corner, subsequently front-left (fl), front-right (fr), rear-left (rl), rear-right (rr). Hubs translate vertically with respect to the sprung (main body). Suspensions situated between sprung and unsprung masses are modeled as linear viscous dampers and spring with constants coefficients. In case of tire, simple linear spring with constant coefficient is used [3]. For simplicity, small pitch and roll angles are assumed, which results in linearization of sin(φ) to φ.

Equations describing 7-DOF vehicle with passive suspension:

- Sprung mass (chassis) motion:

$$M_{ch}\ddot{Z_{ch}} = F_{sfl} + F_{dfl} + F_{sfr} + F_{dfr} + F_{srl} + F_{drl} + F_{srr} + F_{drr} \tag{1}$$

M_{ch} is a chassis mass, $\ddot{Z_{ch}}$ vertical acceleration along z axis. Each corner is represented by pair of forces, one is a spring force while second comes from damper. For example F_{sfl} is a spring force and F_{dfl} is a damper force, both located at the front left corner of the vehicle.

$$I_{pitch}\ddot{\theta} = (F_{sfl} + F_{dfl} + F_{sfr} + F_{dfr})lL + (F_{srl} + F_{drl} + F_{srr} + F_{drr})lR \tag{2}$$

I_{pitch} is a moment of inertia about the pitch center (y axis), $\ddot{\theta}$ corresponds to pitch acceleration, lF and lR are distances between center of gravity and front axle, rear axle.

$$I_{roll}\ddot{\varphi} = (F_{sfl} + F_{dfl} + F_{srl} + F_{drl})\frac{wC}{2} - (F_{sfr} + F_{dfr} + F_{srr} + F_{drr})\frac{wC}{2} \tag{3}$$

I_{roll} is a moment of inertia about the roll center (x axis), $\ddot{\varphi}$ corresponds to roll acceleration and wC is a model width (it is assumed that center of gravity is located on the roll center) .

- Unsprung mass (wheel) motion:

$$M_{WhFL}\ddot{Z_{FL}} = F_{tireFL} - F_{sfl} - F_{dfl} \tag{4}$$

$$M_{WhFR}\ddot{Z_{FR}} = F_{tireFR} - F_{sfr} - F_{dfr} \tag{5}$$

$$M_{WhRL}\ddot{Z_{RL}} = F_{tireRL} - F_{srl} - F_{drl} \tag{6}$$

$$M_{WhRR}\ddot{Z_{RR}} = F_{tireRR} - F_{srr} - F_{drl} \tag{7}$$

M is a mass of a wheel hub in particular corner of chassis, Z corresponds to vertical acceleration and F_{tire} is a tire force.

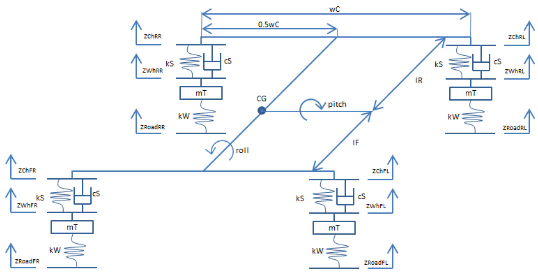

Fig. 1. Full car –mathematical model

2.2 Virtual.Lab model

CAD model was created in the LMS Virtual.Lab software. Idea behind that was to check and validate results from that model with the one designed using mathematical equations in Matlab environment. Obtaining the same results, could significantly reduce time needed to create and computer simulation (as it is faster to create a model in CAD environment instead of developing equations).

In order to obtain reliable outputs from the simulations, simplified full car model in Virtual.Lab was created as close as possible to its mathematical version. Main body (chassis) was constrained with use of bracket joint (special type of joint which gives user opportunity to constrain body between 6 to 0 DOF), so it could translate along z axis and rotate along x and y axis, while wheel hubs were constrained with translational joint (allows translation only along one axis).

As it was in case of Matlab, CAD models had the same type of suspension (with linear viscous damper with linear spring) and tire (linear spring, no damping), they were created by TSDA force (Translational – Spring – Damper - Actuator) [4].

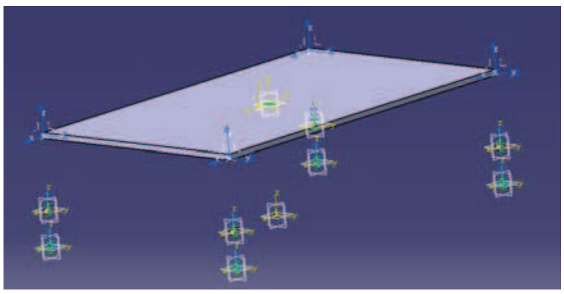
Fig. 2. Full car – Virtual.Lab model

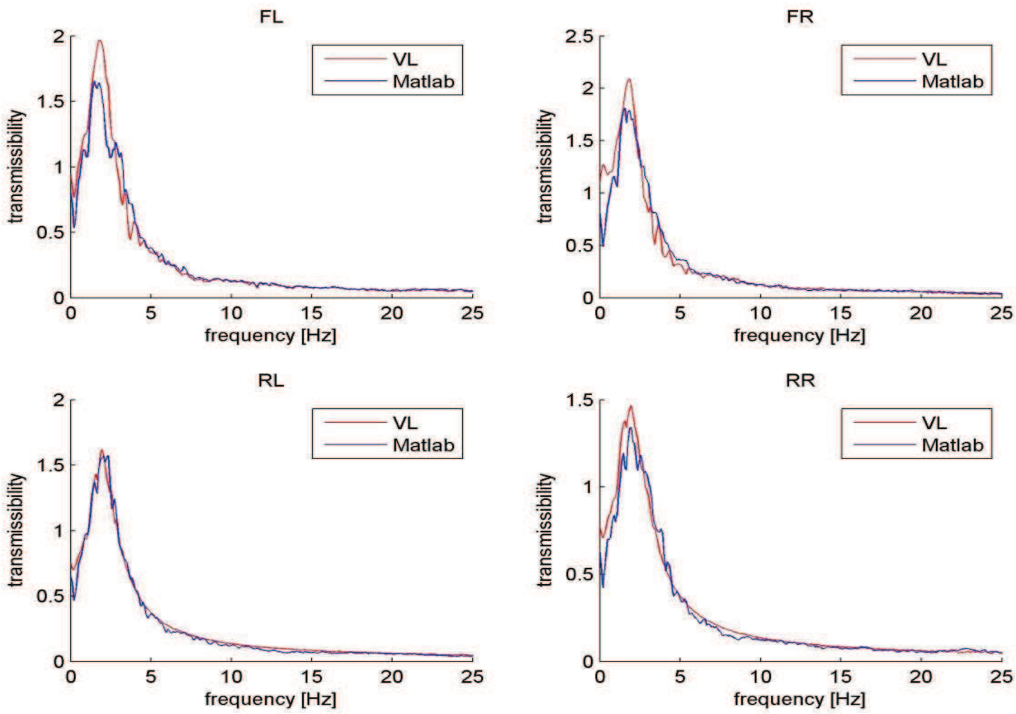

Fig. 3. Full car simulation comparison

2.3 . Results validation between mathematical and Virtual.Lab model

Both models (from VirtualLab and Matlab) were simulated with the same real road data, which was measured by LMS International equipment installed on the car - Mercedes C350.

Full car model parameters used in simulation are provided in Table 1. CAD model shows some differences compared to the mathematical one (fig. 3). Apart from the difference in magnitude, Matlab and VL results are slightly shifted along the frequency axis. Such a phenomena can be explained by the errors in roll and pitch rotation. Unfortunately it was impossible to eliminate yaw.

Table 1. Car parameters for Matlab and VL

mT = 70; % hub mass mC = 1640; % car mass kW = 240000; % tire stiffness kS = 64000; % spring stiffness cS = 700; % damping wC = 1.6; % car width lR = 1.4904; % distance from rear to CG lF = 1.2696; %distance from front to CG Ir = 540; % roll inertia Ip = 2800; % pitch inertia

3. SHOCK ABSORBER TEST

3.1 EUSAMA test

EUSAMA (European Shock Absorber Manufacturers Association) evaluated test for determination suspensions' condition [5] Adhesion (EUSAMA value) is a minimal percentage of instantaneous remnant vertical tire contact force between the tire and the road surface.

$$EUSAMA = \frac{F_{min}}{F_{static}} * 100\% \qquad (8)$$

During measurement each axle is tested separately on shaker/test bed. First equipment weights static load of each wheel, then plate of a shaker vibrates with chirp sinus starting from 0Hz to 25Hz and 6mm stroke. Minimum EUSAMA value (adhesion) occurs at the resonant frequency of the unsprung mass (between 10– 20Hz, depending on parameters of a suspension).

Table 2. EUSAMA value interpretation [5]

EUSAMA value	Interpretation
61% - 100%	Excellent dynamic wheel contact
41% - 60%	Good dynamic wheel contact
21% - 40%	Poor dynamic wheel contact
0% - 20%	Damper is classified as broken

Damper is classified as broken when the EUSAMA value is lower than 21% (Table 2) or difference between left and right wheel for axle is bigger than 50% [5].

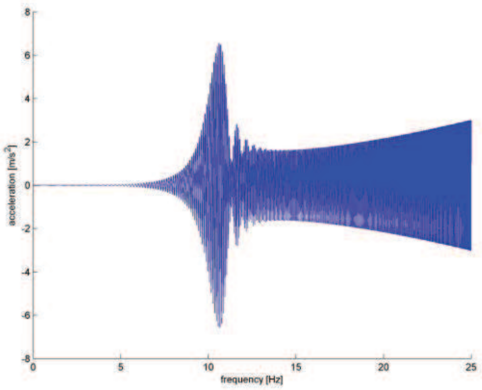

Fig. 4 Wheel hub response

3.2. Minimum phase shift test

Minimum phase shift test is introduced by GOCA (Belgian vehicle inspection) [6], because EUSAMA was not reliable for super light vehicles, vehicles with run flat tire, and it could have happened that heavy vehicles despite a bad working shock absorber pass the EUSAMA test [6].

Phase shift is an angular difference between the sinusoidal position of the suspension tester platform and the sinusoidal vertical tire contact force between the tire and suspension tester platform. This test uses the same measurement equipment and input chirp signal (sinus signal with amplitude 3mm and frequency varying from 0 to 25Hz) as it is in case of EUSAMA test [6].

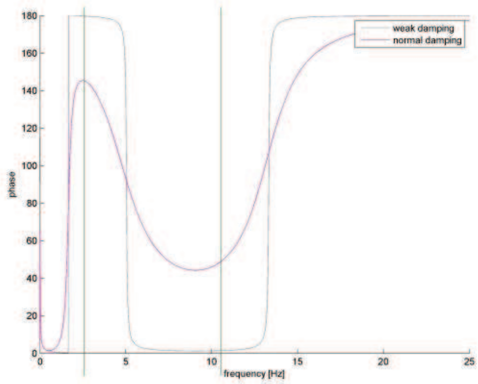

Fig. 5 Minimum phase shift – ideal

Resonant frequency of the sprung mass is located between 1 and 3 Hertz. If no damping occurs, the phase angles are equal to:

$$\varphi_{23} = \varphi_2 - \varphi_3 = 180^0 \tag{9}$$

$$\varphi_{13} = \varphi_1 - \varphi_3 = 0^0 \tag{10}$$

$$\varphi_{12} = \varphi_1 - \varphi_2 = -180^0 \tag{11}$$

where:

φ_1 – phase angle of sprung mass
φ_2- phase angle of unsprung mass
φ_3- phase angle of suspension tester platform/road
φ_{12}- phase shift between φ_1 and φ_2 (transmissibility phase)
φ_{13}- phase shift between φ_1 and φ_3
φ_{23}- phase shift between φ_2 and φ_3

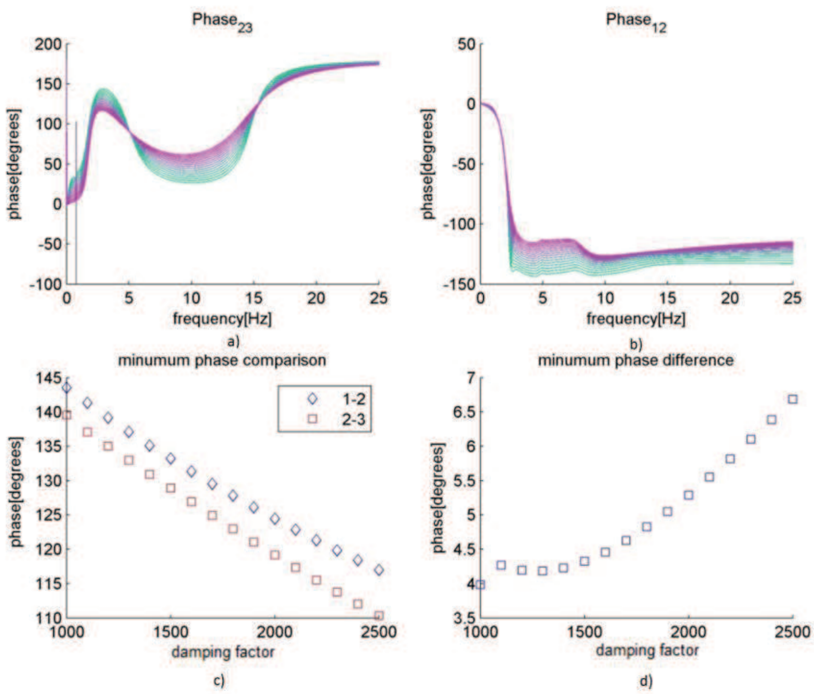

Fig. 6 Phase difference – full car model (front left wheel) – a) phase shift between $\boldsymbol{\varphi_2}$ and φ_3, b) phase shift between φ_1 and φ_2, c) comparison between minimum phase shift values for $-\boldsymbol{\varphi_{12}}$ and $\boldsymbol{\varphi_{23}}$ with respect to damping factor, d) difference between minimum phase shifts $-\boldsymbol{\varphi_{12}}$and $\boldsymbol{\varphi_{23}}$ with respect to damping factor

At that frequency, relative displacement between sprung and unsprung mass is maximum. Increasing damping factor results in decreasing mentioned distance and increasing φ_{23} [7]

Resonant frequency of sprung mass depending on the parameters varies between 10 and 20 Hertz.

The phase angle between indicates the strength of the shock absorber in the vehicle. Term “minimum phase shift” corresponds to the lowest value of the phase angle between sprung and unsprung mass. Corresponding to the criteria introduced by GOCA, shock absorber is classified as broken if that value is lower than 40° [6].

3.3. Relations between transmissibility and minimum phase shift test

Fig. 6a and 6b presents phase shifts evaluated for $\varphi_{23}$and φ_{12}.

The transmissibility calculated between wheel hub and chassis cannot be directly connected to the minimum phase shift criterion. However according to the equations (eq. 9, eq. 10, eq. 11), some similarities can be found. Minimum value of transmissibility between 1Hz and 3Hz should be exactly opposite to the phase shift between wheel hub and tester platform within the same frequency range (fig 6c).

As it is illustrated in fig. 6d the difference between phase shift and transmissibility from 1Hz to 3Hz is equal up to 10 degrees. In most cases it can be expressed in form of quadratic or cubic function. The error between those two quantities could be even assumed to be zero, for simplicity.

Unfortunately there is no easy and clear way how to move from transmissibility value of phase shift from 1Hz to 3Hz, to minimum phase shift value.

4 ROAD PROFILE

Road surface is considered as random surface. As with any random signal such profile can be represented with use of Fourier transform into sum of sine waves with varying amplitudes and phase shifts. A plot of amplitude with respect to the frequency can be represented in form of power spectral density.

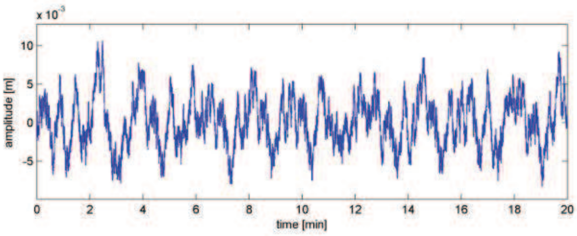

Fig. 7 Road signal (road class A – highway), dt = 200ms

It is known that PSD of the roughness of a road profile has the decreasing tendency with increasing wavelength. In the table below one can see the approximations of PSD, obtained by fitting the experimental data [8].

Table 3. PSD approximation – analytical [8]

Name	PSD approximation	Wavenumber
ISO (1995-09-01)	$G_d(n) = Cn^{-w}$	$0 \le n \le \infty$
BSI (1972)	$G_d(n) = \begin{cases} Cn^{-w_1} \\ Cn^{-w_2} \end{cases}$	$0 \le n \le n_0$ $n_0 \le n \le \infty$
Two Split	$G_d(n) = \begin{cases} Cn^{-w_1} \\ Cn^{-w_2} \\ Cn^{-w_3} \end{cases}$	$0 \le n \le n_1$ $n_1 \le n \le n_2$ $n_2 \le n \le \infty$
Sayers (1986)	$G_d(n) = \frac{C_1}{n^4} + \frac{C_2}{n^2} + C_3$	$0 \le n \le \infty$
Gillespie (1985)	$G_d(n) = \frac{C(1 + (\frac{0.066}{n^2}))}{n^2}$	$0 \le n \le \infty$
Marcondes et al. (1991)	$G_d(n) = \begin{cases} C_1 \exp(-kn^p) \\ C_2(n - n_0)^q \end{cases}$	$0 \le n \le n_0$ $n_0 \le n \le \infty$
Sussman (1974)	$G_d(n) = \frac{C}{\alpha^2 + n^2}$	$0 \le n \le \infty$
Macvean (1980)	$G_d(n) = \frac{C}{(\alpha^2 + n^2)^2}$	$0 \le n \le \infty$
Xu et al. (1992)	$G_d(n) = \frac{A}{2\alpha \exp(\frac{-n^2}{2\alpha^2})}$	$0 \le n \le \infty$
Kozin and Bodanoff (1961)	$G_d(n) = \frac{A}{\alpha \exp(\frac{-n^2}{\alpha^2})}$	$0 \le n \le \infty$

Note: C, C1, C2, C3 – unevenness index, ω – waviness of the road surface, p, k, q, α and β are real positive constants.

MoDe project is focused on frequency range between 0.2 Hz and 20Hz, which is normally considered in vehicle dynamics for comfort, safety and road holding analyses as ride. Mentioned frequency corresponds to the wavelength from 100m to 1m. However for wavelengths higher than 100m PSD function become flat. It means that road input excitation is not a white noise, but Brownian or red one [8].

International Organisation for Standardization introduced road roughness divided into eight classes

(ISO 1995-09-01, starting with A – highway and ending on H – offroad).

ISO standards suggest w = 2 for road undulations, for disturbances with wavelength greater than 6 meters, and w = 1.37 for irregularities with a wavelength smaller than 6 meters [8].

Road input with desired ISO class was generated in time domain and low pass filtered for frequencies below 25Hz. Graphs of estimated PSD, PSD of generated signal before and after filtering, and PSD of real measured signal are plotted on fig. 8.
Fig. 8 represents comparison between four PSD.
- ideal PSD – corresponds to the PSD approximation according to ISO standard
- raw PSD – PSD of artificially generated road signal
- filtered PSD – raw PSD + low pass filter (25Hz)
- measured PSD – PSD from measured road signals on Mercedes C350.

Table 4 ISO – PSD approximation coefficients [8]

Road class	C factor
A	1.6e-7
B	6.4e-7
C	2.56e-6
D	1.024e-5
E	4.096e-5
F	1.6384e-4
G	6.5536e-4
H	2.62144e-3

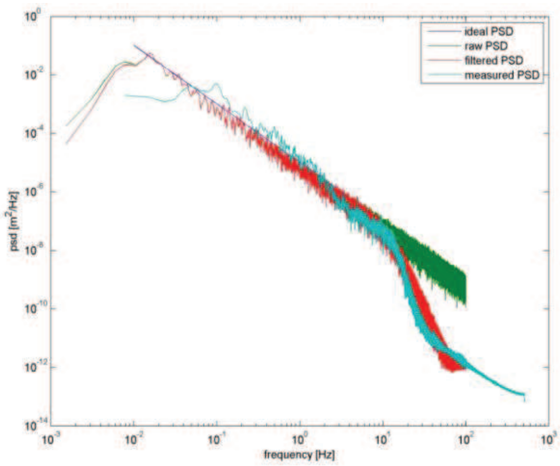

Fig. 8 Comparison between ideal, raw, filtered and measured PSD

5. GAMMA DEGRADATION

The gamma process is ideally suited to model graduate damage that monotonically accumulates over time, such as wear, corrosion, erosion, and creep of materials, which are common causes of failure of engineering components [9]

However two types of gamma degradation models can be distinguished:
- Threshold model supposes that the item or system fails whenever its degradation level reaches a certain critical deterministic or random threshold [9].
- Shock model supposes that the item or system is subjected to external shock which may be survived or lead to failure. The shocks usually occur according to Poisson process whose intensity depends on degradation and environmental factors [9].

It is possible to combine those two models into one (as Lemoine and Wenocur suggested) and obtain degradation – threshold – shock model (DTS model).

6. DEGRADATION DATA – SIMULATION (GUI)

6.1. Road profile

At the beginning of the program user loads a symbolic road profile from the text file (*.txt). File should consist of 3 columns and minimum one row.

Table 5 Symbolic road profile 1

B	50	20
A	120	20
C	50	10
D	30	5
B	30	15
E	35	5

First column is a road class (according to ISO standards – letter from A to H), second one is an average speed of the vehicle (km/h) while last one time (min).

For example (generated road profile is illustrated in fig. 9 It is assumed that road profile is the same for the side wheels. However signal between front and rear is shifted by the time delay derived from average velocity and distance between wheels. Generated profile is assumed to have constant surface condition of dry road without any influence of snow or rain.

6.2. Transmissibility

After road profile is generated, program calculates damping degradation (starting from healthy to worn damper with previously determined damping factors) for each wheel separately. Subsequently full car Matlab model is simulated n times, (where n is a number of measurements) with road signal divided into equal length parts each with damping changing damping factor.

6.3. Minimum transmissibility

As an output from simulation, program calculates the minimum values of each transmissibility phase (between 0Hz and 25Hz). Comparing to the transmissibility amplitude, the minimum phase is monotonically decreasing with dropping damping factor.

Normally the minimum value of phase transmissibility should be located between 1Hz and 3Hz, but it can move on the frequency axis when influenced by other shock absorbers.

Script generates random road signal according, and subsequently calculates phase shifts (fig. 11).

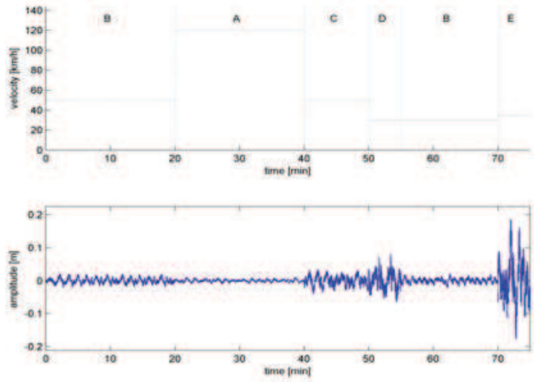

Fig. 9 Generated road profile

Result of minimum phase values after simulation with 400 measurement points are shown in the plot fig 10.

It is easy to notice that main assumption about degradation is satisfied – the smallest value of the phase shift between wheel and chassis is always decreasing.

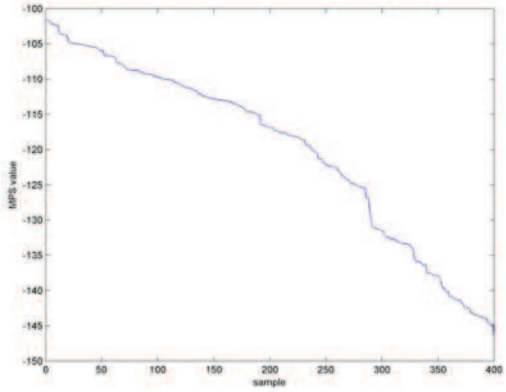

Fig. 10 MPS value during the simulation

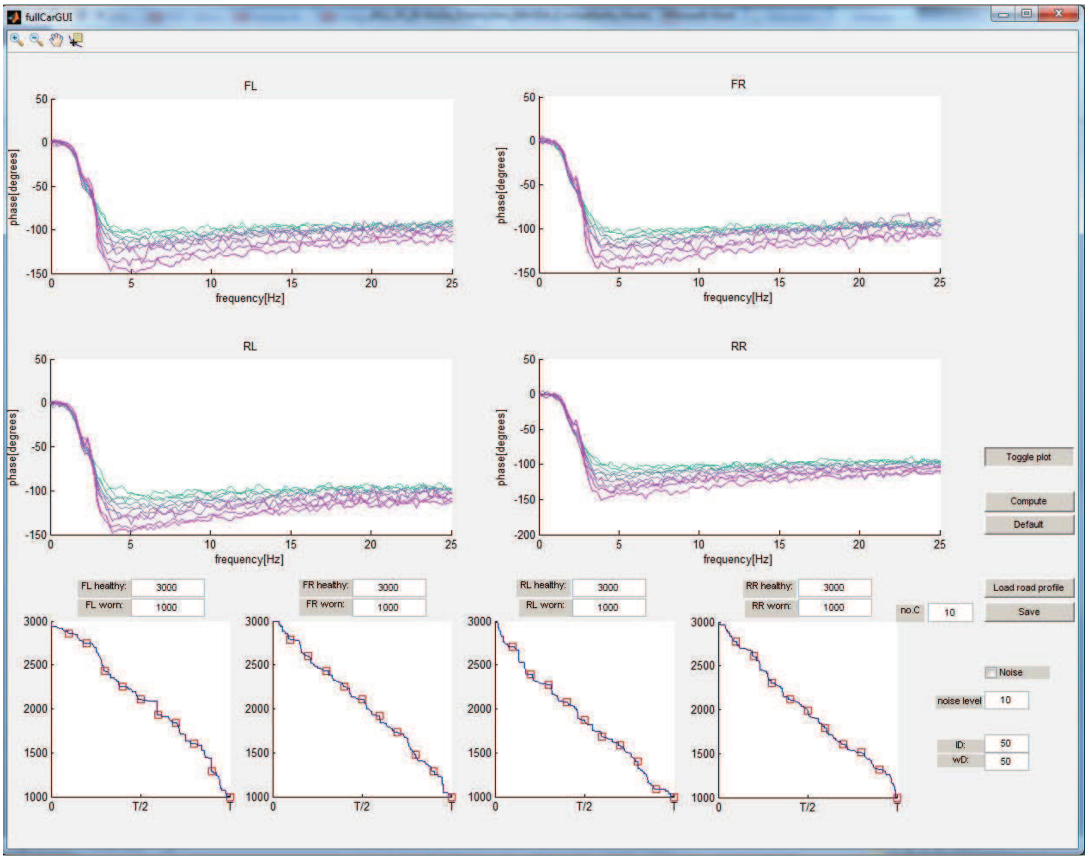

Fig. 11 Degradation data simulation (GUI)

7. CONCLUSIONS

Project originally was created for Volvo trucks, however all the measurement were performed on the Mercedes C350, which means that all the steps made so far, should be repeated with focus on a truck model.

After validation between Matlab and Virtual.Lab environment results were still different.

It is necessary to point out that output data from simulation is the same from those two programs in case no rotation.

Further work should be focused on evaluating gamma degradation based on real data from the manufactures.

Eventually, minimum value of transmissibility ought to be connected to the EUSAMA or minimum phase shift method to determine if damper is healthy or broken.

8 BIBLIOGRAPHY

[1] Mechefsk C. K., *Machine Conditioning, Monitoring and Fault Diagnostics*, Vibration and Shock Handbook, CRC Press, 2005, pp. 25-1 – 25-9.

[2] Randall R. B., *Vibration-based Condition Monitoring: Industrial*, Aerospace and Automotive Applications, Wiley, 2011, pp. 1-23.

[3] Abramov S., Durieux O., Mannan S.: *Semi-Active Suspension System Simulation Using SIMULINK*, Int. J. of Engineering Systems Modelling and Simulation – Vol. 1 No. 2/3, 2009, pp. 101-114.

[4] LMS Virtual.Lab on-line help, Revision 9-SL3, 2010.

[5] Tsymberov A.: *An Improved Non-Intrusive Automotive Suspension Testing Apparatus with Means to Determine the Condition of the Dampers*, SAE Technical Paper 960735, 1996

[6] Buekenhoudt P.: *Minimum phase shift*, CITA International Motor Vehicle Inspection Committee Conference and 15th General Assembly, 2011.

[7] Goktan A. G., Yetkin A., *Road Load Data Estimation on Multiaxial Test Rigs for Exhaust System Vibration*s, SAE Technical Paper 2002-01-0805, 2002

[8] Soria L., delli Carri A., Peeters B., Anthonis J., Van der Auweraer H.: *Active Suspension Systems for Passenger Cars: Operational Modal Analysis as a Tool for the Performance Asseement*, Modal Analysis Topics, Volume 3, Springer New York, 2011, pp. 313-323.

[9] Lehmann A.: *Failure Time Models Based on Degradation Processes*, Advances in Degradation Modeling, Birkhäuser Boston, 2010, pp. 209-231.

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