Testing and Simulation of a Motor Vehicle Suspension

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Abstract: - The paper presents experimental results on modelling and simulation of vehicle suspension. Testing the performance practice of suspension dampers is performed using a device called Spider8, which is achieved by a mathematical model to assess the behaviour and performance suspension behaviour experimentally tested through simulation environment under MATLAB/ Simulink. From the experimental data damper, model coefficients are determined suspension. Simulation results are compared with experimental data, after which the developed model is validated.

Keywords: - Testing suspension shock, Modelling and simulation.

I. INTRODUCTION

A long time, efforts have been made to make the suspension system to function optimally by optimizing the parameters of the suspension system, but to limit internal passive suspension system the improvement is effective only in a certain frequency range. Compared with passive suspension, the active suspension can improve the performance suspension system on a comprehensive frequency. Semiactive suspensions were proposed in the early 1970's [1] and can be almost as effective as fully active suspension in order to improve the quality of vehicle behavior. When the control system fails, the semiactive suspension may continue to operate on a passive condition. Compared with active and passive suspension systems, the semiactive suspension system combines the advantages of both active and passive suspensions. It provides better performance compared with passive suspension and it's economic, safe, and does not require any high power components or an electrical power source of high power [2]. To the semiactive suspension the adjusting the damping force can be obtained by adjusting the orifice area from the damper passages for filling oil, and so the fluid flow resistance changes. In order to determine the dynamic response of the suspension of a road vehicle (type Dacia Logan) it has been fitted with dampers provided with strain gauges. It was conceived and carried out the testing of automotive suspension, which was conducted estimating parameters of the mathematical model. An experimental test mechanism is mounted to determine the functional properties of the suspension, but also to obtain data necessary to estimate the dynamic model parameters. The damper car (Dacia Logan) is fixed in a mechanism for automated testing (Spider8). In order to determine the dynamic response suspension, front suspension has been equipped with dampers fitted with brand tens metric. To simulate the behaviour of the suspension of motor vehicles under the control of vibration there has been developed a model that more faithfully reproduces the actual behaviour. Simulation results in MATLAB /Simulink based on the mathematical model developed are compared with the experimental data. The comparison made validate the parameters measured in the phase of testing suspension. Vibration control of vehicle suspension system has been an active subject of research because it can ensure a better performance for comfort and safety.

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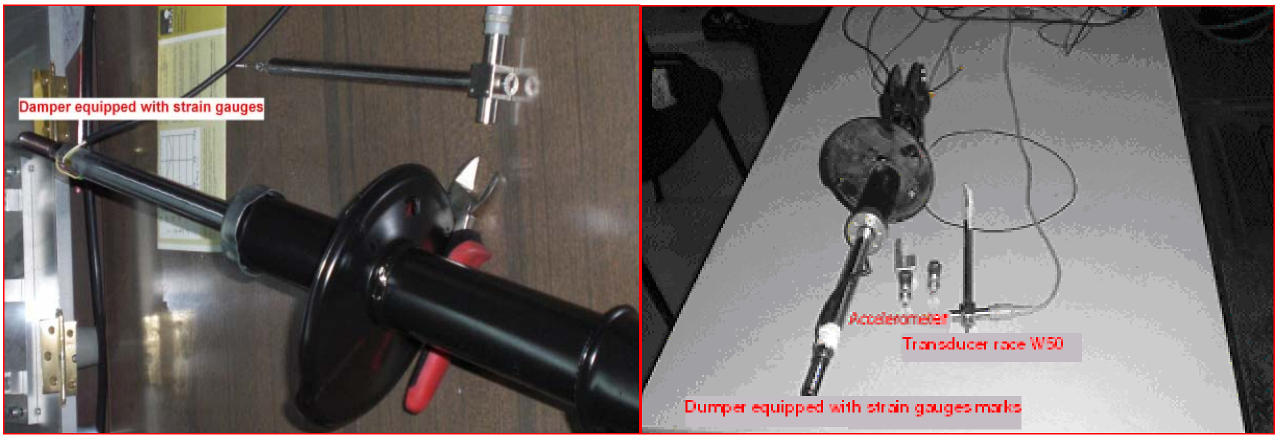
II. EXPERIMENTAL DETERMINATION OF STATIC RESPONSE OF SUSPENSION

A. Description of Device Spider 8. Spider 8 is an electronic system for measuring the number of analog data, digital specialised for the purchase of mechanical quantities such as forces, mechanical tension, pressure, acceleration, speed, movement and temperature [11]. The device is equipped with means of measuring analog voltage signals, which allows measurement of any parameters, if there is an interconnection system as a system transducer-signal conditioner to convert the voltage signal of that parameter. Spider8 connects to computer via parallel port RS232, or USB. The acquisition system includes embedded specialized modules for mechanical measurement of certain sizes. Each channel has its own measurement converter A/D (analog/digital), which can be set to the sampling frequency in the range 0.1...9600 samples/second. Converters working in parallel are synchronized by the measure, giving the possibility of the simultaneous acquisition on these 8 channels [3], [4], [6].

B. Damper Equipped with Strain Gauges(Fig. 1). To determine the characteristic of strength, it was chosen the option to direct determination of the force of the rod damper developed by conducting an assembly with four strain gauges type LY5mm/120 Ohm applied rod actuators, approx. 5 mm superior to the shoulder. Calibration, as the force transducer of the damper was done on the Universal Testing Machine Mechanical of endowment in the Faculty of Electric Engineering, Environmental and Industrial Science of University of Craiova, the transducer using force Hottinger type U2B 10 kN. Calibration was done in the field ±1500 N, with the direct comparison method for both areas (tensile and compressive), applying forces known and measured with the transducer Hottinger type U2B 10kN. During calibration both transducers were coupled to the transducer measurement location was chosen as the optimum measurement parameters for suspension car features [1]:



a) b)

Fig. 1: Explanation of the experimental static response of the suspension: a) a silencer-equipped with 4 strain gauges in full bridge for measuring tensile force / compression; b) Measuring transducers.

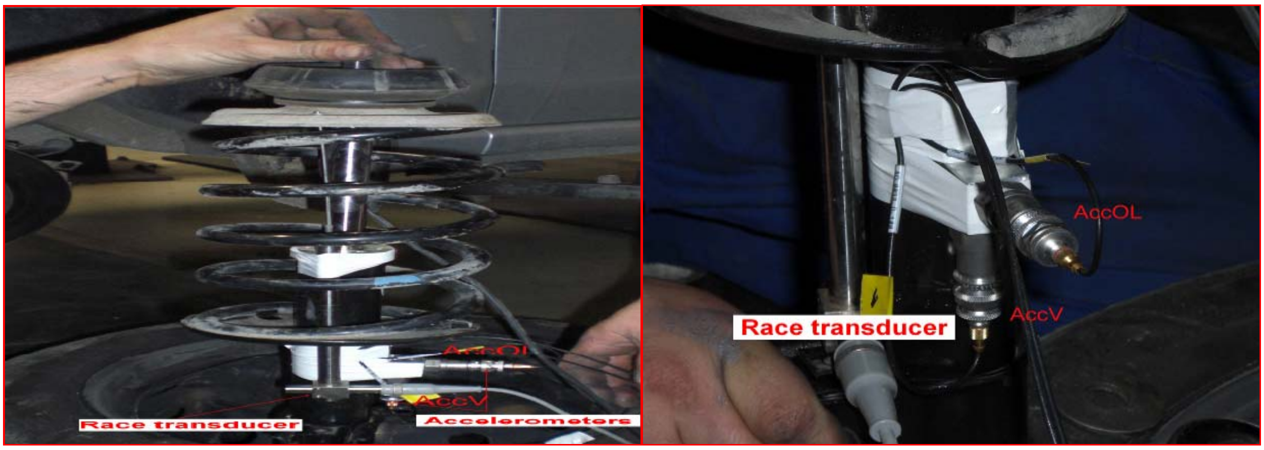
acquisition Spider8. From the static characteristic resulted after the experimentation there can be noticed that the ± 1500 N, the feature is perfectly linear, its pant being 50.65 N/mV.

III. EXPERIMENTAL DETERMINATION OF SUSPENSION DYNAMIC RESPONSE

A. Performing Tests. In order to determine the dynamic response of the suspension and the damper, the front suspension of a Logan car was equipped with a silencer with strain gauges mark, previously presented.
Equipment and measuring transducers used are: the purchasing system Spider 8, signal conditioner NEXUS 2692-A-0I4, accelerometer Bruel & Kjaer type 4391 (3 pcs.) Inductive transducer of linear W50 race, equipped

- Force developed in the damper rod (measured with strain gauges mounted on the rod);
- The damper race (used the transducer W50 positioned parallel with damper rod through the arch of suspension);
- Acceleration in the vertical direction of the front axis (it was used the accelerometer AccV mounted on the casing of the damper on vertical direction);
- Horizontal acceleration - longitudinal of the front suspension axle (along the car axis, was used the accelerometer AccOL mounted on the damper carcass on horizontal-longitudinal direction);
- Acceleration in the vertical direction of the flexible platform.

Moving rod of the race transducer was caught by the superior platen of the suspension and the body of the



a) b)

Fig. 2: Positioning of transducers measure suspension front: a) spring b) a silencer.

with a silencer and notebook brands tens metric IBM ThinkPad R51. Tests were conducted on the premises SC REDAC SRL Craiova, the test stand type MB6000 Beissbarth, comprising: oscillating platform, display panel, rollers break for checking the brakes space. The transducer was fixed on damper casing, thus transducer W50 measuring the damper race (Fig.2). The accelerometer AccV measures the acceleration on the vertical direction, the same acceleration on the vertical on the front car axle, while the accelerometer AccOL

a) b)

Fig. 3: Detail of: a) front suspension and positioning measurement transducers; b) the location of the platform side of the wheel vibratory equipped with transducers measure.

measures the acceleration on horizontal-longitudinal direction of the damper body on the longitudinal axis direction of the car [15]. The accelerometer is represented to measure the acceleration in vertical direction, generated by the platform (Fig. 3). The accelerometer catching on the platform was made in rigid assembly by sticking with Superglue adhesive type. Measurements were made when the oscillatory motion was generated by oscillating platform left. Determination of dynamic response has been made for the left wheel equipped with transducer measure. Sampling rate of data acquisition Spider8 was 2400 samples/second. The following parameters were noticed: damper race (CRS, measured in mm), the shock strength (measured in F, N), acceleration in vertical direction, at damper level (measured in AccV, m/s^2) acceleration on the direction horizontal - longitudinally at the damper level (measured in AccOL, m/s^2) and acceleration on the direction vertical to the flexible platform (measured in AccVP, m/s^2).

After each test, the data acquired were viewed and stored in files of ASCII data for further processing. The platform

Fig. 4: The calculation technique, measurement and data acquisition.

generates an oscillating sinusoidal motion with the following characteristics [8]:

- for a period of time of about 1 s it is generated a sinusoidal motion of frequency and acceleration increasing uniformly from 0 frequency to a value of

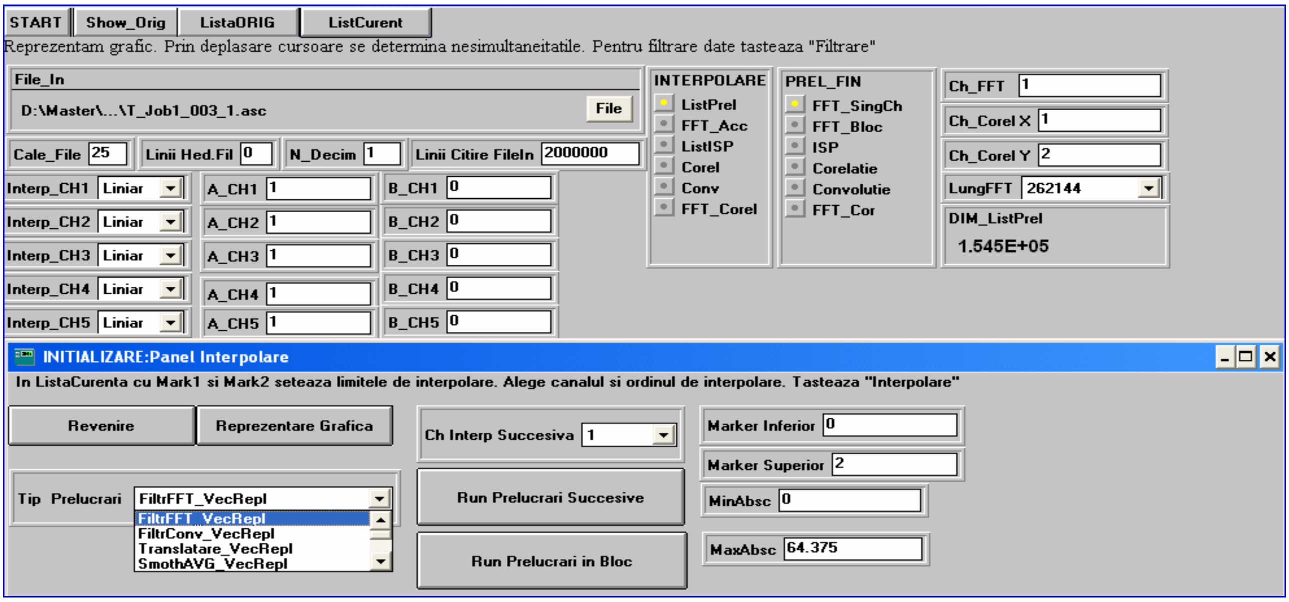


Fig. 5: Main panel of the program PrelVib.tst.

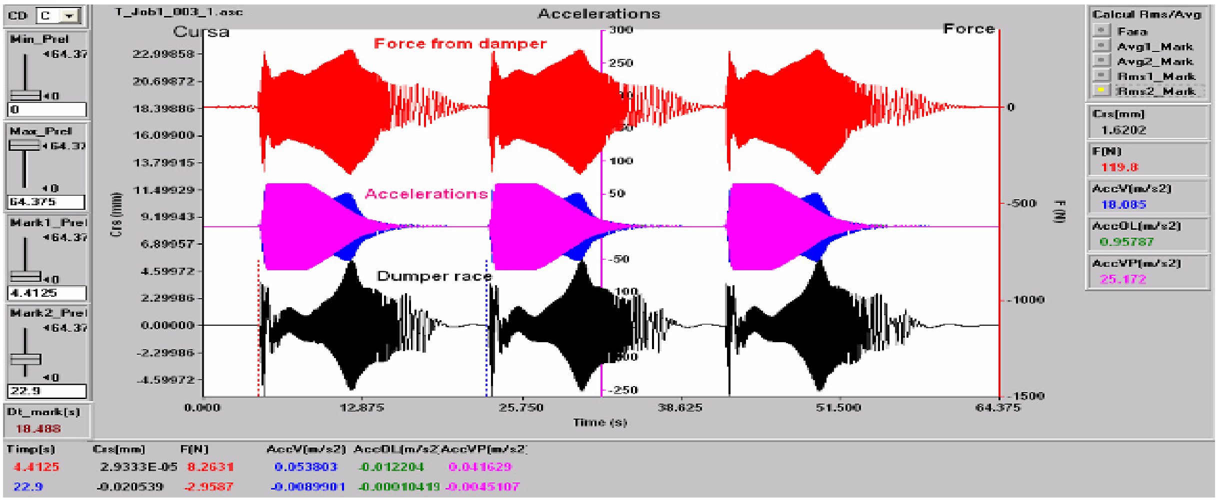


Fig. 6: Features original registration for the left front wheel excitement.

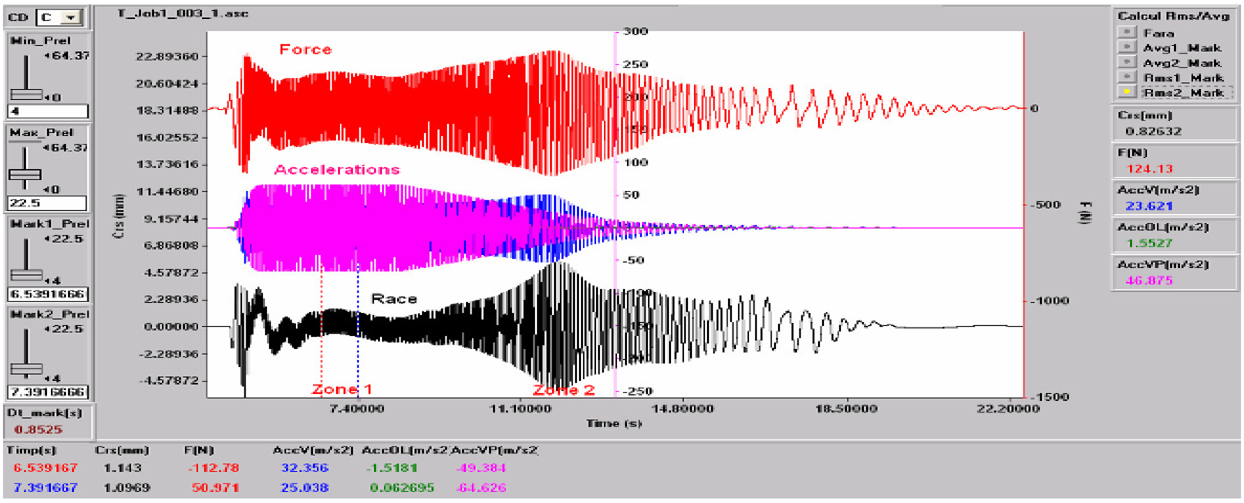


Fig. 7: Zoom- first pulse characteristics registration.

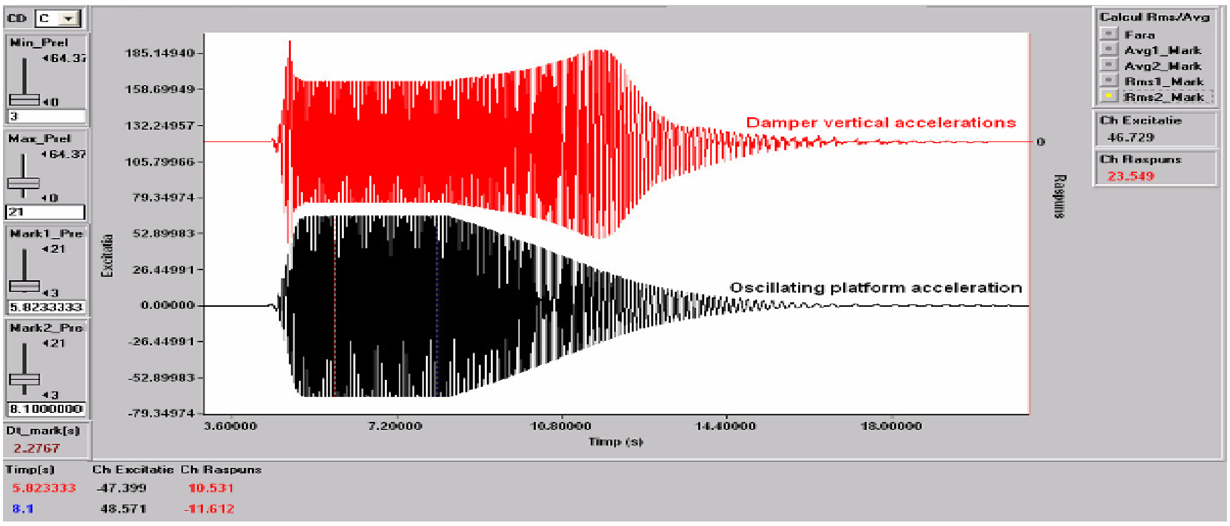


Fig. 8: The first pulse of the vibration characteristics for recording original sizes AccV/AccVP.

approximately 24.5 Hz and an acceleration value of about 50 m/s^2;

- for a period of time approximately 8 s motion generates a sinusoidal constant frequency of about 24.5 Hz and constant acceleration of 50 m/s^2;

- for a period of time about 8 s it is generated a sinusoidal motion of frequency and acceleration decreasing uniformly to 0.

Interpolare - highlights the "Panel Interpolare" panel that allows procedures such as: filtration through FFT, filtration by convolution, vector translation with the

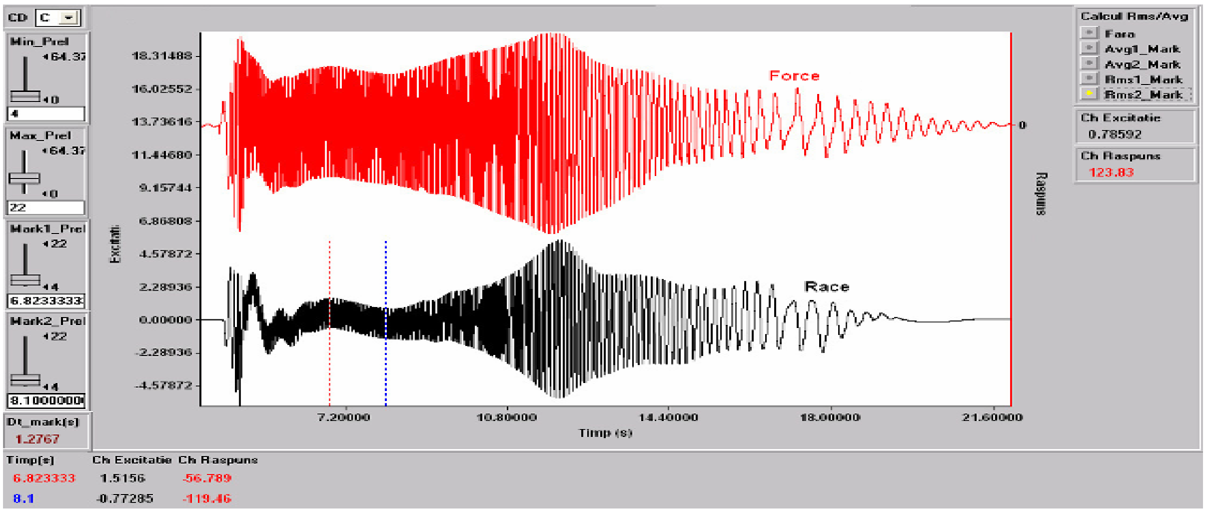


Fig. 9: The first oscillation pulse of the original characteristics registered for F/Crs sizes (the first impulse).

B. Processing Experimental Data. Under the Test Point programming environment was developed the program PrelVib.tst for determining the characteristics of the frequency of the suspension and the damper of the car (Fig. 5).

subvector's average whose limits are submitted by *MarkerInferior* and *MarkerSuperior*, *SmothAVG*, linear or polynomial interpolation. All this processing are submitted to the transmitted channel through the selector "Run Prelucrari Succesive" or they can be applied to all

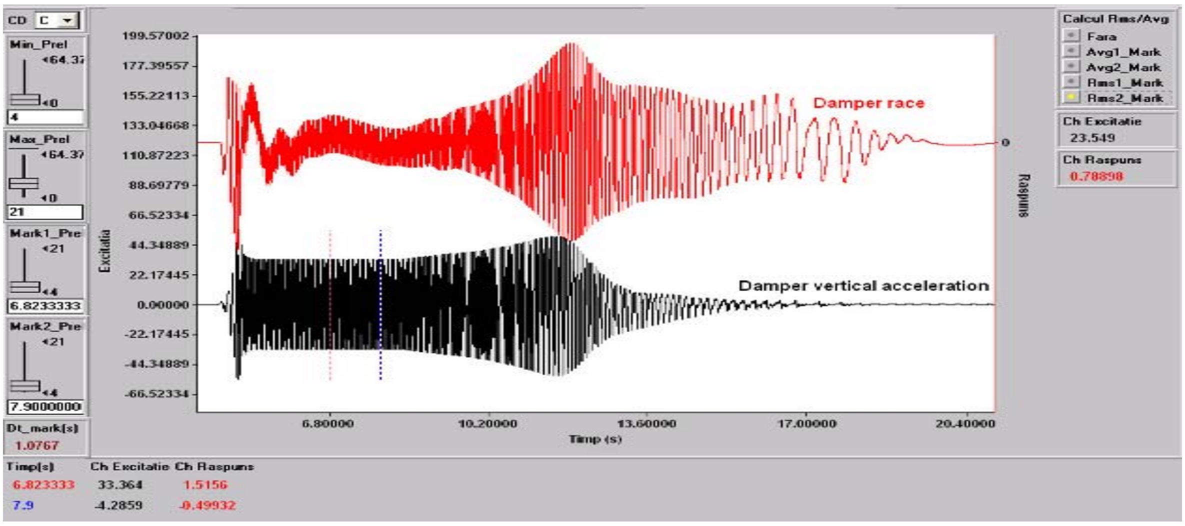


Fig. 10: The first pulse of the vibration characteristics for recording original sizes crs/AccV.

ShowOrig - highlights a panel in which to set name parameters and measurement units for file data. You can select the number of features that can be viewed at this stage. It makes visible the objects *Current list, PREL, ListCurentPrel.*

ListaOrig - allows viewing of graphical original characteristics. During processing they may be modified by digital processing, depending on needs, but ListaOrig maintains unaltered the original features, to which you can return if necessary.

ListCurent - object working with current characteristics. They can be modified by the *Interpolare* object. At each change in the current list the features with latest changes are maintained.

current channels through "Run Prelucrari in Bloc".

Prel-Fin - allows high level processing of current characteristics. It can perform the following types of processing:

FFT_SingCh, calculates the Fourier transform of the input channel transmitted by Ch_FFT using Fast Fourier Transform (FFT) techniques

FFT_Bloc, calculates the Fourier transform of all input channels using FFT techniques;

ISP, provides its users a panel that allows the functions calculation of frequency response between any two channels from the list of current features.

Corelare and Convolutie calculates the correlation and convolution functions of two channels specified by

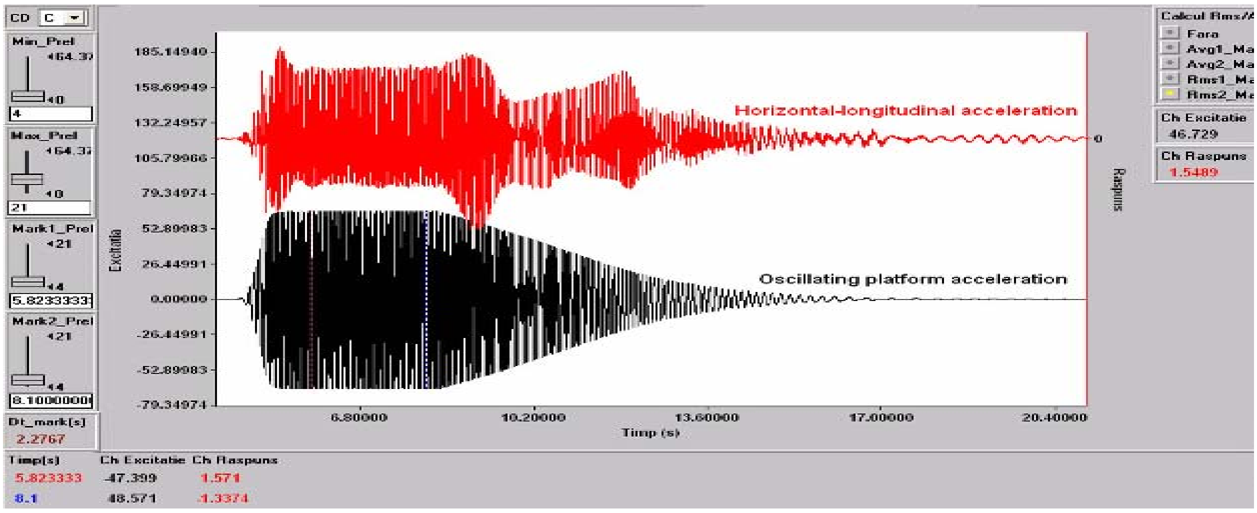


Fig. 11: The first pulse of the vibration characteristics for recording original sizes AccOL/AccVP.

Ch_CorelX and Ch_CorelY.

ListCurent provides the user with a graphic with two cursor, the selector CalculRms/Avg and displays with the following use.

MinPrel and *MaxPrel* - Slider for selecting the upper and lower limits of data representation. It allows zoom on abscissa.

Mark1 and *Mark2* - Slider for the horizontall move of two cursors. They allow the reading of the current value of represented parameters, their values being transmitted to the display set of objects in the bottom of the graph. The combination of drawn and displays is made by color.

Calcul RMS / Avg - five position selector whic runs:

1. *Fara* - hides the pointer objects.
2. *AVG1_Mark* - calculates the average characteristics on a number of samples sent in DT-1_Mark left and right centered to appropriate sample of Mark1. The average values are submitted in the pointer objects of the selector: Crs (MMI), AccV (m/s2), AccOT (m/s^2).
3. *AVG2_Mark* - calculates the average characteristics on a number of samples between Mark1 and Mark2. Average values are transmitted in the same object pointer.
4. *RMS1_Mark* - calculates the effective (RMS) features on a number of samples sent in DT-1_Mark centered left and right centered to appropriate sample of Mark1. Average values are submitted in the pointer objects of the selector: Crs (MMI), AccV (m/s2), AccOT (m/s^2).
5. *RMS2_Mark* – calculates the effective characteristics on a number of samples between Mark1 and Mark2. Average values are transmitted in the same pointer objects.

C. Analysis in Domain of Time. There were recorded three plusses of oscillating movement generated by oscillating platform (Fig. 6 and Fig. 7). There have been determined effective values (RMS) of measured parameters for two areas of interest: the frequency and constant acceleration of the platform and the oscillating frequency and decreasing the acceleration platform oscillating, amplifying vibratory maximum response of the suspension. Effective value (RMS) was calculated with the relationship (Table 1):

$$x_{rms} = \sqrt{\frac{1}{T}\int_0^T x^2(t)dt} \; . \qquad (1)$$

Table 1: The values of parameters measured in areas of interest.

Zone	Frequency Hz	Type Value	Race (mm)	Force (N)	AccV (m/s^2)	AccOL (m/s^2)	AccVP (m/s^2)
Zone1	24.87	RMS	0,82	124,1	23,62	1,55	46,87
		Top-top	2.237	425.1	78.42	4,33	151.1
Zone2	13.71	RMS	3,63	225,1	32,2	3,13	16,2
		Top-top	10.77	649.8	104.3	13,94	46.62

Car suspension is characterized by two features relevant frequency response [9]:

- function of the frequency response of the suspension (the transmission of vibration from the platform deck libratory car - depreciation/ amplification through the tires);
- the response in frequency of the damper - characteristic of frequency response of force-race at the damper (Table 2).

Table 2: The values of the relevant characteristics of frequency response, determined for the two areas of interest.

Zone	Frequency (Hz)	Type value	Force/Race (N/mm)	AccV/ AccVP (m/s^2/ m/s^2)
Zone 1	24.87	RMS	151,37	0,505
		Top-top	190,01	0,518
Zone 2	13.71	RMS	62.02	1,99
		Top-top	60.28	2,23

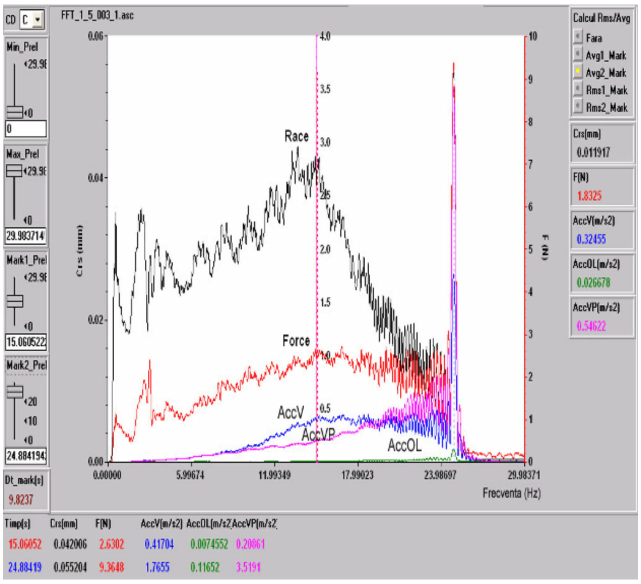
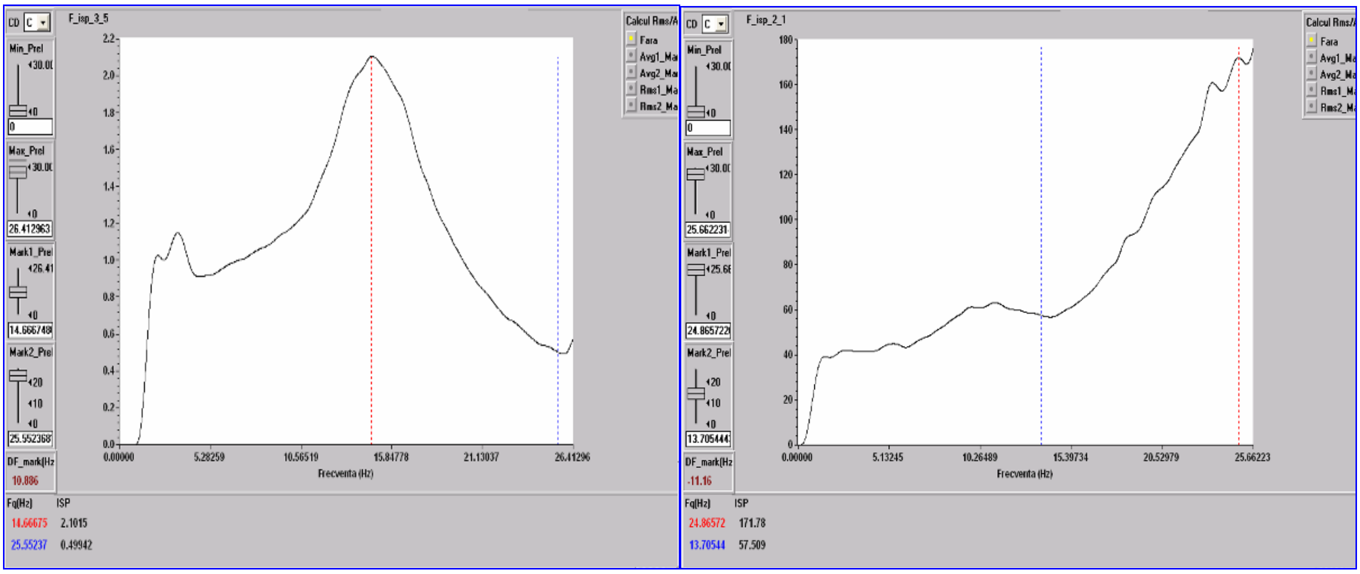


Fig. 12: Spectral characteristics of the parameters measured.

The pairs of the characteristics will serve to determine the response functions in frequency (Fig. 9-12). Characteristics present a zoom of the first pulse of oscillation.

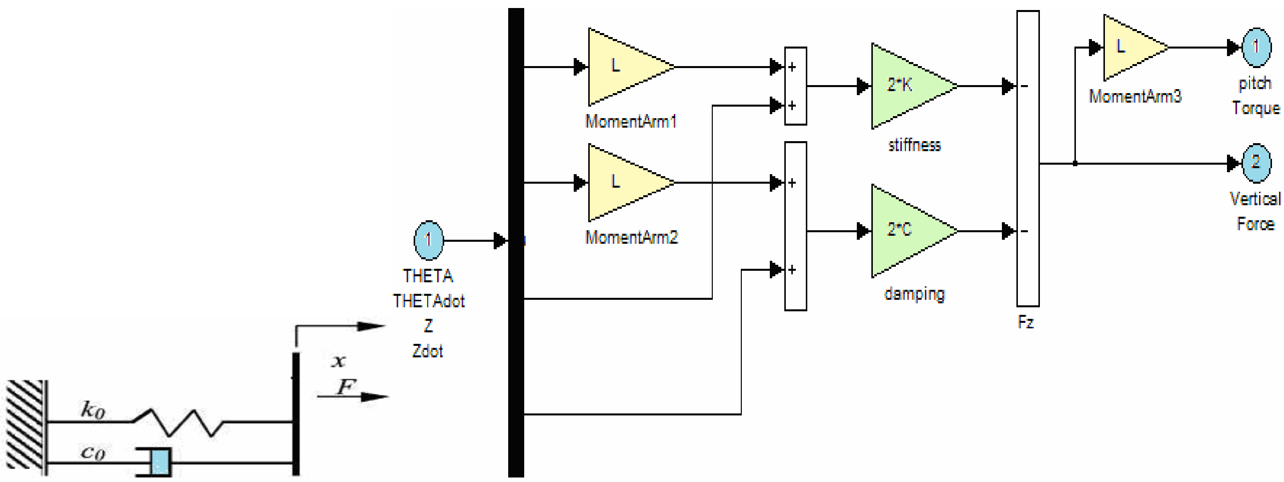
D. Analysis in Frequency Domain. The analysis in frequency of the determined parameters was carried out. In a recording made with the sampling frequency of 2400 Hz/channel and duration of 64.37 s, corresponding to a resolution of 9.1 MHz frequency. Spectral characteristics of the measured parameters are observed areas of interest, the cursor, which are positioned in those areas (Fig. 12).

E. Frequency Response. The response function in frequency of the suspension was determined, characterized by the Fourier transformers ratio of vertical acceleration of the front axle side and vertical acceleration of the oscillating platform. The cursors are positioned in the two marked areas. Comparing the data from Table 2 and the graphics of the FFT (Fig. 12) and ISP (Fig. 13.a), there can be seen a very good correlation



a) b)

Fig. 13: Function Frequency Response: a) to suspension for AccV(m/s^2) /AccVP(m/s^2); b) to damper for F(N)/Crs (mm).



a) b)

Fig. 14: Block chart suspension vehicles: a) the structural b) Simulink diagram.

analysis in time and frequency domains.

Like the analysis in the frequency response of the suspension, the comparison of data from Table 2 and the graphics of the FFT (Fig. 12) and ISP (Fig. 13.b), is a very good correlation analysis performed in the time and frequency. The cursors are positioned in the two areas marked as specified above [13].

IV. MODELLING AND SIMULATION OF SUSPENSION

A. Mathematical Model. The damper model must be continuous in all its components. Structural scheme is shown in Fig.14a. Vibrating applied force to this dynamic system is a function of time variable *t* and is called *F*(*t*).

In the absence of the mobile mass and therefore the forces of inertia, this force *F* is balanced by the three functions described in which the independent variable is the displacement *x*(*t*) or the speed *v*(*t*)=*dx/dt* [2]. To simplify the writing it is the omitted the time variable *t*, but its presence must be understood. The three components that balances force *F* are: *the linear elastic* $f_e(x)$ determined by resort characterized by the coefficient of stiffness k_0; *the linear viscous hydrocarbon* $f_v(x)$ characterized by the coefficient of viscosity c_0 *component histerezis*, $hz(x)=\alpha z$, is characterized by the coefficient of histerezis and non-linear function of histerezis *z*(*x*).
When the mobile mass is zero inertia forces disappear and the equation of equilibrium of forces expressed through three components, $F=f_e+f_v+hz$ is explained by the relationship:

$$F = c_0\dot{x} + k_0 x + \alpha z \,. \qquad (2)$$

In relation (2) function histerezis *z*(*x*) is obtained as solution of the next nonlinear differential equations:

$$\dot{z} = -y|\dot{x}|z|z|^{n-1} - b\dot{x}|z|^{n} + a\dot{x}\,, \qquad (3)$$

where α, *b* and *an* are parameters related *to the loop of histerezis* and *y=z* in the absence of an external disturbance x_0 such as road length related.
Adjusting the parameters *b* and *a* of the model α , it is possible to control the discharge non-linearity facility and transition from one region to pre-critical to post-critical. From the equations (2) and (3) of the suspension model there is shown that this dynamic system modelled can be split into two parts: a linear part L described by the equation (2) and a non-linear N described by equation (3), interconnected as the block diagram in Figure 16 in which the non-linear (which shapes the loop histerezis) is placed into the negative reaction of the system, while the line is placed on the direct path of the system.

Block scheme in Figure 14.a allows transposition Simulink model for the sub if the equation is the nonlinear form

$$z = h(x), \qquad (4)$$

and the linear equation has the form [7]

$$F - az = f_e + f_v \,. \qquad (5)$$

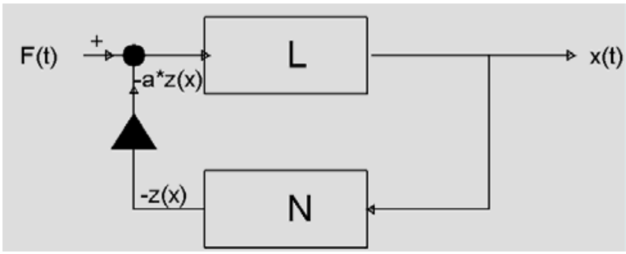


Fig. 15: Structural scheme of vehicle suspension.

B. Simulink Model. Transposition in Simulink of the linear part L (Fig. 16) requires only a few blocks of calculation [5]:
- a *derivative block* and *an amplifier* c_0 that has entry *x* and exit c_0dx/dt; a *block amplifier* k_0 that has the entrance *x* and exit k_0x;

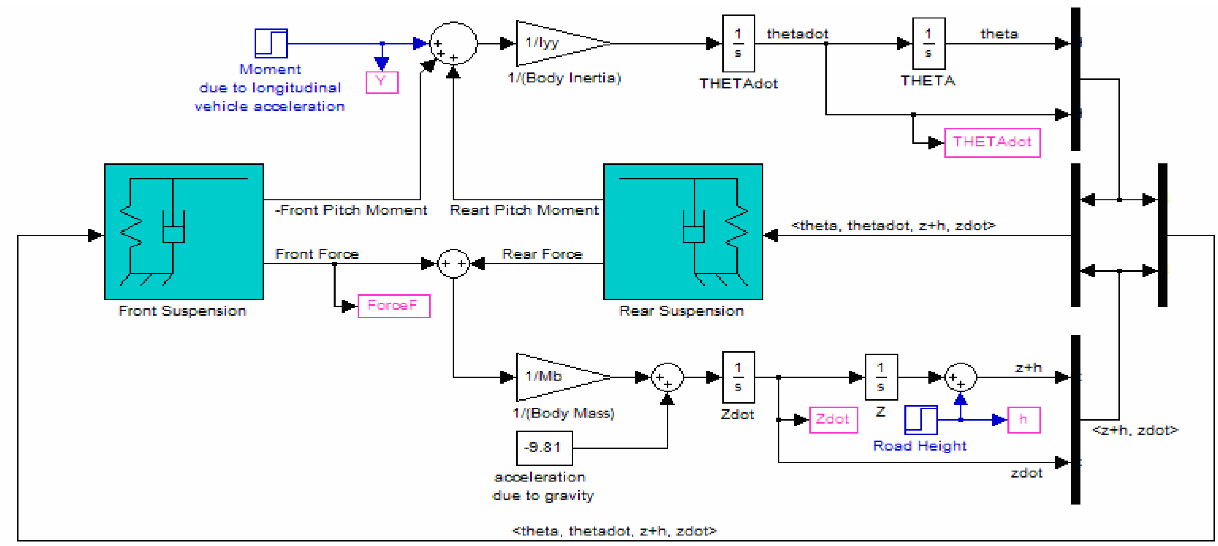


Fig. 16: Scheme of the linear model in Simulink suspension.

-an *adder block* which sums up F, k_0x,c_0dx/dt and az.
Simulink model of the nonlinear equation shows specific loop of the dumper and is described by differential equation (3) which shows that the output $z(x)$ for n=2 and y=z, is calculated by the relationship [10]:

$$z = -\int\left(|\dot{x}||z|z^2 + b\dot{x}|z|^2 - a\dot{x}\right)dt\,. \qquad (6)$$

Input signal of non-linear block N displacement x (representing the entire output model) and is received by the exit block L [12].
$\dot{x}$ speed in equation (3) is obtained by derivation of entry a nonlinear block N i.e. $v(t)=dx/dt.=\dot{x}$.
Relationship (4) will be the basis of the Simulink model of the non-linear in N. To translate a Simulink model given by equation (6) blocks are used Simulink integration, hoisting power, recording, etc. These blocks and the connections between them result from relations (2) and (6). Connections between Simulink blocks of the entire system, made in accordance with (2) and (6) are presented in the simulation scheme in Figure 16. There are two signal generators to simulate the time variation of force $F(t)$ input model [14]. Simulation scheme is equipped with recorders evolution while labour input, the displacement $x(t)$ and velocity $v(t)=dx/dt=\dot{x}$ from the system output.

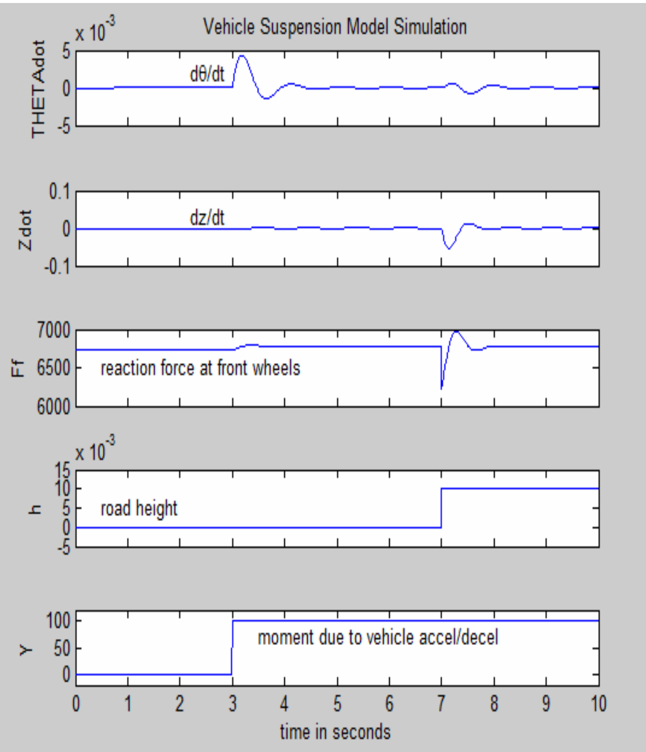


Fig. 17: Results of simulation for the linear model parameters resulted from experimental determinations.

This example describes a simplified half-car model that includes an independent front and rear vertical suspension as well as body pitch and bounce degrees of freedom. We provide a description of the model to show how simulation can be used for investigating ride and handling characteristics. In conjunction with a powertrain simulation, the model could investigate longitudinal shuffle resulting from changes in throttle setting. We model the front and rear suspension as spring/damper systems. A more detailed model would include a tire model as well as damper nonlinearities such as velocity-dependent damping with greater damping during rebound than compression. The vehicle body has pitch and bounce degrees of freedom, which are represented in the model by four states: vertical displacement, vertical velocity, pitch angular displacement, and pitch angular velocity.

The Damper dynamic study on different orders [15]: input step of pressure variation, the input step of flow variation, showed internal phenomena that occur in the damper. Using a predefined set of values it was realized the representation by 3D viewing of the results using Matlab environment (Fig.18 and Fig.19).

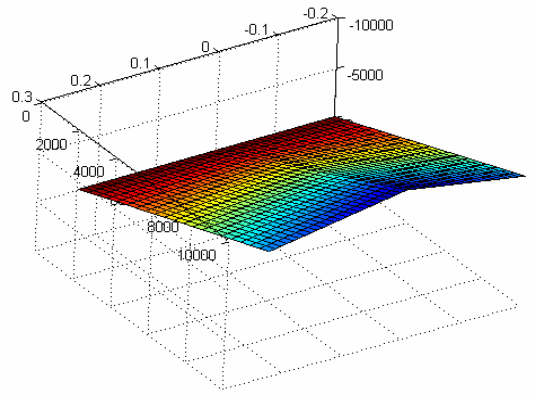


Fig. 18: Distribution of speed along the damper.

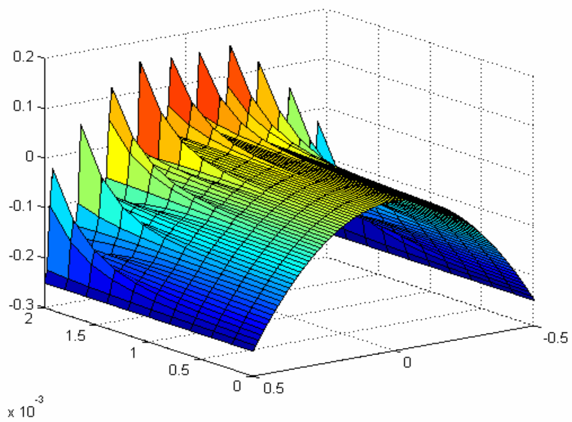


Fig. 19: Speed distribution for a step of pressure against time – moment t=0,005.

V. CONCLUSIONS

Results of linear mathematical simulation model developed (Fig. 17) for road cars suspension was compared with responses obtained during the experimental testing of the suspension (Fig.9-12). There results a good concordance between experimental data and those provided by the mathematical model in the transformed schema simulation illustrated in Figure 16. This leads to the conclusion that the model can be accepted and boo held for computer simulation of vehicle suspension. It was studied the dynamics of a damper to a step input of pressure variation (Fig. 19).

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