Adjustable shock absorber characteristics testing and modelling

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Abstract. The paper shortly presents the design of an electromagnetically adjustable shock absorber and its damping characteristics obtained from manufacturers. Next part presents the methodology and test stands typically used for shock absorber testing and test data processing to obtain force vs velocity characteristics. Next part discusses some new functional requirements and performance due to adjustability of damping level. Next part of the paper presents special dedicated software implemented in the Matlab/Simulink and on the dSpace platform for rapid prototyping of control system. The results of damping characteristics, time delay and hysteresis testing with its comparison for 3 types of passenger car adjustable shock absorbers obtained with use of developed methodology and testing control software are presented in the further part.

1. Introduction

Shock absorbers can be classified according to their abilities of changing damping force characteristics into two groups: passive and adjustable. The latter group can be divided into manually adjustable dampers and automatically adjustable ones with the use of electric signals. The former group can be adjusted using mechanical valves, which can be done while servicing the vehicle or, if done with a use of cable, also while driving. Electromechanical valves can be also used with manual control.

The adjustable shock absorbers are currently used mainly with an electronic control system which adjusts damping force automatically and then the name for this group can be adaptive damper or rather adaptive damping system. In terms of possibilities to change the damping level during one vibration cycle this group can be divided into adaptive and semi active systems. The adaptive damping system is not able to change the damping level at least fast enough to react to body mass resonance frequency. The semi-active system has to react fast enough to change the damping level at least four times per one cycle of vibration at body mass resonance frequency.

There are mainly two groups of automatically adjustable dampers used nowadays in automotive industry: magnetorheological dampers and electromagnetically adjustable dampers with by-pass channel between the upper chamber and the reservoir. These dampers can have an on/off or continuous control of fluid flow in the bypass channel. In this paper the results of characteristics testing for the latter group are presented.

2. Adjustable shock absorber characteristic testing – instrumentation and methodology

To model the influence of passive shock absorber on vehicle suspension dynamics usually the so-called velocity diagrams (damping force in function of piston velocity) are used. That is the fundamental shock absorber characteristics, which is often linearized. In the case of the adjustable shock absorber also the damper response time is a very important parameter. It affects possibilities to control damping level in a wide frequency range.

During damper work a hysteresis can occur. Hysteresis occurs when the damping forces are higher as the damper is decelerating compared to when it is accelerating and it decreases the effectiveness of the shock absorber. It is important when analysing the influence of a single bump and suspension parameters on a vehicle dynamics.

2.1. Test stands

To determine the previously mentioned characteristics it is necessary to perform tests allowing to measure:

- damping force – it is usually done with force sensors (force transducers or load cells) based on strain gauges,
- piston stroke – a relative travel of the piston rod against the tube or kinematic excitation producing compression or rebound of shock absorber – it is measured usually with any fast enough displacement sensor.

Such test are performed with the use of indicator test stands, which are able to extort vertical-backward movement of the lower mounting. The upper mounting of the piston rod is mounted on the test stand frame through a force transducer. There are two possibilities to extort this vertical-backward movement [2]:

- with the use of mechanical system – eccentric crank-shaft unit having variable length,
- with the use of servo hydraulic excitatory using special control system allowing to control piston displacement.

It is important to analyse the necessary power of the test stand, maximum velocities and forces. As a damping force increases with a damper compression and rebound velocities, it is necessary to take into consideration a power increase necessary to realize assumed stroke and velocity. For example: if for 1.05 m/s rebound velocity damping force is 3.135 kN, then the damping power is 3.3 kW. For electrohydraulic exciters it means necessary high fluid flow and pressure. Often special additional hydraulic accumulators are need to be used in case of high velocities to cover the oil demand for a limited number of strokes.

2.2. Measurement and shock absorber control system

According to the need of measurement the influence of current value controlling the by-pass valve opening on shock absorber characteristics and the step response of shock absorber it is necessary to use a dedicated shock absorber control and data acquisition system. Such a system was designed by the authors to perform following tasks:

- to change the damping level to determine shock absorber characteristics for various level of bypass channel valve openings,
- to shorten the time of determination of full shock absorber characteristics switching automatically damping level during tests,
- to enable determination of step response after step input of shock absorber control signal (current).

A designed control algorithm was based on earlier authors works [9] and had a following functionality:

- ability to adjust valve opening level by setting the valve coil current with use of pulse width modulation (PWM signal) at frequency specific for given model of shock absorber ,
- ability to automatically change bypass valve opening level with predefined value (Fig. 2) after exactly one compression/rebound cycle at:

   – the smallest compression/rebound velocity for determining shock absorber characteristics,
   – the highest compression/rebound velocity for determining step response time,
- manual control of bypass valve opening level,
- real time signal processing and force-displacement and force-velocity charts plotting with ability to compensate damping force and displacement offsets.

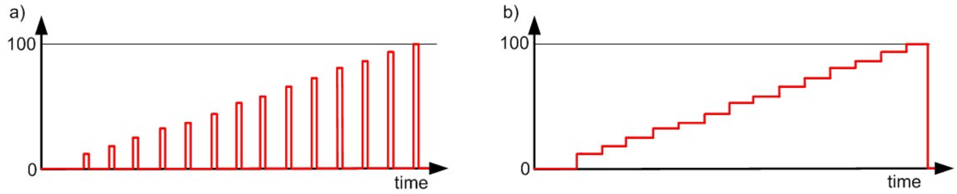

Figure 1. The types of incrementally increasing after every cycle control signal used to test: a) step response, b) non-liner static characteristic

2.3. Tested shock absorbers and testing and data processing procedure

For the research of damping characteristics, its step response and hysteresis parameters three various adjustable shock absorbers with bypass channel were chosen, marked for the purposes of this research as A, B and C type. Shock absorbers of type A and B were used in a front suspension of C class (compact) passenger cars and C type in a front suspension of premium D class passenger car. Shock absorber A was produced by ZF Sachs company and B and C by Tenneco company.

Damper characteristics were determined with use of servohydraulic test stand at constant strokes - mainly 50 mm according to VDA (GermanAutomotive Manufacturesrs and Suppliers Association) methodology of characteristic determining [1], but also 10 mm to get values for small velocities. Using various frequencies it was able to get some data points for following velocities: 0,06; 0,13; 0,26, 0,39, 0,44 m/s.

While measuring both characteristics – force–displacement and force–velocity were presented in a real time and data were collected for further data processing. Characteristics were tested with prepared software – control signal of bypass valve was automatically changed resulting in such a form of characteristics as presented in a figure 2.

All the tested shock absorbers have hysteresis which is important in terms of dynamic processes and also when modelling their characteristics for damping control algorithm development. Analysis results – Table 1 – show that the hysteresis increases with increasing damping forces. It was also observed that the frequency at which test were performed was changing the shape of hysteresis, too: the higher frequency, the bigger hysteresis. For high test frequencies – 6..7 Hz the shape of hysteresis is more regular and close to ellipsoid, for frequencies 1,5 Hz the shape of hysteresis is more irregular. In that case for highest compression/rebound velocities hysteresis is negligible, for velocities around 0 is the biggest and need to be taken into consideration.

Table1. The comparison of height and width of hysteresis for three tested shock absorbers

		Test parameters:			
	Shock absorber	1Hz, 10mm v_{max}= 0,06m/s	0,82Hz 50mm v_{max}= 0,26m/s	1,6Hz 50mm v_{max}= 0,52m/s	6Hz 10mm v_{max}= 0,37m/s
Height of hysteresis at compression/rebound velocity v=0m/s [N]	A	60	200	750	850
	B	100	500	1400	850
	C	300	500	1400	1700
Width of hysteresis [m/s]	A	0,01	0,05	0,1	0,2
	B	0,05	0,1	0,01	0,15
	C	0,05	0,1	0,02	0,15

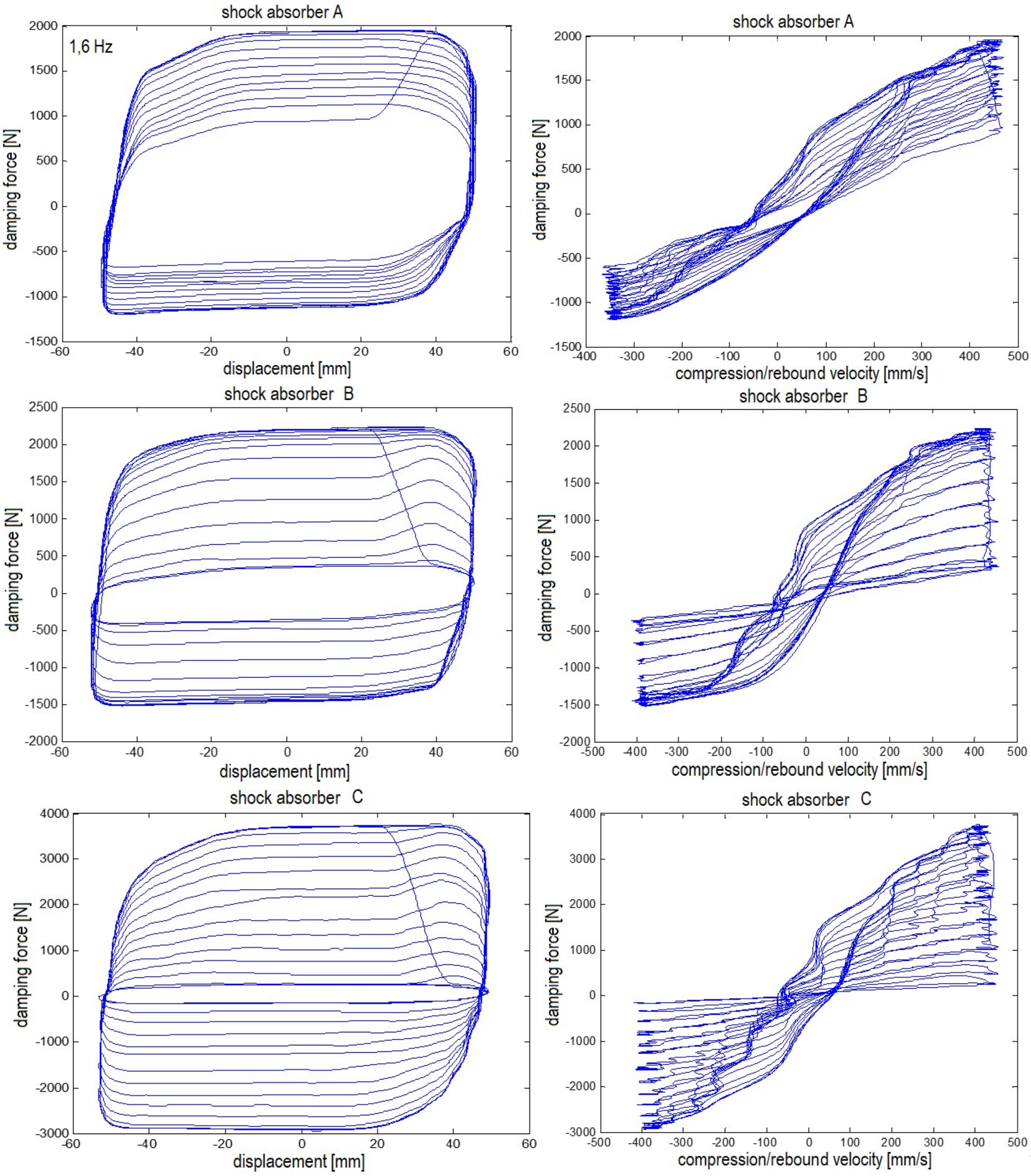

Figure 2. Force-displacement and force-velocity characteristics of tested shock absorbers for the full range of damping force (from minimum to maximum) in 14 increment steps of control signal

2.4. Nonlinear static characteristics

In terms of adaptive damping control algorithm and model development it is important to determine non-linear static characteristics on the base of obtained data. For all the tested shock absorbers these characteristics were prepared averaging compression and rebound damping force values for the same compression/rebound speed occurred for the beginning and end of compression/rebound movement (positive and negative displacement of lower shock absorber mounting point from middle (zero) position. Resulting static characteristics of three tested shock absorbers are presented in the figure 3 for comparison purposes.

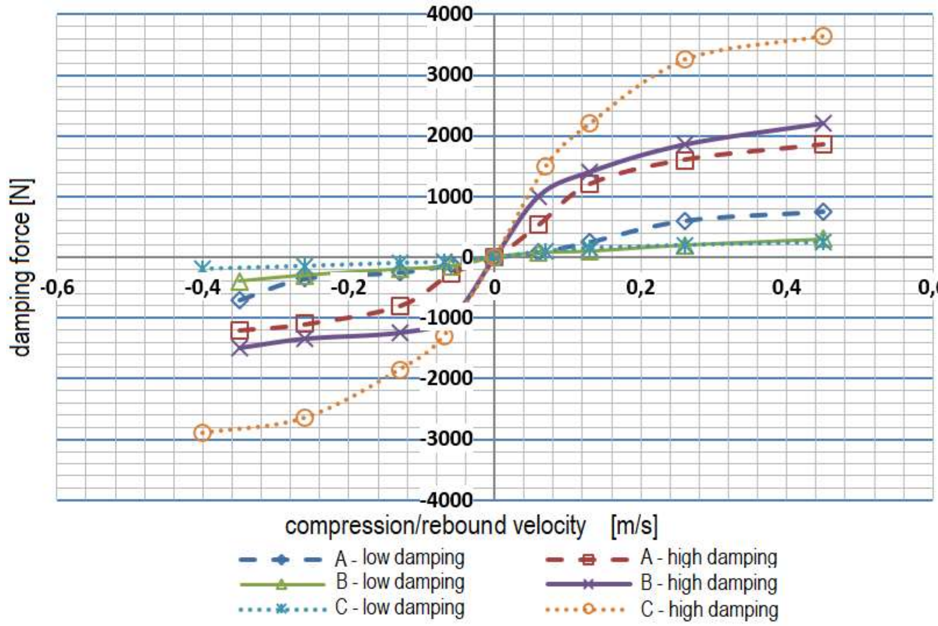

Figure 3. The comparison of force-velocity characteristics of three tested shock absorbers

2.5. Time response values

In order to determine step responses of the tested shock absorbers a special control algorithm was developed (described in paragraph 2.2). This control algorithm is able to change a shock absorber control signal (PWM signal duty cycle) with the smallest changes in compression/rebound velocity during tests with sinusoidal kinematic excitation. It allows to measure time delay with smallest possible influence of a shock absorber velocity on the damping force. This situation is around middle (zero) position of lower part of shock absorber (in the middle of shock absorber stroke) – Figure 4.

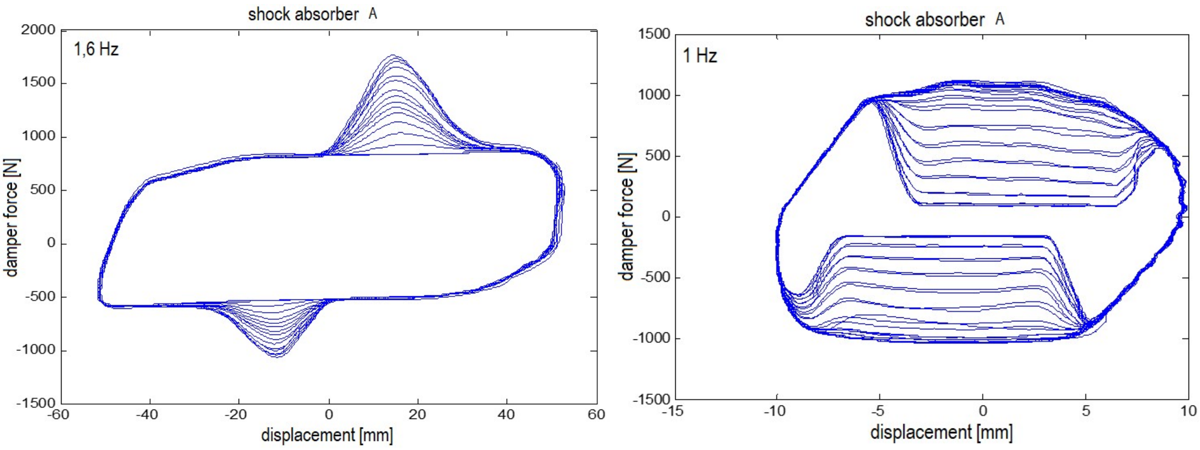

Figure 4. Force-displacement and force-velocity characteristics of tested shock absorbers

For testing time responses of shock absorbers stroke 50 mm and frequency 1,6 Hz were used and also stroke 10 mm and frequency 7 Hz. Step changes of the control signal (PWM duty cycle from 0 to 100% and from 100 to 0%) were performed by designed software and simultaneously following signals were recorded: control signal (PWM duty cycle), current [A], damping force [N], damper compression/rebound velocity [m/s].

An analysis of time histories of recorded signals is necessary to determine time responses of shock absorber after control signal step input:

- the required time for the current signal to reach 5% of the final state of current signal,
- the required time for the damping force signal to reach 5% of the final state of damping force signal,
- the required time for the current signal to reach 90% of the final state of current signal,
- the required time for the damping force signal to reach 90% of the final state of damping force signal.

Exemplary time histories analysed in order to determine time responses are presented in figure 5. As shown in the figure, there are two time delays, the first one between control signal and current signal and second one between current and damping force signal. The first one results from electromagnetic properties of current controller [1]. The other one results from mechanical properties of shock absorber and its valve. For a control algorithm an overall step response (between PWM signal and damping force) is important.

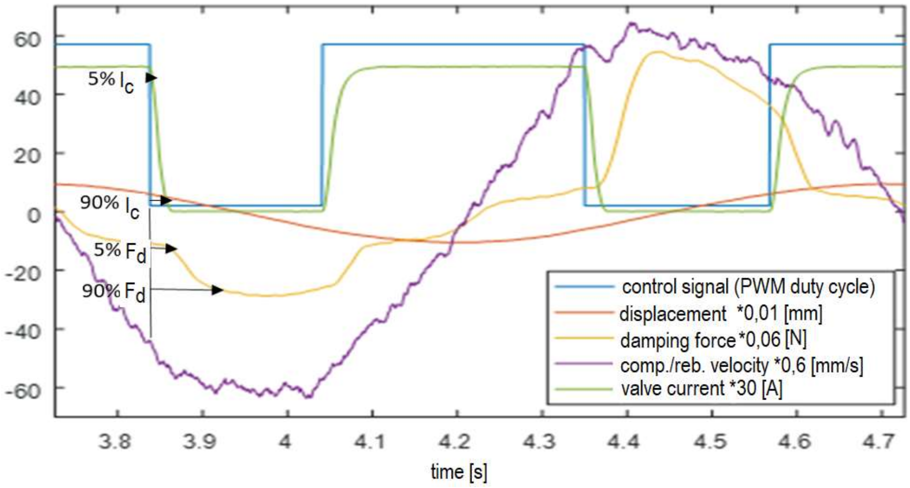

Figure 5. Force-displacement and force-velocity characteristics of the tested shock absorbers

An analysis of all 3 tested shock absorbers allowed to determine step responses for four cases:

- compression with an adjusted increase of damping force (positive change in control signal),
- compression with an adjusted decrease of damping force (negative change in control signal),
- rebound with an adjusted increase of damping force (positive change in control signal),
- rebound with an adjusted decrease of damping force (negative change in control signal).

The overall determined time delays between the control signal (PWM duty cycle) and the damping force are presented in a chart in the figure 6. The observed average time delays were:

- 1.8 ms for delay of the current signal to reach 5% of the final state of current signal,
- 18 ms for the current signal to reach 90% of the final state of current signal
- 17 ms for the damping force signal to reach 5% of the final state of damping force signal
- 55 ms for the damping force signal to reach 90% of the final state of damping force signal.

The average step response for tested dampers is 55 ms. For setting damping force increase during compression this time was about 12 ms longer (about 67 ms) and for setting damping forces decrease about 12 ms shorter (about 43 ms). For damping force increase in some cases setting damping force increase during compression resulted in a longer step response than for setting damping force increase during rebound.

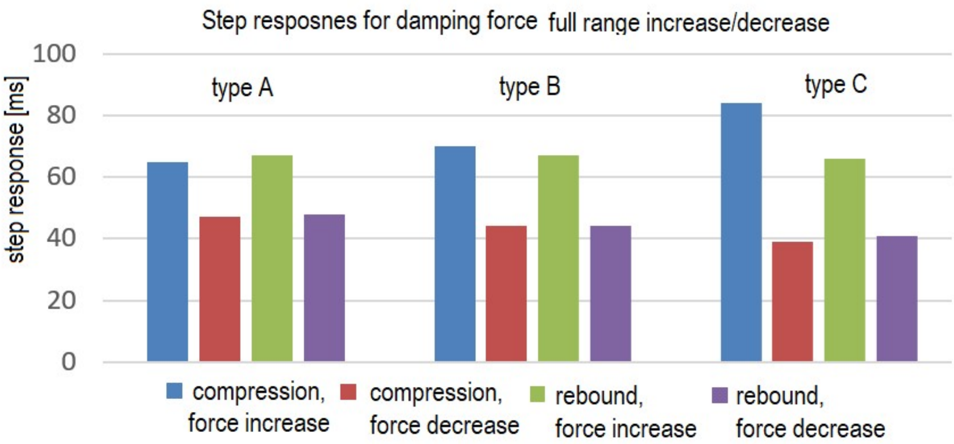

Figure 6. Step response values for three tested shock absorbers for compression/rebound and damping force increase/decrease

3. Shock absorber model

Physical models of shock absorbers are based on a detailed description of the shock absorber's internal structure. These models allow to calculate force vs. velocity curves given input parameters characterizing the damper. Parameters required includes dimensions of damper components, properties of hydraulic fluid, and known internal gas pressures [3,4,10]. These models describe the behavior in a large range of operating conditions very accurately, but they are computationally complex, requiring time consuming computations when implemented in a full vehicle simulation.

Generally for vehicle dynamics or control algorithm development rather simple models of shock absorbers damping forces are used [2], frequently nonparametric models that establish a relationship between the shock force and velocity by purely mathematical means. They are functional models. A shock absorber is characterized by a „black-box" system for a limited range of operating conditions that are tested. Also parametric models can be used, which are usually rheological models such as the Bingham model, the Bouc-Wen model, the Li model, the Spencer model, the Gamota-Filisko model which are based in their construction on three basic models: Hookean solid, plastic body of St. Venant and viscous body of Newton [11].

In this work, for the purposes of using shock absorber model in a damping control system developing, the functional model consisting of following modules was proposed:

1. nonlinear static characteristic,
2. hysteresis,
3. internal friction
4. response time of a shock absorber.

The nonlinear static characteristic module is fundamental for damper characteristics modelling – the average damping force for the given compression/rebound velocity. This model is based on a determined force-velocity characteristics of tested shock absorbers. Implementing model in a Matlab/Simulink software the interpolation block was used for interpolation the value of the damping force.

The tested shock absorbers have the possibility to continuously change damping force in the range between minimum and maximum values - for this reason also interpolation is necessary for the value of the current of the valve coil. It can be also done with interpolation block (2-dimensional) for the three dimensional shock absorber characteristics [9]. In this work another approach was proposed. Observing linear relation between controlling current and damping forces, the middle characteristic between minimum and maximum damping forces was found and formulas to calculate smaller and bigger damping forces were proposed. One for rebound forces and one for compression as the ranges of damping forces for both directions are different. For example, for shock absorber A, these formulas were following:

$$F_{dS} = F_{d\_m} \cdot K_I \tag{1}$$

where:

F_{dS} – interpolated value of damping force from static characteristic for given current,
F_{dS_m} – the middle static characteristics damping force (for middle value of valve coil current)
K_I – the coefficient to increase or decrease damping force according to value of valve coil current and state of damper work - compression or rebound. For A type shock absorber formulas for calculating K_I values according to current value I_c were determined for compression and rebound respectively as:

$$K_{I\_C} = -0{,}55 I_c + 1{,}59 \quad \text{and} \quad K_{I\_R} = -0{,}71 I_c + 1{,}74. \tag{2}$$

The hysteresis module is modelling shock absorber hysteresis important for high damping forces and high velocities. The model implemented in this module is based on a work [6] were a simple model was proposed to model the hysteretic force–velocity characteristic of the MR damper. The model presented in [6] is given by formulas:

$$F_d = c\dot{x} + kx + \alpha z + F_T \tag{3}$$

$$z = \tanh(\beta \dot{x} + \delta sign(x)) \tag{4}$$

where:

$x, \dot{x}$ – damper deflection and speed of deflection,
z – hysteretic variable given by the hyperbolic tangent function,
α – scale factor of the overall hysteresis, determines the height of the hysteresis
c – damping coefficient interpolated form nonlinear static characteristics,
k – stiffness coefficients which is responsible for the hysteresis opening found from the vicinity of zero velocity, modelled in the hysteresis module, a large value of k corresponds to the hysteresis opening of the ends
F_T – damper force offset,
β – scale factor of the damper velocity defining the hysteretic slope. The large value of β gives a step hysteretic slope,
δ – scale factor, δ and the sign of the displacement determine the width of the hysteresis through the term δsign(x), a wide hysteresis is resulting from a large value of δ.

In the hysteresis module were modelled values of products kx and αz. Values of F_T and $c\dot{x}$ were modelled outside the hysteresis module. Values of $c\dot{x}$ were modelled by the first module interpolating damping forces form nonlinear static characteristics and F_T values were modelled by the third module - the internal friction module.

On the base of an analysis of dynamic characteristic of tested shock absorber A, a formula for the relation between scale factor α and valve coil current I_c was developed:

$$\alpha = \alpha_0(-2{,}15 I_c + 4{,}45) \tag{5}$$

where:

α_0 - scale factor α of the hysteresis for middle static characteristics damping force,
I_c - valve coil current.

The internal friction module modelling the force F_T consists of two elements – the value of friction force and a $signum$ function due to model friction force with opposite sign to damping force.

The module modelling response time of a shock absorber is based on the model presented in [7,8] and consists of two blocks modelling for the damping force increase:

- dead time T_0,

- time delay with time constant T_Z.

Considering that the time response of tested shock absorbers depends on the stroke direction and on the valve operating state, including switching direction (from soft to hard or vice versa), the four different time delays with use of for different values of T_0 and T_Z are calculated in model and appropriate is used according to compression/rebound movement and switching direction. For test A type shock absorber these values were determined to be the same for compression and rebound directions and for switching from soft to hard they were T_0=4ms i T_Z=5ms and for switching from hard to soft: T_0=2ms i T_Z=3ms. On the figure 7 force-displacement and force-velocity characteristics of damper A type are presented for switching from soft to hard or vice versa for tests with damper stroke amplitude 50mm and frequency 1,25 Hz.

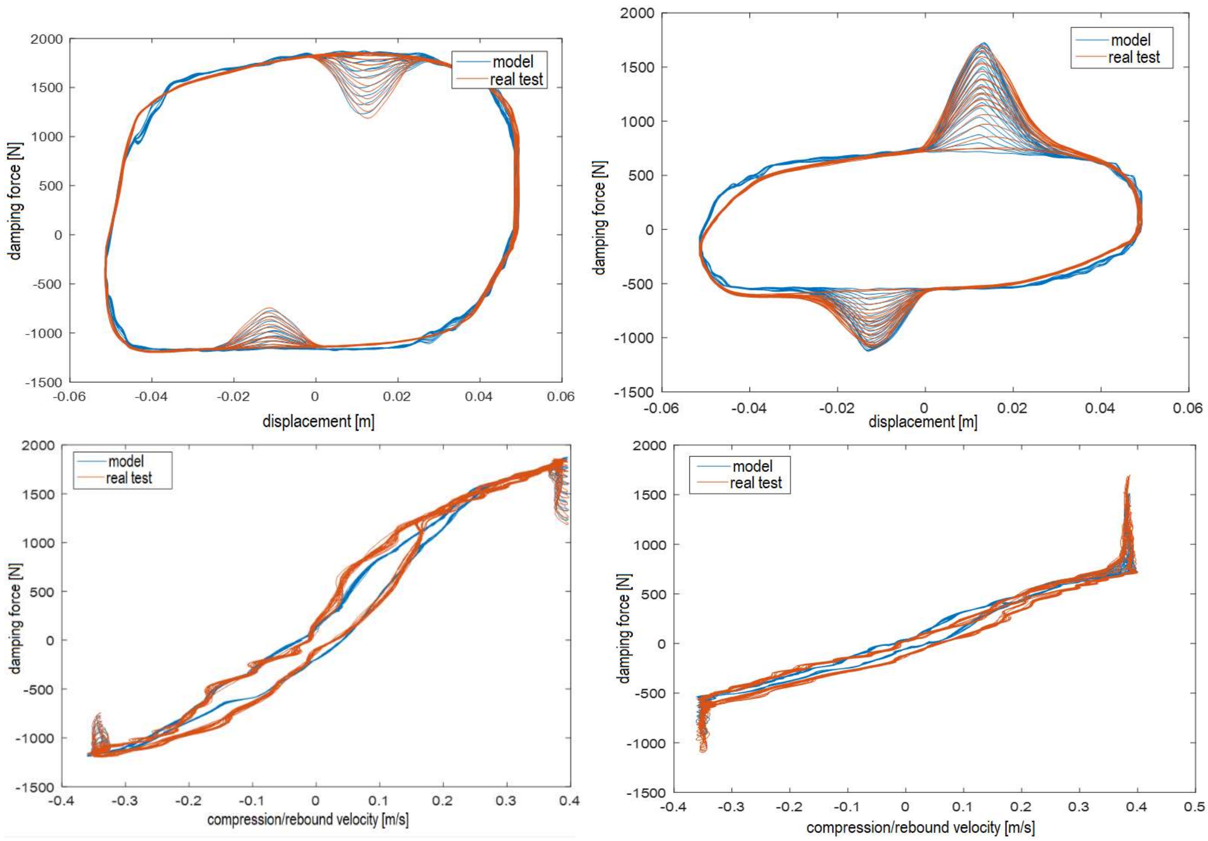

Figure 7. Force-displacement and force-velocity characteristics of tested shock absorbers during testing and modelling step force increase

For these characteristics coefficient E of relative error between real test damping forces F_{t_B} and model damping forces F_{t_M} was calculated:

$$E = \frac{\sum|F_{t\_B}| - \sum|F_{t\_M}|}{\sum|F_{t\_B}|} \tag{6}$$

Its value for switching from hard to soft was E =0,8%. Average error of estimating maximum damping forces during transition phase was 56 N in absolute values. For switching from soft to hard these values were E =1,2% and 67N.

4. Conclusions

This paper has presented a methods for testing nonlinear static characteristic and step response of three electromagnetically adjustable shock absorber. The most important aspect of developed test methodology was special data acquisition and a tested shock absorber control software. This software was necessary to make easier and faster tests of static characteristics and to allow step response testing. The switching damping force level was controlled by predefined control signal increments and position of shock absorber lower mounting.

The results of the force-displacement and the force-velocity characteristics of tested shock absorbers for the full range of damping force (from minimum to maximum) in 14 increment steps of control signal were presented and results of analysis of step response values for four cases - for compression/rebound and damping force increase/decrease.

An analysis of the hysteresis showed that the hysteresis increases with increasing damping forces. It was also observed that the frequency at which tests were performed was changing the shape of hysteresis, too: the higher frequency the bigger hysteresis. For high test frequencies – 6..7 Hz the shape of hysteresis is more regular and close to ellipsoid, for frequencies 1.5 Hz the shape of hysteresis is more irregular. In that case for highest compression/rebound velocities the hysteresis is negligible, for velocities around 0 it is the biggest and needs to be taken into consideration.

The paper has also presented the functional shock absorber model consisting of four modules - nonlinear static characteristic**,** hysteresis, internal friction and response time of a shock absorber. This model is useful for simulation of vehicle dynamics and suspension damping control algorithms. The model was implemented in the Matlab/Simulink systems and presented results proved its possibilities for adequately modelling damping forces including dynamics changes due to changes in a control signal (step response) and direction of movement (hysteresis).

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