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EFFECT OF FAIL-SAFE MODE ON THE DAMPING CHARACTERISTICS OF A BYPASS-CONTROLLED SHOCK ABSORBER

Summary. The article provides and discusses results of tests of a bypass-controlled hydraulic shock absorber studied at a test stand (mechanical inductor), making it possible to plot work charts and determine velocity characteristics. Measurements of the shock absorber performance were conducted at different values of velocity and forced travel. The measurements were carried out for two statuses, i.e. for the emergency power failure condition and for normal setting. The results thus obtained can be used in simulation tests as input shock absorption characteristics for different technical conditions.

1. INTRODUCTION

The suspension system of the assembly is used to support the weight of the vehicle, to absorb road shocks, as well as to maintain the movement of the tires with the road. During mechanical work, the suspension system is in a dynamic balance, still adaptation to changing driving conditions. The vehicle suspension must be analyzed before production and its parameters selected. This requires an analysis of the shock absorber system under dynamic load conditions to see how it works in the worst-case scenario.

A safe vehicle must be able to operate and maneuver in a wide range of road conditions. Good contact with the road should be maintained while fast maneuvering. Suspension is the term given to a system of springs, shock absorbers, and rods connecting a vehicle with wheels. The shock absorber has a significant effect on handling and safety. Shock absorbers are also critical to the interaction of tires off the road, which includes the tendency of tires to slip off the road. It affects braking, steering, cornering, and overall stability. Generally, shock absorbers should be replaced when the mileage exceeds a certain value. To make sure what the technical condition is, the best solution is to check the condition of the shock absorber. There are several methods for testing the damper condition. One of the tests is the analysis of damping characteristics using a dynamic test on a special stand. It is complicated and expensive because the machine is very expensive but provides reliable test results.

Tests of shock absorbers are performed at test stands which enable measurement of motion parameters (displacement and input function velocity) as well as the force (shock absorber resistance) with which a shock absorber responds to a kinematic input function. Such stands are typically used in shock absorber durability tests. Testing of shock absorbers at such a stand makes it possible to establish graphs depicting the damping forces in the function of displacement and linear velocity of the shock absorber's piston rod against its housing (Fig. 1).

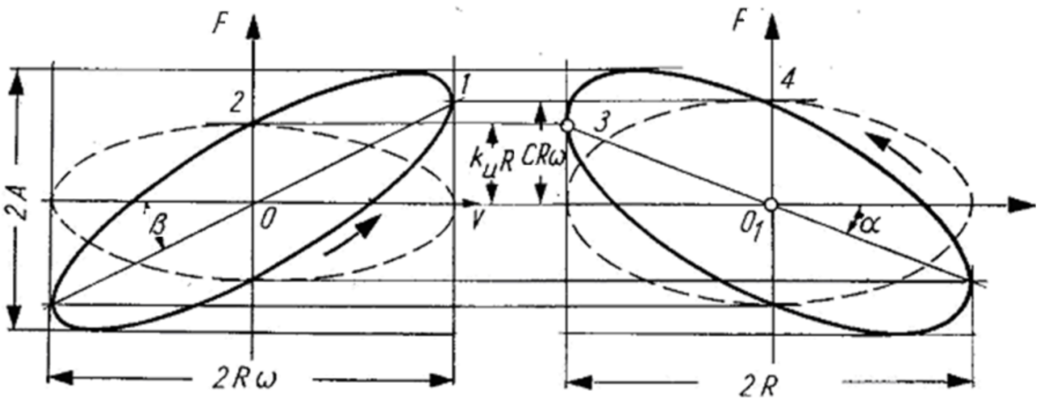

Fig. 1. Theoretical diagram of shock absorber operation (damping force in function of displacement 2R and linear velocity 2ωR [8]

In many cases, it is enough to simplify the damping characteristics of a shock absorber so as to bring them down to a linear coefficient to study the dynamics of suspension systems in automotive vehicles. The said coefficient (shock absorber damping constant c) equals the ratio of the force given by the point of intersection of the work curve and the axis of abscissae (see Fig. 1) to a product of forced angular frequency ω and arm length R. Symbol A on fig.1. is amplitude of force.

$$c = \frac{cR\omega}{R\omega} = \frac{F_4}{R\omega} = tg\beta \tag{1}$$

where c -shock absorber damping constant, ω - forced angular frequency, F_4 - force -point 4 on diagram, and R - arm length.

The value of conventional elasticity constant k_u equals the tangent of the inclination angle of a line cutting through the beginning of the coordinate system and through the point of tangency of the characteristic curve with lines parallel to the X-axis (see Fig. 1) for the extreme forced travel values.

$$k_u = \frac{F_3}{R} = tg\alpha \tag{2}$$

where k_u - conventional elasticity constant, F_3 - force - point 3 on diagram, and R - arm length.

Regarding the tests of shock absorbers of contemporary cars with non-symmetrical damping characteristics adjusted by overflow valves, one obtains work graphs that diverge from the ellipsis. In many cases, the force of the shock absorber's resistance against displacements can be described with satisfactory accuracy by polynomials of the n^{th} order (typically 2^{nd} or 3^{rd}) [1-4, 12-17].

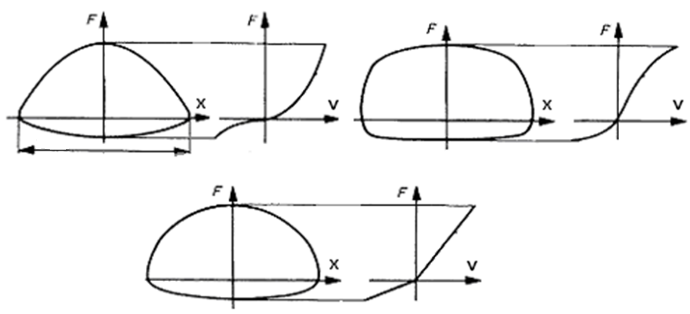

Fig. 2. Sample graphs of automotive shock absorber operation and corresponding damping characteristics (progressive, degressive, and linear): x-displacement, v-linear velocity, and F-force

The apparatus used to study shock absorbers is commonly referred to as test stands, and they make it possible to measure the values of forces, displacements, and velocities on variable input function parameters (angular velocity and/or piston rod travel values). Such a test stand is to be found at the

Laboratory of Motor Vehicle Dynamics of the Faculty of Transport of the Silesian University of Technology [8-10].

The aim of the work was to determine the damping characteristics established for the two operating statuses, including for the emergency status, to make it possible to run simulations by taking the shock absorber's technical condition into account.

2. METHODS AND MATERIALS

The object of the tests was a shock absorber by Monroe, typically installed on Audi A6. The measurements were performed in the *fail-safe* mode, i.e., when the solenoid valve controlling the oil flow through the bypass was not in operation, and for comparison, with the 1.7 A and 9 V power supply on, i.e. close to the level theoretically defined by the manufacturer as the near-maximum damping value. The current value limitation below the maximum of 2 A was owing to the test object considerably heating up (caused by the operating parameters that were more encumbering than those of standard vehicle operation and on account of replacing the PWM control signal with direct current) [5-7, 11]. The shock absorber subject to the tests as well as the bypass control valve has been schematically shown in Fig. 3.

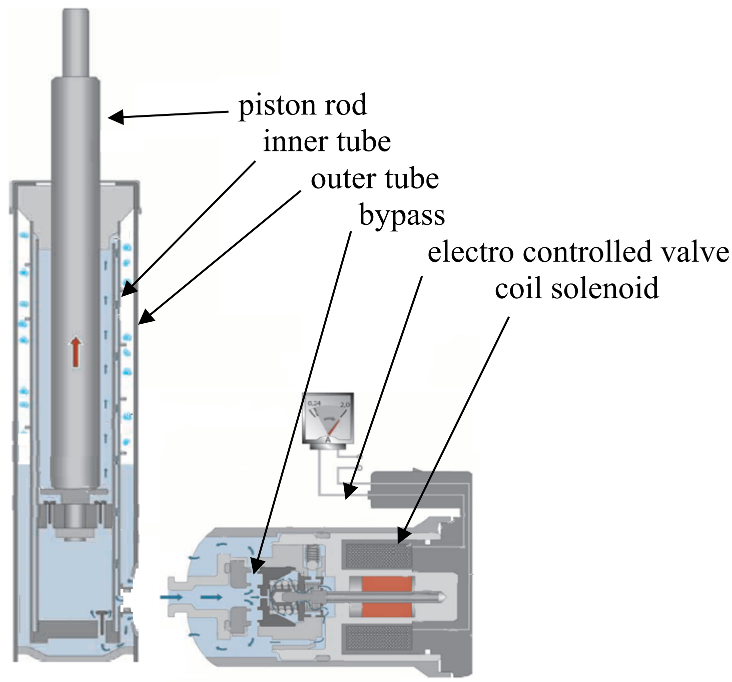

Fig. 3. Bypass-controlled shock absorber subject to tests and control valve

3. MEASURING STATION

In the measurement system addressed in the paper, a type CL 16 bidirectional strain gauge with the operating range of ±2.5 kN was used for direct force measurements, whereas the PTx 100 series transformer-type linear displacement transducer together with the MPL 104 displacement gauge was used to measure displacements. The signals received from the transducers were recorded using the SigLab 20-220A two-channel analyzer and stored on a computer hard drive in a format compatible with the Matlab software. A diagram schematically representing the measurement system used to measure the changes in the forces generated by the shock absorber and the instantaneous displacement values at the test stand has been provided in Fig. 4.

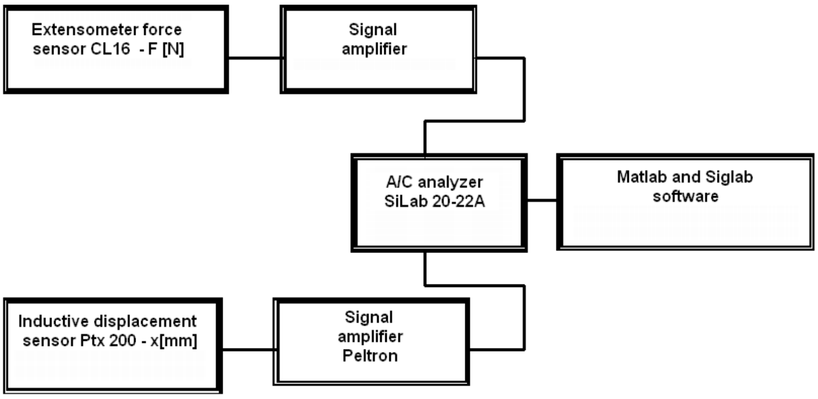

Fig. 4. Measurement system diagram

Once the bypass-controlled shock absorber had been mounted on the test stand, the shock absorber was warmed up before the actual measurements were taken. The test procedure first involved heating up the shock absorber by running it for two minutes. Depending on the different input function velocities changed according to the frequency, the time of a single measurement ranged between 20 and 60 s, thus ensuring that at least 20 operating cycles were recorded. The tests were performed on the shock absorber travel of 60 mm. The bypass-controlled shock absorber mounted on the test stand has been shown in Fig. 5.

Fig. 5. Bypass-controlled shock absorber mounted on the test stand

For the shock absorber mounted on the test stand, time curves of force signals and of linear displacement were recorded. The signals thus recorded were smoothed using a low-pass filter. For signal filtering in Matlab, FIR1 function was used. FIR1 implements the classical method of windowed linear phase FIR digital filter design. Lowpass, linear phase FIR filter with cutoff frequency and Hamming-windowed was used. The time curves recorded for the signals before and after filtering have been shown in Fig. 6.

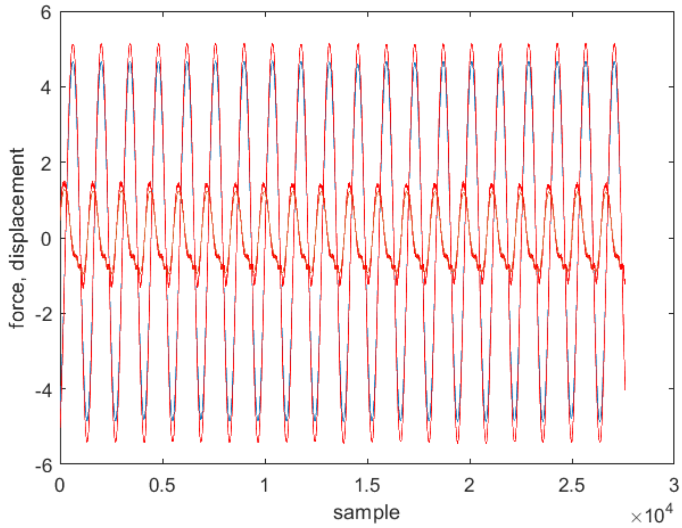

Fig. 6. Time curves recorded for force signals and displacement before (red) and after filtering (blue)

Further proceedings included determination of averaged loops depending on the input function velocity, each time based on at least 15 complete recorded operating cycles comprising the motion of compression and expansion. Fig. 7 depicts the averaged loops of force changes in the function of displacement (a) and of linear velocity (b).

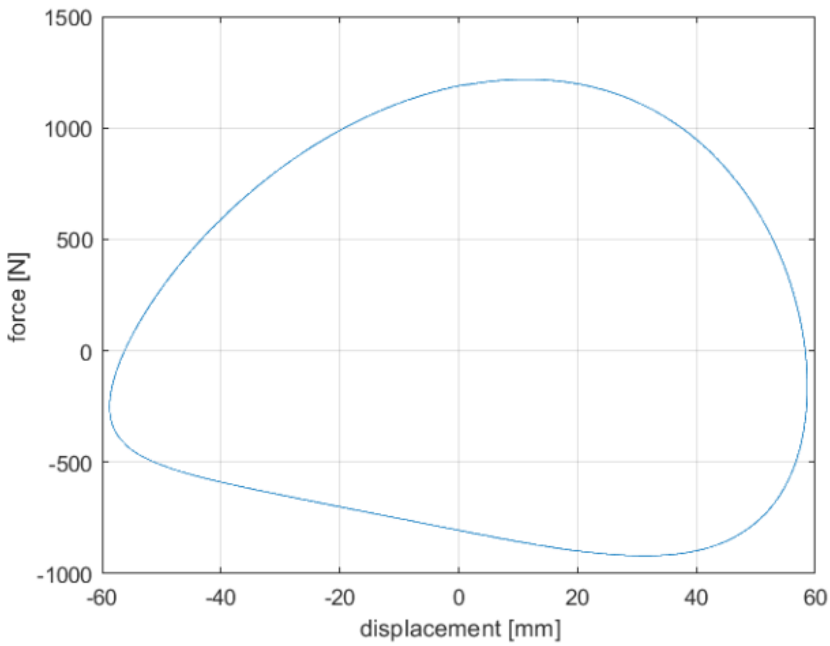

Fig. 7a. Averaged loops of force changes in the function of displacement

4. RESULTS

According to the procedure discussed in this paper, tests were performed for selected frequencies and travel values for two statuses, i.e. the *fail-safe* status, in which the solenoid valve controlling the oil flow through the bypass was not in operation, and – for comparison – in a condition with the 1.7 A power supply enabled. Some typical results thus obtained have been provided in Figs. 8–11.

The tests were performed for selected frequencies (Tab. 1) and constant amplitude value (stroke) 60 mm. The maximum values of the linear velocity of excitation resulting from these parameters are presented in Table 1. The colors of loops in Figs. 8-11 correspond to the frequency of excitations shown in Table 1.

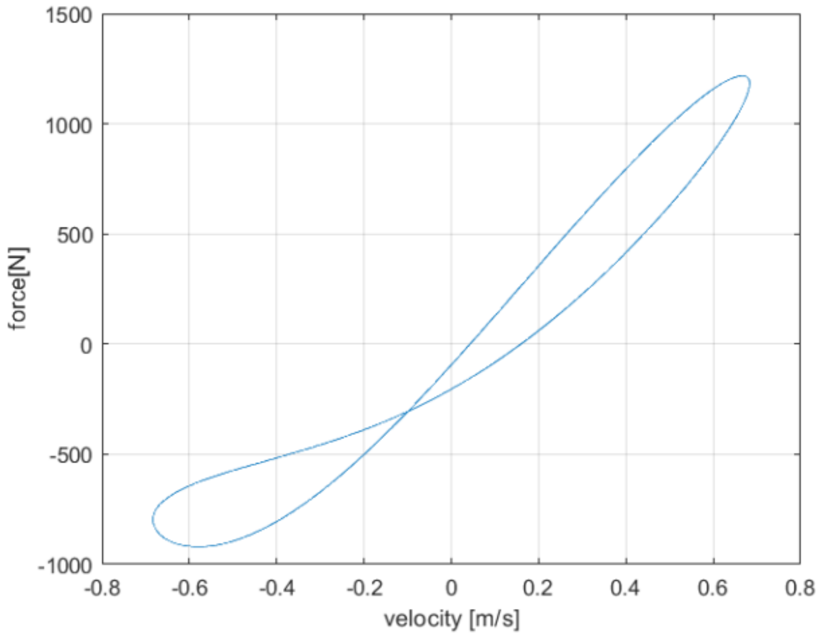

Fig. 7b. Averaged loops of force changes in the function of linear velocity

Table 1

Values of input selected parameters

Frequency [Hz]	Angle velocity [1/s]	Amplitude [m]	Linear velocity [m/s]
0,440	2,761	0,06	0,1598
0,684	4,296	0,06	0,2651
0,919	5,770	0,06	0,3612
1,148	7,210	0,06	0,4496
1,382	8,681	0,06	0,5341
1,618	10,160	0,06	0,6123
1,853	11,639	0,06	0,6831

Based on the plotted graphs of force changes in the function of linear velocity, point characteristics were determined as force values corresponding to the maximum linear velocity. The characteristics thus established were approximated by polynomials functionally describing changes to the damping force (Fig. 12). The characteristics described by that means can be used for purposes of simulation tests, e.g. in multibody system analyses.

5. DISCUSSION

Currently, among the test methods for shock absorbers installed in automotive vehicles, the EUSAMA method is most often used. The advantage of its use is the unification of construction of test stands, large statistical material, and no need for comparative database. Its disadvantage is a simplified multi-state qualification system, and therefore a simple system for assessing the technical condition of the shock absorber, large width of individual condition ranges, no damage identification, and ambiguity of results for rear axle shock absorbers, especially for vehicles with low own weight.

Other methods (BOGE, MAHA, and HOFFMAN) currently have limited applications. The recommendation in some cases of ambiguous assessment is the control of shock absorbers at test stands allowing for verification and comparison with reference characteristics. This is the only method that allows for unambiguous qualitative and quantitative verification. However, it is difficult in practical application because it requires disassembly of the shock absorber from the vehicle and testing on an expensive test stand.

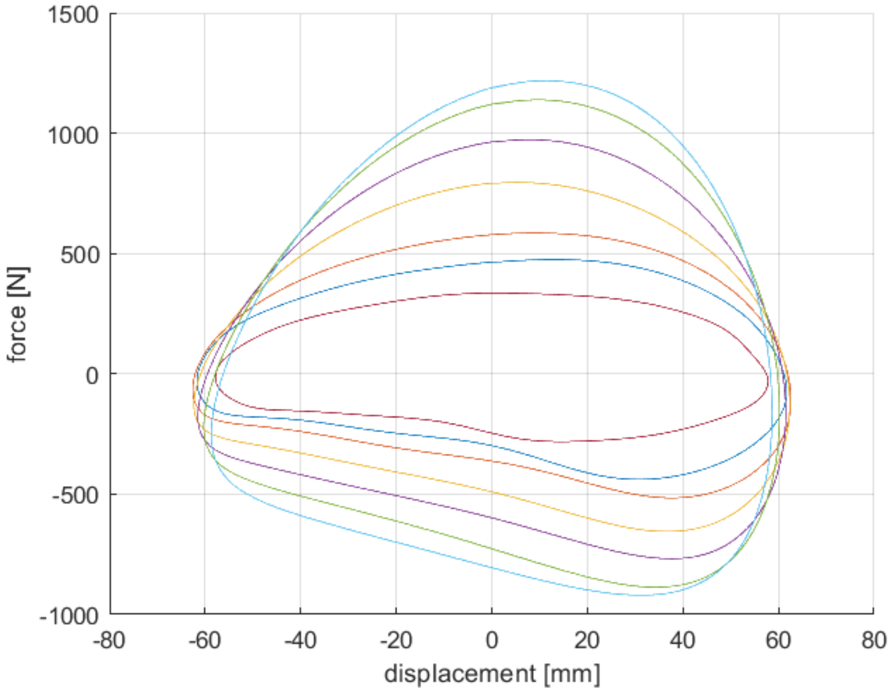

Fig. 8. Graph of force changes in the function of displacement for the shock absorber operating in the fail-safe mode; colors correspond to values of input function frequency (red - 0,44 Hz, dark blue - 0,68 Hz, orange - 0,92 Hz, yellow 1,15 Hz, magenta-1,38 Hz, green - 1,62 Hz, and light blue – 1,85 Hz)

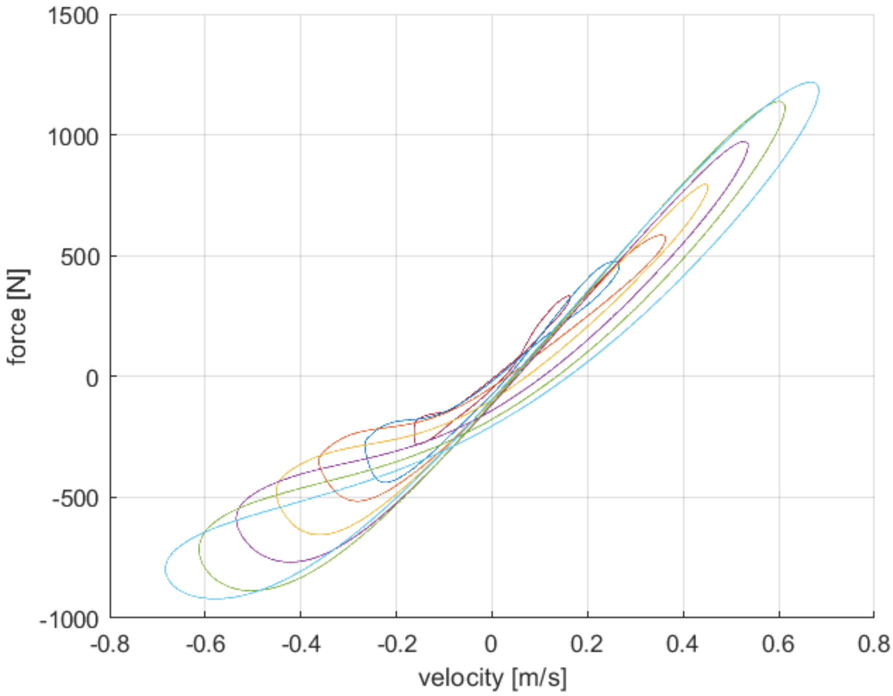

Fig. 9. Graph of force changes in the function of linear velocity for the shock absorber operating in the fail-safe mode; successive colors correspond to increasing values of input function frequency (red - 0,44 Hz, dark blue - 0,68 Hz, orange - 0,92 Hz, yellow – 1,15 Hz, magenta-1,38 Hz, green -1,62 Hz, and light blue – 1,85 Hz)

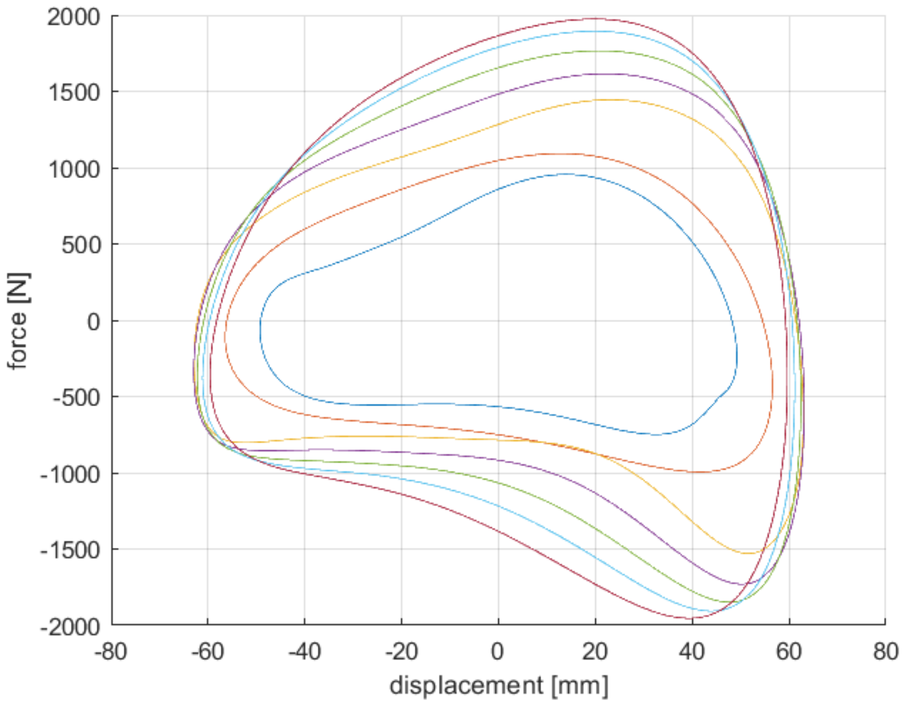

Fig. 10. Graph of force changes in the function of displacement for the shock absorber operating in the power supply mode; successive colors correspond to increasing values of input function frequency (red – 0,44 Hz, dark blue - 0,68 Hz, orange - 0,92 Hz, yellow – 1,15Hz, magenta-1,38 Hz, green - 1,62 Hz, and light blue – 1,85 Hz)

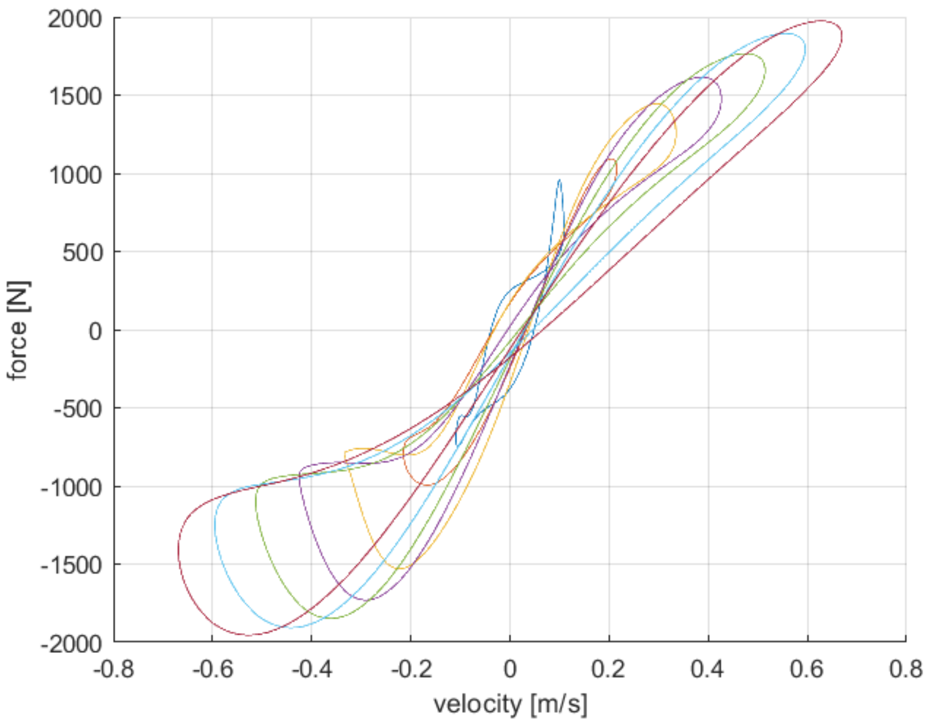

Fig. 11. Graph of force changes in the function of linear velocity for the shock absorber operating in the power supply mode; successive colors correspond to increasing values of input function frequency (red – 0,44 Hz, dark blue - 0,68 Hz, orange - 0,92 Hz, yellow – 1,15 Hz, magenta-1,38 Hz, green -1,62 Hz, and light blue – 1,85 Hz)

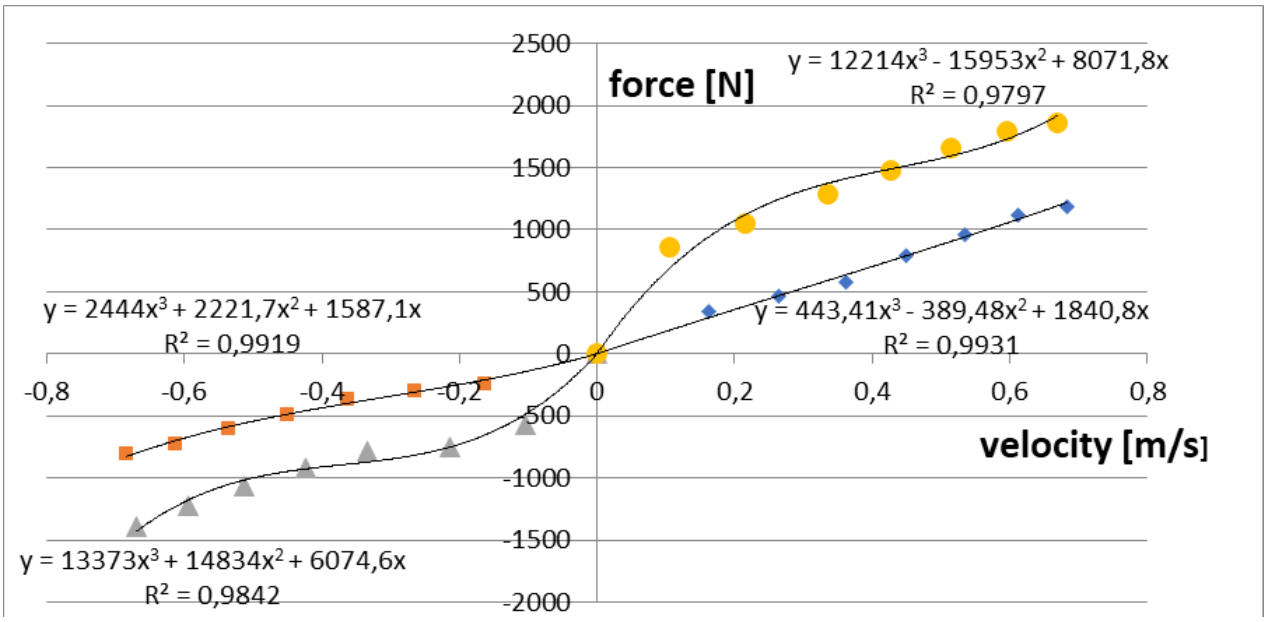

Fig. 12. Established point characteristics and polynomials approximating these characteristics – colors red and blue represent the fail-safe mode, whereas yellow and gray correspond to the power supply mode

6. CONCLUSIONS

The tests performed under the study in question made it possible to plot the graphs of changes to the damping force of a bypass-controlled shock absorber in the function of displacement and in the function of linear velocity. With reference to the graphs thus obtained, damping characteristics were determined as functions approximating the damping force against linear velocity. The characteristics described in this way are most frequently used in simulation studies of dynamics of automotive vehicles. The vibration damping characteristics thus obtained can be described as non-linear and asymmetrical. The graphs plotted with reference to the tests imply that the shock absorber's resistance force is lower during compression than during expansion, which stems from the characteristics assumed for the condition of reduction of the forces occurring in the shock absorber compression motion phase to be fulfilled. The characteristics established for the two operating statuses, including for the emergency status, make it possible to run simulations by taking the shock absorber's technical condition into account. The description of the approximating functions envisaged in the form of polynomials may be a description of damping characteristics, e.g. in simulations of multibody-type systems (Adams /Car, etc.).

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