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LABORATORY RESEARCH OF THE CONTROLLABLE HYDRAULIC DAMPER

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The main focus is on the hydraulic shock absorbers employed in vehicles, in the context of parameters associated with the damping force control for the given direction of action and with the energy dispersion. Two types of shock absorbers are examined: those with constant and variable parameters of damping force adjustment. Shock absorbers that were subject to research investigations included a commercially – available shock absorber with fixed parameters and an experimental absorber with variable parameters. The paper summarises the problems experienced during modelling of hydraulic shock absorbers when the equivalent damping ratio is sought. Damper properties and the methods of determining them are outlined. The laboratory tests performed on these shock absorbers are considered in more detail and relevant characteristics are provided for the sake of comparison. The reasons why shock absorbers with fixed parameters have been in widespread use for many years are now investigated, highlighting the aspects which prompt the research work on shock absorbers with variable parameters.

Key words: hydraulic damper, energy dispersed, semiactive vibration control.

1. Introduction

Shock absorbers are used to dampen or reduce the amount of vibrations of the sprung weight of a vehicle. Conventional hydraulic and hydro-pneumatic shock absorbers are now in most widespread use. They are double-acting and their actions are unsymmetrical: stronger for the release and weaker for the bending movement. Hydraulic shock absorbers with telescopic structure (single or twin tube) are commonly applied. Hydraulic shock absorbers disperse the energy of vibrations or bounces by overcoming the flow resistance involved in the liquid flows through calibrated channels or orifices. Shock absorbers with fixed parameters have been long employed in vehicles, despite certain deficiencies: high vulnerability due to contamination of the working fluid, operating characteristics varying with temperature, the necessity to maintain tight-proof connections, problems with damping control depending on the road conditions.

Along with improvements of the conventional shock absorbers, research work is now under way to design shock absorbers with variable parameters, also known as controllable dampers which utilise devices that enable the variations of the operating characteristics during the vehicle ride. This group includes shock absorbers utilising fluids controlled by mechanical, electromagnetic, electronic systems allowing for continuous or stepwise variations of the damping characteristics. The control system presets the cross-section available for flow in throttling valves, thereby controlling the flow rate. Alternatively, the controlled parameter is rate of fluid filling (in the working volume or an additional air bag). Thus the damping force can be adjusted to the riding speed or the road quality. However, these solutions are now more costly and less reliable than the conventional dampers. The control of damping properties seems to be a most desirable feature so it is reasonable to expect that all the shortcomings will be progressively overcome.

Hydraulic shock absorbers with fixed and variable parameters of the damping force adjustment are compared in further sections. A group of shock absorbers with fixed parameters includes a commercially available telescopic (single tube) shock absorber employed in motorcycles. This is a variation of a hydraulic piston cylinder with a single-acting piston rod, operating as a passive device. Unlike most conventional acting devices, it has no external openings providing any connection with the working chambers. That is why the liquid might flow only from one chamber to the other through the valve assembly fitted in the piston. The valve assembly is made from carefully selected and hydraulically connected throttling gaps and return valves. From the standpoint of hydraulics, we get a system of non-controllable throttling and return valves, their properties dependent on the velocity and direction of the piston movement. The other group of shock absorbers includes an experimental damper comprising a hydraulic cylinder, a servo-valve and an electronic module. The servo-valve here is an equivalent of the valve assembly in the piston of conventional hydraulic shock absorbers, but unlike conventional solutions it enables a continuous adjustment of damping force adjustment parameters through the control of resistance of a liquid flow effected by the electronic module. In this case the chambers are connected to an external control unit (a servo-valve) so they can be connected via a regulated valve cross-section.

2. Problems associated with modelling of hydraulic shock absorbers

Modelling and simulations of vibration reduction units utilizes chiefly the systems of several DOFs. They are governed by equations of motion formulated in accordance with the Newton principles of dynamics or the Lagrange equations.

Certain physical parameters in the equations of motion are directly measurable, others are derived from the manufacturer's data, otherwise the identification procedure is required.

Damping properties of shock absorbers are determined on the basis of simulation tests made on mathematical models or in the course of experiments on real objects. In order to simulate the vibration reduction device utilising a hydraulic shock absorber, it is required that a number of parameters associated with absorber structure and properties of the working fluid should be established first. The damping force is generated by the flow resistance of the hydraulic medium forced through from one chamber to the other via the shaped throttling holes [1, 7]. In the general case the damping force F_d is expressed as

$$F = c\dot{x}^n + F_{st} \tag{2.1}$$

where: c – damping ratio, $\dot{x}$ – piston velocity with respect to the cylinder, n – coefficient associated with design objectives.

The factor c present in Eq. (2.1) is a nonlinear function of displacement and velocity, temperature of the working fluid as well as geometry and mechanical parameters of the valve assembly in the piston.

Characteristics of dampers used in most practical applications for the given operating ranges are:

- progressive, for $n > 1$,
- linear, for $n = 1$,
- decreasing, for $0 < n < 1$.

Accordingly, the damping characteristics can be adjusted to the particular needs. A damper can be designed with a progressive-decreasing characteristics, which means that in one direction (tension) the relationship between the damping force and velocity is progressive, whilst in the opposite direction (compression) it is a decrease. However, the theoretical analysis of working fluid flow through a system of throttling holes and return valves for certain solutions might prove to be extremely complicated and time-consuming, requiring advanced and extended nonlinear mathematical models taking into account the flow processes and thermodynamic phenomena in the working fluid as well as a large number of parameters characterising the geometry and mechanical parameters of the valve system in the piston. To obtain reliable values of these parameters is another time-consuming task. Thus the equivalent damping factor is difficult to obtain, even though a simplified mathematical model might be applied. Valve systems in shock absorbers might vary in their structure and operating principles. The difference lies in the number of available flow paths, which depends on the piston stroke and the direction of its motion. Factors that affect the damping force include the difference in the effective areas of the piston, responsible for the excess of shortage of volume in one of the working chambers. That is why a compensating volume filled with compressed gas has to be provided at the cylinder end [2, 3].

In conventional solutions the damping force is a nonlinear function of several design parameters, which further complicates the modeling task. Besides, response force of shock absorbers is modeled as the force of viscous damping and friction resistance in sealed connections.

3. Damper properties

In the traditional approach the damper behaviours are described by two characteristics:

1) hysteresis loop – obtained by measurements of response force as the function of displacement (mainly used to estimate the energy dispersion),
2) damping force vs. relative piston velocity – chiefly used in the automotive industry as it is convenient for simulations of vibration reduction systems.

The first characteristic enables the estimation of energy dispersed by the damper during one cycle. Its shape and the coordinate value indicate the type of the acting damping forces (viscous, Coulomb, structural) [6]. The other is commonly applied in the automotive industry as its shape helps to predict the damper behaviour in the given conditions: vehicle identity, type of ride (sports car, tourist car) and road conditions. These issues are addressed in more detail in [3, 4].

The identity of the applied kinetic excitations has a major bearing on the results of laboratory tests. In the course of the experimental programme the following exciting signals were considered:

- sine signals – to determine the damping force vs. displacement,
- triangular signals (tooth-shaped, symmetric waveform) – to determine the static relationship between the damping force and velocity.

Application of a triangular signal allows to obtain the constant piston velocity with respect to the cylinder, hence the damping force can be measured in steady-state conditions. The distribution of the experimental data is most uniform and dynamic errors due to working fluid inertia can be avoided.

4. Laboratory tests

The laboratory tests were conducted in the Department of Process Control AGH-UST using the specialised experimental set-up, shown in Fig. 1, comprising a vertical framework structure with horizontal platforms, an electro-hydraulic shaker 1 placed in the lower part of the frame and driving the lower platform 2, the hydraulic supply station (behind the framework structure) and the measurement-control cab (not seen in the picture) with all the vital equipment. The shaker with an interface allows to generate displacements of precisely controlled shape, amplitude and frequency. Signals reproducing the real-life conditions can be generated, too. The tested shock absorber assembly 3 is fixed in between the platforms, the lower one being connected to the shaker. Depending on the test program and the structure of the tested unit (1 DOF, 2 or 3 DOF), the upper platform 4 might move freely in the vertical direction or be immobilised. Damping force is measured with the use of tensometric force sensor 5. The experimental set-up is described in more detail elsewhere [5].

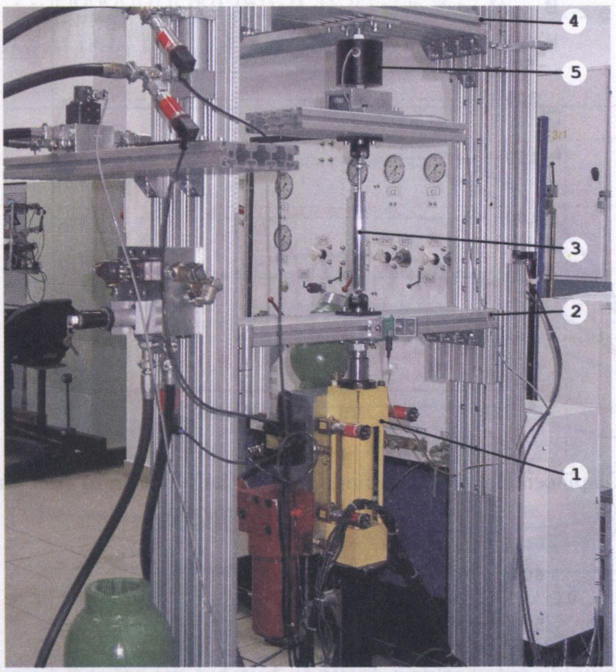

FIG. 1. Experimental stand with a commercially available shock absorber mounted in between the platforms.

Tests were performed on two hydraulic shock absorbers: one with fixed, the other with variable parameters of the damping force adjustment. A commercially available shock absorber intended for motorcycles is considered in the first part. In further sections a novel damper design is proposed with a continuously controlled damping force. The analogous laboratory tests were conducted to support the theoretical considerations. Of particular interest is the amount of energy dispersed by the tested shock absorbers.

4.1. Testing of a commercially-available shock absorber

The commercially-available shock absorber was tested to find the fundamental characteristics listed in the Sec. 3. In the first place the response force was determined as a function of relative piston velocity. During the experimental program the upper platform connected with the piston rod joint was immobilized whilst the lower platform, connected with the cylinder joint, would reproduce the triangular displacements generated by an electro-hydraulic shaker. The period of the generated signal would cyclically decrease after two cycles with a con-

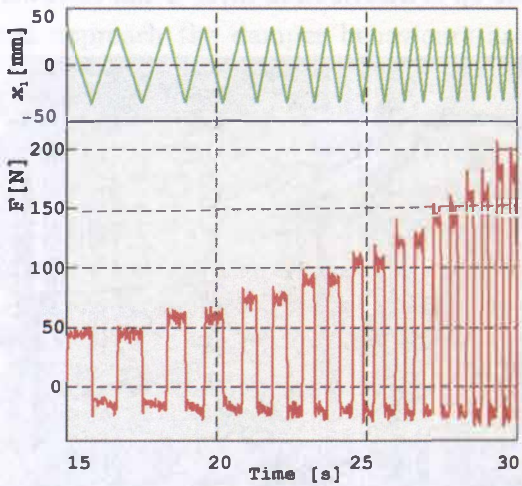

FIG. 2. Displacement and damping force in the function of time in the part of the experimental program.

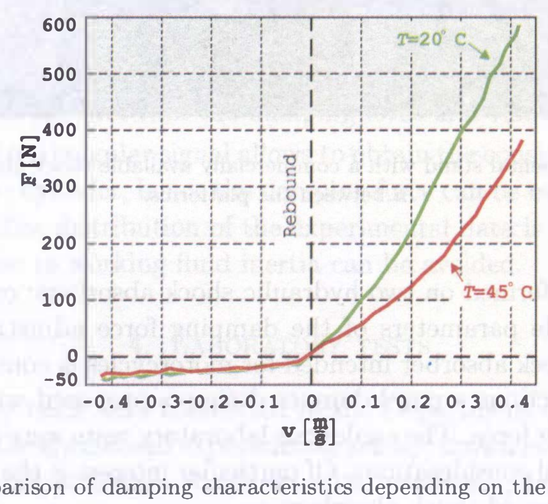

FIG. 3. Comparison of damping characteristics depending on the piston velocity for two oil temperatures.

stant velocity. The whole experiment lasted 60 s, to eliminate the changes of oil temperature as a cylinder. Some portions of thus registered damping force and piston displacement as a function of time are shown in Fig. 2. Characteristics obtained on the basis of experimental results are shown in Fig. 3. It is readily apparent that the damping force is affected by the temperature of the working fluid. Its increase from 20 to 45° C brings about a nearly two-fold reduction of the damping force.

In further experiments the damping force was determined as a function of displacements. The shaker would reproduce the displacements in the shape of a sine signal with a constant amplitude 15 mm, corresponding to one half of the nominal stroke of the shock absorber. The frequency was varied cyclically from 0.5 to 4 Hz, yielding the family of characteristic curves shown in Fig. 4. The working fluid temperature during the tests was $T = 45°$ C.

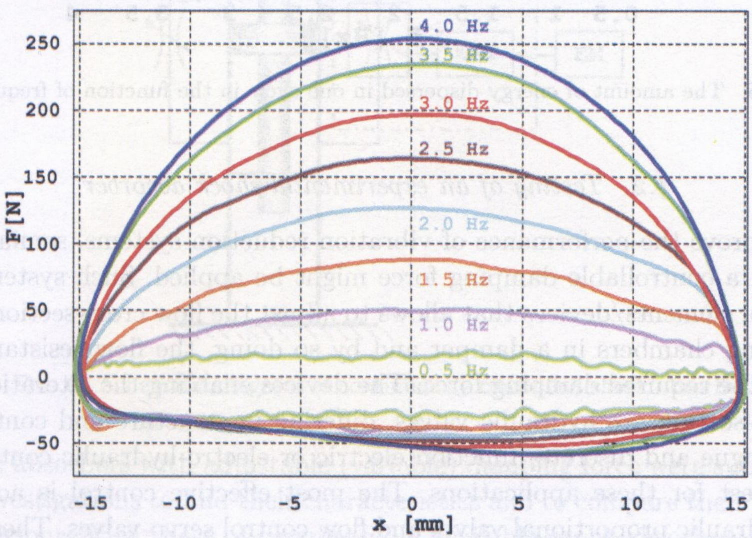

FIG. 4. Damping force as the function of piston displacement obtained by altering the vibrations frequency.

The area bounded by the curve for the given frequency is the measure of the amount of energy dispersed by the shock absorber during one period of amplitude. It is equal to the work performed by the piston. The energy is determined for each preset frequency of applied excitations as the sum of elementary trapezoids with the height dx, in accordance with the formula

$$E = \sum_{i=1}^{n-1} \left[(x(i) - x(i+1)) \cdot \frac{F(i) + F(i+1)}{2} \right]. \tag{4.1}$$

The experimental characteristics of the energy dispersed by a shock absorbed as a function of vibration frequency is shown in Fig. 5.

FIG. 5. The amount of energy dispersed in one cycle in the function of frequency.

4.2. Testing of an experimental shock absorber

To improve the performance of vibration reduction systems, semiactive systems with a controllable damping force might be applied. Such systems utilize the control elements/devices that allows to adjust the flow cross-section between the working chambers in a damper and by so doing, the flow resistance is adjusted to the required damping force. The devices enabling the alterations of the flow cross-section are hydraulic valves, differing in structure and control strategy. Analogue and discrete- function electric or electro-hydraulic control valves are the best for these applications. The most effective control is achieved in electrohydraulic proportional valves and flow control servo-valves. These devices convert the electric low-power input signal into an amplified hydraulic flow-rate signals, proportional to the input signal. Application of such devices and carefully chosen electronic modules in vibration reduction systems affords us the means of continuous control of the damping force. Furthermore, such solutions allow to recover the energy of vibrations, instead of energy dispersion. Research is now under way to recover the energy of vibrations of unsprung mass and to utilise it to power-supply the active vibration reduction systems.

The variable damping characteristics of the shock absorber allows to adjust its parameters to various types of disturbances (road surface profiles, velocity, loading and others).

Figure 6 shows a schematic diagram of an experimental shock absorber comprising a symmetric hydraulic cylinder and an electrohydraulic driving unit. The control unit, consisting of an electrohydraulic servo-valve SV and an electronic module EM, enables the continuous adjustment of the flow intensity (and hence the flow resistance) between the chambers in the cylinder. Flow intensity in the hydraulic amplifier HA is changed by altering the current control of the electromechanical transducer EMT. The control current is usually expressed as the percentage fraction of the nominal current. The cylinder is equipped with a manual-control, high-precision throttling valve TV. When the servo-valve SV is closed, this throttling valve allows for continuous adjustment of the damping force. During the experimental program this valve remained closed.

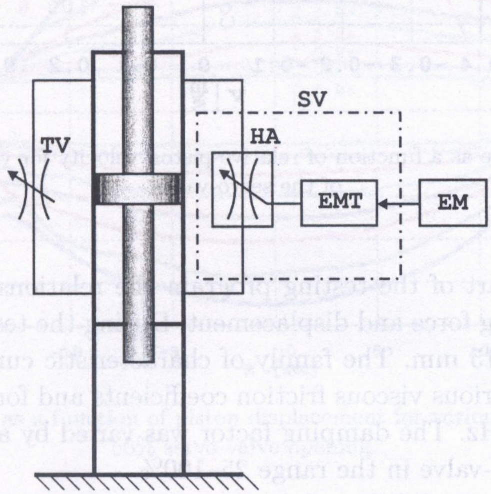

FIG. 6. Schematic diagram of continuously controlled shock absorber.

Shock absorbers with adjustable (variable) damping force were subject to research investigations to find their characteristics and to compare their properties and behaviours with those of commercially available shock absorbers with fixed parameters. The relations between the damping force and the relative piston velocity for various control levels of the servo-valve are shown in Fig. 7. It is readily apparent that the relationship is nonlinear. This function can be linearized by distinguishing three velocity ranges and neglecting the coefficient of static friction. Accordingly, we get the damping force vs. velocity characteristics made up from three straight-line sections with variable slope. This issue is addressed in more detail in [1]. It appears that the approximate coefficient of viscous friction for this unit does not depend linearly on the control opening of the servo-valve. When the servo-valve is wholly opened and the piston has the minimal velocity ($v_{\min} = 0.024$ m/s), the motion resistance depends on the friction in sealing elements. This is a most undesirable effect and hence it has to be eliminated by applying low-friction sealing joints.

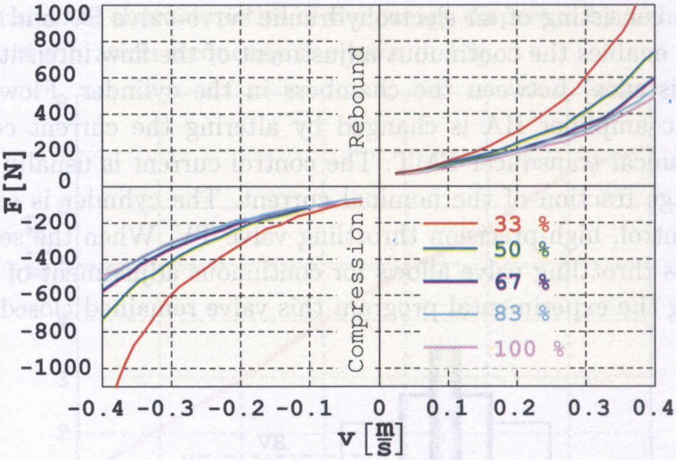

FIG. 7. Damping force as a function of relative piston velocity for various control opening of the servo-valve.

In the second part of the testing program the relationship was determined between the damping force and displacement. During the tests, the amplitude of the sine signal was 15 mm. The family of characteristic curves shown in Fig. 8 were obtained for various viscous friction coefficients and for the constant vibration frequency of 3 Hz. The damping factor was varied by adjusting the control opening of the servo-valve in the range 25–100%.

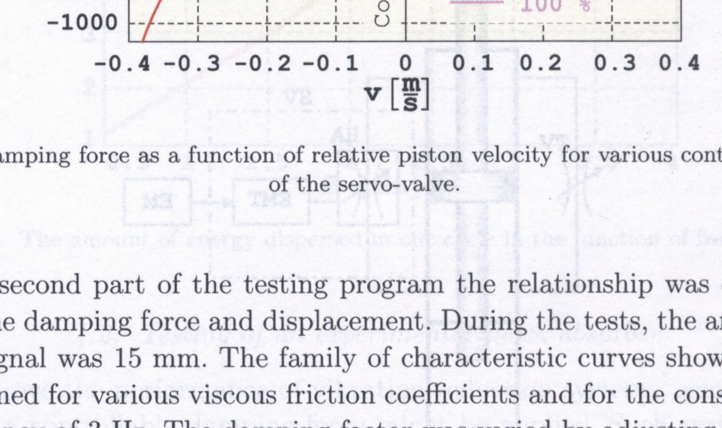

FIG. 8. Damping force as a function of piston displacement for various control openings of the servo-valve, vibrations frequency 3 Hz.

The family of characteristic curves shown in Fig. 9 was obtained in the conditions when the frequency of piston displacement was altered from 0.5 to 4 Hz, for the constant viscous damping factor ($c \approx 1200$ Ns/m, servo-valve open in 50%). However, in the range for which the linearization was performed, the characteristic curves might be better approximated by ellipses (as in the case of ideal model of viscous damping [6]).

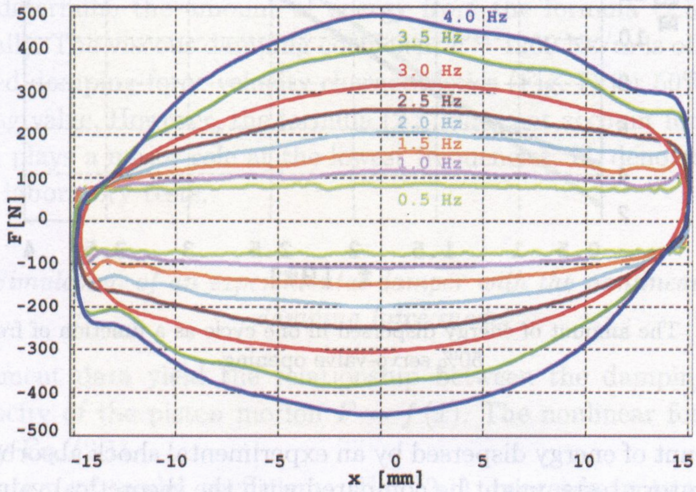

FIG. 9. Damping force as a function of piston displacement for various vibration frequencies, 50% servo-valve opening.

Figure 10 shows the amount of energy dispersed in one cycle as a function of the servo-valve opening and Fig. 11 – in the function of frequency.

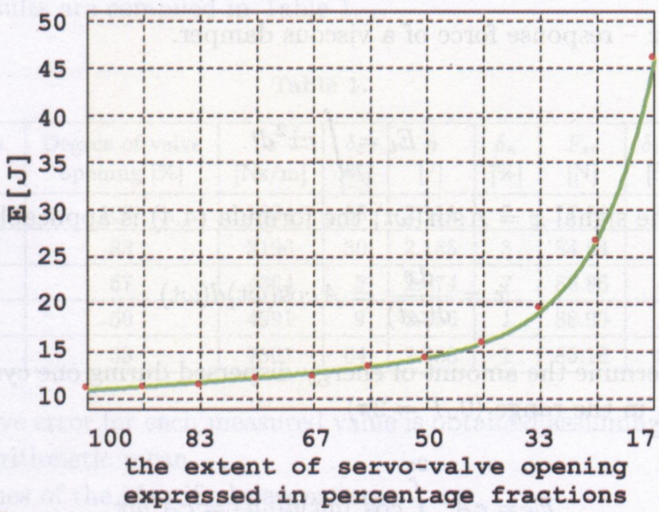

FIG. 10. The amount of energy dispersed in one cycle depending on the servo-valve opening.

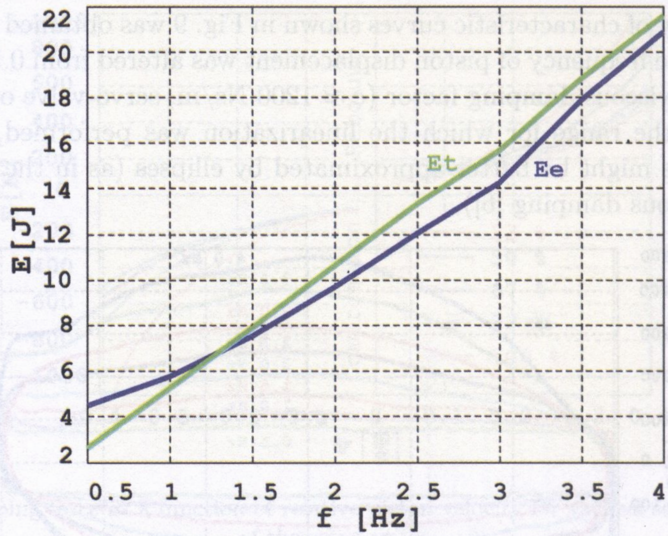

FIG. 11. The amount of energy dispersed in one cycle as a function of frequency, 50% servo-valve opening.

The amount of energy dispersed by an experimental shock absorber, obtained during laboratory tests, might be compared with the theoretical value. As in the case of commercially available shock absorbers, this energy is equal to the work performed by the damper and is expressed by Eqs. (4.2) or (4.3)

$$E_t = \int F_d dx, \tag{4.2}$$

where $F_d = c\dot{x}$ – response force of a viscous damper.

Hence

$$E_t = \int c\dot{x}^2 dt. \tag{4.3}$$

For the sine signal $x = A\sin(\omega t)$ the formula (4.4) is applicable

$$\dot{x} = \frac{dx}{d(\omega t)} = A\cos(\omega \mathrm{t})d(\omega \mathrm{t}). \tag{4.4}$$

Let us determine the amount of energy dispersed during one cycle of damper operation, i.e. in the range $\langle 0, T = 2\pi \rangle$

$$E_t = cA^2 \int_0^{2\pi} \cos^2(\omega t)d(\omega t) = cA^2\omega\pi. \tag{4.5}$$

The amount of energy dispersed by the shock absorber is proportional to the frequency and squared amplitude (i.e. excitation parameters in semiactive vibration reduction systems) and to the viscous damping coefficient (a controllable parameter).

Figure 11 compares the amounts of energy determined experimentally and theoretically, in accordance with the formula (2.5). The amplitude and pulsation utilized to determine the amount of energy from the formula (2.5) are found experimentally. The viscous damping coefficient $c = 1200$ Ns/m is obtained from the linearized damping force-velocity characteristics (Fig. 7) for 50% opening of the throttling valve. However, the formula (2.5) does not account for the friction force, which plays a major role at the lowest frequencies. E_e denotes the energy obtained in laboratory tests.

4.3. Simulation of an experimental damper with the nonlinear damping force model

Measurement data yield the relationship between the damping force and relative velocity of the piston motion $F = f(\dot{x})$. The nonlinear force model is expressed by Eq. (2.1).

Identification of model coefficients, Eq. (2.1), proceeds in two parts. The processes of damper compression and rebound are analysed separately.

4.3.1. Rebound stage. For each measured servo-valve setting a curve is chosen in accordance with Eq. (2.1) and the coefficients c, n, F_{st} are derived accordingly. The results are compiled in Table 1.

Table 1.

No.	Degree of valve opening [%]	c [Ns/m]	δ_c [%]	n [/]	δ_n [%]	F_{st} [N]	δ_{Fst} [%]
1.	100	2834	38	2.111	5	77.82	9
2.	83	3196	30	2.165	3	84.54	1
3.	67	4364	5	2.374	7	88.86	4
4.	50	4991	9	2.256	1	88.99	5
5.	33	7501	64	2.208	1	85.16	0

The relative error for each measured value is obtained assuming that the real value is the arithmetic mean.

Mean values of the identified parameters:

$$n_{\text{av}} = 2.22, \quad F_{st\ \text{av}} = 85\ [\text{N}], \quad c_{\text{av}} = 4577\ [\text{Ns/m}].$$

Relative error is derived from formula (4.6)

$$\delta_{x\%} = \frac{|x - x_{\text{av}}|}{x_{\text{av}}} \cdot 100\%. \tag{4.6}$$

It is apparent (see Table 1) that the coefficient F_{st} and the power exponent n depend on the valve opening in a minor degree only. On the other hand, the value of coefficient c is strongly dependent on the degree of valve opening. For the purpose of next analyses, n and F_{st} are regarded to be constant and the relationship between the damping force and the relative velocity of the piston motion is investigated anew. The assumed force model is governed by Eq. (4.7)

$$F = c\dot{x}^{2.22} + 85. \tag{4.7}$$

Results are summarised in Table 2.

Table 2.

No.	Valve opening [%]	c [Ns/m]
1.	100	3078
2.	83	3363
3.	67	3806
4.	50	4856
5.	33	7591

4.3.2. Compression stage. A similar analysis was performed for the compression stage. The results are compiled in Table 3.

Table 3.

No.	Degree of valve opening [%]	c [Ns/m]	δ_c [%]	n [/]	δ_n [%]	F_{st} [N]	δ_{Fst} [%]
1.	100	2219	57	1.873	13	85.85	5
2.	83	2525	51	1.933	10	79.63	12
3.	67	3196	37	2.059	4	84.12	7
4.	50	4609	10	2.170	1	86.15	5
5.	33	12960	154	2.701	26	117.6	30

The procedure was similar to that used for the rebound stage. The mean values of the power exponent n and the static force F_{st} were determined accordingly.

The force model for the compression stage is expressed by the formula:

$$F = c\dot{x}^{2.15} + 90.67. \tag{4.8}$$

Damping ratios obtained for various levels of valve opening are compiled in Table 4.

Table 4.

No.	Valve opening [%]	c [Ns/m]
1.	100	2847
2.	83	3011
3.	67	3432
4.	50	4481
5.	33	7680

Experimental data and the results derived from (4.7) and (4.8) are compared in Fig. 12.

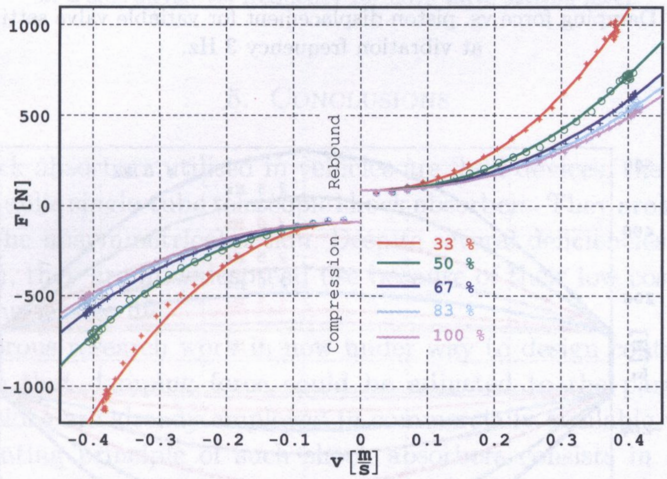

FIG. 12. Comparison of experimental characteristics and those derived from the nonlinear model.

In further considerations the servo-valve setting expressed in percent is replaced by a corresponding damping ratio.

The relationship between the damping force and displacement was established in the next step. Sine signals with the frequency 3 Hz and amplitude 15 mm were applied as displacement signals. The force is derived from formulas (4.7) and (4.8).

Figures 13 and 14 show the damping force vs. displacement characteristics for variable damping ratio (Fig. 13) and frequency (Fig. 14).

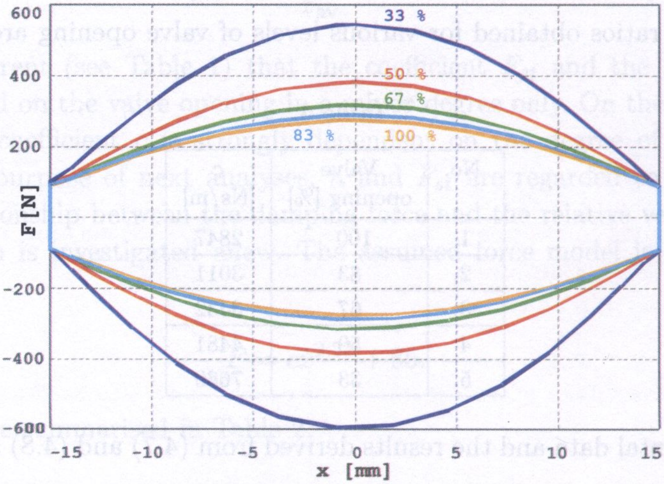

FIG. 13. Damping force vs. piston displacement for variable valve setting levels, at vibration frequency 3 Hz.

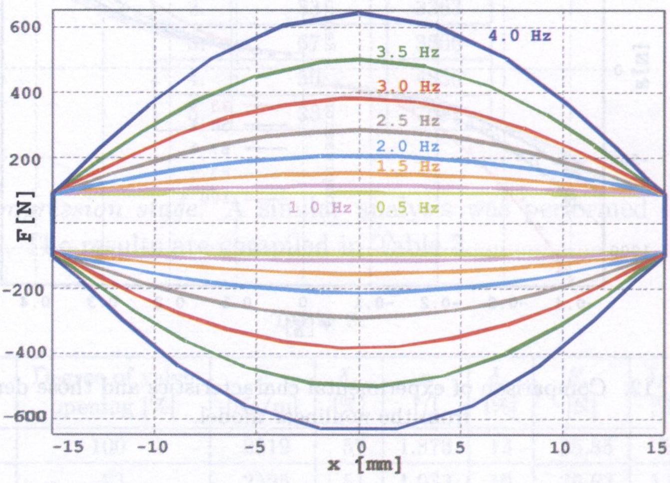

FIG. 14. Damping force vs. piston displacement for variable vibration frequency, at 50% valve setting level.

The amount of energy dissipated by the damper during one cycle of a sine signal, obtained from (4.1), is shown in Fig. 15.

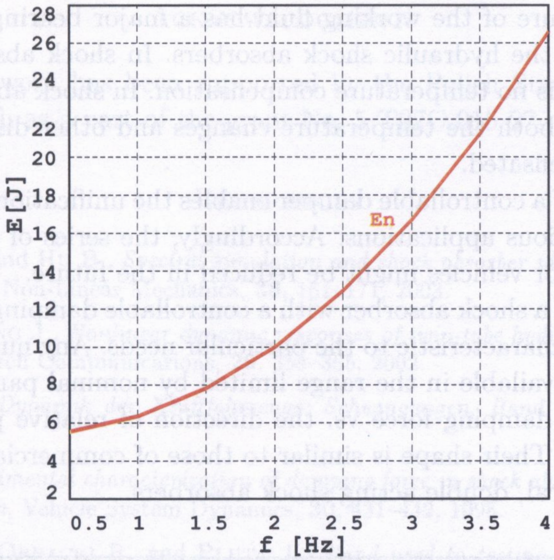

FIG. 15. The amount of energy dissipated by the damper during one cycle of a sine signal vs. frequency for 50% valve setting level.

5. Conclusions

Most shock absorbers utilised in vehicles are fluid devices, the largest group includes hydraulic single-tube telescopic shock absorbers. They are mostly double-acting with the unsymmetrical action. Despite several deficiencies (no damping force control), they are in widespread use because of their low costs, high reliability and long service life.

Most rigorous research work in now under way to design controllable shock absorbers, so that damping force could be adjusted to the particular needs. Certain solutions are already employed in commercially available vehicles.

The operating principle of such shock absorbers consists in absorption of energy of vibrations or shocks by overcoming the flow resistance experienced when a fluid flows through a valve or a group of valves. In shock absorbers with fixed parameters the valve system is placed inside the cylinder, in the mobile piston, which precludes the control action. In the case of controllable shock absorbers, their operating characteristics can be varied using external valves controlling the flow intensity (flow rate) between the chambers. Electrohydraulic flow-control servo-valve allow for the most effective control.

Application of flow-control servo-valves to control the flow resistance between the chambers in the shock absorber allows the use of the semiactive control algorithm.

The temperature of the working fluid has a major bearing on the damping characteristics of the hydraulic shock absorbers. In shock absorbers with fixed parameters there is no temperature compensation. In shock absorbers with variable parameters, both the temperature changes and other disturbances can be adequately compensated.

Engineering of a controllable damper enables the unification of damping characteristics for various applications. Accordingly, the series of types of available shock absorbers for vehicles might be reduced in the future.

Application of a shock absorber with a controllable damping force makes possible to tailor its characteristic to the particular needs. Any number of operating characteristic is available in the range limited by nominal parameters. Selected characteristics of damping force vs. the direction of relative piston motion are shown in Fig. 16. Their shape is similar to those of commercially available ones with unsymmetrical, double acting shock absorbers.

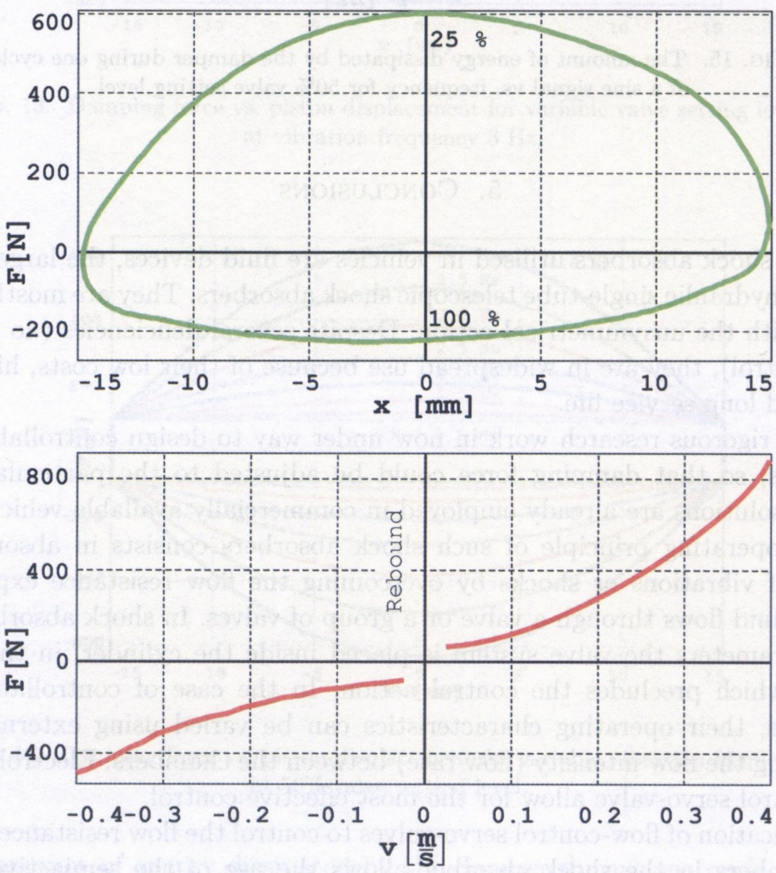

FIG. 16. Shaping the characteristics of a controllable servo-valve.

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