semi-active suspension, damper characteristic, damping ratio

**Grzegorz ŚLASKI**
Institute of Machines and Motor Vehicles, Poznan University of Technology
ul. Piotrowo 3, 60-965 Poznan, Poland
*Corresponding author*. E-mail: grzegorz@slaski.eu

# DAMPING PARAMETERS OF SUSPENSION OF PASSENGER VEHICLE EQUIPPED WITH SEMI-ACTIVE DAMPERS WITH BY-PASS VALVE

**Summary.** The paper presents results of experimental tests of characteristics of semi-active dampers with by-pass valve and results of calculations evaluating suspension damping ratio taking into consideration also installation ratio and vehicle sprung mass changes. The asymmetry of damper characteristic is also investigated and changes in damper damping coefficients versus damper velocity. The papers also compares these values of passive damper used in this car with tested semi-active damper.

# PARAMETRY TŁUMIENIA ZAWIESZENIA SAMOCHODU OSOBOWEGO WYPOSAŻONEGO W AMORTYZATORY PÓŁAKTYWNE Z ZAWOREM OBEJŚCIOWYM

**Streszczenie.** Artykuł prezentuje wyniki badań eksperymentalnych amortyzatorów półaktywnych z zaworem obejściowym, a także wyniki obliczeń oszacowujących bezwymiarowy współczynnik tłumienia zawieszenia z uwzględnieniem przełożenia kinematycznego zawieszenia oraz zmian masy resorowanej pojazdu. Rozpatrzono także asymetrię charakterystyki amortyzatora oraz zmiany współczynnika tłumienia amortyzatora w funkcji prędkości jego pracy. Artykuł porównuje także te wartości dla amortyzatora pasywnego używanego w tym samochodzie z amortyzatorem półaktywnym.

## 1. INTRODUCTION

The aim of a vehicle suspension is to provide an isolation of a vehicle body from road irregularities and to ensure good road holding. The first goal lies within the area of ride analysis and concerns a problem of how to reduce a discomfort experienced by vehicle occupants. The second one lies within the area of handling analysis. The design goal is to minimize both the acceleration of the body and the dynamic tire load, while operating within the constraints of suspension rattle space for a given suspension parameter set.

The tool used for a preliminary design of a suspension parameter set is a linear quarter car suspension model. It allows to find optimal linear damping coefficient and spring stiffness. But real damper has a nonlinear and asymmetric characteristics.

Semi-active suspensions uses dampers not only with nonlinear and asymmetric characteristics but with characteristics having some area of forces between lowest and highest limits of damper characteristics.

The author of this paper did some experimental test to find characteristics of passive and semi-active damper and some calculations to estimate damping ration of a suspension of specific passenger car suspension.

## 2. DAMPING PARAMETERS

There are at least two important damping parameters used to describe a damped system. One of them is a damping coefficient $C$ , second is a damping ratio $\zeta$ .

The damping coefficient can be specified on the base of a damper force characteristics, which indicates how the damping force varies with the damper compression and extension velocities. It can also indicate how the force varies with any other factors important for damper.

But the most important parameters specified by this characteristics and that can be correlated with the ride and handling quality are an overall mean damping coefficient, asymmetry factor and progressivity factor.

The damping coefficient values are measured in Ns/m. Multiplying damping coefficient by damper speed we get value of the damping force. This parameter is good to characterize the damper as a suspension component or even to characterize the damping coefficient of suspension but it does not give information on the behavior of this suspension.

To know the influence of value of damping coefficient $C$ on suspension behavior, we need to compare this coefficient with the mass $m$ and the spring element stiffness $K$ of analyzed suspension. These parameters are related by a damping ratio described as:

$$\zeta = \frac{C}{C_{crit}} = \frac{C}{2\sqrt{mK}} \qquad (1)$$

The damping ratio is a proportion of system damping coefficient $C$ to its critical damping coefficient $C_{crit}$. System is critically damped when the damping ratio $\zeta$ prevents overshoot.

Passenger cars usually have an effective mean damping ratio around 0.3. This value gives less vehicle control but bigger vehicle comfort. Racing cars can have a damping ratio approaching 1.0. Depending on road conditions the optimum ride may occur for damping ratio around 0.2 and optimum handling occur for a damping ratio for example even equal 0,8.

The actual value depends upon ride/handling compromise adopted to the particular car.

A damping ratio influences also the damped natural frequency. For a damping ratio of 0.2 the damped natural frequency is only about 2% less than the undamped value but for damping ratio of 0.8 it is 40% less than when undamped.

When the damper is in condition of excessive wear the damping ratio may drop below 0.1 value. This value gives poor control of oscillations.

## 3. EXPERIMENTAL TESTS OF DAMPERS FORCE CHARACERISTICS

For experimental tests of a passive and a semi-active damper characteristics special test rig was used with dSpace data acquisition and control system. A dedicated control program was build and used with dSpace hardware and software during tests of semi-active damper due to a necessity to control valve by an electric current [4].

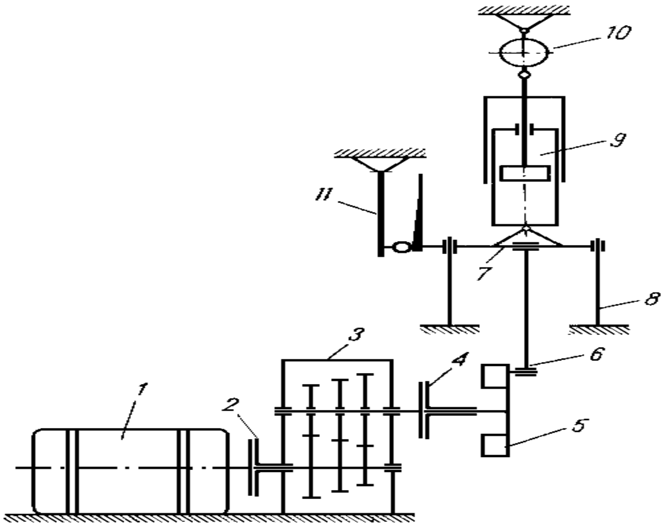


Fig. 1. Kinematic diagram of test rig for shock absorbers characteristic measurements (1 – electric motor, 2 – clutch, 3 – transmission, 4 – safety clutch, 5 – flywheel with eccentricity, 6 – connecting rod, 7 – slider, 8 – guide bar, 9 – tested shock absorber, 10 – force transducer, 11 – shock absorber displacement transducer)

Rys. 1. Schemat kinematyczny stanowiska do badania amortyzatorów wymontowanych z samochodu (1 – silnik elektryczny, 2 – sprzęgło, 3 – skrzynia biegów, 4 - sprzęgło bezpieczeństwa, 5 – koło zamachowe z mimośrodem, 6 – korbowód, 7 – suwak, 8 – prowadnica, 9 – badany amortyzator, 10 – pierścień stalowy z naklejonymi tensometrami do pomiaru siły, 11 – belka z naklejonymi tensometrami do pomiaru przemieszczenia)

### 3.1. Damper characteristics

The data obtained during tests, after processing, allowed to get the passive and the semi-active damper characteristics. An effective suspension damping force characteristics is a characteristics of a damping force when measured at the wheel. This is as opposed to simply measure shock absorber damping force characteristics alone. To get the values damping forces of the shock absorber need to be multiplied by a shock absorber installation ratio. For the investigated vehicle and its rear suspension this ratio is equal to 0,68. The measured and calculated characteristics is presented in a figure 2. The forces presented in this figure are 0,68 times smaller than the damper produces and the velocities are bigger 1/0,68=1,47 times than velocities of the damper.

A dotted line presents the passive damper characteristics and solid lines characteristics of the semi-active damper for various levels of the control signal (a various electric current of the solenoid valve).

Positive velocities and forces are for the extension of the damper and negative values are for the compression.

The characteristics is asymmetric and the relation between damper velocity and force is not linear. The passive damper gives compression forces almost the same as the semi-active damper at the highest value of electric current (the semi-active damper is the most soft for this value of a control signal). Extension forces of the passive damper are in the middle range of highest and smallest values of damping forces of the semi-active damper.

Highest values of extension forces are almost three times bigger than the lowest values. For compression values a difference between forces is much smaller and is not bigger than two times. The characteristics for all levels of solenoid current are nonlinear.

During some simulations a symmetric characteristic is used. To find one value of force for given damper speed it is possible to calculate mean value of the damper force. The result of such a calculation for all the area of damper speed between 0 and 0,88 m/s is presented in a figure 3.

It is interesting that a mean damping force for passive damper lays in the middle range between smallest and highest forces for the semi-active damper.

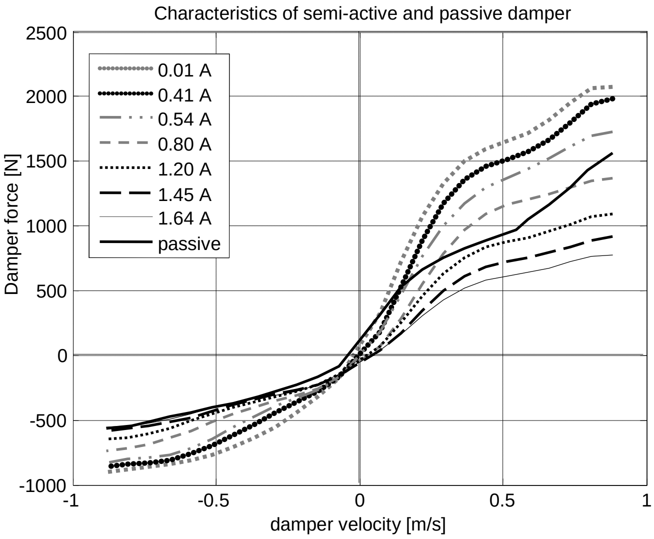


Fig. 2. The characteristics of semi active and passive damping forces of the rear suspension of a tested vehicle (calculated for the wheel)
Rys. 2. Charakterystyka amortyzatora półaktywnego i pasywnego tylnego zawieszenia badanego pojazdu (obliczone dla koła)

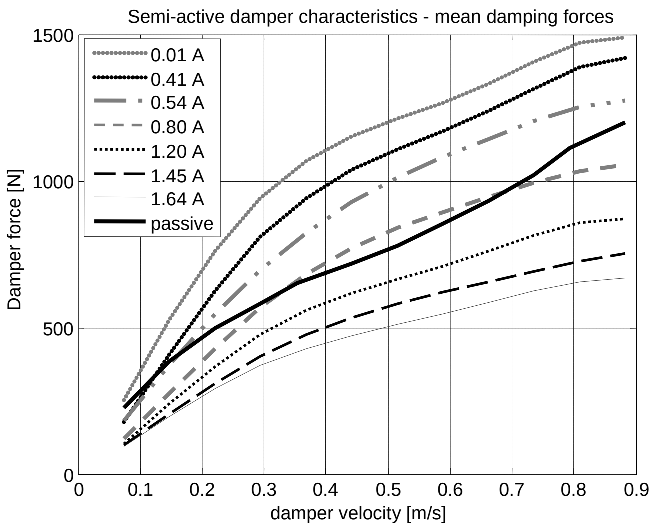


Fig. 3. Characteristics of mean forces of the semi-active and the passive rear suspension shock absorber
Rys. 3. Charakterystyka średnich sił tłumienia dla półaktywnego i pasywnego amortyzatora tylnego zawieszenia

### 3.2. Damper asymmetry

As presented in the figure 2 forces exerted by a damper at a given speed are highly unequal. Typically the extension force $F_e$ is three to four (five) times the compression force $F_c$. The mean force $F_m$ as presented in figure 3 is the average value of the compression force and the extension force $(F_m = 0.5(F_c + F_e))$ , thus:

$$F_e = (1 + e_D)F_m \tag{2}$$
$$F_c = (1 - e_D)F_m \tag{3}$$

and

$$e_D = \frac{F_e - F_c}{F_e + F_c} \tag{4}$$

Where $e_D$ is a damper force transfer factor for that particular speed. The transfer factor varies with particular damper and with operating speed but is typically 0.5 – 0.6 [1].

In the other work [6] the damper asymmetry ratio $\lambda$ is used. This value is defined as:

$$\lambda = \frac{F_e}{F_C} \tag{5}$$

Relation between $e_D$ and $\lambda$ is:

$$e_D = \frac{\lambda - 1}{\lambda + 1} \tag{6}$$

or

$$\lambda = \frac{1 + e_D}{1 - e_D} \tag{7}$$

Typical values of $\lambda$ are between 3 and 5 for passenger cars and between 3 and 7 for trucks.

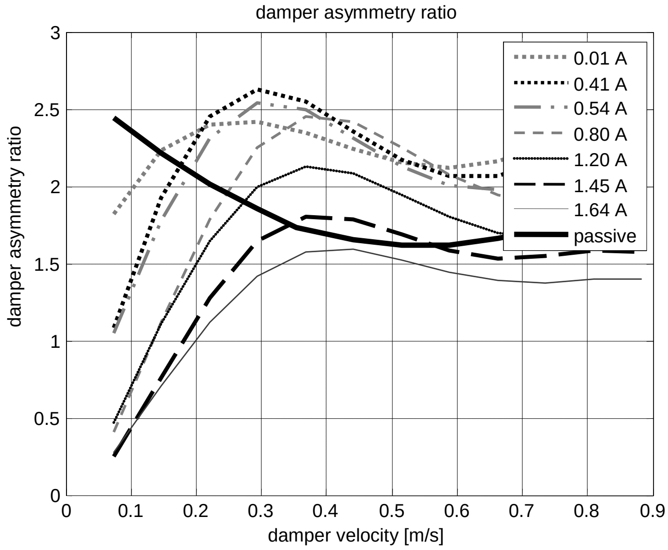


Fig. 4. Investigated the passive and the semi-active shock absorber asymmetry ratios $\lambda$ versus damper speed

Rys. 4. Współczynnik asymetrii $\lambda$ dla badanych amortyzatorów pasywnego i półaktywnego

The semi-active damper has a bigger asymmetry for the sport level of the control signal (0A) – its value is around 2,2 and smaller values for the comfort level of the control signal (1,64A) – this value is around 1,5. The asymmetry changes with the speed of damper.

### 3.3. Damping coefficient

On the base of measured and approximated relation between the damper speed and the force it was possible to calculate a damping coefficient of the damper. This value is very often the only one parameter which we use in linear models of damping force used in quarter car models. Figure 5 presents values of the damping coefficient calculated from the shock absorber characteristics.

Values of damping coefficient are changing with damper speed and the direction of damper work. They are much bigger for the small values of damper speed. John Dixon in [1] explain this shape of damper characteristics with the fact that the damper speed for such a maneuver like a rapid steering or a lane change are quite small. Due to this fact damper coefficient must be bigger to give sufficient damping force.

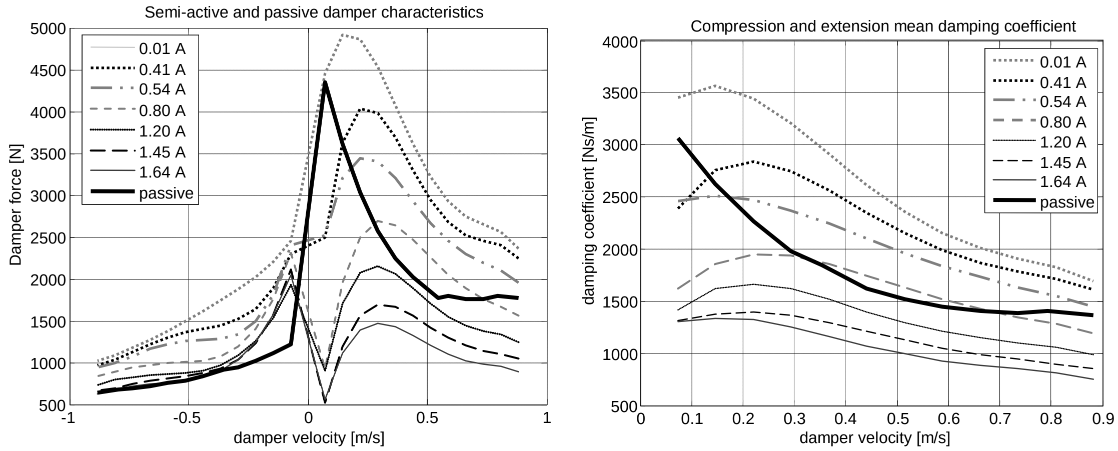


Fig. 5. Damping coefficient and mean damping coefficient versus damper speed for the passive and semi-active rear suspension of a passenger car

Rys. 5. Współczynnik tłumienia i średni współczynnik tłumienia dla pasywnego i półaktywnego tylnego zawieszenia samochodu osobowego

### 3.4. Damping ratio

Damping ratio is dimensionless and to calculate its value it is necessary to know the mass and the stiffness of suspension parameters. It is important to use stiffness parameter of the suspension not of the spring. The difference is made by the place of installation and mathematically is described by the installation ratio. The mass is the sprung mass and has to be the mass of one corner of the vehicle supported by analyzed suspension.

The mass of the one corner of analyzed vehicle – its rear suspension - changes during normal operation due to changes of number of passengers and presence of any luggage in the luggage compartment. For the analyzed car this mass has changed from 230 kg to 400 kg [3].

The stiffness of the spring of the analyzed suspension is k=20500 N/m and installation ratio is 1,05. It gives value of the suspension stiffness equal k=22600 N/m [5].

On the base of above values it is possible to calculate the damping ratio for this suspension for various damping coefficients due to controllability of semi-active damper and of course to calculate damping ratio for the suspension with passive damper for comparison to both of them.

The results of calculations based on asymmetric damper characteristics and based on mean force calculated characteristics are presented in figure 6. It is interesting to see the other - smaller – area of limits of damping ratio for calculations made using mean force values.

In the figure 7 are presented damping ratios for the passive and semi-active suspension for the sprung mass corresponding to gross vehicle weight.

The difference between damping ratios between unloaded and loaded vehicle are about 25%. The car without load has a bigger values of damping ratios. This problem was in detail discussed in the other work of the author [3].

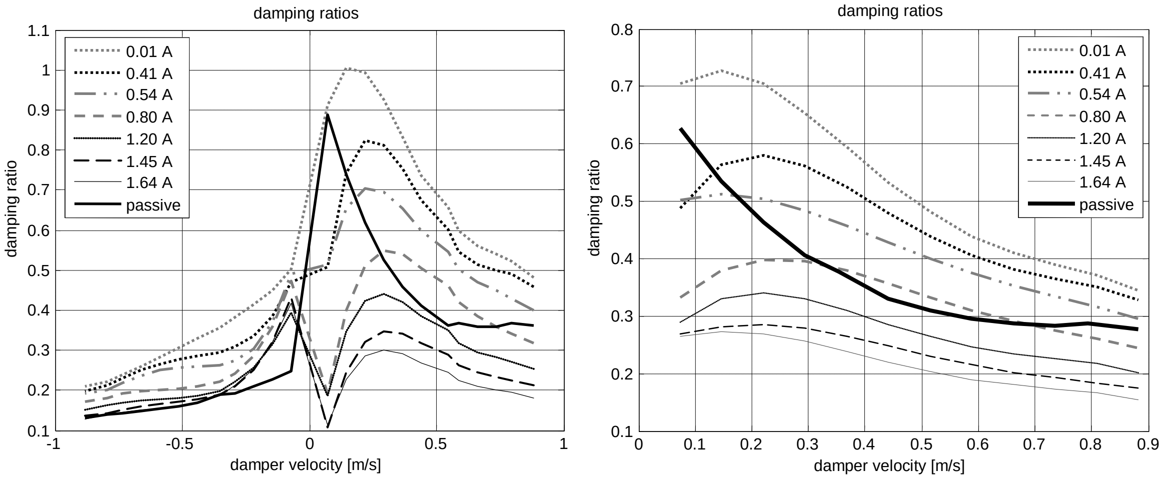


Fig. 6. Damping ratios of the passive and the semi-active suspension of investigated vehicle with sprung mass corresponding to curb vehicle weight

Rys. 6. Bezwymiarowy współczynnik tłumienia dla zawieszenia pasywnego i półaktywnego dla nieobciążonego samochodu

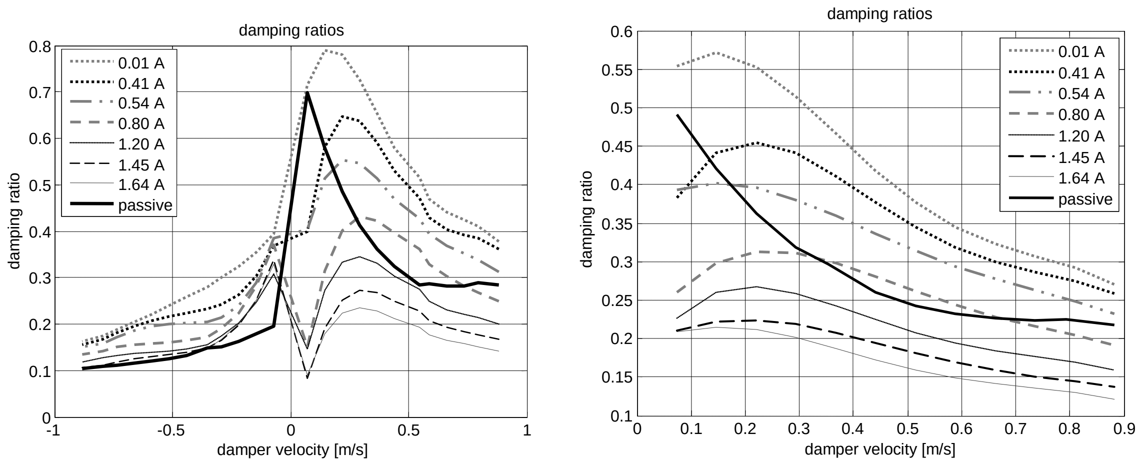


Fig. 7. Damping ratios of the passive and the semi-active suspension of investigated vehicle with sprung mass corresponding to gross vehicle weight

Rys. 7. Bezwymiarowy współczynnik tłumienia dla zawieszenia pasywnego i półaktywnego dla masy resorowanej odpowiadającej masie całkowitej samochodu

## 4. CONCLUSIONS

The goal of the conducted research was to find the values of parameters describing passive and semi-active dampers for the purposes of use this knowledge in the process of designing semi-active suspension controller and model.

The research was made on a real shock absorber used in the passenger car, and gave the knowledge on a real shock absorber characteristics and parameters.

Analysis of shock absorber characteristics shows that real shock absorber characteristics is asymmetric and nonlinear. It is possible to find mean value of damping forces when using symmetric model of shock absorber characteristic. But it gives much smaller values from extension values and much bigger from compression values.

Nonlinearity of damper's characteristics occurs in bigger values of the damping coefficient and the damping ratio for the small values of damper speed and smaller values for the bigger velocities. Values for small velocities are almost two times bigger than for bigger velocities. The influence of control signal - electric current - is almost linear in limit values of this signal (from 0 to 1,6 A).

Very interesting fact is, that however the shock absorber has adjustable damping force, this force changes in limits of usually used values of damping ratios (0,2 – 0,4) with 0,2 value possibly optimal for comfort and 0,4 possibly optimal for handling (depends on particular car and situation).

The study gave a lot of interesting insights that require detailed examination. Further research will be concentrated on the way of linearization of damping characteristics to linear relation between force and damper velocity and comparison of differences in modeling with use of nonlinear and asymmetric damping force model and full linear suspension model. It is interesting that also in specialized literature there is no detailed information about the reasons and effects of usage of asymmetric shock absorber characteristics.

**References**

1. Dixon J.C.: *The Shock Absorber Handbook*. Professional Engineering Publishing Ltd and John Wiley and Sons Ltd, West Sussex, 2007.
2. Milliken, W.F., Milliken, D.L.: *Race Car Vehicle Dynamics*. Society of Automotive Engineers, Warrendale, 1995.
3. Pikosz H., Ślaski G.: *Problem zmienności obciążenia eksploatacyjnego pojazdu w doborze wartości tłumienia w zawieszeniu*. ARCHIWUM MOTORYZACJI, 1/2010, pp. 35-44.
4. Pikosz H., Ślaski G.: *Badania charakterystyk pracy amortyzatorów o regulowanym elektronicznie tłumieniu – metodyka i narzędzia badań*. Zeszyty Naukowe Instytutu Pojazdów Politechniki Warszawskiej nr 1(77) 2010, pp. 321-332.
5. Pikosz H., Ślaski G.: *Charakterystyki elementów sprężystych i tłumiących zawieszenia samochodu osobowego oraz zastępcze charakterystyki ich modeli*. Logistyka 2/2010, pp. 1 -10 (dodatek CD).
6. Stańczyk L., Łomako D.: *Komputerowe obliczenia zespołów samochodów i ciągników.* Wydawnictwo Politechniki Świętokrzyskiej, Kielce, 2004.

Received 15.04.2010; accepted in revised form 09.06.2011