Nonlinear analysis of shock absorber with amplitude dependent damping

Jan Łuczko[1], Urszula Ferdek[2] and Waldemar Łatas[3]

Institute of Applied Mechanic, Cracow University of Technology
Al. Jana Pawła II 37, 31-864 Kraków, Poland
[1] *e-mail: jluczko@mech.pk.edu.pl*
[2] *e-mail: uferdek@mech.pk.edu.pl*
[3] *e-mail: latas@mech.pk.edu.pl*

Abstract

Analysis of vehicle suspension with a non-linear passive damper is performed in this paper. The model of mono-tube hydraulic shock absorber with stroke dependent damping is proposed. The construction of this damper, compared to the classic mono-tube shock absorber construction, has got additional chambers that increase the oil flow in the case of relatively small displacement of the piston. The proposed damper, used in the suspension of the car, improves comfort of the ride providing a high level of safety. The proposed model allows to investigate the influence of the important damper construction parameters on its characteristics. Utilising the numerical integration methods the basic characteristics of the system were determined. The response of the system to harmonic excitation with an amplitude decreasing with decreasing amplitude with increasing frequency, was investigated.

Keywords: hydraulic damper, shock absorber, quarter-car model, damping force, vehicle suspension

1. Introduction

In order to ensure comfort of the ride and high level of safety, increasingly advanced construction solutions are used in the suspensions of cars. One of the basic suspension components responsible for damping is the shock absorber. A new construction solution for hydraulic dampers requires a detailed analysis of its effectiveness.

In order to assess the effectiveness of the damper, analyses of various vehicle models e.g. quarter-car or half-car [3], have been performed. The analysis of the car model requires introduction of a relatively simple hydraulic damper model, properly describing its basic properties and allowing simultaneously investigation of the influence of essential parameters within the wide range of their changes. Tests of modelling mono-tube dampers [1], twin-tube dampers [2, 4] and others have been undertaken. They mainly differ in the approach describing the oil flow through valves.

In this paper the quarter-car model with a mono-tube hydraulic shock absorber is examined. The characteristic of the damper model is dependent on the stroke of the piston The influence of the excitation parameters as well as constructional ones on the vibration damping effectiveness was investigated. The criterion of quality activities included indices responsible for the comfort and safety of the ride.

2. Quarter-car suspension model

The quarter-car model is presented in Fig. 1.

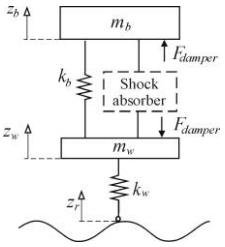

Figure 1: Quarter-car model

The vibration of the system around the static equilibrium position can be written using the following differential equations:

$$m_w \ddot{z}_w = k_b(z_b - z_w) + k_w(z_r - z_w) + F_{damper} \tag{1}$$

$$m_b \ddot{z}_b = -k_b(z_b - z_w) - F_{damper} \tag{2}$$

The coordinates z_w and z_b define the position of the so-called non-spring-supported mass m_w (mass of the wheel, axle of the vehicle and some elements of the powertrain system) and position of the spring-supported mass m_b (¼ remaining mass, mainly body car), parameter k_b is the stiffness of the spring, while k_w defines the stiffness of the wheel. The influence of the damper on the spring-supported and non-spring-supported mass is described by the parameter F_{damper}. The coordinate $z_r(t)$ is the applied kinematic excitation, describing the road surface irregularities:

$$z_r(t) = a(\omega)\sin\omega t \tag{3}$$

where ωa=const (condition constant maximum velocity).

3. Model of a variable damping shock absorber

The physical model of the mono-tube shock absorber described in this paper is presented in Fig. 2.

Two chambers filled with oil are in the main cylinder: rebound chamber K_1 above the piston and chamber K_2 below the piston. An additional cylinder is rigidly connected to the piston rod and the resiliently attached piston divides this cylinder into two chambers K_3 and K_4. A floating piston is in the main cylinder, separating the chamber K_2 filled with oil from the chamber K_5 filled with gas. Two phases of the piston rod motion are essential in the damper operations: the compression phase and rebound (expansion) phase.

The resistance force depends mainly on the resultant pressure force acting on the piston, it means on the oil pressure in chambers K_1 and K_2. The coordinate z_w defines the position of the cylinder, while the positions of the pistons determine the coordinates: z_b – main piston, z_{fp} – floating piston and z_{ap} – auxiliary piston. Taking into consideration friction force F_{f1} between the piston and cylinder, the damping force can be described as:

$$F = (p_1 - p_0)A_{tp} - (p_2 - p_0)A_p + F_{f1}\,\mathrm{sgn}(\dot{z}_b - \dot{z}_w) \tag{4}$$

where p_i is the oil pressure in the chambers K_i (i=1,2,3,4), p_0 is the working pressure of the gas. The parameters A_{tp}, A_p define the top and the bottom area of the piston, while A_{ap} is the area of the additional piston.

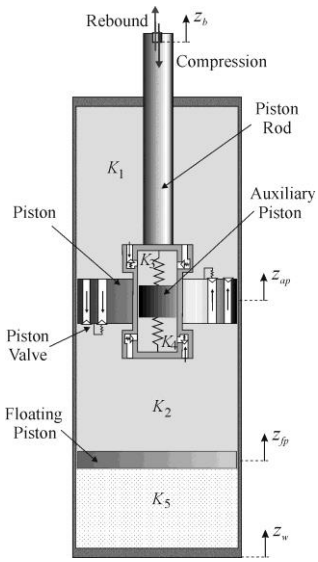

Figure 2: Model of the mono-tube shock absorber

In order to determine the characteristic of the damping force, the equations (1-4), the equations of motion of the floating and auxiliary piston and nonlinear differential equations describing the pressure changes in the chambers K_i should be solved. The operation of the damper depends mainly on the used strategies of the flow control, which takes into account the nonlinear functions defining the mass flow rates.

4. Results of numerical simulations

Utilising the numerical integration methods the basic characteristics of the damper are determined. The influence of the excitation parameters (Fig. 3) and constructional ones on the damping force dependence on the piston displacement and relative velocity, is investigated.

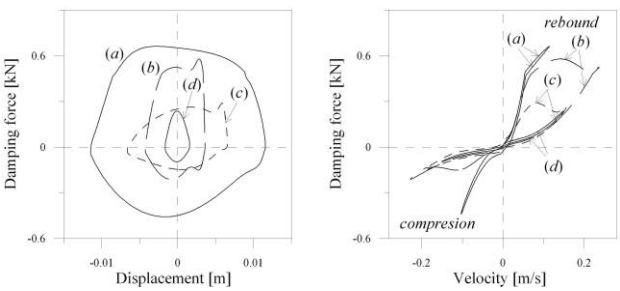

Figure 3: Influence of the excitation frequency on the damping force characteristics: (a) f=1.5 Hz, a=1.6 cm; (b) f= 4 Hz, a=0.6 cm; (c) f=10 Hz, a=0.24 cm; (d) f=15 Hz, a=0.16 cm

The examples diagrams of the pressure time waveforms and the mass flow rates corresponding to the characteristic (d) shown in Fig. 3, are presented in Fig. 4.

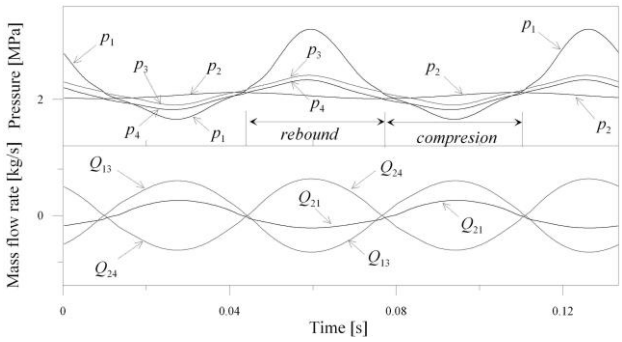

Figure 4: Pressure time histories and mass flow rates

The influence of model parameters, characterising mainly the area of the orifice cross-section on the RMS values of velocity and acceleration of the spring-supported mass and also on the values of dynamic reaction exerted on the wheel, is investigated in this paper.

5. Conclusion

The developed in this paper the quarter-car model allows to investigate the most important properties of the hydraulic shock absorbers. The numerical analysis of the damper model indicates that the characteristic of the damping force substantially depends on the amplitude and the excitation frequency. Within the range of small amplitudes and high frequencies, the system behaves like a shock absorber of 'soft' characteristic, which improves driving comfort, while within the resonance ranges the increased damping force provides a high safety of the ride.

References

[1] Farjoud A., Ahmadian M., Craft M. and Burke W., Nonlinear modeling and experimental characterization of hydraulic dampers: effects of shim stack and orifice parameters on damper performance, *Nonlinear Dynamics*, 67.2, pp. 1437-1456, 2012.

[2] Ferdek U. and Łuczko J., Modeling and analysis of a twin-tube hydraulic shock absorber, *Journal of Theoretical and Applied Mechanics*, 50.2, pp. 627-638, 2012.

[3] Ferdek U. and Łuczko J., Performance comparison of active and semi-active SMC and LQR regulators in a quarter-car model, *Journal of Theoretical and Applied Mechanics*, 53.4, pp. 811-822, 2015.

[4] Funke T. and Bestle D., Physics-based model of a stroke-dependent shock absorber, *Multibody System Dynamics*, 30.2, pp. 221-232, 2013.