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DYNAMICS OF A QUARTER CAR SYSTEM WITH AMPLITUDE SELECTIVE DAMPING

JANUSZ GOŁDASZ, MARCIN KNAPCZYK

BWI Group, Technical Center Kraków, ul. Podgórki Tynieckie 2, 30-399 Kraków, PL
e-mail: janusz.goldasz@bwigroup.com

Summary. An amplitude-selective damping is a phenomenon in hydraulic shock absorbers where a pressure drop is a function of physical displacement even at a constant flow rate. The combination is often claimed to provide less damping for small displacement inputs as well as ordinary ride on vehicles. At the same time the system offers good ride and handling metrics by augmenting the damping force during such vehicle manoeuvres. Therefore, the purpose of this study is to investigate the dynamic performance of a simple two-degree-of-freedom (2DOF) system with an amplitude-selective damping type shock absorber. In the paper the authors derive & analyze a model of the shock absorber on a component level, then study the influence the device has on the suspension behaviour within the prescribed range of frequencies.

1. INTRODUCTION

From the engineering perspective, a typical passive shock absorber characteristic is a compromise between handling properties of a car, passenger comfort and safety as well as NVH (Noise, Vibration, and Harshness). Despite their relative simplicity and maturity of the design a good damper design is always an engineering challenge. At the same time it seems there is a need for a simple, inexpensive shock absorber solution for ordinary vehicles. Some notable examples include frequency (inertia) sensitive valves, position (stroke) dependent systems or amplitude-selective-damping assemblies [1]. The purpose of the present paper is to understand the behavior of a quarter car vehicle model equipped with an amplitude-selective-damping (ASD) valve. Therefore, the authors proceed with a brief description of a patented ASD piston design in Section 2.1, then in Sections 2.2 and 2.3 outline the ASD type shock absorber model as well as the simplified suspension model. Finally, the numerical results are illustrated in Section 2.4, and the conclusions drawn in Section 3.

2. MODELING AND SIMULATIONS

2.1. Amplitude-selective-damping piston description

As shown in Figure 1, the shock absorber assembly houses one piston assembly separating the main compression (lower) and rebound (upper) chambers. The piston assembly comprises a chamber with a floating piston in the form of a light displaceable rigid disc [2]. The floating piston divides the fluid chamber assembly into a secondary compression and rebound chamber, respectively, and forms an additional flow path (bypass) by means of holes in the piston rod and the primary piston assembly. The motion of the floating disc in the chamber is limited by stops. Therefore, the additional bypass & main piston valve together handle vibrations with small amplitudes (pressures), whereas larger amplitude (pressure) vibrations are managed by the main piston valve only. In addition to the floating piston, the chamber contains additional valves providing sufficient damping at low stroking velocities (for body and wheel control). The combination is often claimed to provide less damping for small displacement inputs as well as ordinary ride on vehicles. An example of actual characteristic of a real ASD system is revealed in Figure 2. At the same time the system offers good ride and handling metrics by augmenting the damping force during such vehicle manoeuvres. Moreover, the floating piston may be supported by two opposing springs to force it into a central position in the ASD chamber [1]. The particular ASD valve arrangement is suitable for both twin-tube as well as mono-tube shock absorbers. In addition to that, twin-tube shock absorbers may utilize ASD type valves located at the base valve for controlling damping forces in compression strokes only.

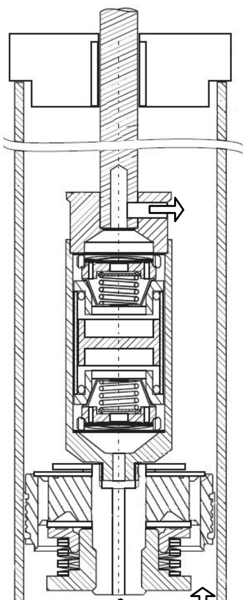

Fig. 11. Amplitude-selective-damping piston [1];
the arrows illustrate the primary & secondary flow paths

Variations of the ASD concept include elastomer impact cushions on the floating piston for contact noise reduction. Another more exotic possibility would be replacing the springs with permanent magnet pieces on the floating piston and the top/bottom of the ASD chamber [3,4,5,6,7].

2.2. ASD shock absorber model

Consider the simplified shock absorber with the ASD type piston in a monotube shock absorber configuration. The configuration is similar to that described above with the addition of two springs supporting the floating piston. For simplicity, the blow-off valves in the ASD chamber are neglected, and the restriction holes at either end of the fluid chamber are assumed large. Moreover, the shock absorber is rid of the piston rod which is typical for symmetric thru-rod shock absorber designs, and the main piston is characterized by a single hydraulic resistance constant R_h. The shock absorber fluid is characterized by the three parameters: the density ρ, the bulk modulus β, and the viscosity μ.

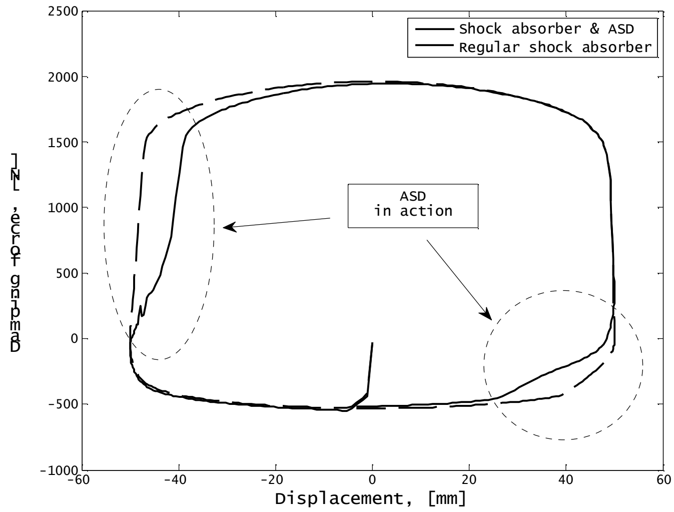

Fig.12. Amplitude-selective-damping system example; phase plane of force vs. displacement (v=0.52 m/s)

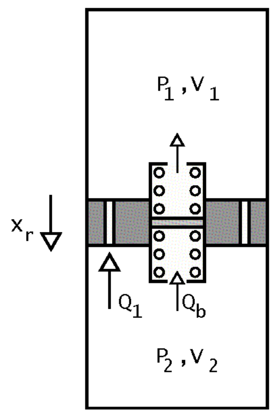

Fig. 13. Simplified amplitude-selective-damping (ASD) shock absorber model

By balancing spring forces as well as hydraulic forces on the floating piston the following relationship between the displacement of the floating piston x_b and the pressure drop across it can be obtained

$$kx_b = A_2(P_1 - P_2) \tag{1}$$

$$x_b = \frac{A_2}{k}(P_1 - P_2) \tag{2}$$

where k is the spring ratio, A_2 denotes the cross-sectional area of the floating piston, and P_1 & P_2 refer to the pressures acting on the floating piston. By multiplying the above equation by the floating piston area and differentiating it with respect to time, the volume V_b that is displaced by the floating piston as well as the corresponding flow rate Q_b can be obtained

$$V_b = \frac{A_2^2}{k}(P_1 - P_2) \tag{3}$$

$$Q_b = \frac{dV_b}{dt} = \frac{A_2^2}{k}\frac{d\Delta P}{dt} \tag{4}$$

Assuming the flow rate through the main piston of the following form

$$Q_1 = \frac{1}{R_h}\Delta P \tag{5}$$

the total flow rate including both Q_b and Q_1 can be set as follows

$$Q = Q_1 + Q_b = \frac{A_2^2}{k}\frac{d\Delta P}{dt} + \frac{1}{R_h}\Delta P \tag{6}$$

Finally,

$$Q_1 + Q_b = \begin{cases} \dfrac{A_2^2}{k}\dfrac{d\Delta P}{dt} + \dfrac{1}{R_h}\Delta P & ; \left|\dfrac{A_2}{k}\Delta P\right| \le x_m \\ \dfrac{1}{R_h}\Delta P & ; \left|\dfrac{A_2}{k}\Delta P\right| > x_m \end{cases} \tag{7}$$

where x_m is the floating piston's maximum displacement. Then, the system of equations governing the pressure variation in either chamber can be written as

$$\begin{cases} \dfrac{dP_1}{dt} = \dfrac{\beta}{V_1}(A_p v - Q_1 - Q_2) \\ \dfrac{dP_2}{dt} = \dfrac{\beta}{V_2}(-A_p v + Q_1 + Q_2) \end{cases} \tag{8}$$

where A_p is the main piston area factor, and v is the stroking velocity. Also, V_1 and V_2 denote the fluid volumes in the upper and lower chambers, respectively. The above system of equations 8 and Equation 7 together form a system of ordinary differential equations to describe the behavior of a shock absorber of the ASD type design.

In order to derive the shock absorber transfer function for the ASD range of pressures, let us assume the state-space vector $X = [P_1 \quad P_2]^T$ is of the following form

$$X = AX + Bv \tag{9}$$

$$A=\begin{vmatrix} -\dfrac{\beta}{V_0}\left(\dfrac{1}{R_h}+\dfrac{A_2^2}{k}\right) & \dfrac{\beta}{V_0}\left(\dfrac{1}{R_h}+\dfrac{A_2^2}{k}\right) \\ \dfrac{\beta}{V_0}\left(\dfrac{1}{R_h}+\dfrac{A_2^2}{k}\right) & -\dfrac{\beta}{V_0}\left(\dfrac{1}{R_h}+\dfrac{A_2^2}{k}\right) \end{vmatrix} \tag{10}$$

$$B=\begin{vmatrix} \dfrac{\beta}{V_0}A_p & 0 \\ 0 & -\dfrac{\beta}{V_0}A_p \end{vmatrix} \tag{11}$$

The damping force can be calculated as follows

$$F_d=(P_1-P_2)A_p \tag{12}$$

From Equations 9 and 12 the transfer function between the damping force and the stroking velocity for can be obtained as follows

$$H(s)=\frac{F_d(s)}{V(s)}=\begin{cases} 1:\dfrac{A_p^2}{s\left(\dfrac{V_0}{2\beta}+\dfrac{A_2^2}{k}\right)+\dfrac{1}{R_h}} & ;\quad \left|\dfrac{A_2}{k}\Delta P\right|\le x_m \\ 2:A_p^2R_h & ;\quad \left|\dfrac{A_2}{k}\Delta P\right|> x_m \end{cases} \tag{13}$$

The first of the transfer function describes the behavior of the ASD type mechanism. For comparison, the other equation shows the obvious transfer function of a linear (proportional) shock absorber.

To conclude, Equation 6 implies the ASD type piston is a dual-mode pressure-rate sensitive system.

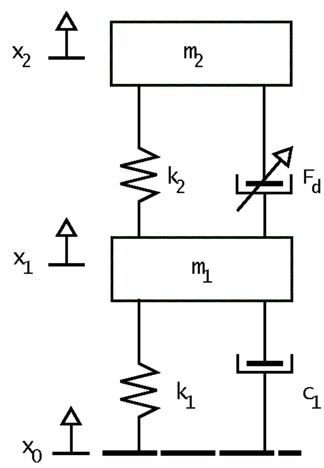

Fig. 14. Two-degrees-of-freedom (2DOF) model

2.3. Two-degrees-of-freedom system model (quarter car)

The equations of motion for the two-degrees-of-freedom system model shown in Figure 4 can be derived as follows

$$\begin{cases} m_1 \dfrac{d^2 x_1}{dt^2} = -k_1(x_1 - x_0) - c_1\left(\dfrac{dx_1}{dt} - \dfrac{dx_0}{dt}\right) + k_2(x_2 - x_1) + F_d \\ m_2 \dfrac{d^2 x_2}{dt^2} = -k_2(x_2 - x_1) - F_d \end{cases} \tag{15}$$

where m_1, m_2 denote the masses of the suspension, x_0 is the road input, and x_1, x_2 are the vertical displacements. Again, the damping force F_d is given by Equation 12 (as well as Equation 8). The parameters of the model are given in Table 1, and they fairly correspond to a typical medium-size vehicle suspension.

Table 2. 2DOF Suspension parameters & ASD shock absorber

Parameter	Value
m_1	70 kg
m_2	500 kg
k_1	390 N/mm
k_2	110 N/mm
R_h	0.6068 kg·s/mm^4
A_p	1662 mm^2
V_0	162000 mm^3
β	900 MPa
ρ	0.85 g/cm^3
A_2	1256 mm^2
k	15 N/mm
x_m (peak-to-peak)	10 mm

2.4. Results

In this section, simulation results are presented for the derived amplitude-selective-damping shock absorber model as well as for the above vehicle suspension containing an ASD type shock absorber. For comparison purposes, the vehicle suspension incl. a linear shock absorber was modeled and analyzed, too. The simulations were conducted for the quarter car model (see Equation 15) subjected to constant speed sine wave excitations within the frequency range from 0.3 Hz to 20 Hz. The suspension simulation results are presented in Figures 7 and 8 for two different peak amplitudes of the road input: 0.15 m/s, 0.5 m/s.

The non-linear dual-mode behavior of the ASD shock absorber can be best seen in Figures 5 and 6. Here, the simulations were conducted at the piston speed v=0.15 m/s and two different frequencies of the excitation: 1 Hz, 15 Hz, respectively, and compared against the linear shock absorber. At the higher frequency, the damping force is clearly dominated by the additional flow through the ASD bypass. Also, note the increased hysteresis width (within the operation regime of the ASD floating piston).

On the suspension level, as seen in Figures 7 and 8, application of the amplitude-selective-damping in a vehicle suspension results in the higher second (wheel) resonant response. The first resonant frequency is not affected by the ASD shock absorber. However, at frequencies above the second resonance, the dual-mode ASD shock absorber results in lower transmissibility of the upper mass, better isolating the suspension form the road inputs. Moreover, the threshold frequency (above which the shock absorber behavior is dominated by the ASD mechanism) varies with the amplitude of the excitation.

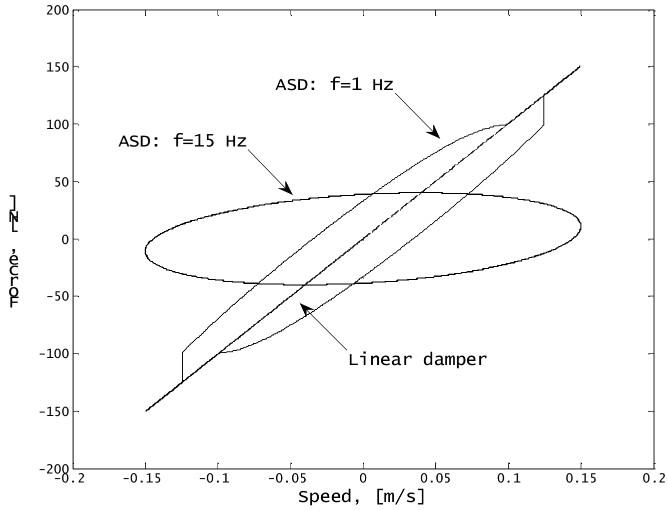

Fig. 15. ASD shock absorber: Force vs. speed; v=0.15 m/s

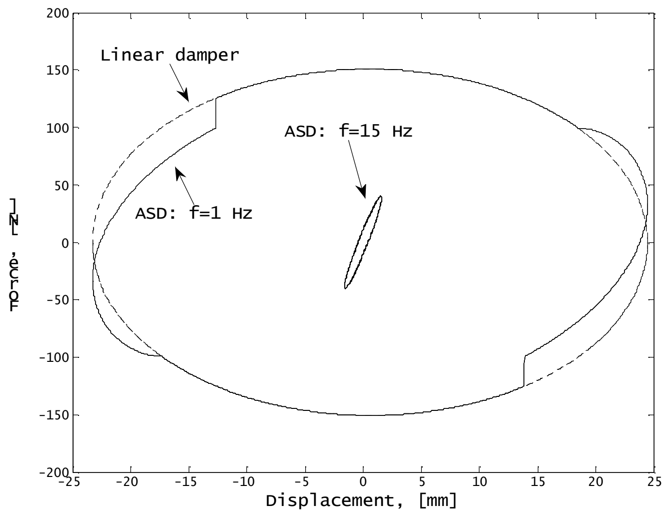

Fig. 16. ASD shock absorber: Force vs. displacement; v=0.15 m/s

3. SUMMARY AND CONCLUSIONS

The purpose of the study was to illustrate the influence the so-called amplitude-selective-damping has on the vehicle suspension. The simulation results indicate the application of an ASD type shock absorber may result in vibration isolation improvement at frequencies above the second (wheel) resonant frequency. The suspension transmissibility at the lower frequencies (primary ride) is not affected by the shock absorber dual-mode behavior.

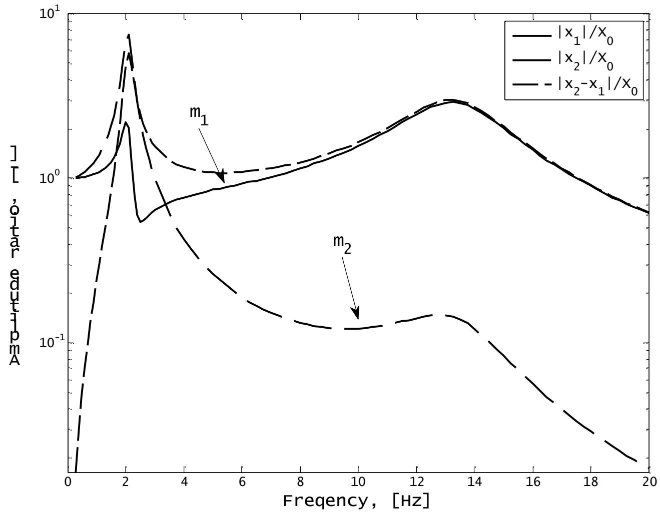

Fig. 17. Transmissibility; base (linear) damping w/o ASD

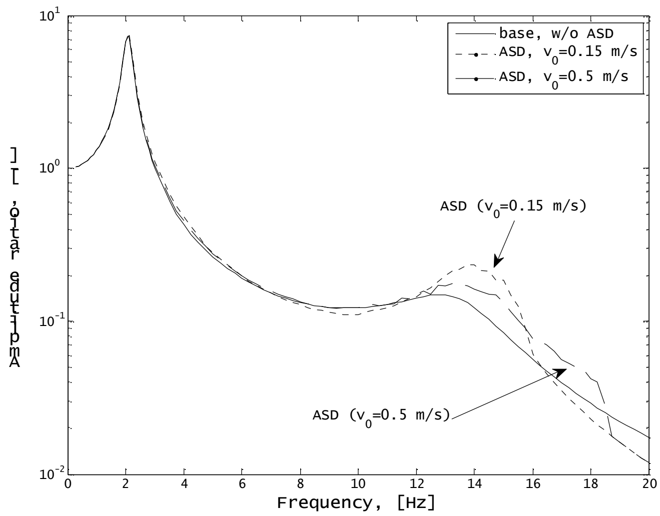

Fig. 18. Transmissibility of the upper mass m_2; ASD damping vs. base system

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