

MODELLING OF AMPLITUDE-SELECTIVE-DAMPING VALVES

SUMMARY

The so-called amplitude-selective-damping (ASD) valving is a relatively new approach for modifying the output of a hydraulic shock absorber. In the automotive industry ASD valves are known to improve isolation from road inputs. During more aggressive maneuvers the systems maintains the performance of a standard (non-ASD) shock absorber. In the paper, the author derives and analyzes a fairly complete state-space model of an exemplary piston-side ASD valve. The model includes key geometric and performance characteristics of the valve. The results are shown in the form of phase plane plots of force-displacement diagrams, respectively, for a twin-tube shock absorber configuration of choice.

Keywords: amplitude-selective-damping valve, twin-tube shock absorber, hydraulic shock absorber, automotive shock absorber modeling

MODELOWANIE ZAWORÓW

O CHARAKTERYSTYCE ZALEŻNEJ OD AMPLITUDY PRZEMIESZCZENIA

W pracy przedstawiono model dwururowego amortyzatora samochodowego z zaworem o charakterystyce zależnej od amplitudy przemieszczenia. Zawór dodatkowy działa równolegle do zaworu głównego tłoka i pozwala na kształtowanie osiągnięć amortyzatora w zakresie małych przemieszczeń oraz średnich i wysokich częstotliwości wymuszenia. Model zawiera kluczowe zmienne geometryczne i materiałowe uwzględniające podstawowe osiągi zaworu w szerokim pasmie przemieszczeń i częstotliwości. Wyniki obliczeń zaprezentowano na płaszczyźnie fazowej siła-przemieszczenie w zakresie prędkości do 260 mm/s i częstotliwości wymuszenia do 12 Hz.

Słowa kluczowe: zawór z tłumieniem zależnym od amplitudy przemieszczenia, amortyzator dwururowy, amortyzator hydrauliczny, modelowanie samochodowych amortyzatorów hydraulicznych

1. INTRODUCTION

The design of a fully-functional automotive damper is an engineering challenge (Dixon 2009) and a compromise between passenger comfort, safety and NVH (N – noise, V – vibration, H – harshness). Therefore, there is always a need for simple shock absorber solutions for ordinary vehicles. One example is the amplitude-selective-damping valve. ASD valves are known to improve isolation from road inputs (Gołdasz and Knapczyk 2010). As such, the purpose of the paper is to understand the performance of an ASD valve built into the piston of a twin-tube shock absorber.

To start with, the shock absorber assembly houses one piston assembly separating the main compression (lower) and rebound (upper) chambers. The piston assembly comprises an ASD chamber with a floating piston in the form of a light rigid disc (Ślusarczyk *et al.* 2008). The floating piston divides the fluid chamber assembly into a secondary compression and rebound chamber, respectively, and forms a leakage 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. The bypass valve and main piston valve together handle vibrations with small amplitudes (pressures), whereas larger amplitude (pressure) vibrations are managed by the main piston valve. The ASD chamber contains valves providing sufficient damping at low stroking velocities (for body and wheel control) (Ślu-

sarczyk *et al.* 2008). Variations of the ASD concept include elastomer impact cushions on the floating piston for contact noise reduction, valves in the floating piston and the like (Gotz 2008).

To examine the performance of an ASD valve, in Section 2 the author outlines the twin-tube shock absorber model as well as the ASD valve model. Finally, the numerical results are illustrated in Section 3, and the conclusions drawn in Section 4.

2. MODELING OF TWIN-TUBE SHOCK ABSORBER WITH ASD VALVES

The following sections contain the generic description of the twin-tube damper model, and the amplitude-selective-damping valve model.

2.1. Twin-tube shock absorber model

The generic model of a conventional (automotive) shock absorber is revealed in Figure 1. In the shock absorber, the fluid volume is separated into three fluid chambers (V_r – rebound chamber, V_c – compression chamber, V_{res} – reserve chamber). The flow between adjacent chambers is by means of orifices and valves located between the rebound chamber and the compression one (piston valve) as well as the compression chamber and the outer reservoir (base valve). The

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displacement of the piston x_1 forces the fluid to flow from one chamber to the other (adjacent) one. For example, in the rebound (upward) portion of the piston stroke, the appropriate fluid volume is transferred from the rebound chamber to the compression chamber, and from the reserve chamber into the compression chamber. Accordingly, in compression the fluid is transferred from the compression volume into the rebound one, and from the compression chamber into the reservoir.

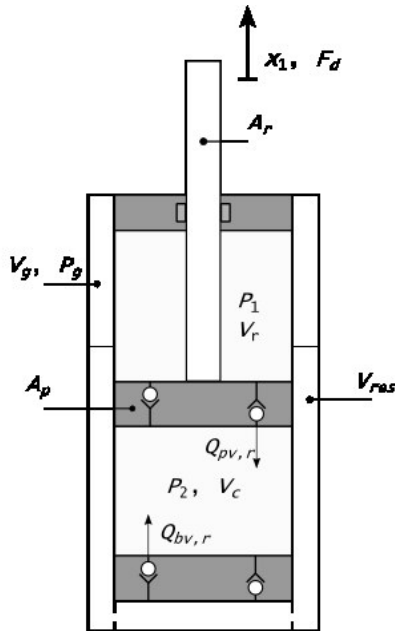


Fig. 1. Generic model of a twin-tube damper

Deriving the hydraulic system in a twin-tube shock absorber as well as analyses of deflected disc type valves have been accomplished on numerous occasions, and is beyond the scope of this analysis; the reader should refer to Lang (1977), Lee (1977), Mollica (1977) or Talbot and Starkey 2002) for studies of damper modeling and methods of calculating the hydraulic resistance of deflected disc valves and blow-off valves used in typical automotive shock absorbers. For example, using the flow continuity equations in the rebound stroke yields the following equations for the rebound pressure P_1 and the compression pressure P_2

$$-Q_{pv,r} + (A_p - A_r)v_1 = \frac{1}{\beta} V_r \frac{dP_1}{dt} \quad (1)$$

$$Q_{pv,r} + Q_{bv,r} - A_p v_1 = \frac{1}{\beta} V_c \frac{dP_2}{dt}$$

where

$$V_r = V_{r,0} - (A_p - A_r)x_1 \quad (2)$$

$$V_c = V_{c,0} + A_p x_1$$

$V_{r,0}$ ($V_{c,0}$) are the initial rebound (compression) volumes. $Q_{pv,r}$, $Q_{bv,r}$ denote the total flow rates through the piston valve and the base valve in the rebound stroke. For comparison, the respective flow rates through the piston valve and the base valve in compression are $Q_{pv,c}$ and $Q_{bv,c}$. The flow rates through the main piston valve and the base valve are calculated according to the modified Bernoulli's equation (Lang 1977, Lee 1997, Mollica 1997, Talbot and Starkey 2002). In the configuration, the ASD valve is built into the piston and operates in parallel to the main valve assembly; the flow rate through the amplitude-selective-damping valve Q_2 is added to the flow-rate through the piston valve Q_{pv} . The area of the piston head is A_p , and that of the piston rod is A_r . The fluid is characterized by the bulk modulus β and the density ρ . Finally, the pressure of gas P_g in the reservoir can be expressed as follows

$$P_g = P_{g,0} \left(\frac{V_{g,0}}{V_{g,0} + \int Q_{bv} dt} \right)^\gamma \quad (3)$$

where γ is the gas constant, Q_{bv} is the base valve flow rate incorporating both $Q_{bv,c}$ and $Q_{bv,r}$. $P_{g,0}$ and $V_{g,0}$ are the initial gas charge pressure and volume, respectively. The Q_{bv} integral outputs the volume of fluid into (or out of) the reservoir chamber at a given time constant t . Finally, the damping force F_d including the friction force F_0 can be obtained in the following manner

$$F_d = P_1(A_p - A_r) - P_2 A_p \pm F_0 \quad (4)$$

2.2. Amplitude-selective-damping valve model

Consider the generic configuration of the ASD valve that is illustrated in Figure 2.

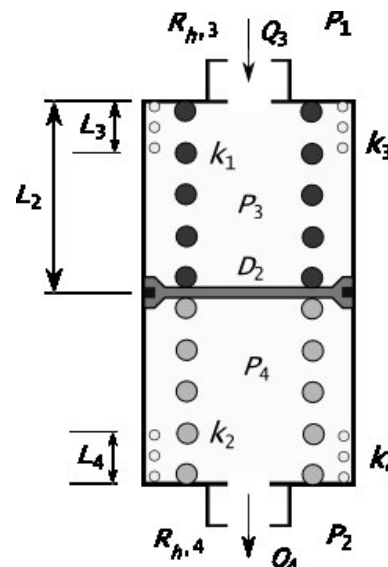


Fig. 2. ASD valve generic model

The assembly incorporates two springs (k_1, k_2) supporting the floating piston. In the examined system, the piston is in contact with one spring at a time only, i.e., in the upward (downward) motion it contacts the upper (lower) spring. The cross-sectional area of the floating piston is A_2 , and it separates the housing cylinder into two sections of equal length (L_2). Also, the motion of the ASD piston is impact-cushioned by two supplementary springs of the respective spring ratios k_3, k_4 and the dimensions L_3, L_4 . The hydraulic restriction of the inlet port is $R_{h,3}$, and that of the outlet port is $R_{h,4}$. The inertia of the piston and the compressibility of the fluid in the ASD valve chambers are omitted, and equal lengths L_3 and L_4 are assumed. Then, by balancing the forces on the floating piston the relationship between the displacement of the floating piston x_2 and the pressure drop across it can be obtained

$$\begin{cases} k_2 x_2 = (P_3 - P_4) A_2; (P_3 - P_4) > 0 \wedge |x_2| < L_2 - L_3 \\ k_1 x_2 = (P_3 - P_4) A_2; (P_3 - P_4) \leq 0 \wedge |x_2| < L_2 - L_3 \end{cases} \quad (5)$$

$$\begin{cases} k_2 x_2 + k_4 (x_2 - (L_2 - L_3)) = (P_3 - P_4) A_2; (P_3 - P_4) > 0 \wedge |x_2| \geq L_2 - L_3 \\ k_1 x_2 + k_3 (x_2 - (L_2 - L_3)) = (P_3 - P_4) A_2; (P_3 - P_4) \leq 0 \wedge |x_2| \geq L_2 - L_3 \end{cases} \quad (6)$$

where P_3, P_4 are the hydraulic pressures on either side of the ASD piston. Accordingly, the volume displaced by the moving piston is as follows

$$V_2 = A_2 x_2 \quad (7)$$

where the displacement x_2 is calculated using the Equations (5) and (6), respectively. The relationship for the flow rate Q_2 through the ASD valve is acquired by differentiating the Equation (7) to obtain

$$Q_2 = \begin{cases} \frac{A_2^2}{k_2} \frac{d(P_3 - P_4)}{dt} & ; (P_3 - P_4) > 0 \wedge |x_2| < L_2 - L_3 \\ \frac{A_2^2}{k_1} \frac{d(P_3 - P_4)}{dt} & ; (P_3 - P_4) \leq 0 \wedge |x_2| < L_2 - L_3 \\ \frac{A_2^2}{k_2 + k_4} \frac{d(P_3 - P_4)}{dt} & ; (P_3 - P_4) > 0 \wedge |x_2| \geq L_2 - L_3 \\ \frac{A_2^2}{k_1 + k_3} \frac{d(P_3 - P_4)}{dt} & ; (P_3 - P_4) \leq 0 \wedge |x_2| \geq L_2 - L_3 \end{cases} \quad (8)$$

Piston displacements larger than L_2 cut-off the ASD valve ($Q_2 = 0$), and the fluid flow occurs in the main piston valve only; the expression (8) refers to the rate of volume of fluid displaced by the moving piston. Finally, the flow rates through the entry and exit ports can be expressed as

$$\begin{aligned} Q_3 &= C_3 A_3 \sqrt{2 \frac{|P_1 - P_3|}{\rho}} \operatorname{sgn}(P_1 - P_3) = \\ &= R_{h,3} \sqrt{|P_1 - P_3|} \operatorname{sgn}(P_1 - P_3) \end{aligned} \quad (9)$$

$$\begin{aligned} Q_4 &= C_4 A_4 \sqrt{2 \frac{|P_4 - P_2|}{\rho}} \operatorname{sgn}(P_4 - P_2) = \\ &= R_{h,4} \sqrt{|P_4 - P_2|} \operatorname{sgn}(P_4 - P_2) \end{aligned} \quad (10)$$

where C_3, C_4 are the hydraulic discharge (flow) coefficients. The expressions (5–6) and (8–10) form a set of equations quantifying the behavior of the ASD valve (and a set of expressions governing the behavior of a twin-tube shock absorber with ASD valves in the piston valve if used with expressions (1)–(3)).

2.3. Test configuration

The geometry and fluid data given in Table 1 refer to a prototype twin-tube shock absorber (base configuration). The pre-calculated steady-state characteristics of the piston valve and the base valve are shown in Table 2 and Figure 3. These parameters were used in the study to simulate the shock absorber behaviour across the prescribed range of frequency and displacement inputs. Note the ASD valve operates in either stroking direction. However, the shock absorber was configured in such a way so as to generate a majority of the forces in compression via the base valve assembly. Therefore, the ASD valve contribution to the shock absorber performance can be seen in the rebound (positive) section of the force-velocity phase plane and the first and second quadrants of the force-displacement phase plane, respectively.

3. RESULTS

In this section, simulation results (calculated with Simulink using the fixed-step `rk4` solver) are presented for the derived twin-tube shock absorber model with the ASD valve. The purpose of the simulations was to examine the performance of the ASD valve subjected to small-stroke excitations as well as to investigate the influence of the key geometric variables on the valve's performance (ASD piston area – A_2 , ASD piston travel – x_2 , leading spring ratio – k_1, k_2 , impact-cushioning spring ratio – k_3, k_4 , hydraulic restrictions – $R_{h,3}, R_{h,4}$). Preliminary sensitivity analysis by the author showed the above variables have a significant impact on the performance characteristics of a shock absorber. The simulations were conducted for the described prototype twin-tube shock absorber subjected to constant velocity sine wave excitations. In the presented graphs the peak velocity of the piston rod was 260 mm/s, and the peak-to-peak displacement amplitude was 10 mm.

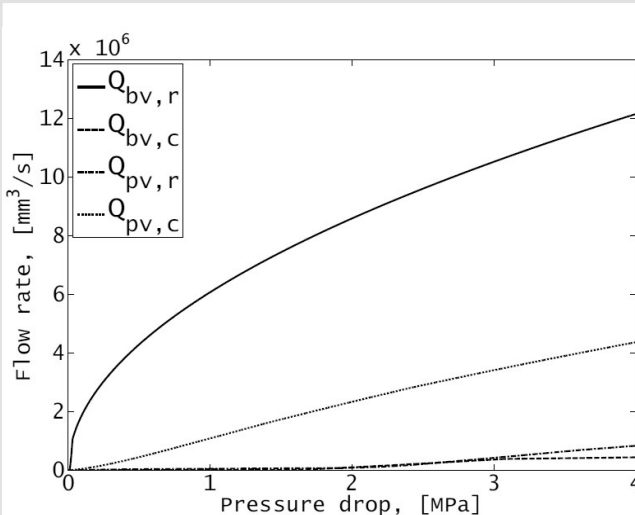
Table 1

Base configuration; geometry and material properties

Symbol	Description	Value
A_r [mm ²]	Piston rod area	120.8
A_p [mm ²]	Piston head area	804.3
$V_{r,0}$ [mm ³]	Initial rebound volume	146949.3
$V_{c,0}$ [mm ³]	Initial compression volume	48254.9
$V_{g,0}$ [mm ³]	Initial gas volume	310068
$P_{g,0}$ [N/mm ²]	Initial gas pressure	0.6
A_2 [mm ²]	ASD piston head area	254.5
A_3 [mm ²]	Inlet port (restriction) area	9.62
A_4 [mm ²]	Outlet port (restriction) area	9.62
L_2 [mm]	ASD piston maximum travel	10
L_3 [mm]	Upper spring free length	5
L_4 [mm]	Lower spring free length	5
k_1, k_2 [N/mm]	Leading spring stiffness ratio	20
k_3, k_4 [N/mm]	Impact-cushioning stiffness ratio	70
C_3, C_4	Discharge (flow) coefficients	0.5
ρ [kg/m ³]	Oil density	850
β [N/mm ²]	Bulk modulus	1500
γ	Gas constant	1.4

Table 2Flow rate vs. pressure: main piston valve Q_{pv} and base valve Q_{bv}

Quantity	Pressure [MPa]				
	0.5	1.0	2.0	3.0	4.0
$Q_{pv,r}$ [mm ³ /s]	4.28e+4	6.05e+4	9.14e+4	4.32e+5	8.42e+5
$Q_{pv,c}$ [mm ³ /s]	4.54e+5	1.09e+6	2.34e+6	3.42e+6	4.37e+6
$Q_{cv,r}$ [mm ³ /s]	4.31e+6	6.07e+6	8.59e+6	10.50e+6	12.14e+6
$Q_{bv,c}$ [mm ³ /s]	1.49e+4	2.11e+4	1.08e+5	3.73e+5	4.47e+5

**Fig. 3.** Piston valve and base valve characteristics

As illustrated in Figures 4–7, the principal contribution to the valve's performance comes from the floating piston area and the leading spring stiffness ratios. The two variables represent the area-squared-to-stiffness-ratio coefficient of the pressure gradient rate and directly influence the flow rate through the ASD valve. In addition to that, the springs k_3, k_4 affect the ASD valve in the impact-cushioned phase, i. e. during the transition from the ASD-dominated behaviour to the flow regime dominated by the main valve characteristics (see Fig. 5–6). The ASD piston travel and the cushioning travel are of secondary yet engineering importance. The influence of hydraulic restrictions at the entry and exit to the valve, respectively, can be best seen in Figure 7; predictably, the biggest impact on the damping force is associated with the highest restriction level at the valve's entry and exit. Finally, Figure 8 shows the steady-state performance of the prototype shock absorber at various piston rod stroking amplitudes.

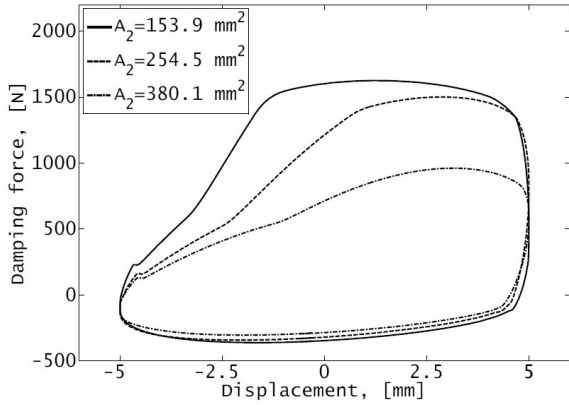


Fig. 4. Floating piston area (A_2)

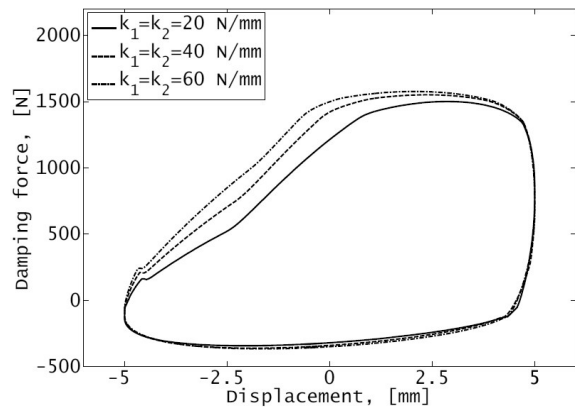


Fig. 5. Leading spring stiffness (k_1, k_2)

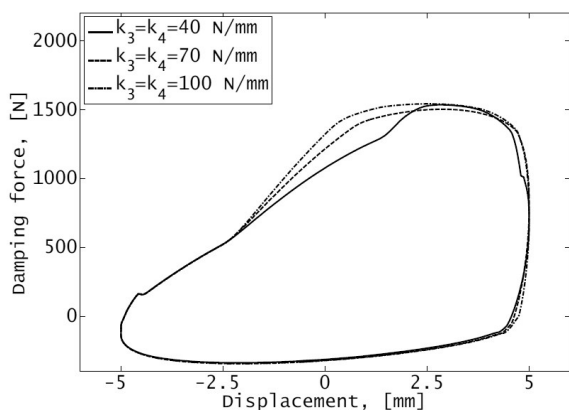


Fig. 6. Impact-cushioning spring stiffness (k_3, k_4)

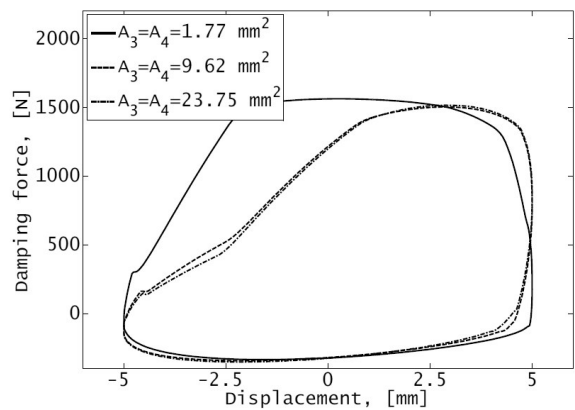


Fig. 7. Inlet (outlet) restriction ($R_{h,3}, R_{h,4}$)

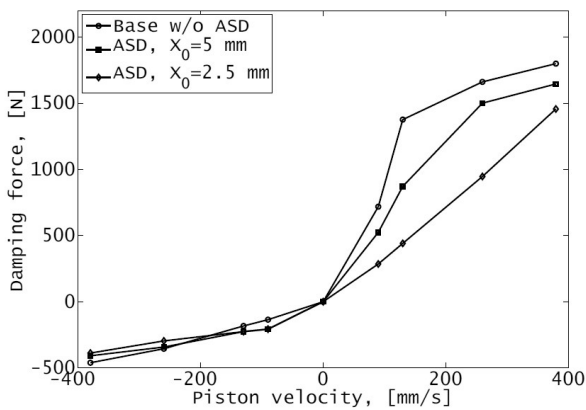


Fig. 8. ASD shock absorber steady-state characteristics

4. SUMMARY AND CONCLUSIONS

The purpose of the present study was to examine the influence of key characteristics of the amplitude-selective-damping valve on the performance of a conventional automotive hydraulic twin-tube shock absorber as well as to outline a fairly complete model of the ASD valve. The results point to the key contributors to the valve's performance, and indicate the application of an ASD valve may be a powerful method for shaping the damper base characteris-

tics. At this stage the study provides partially validated results – only the standard twin-tube damper model (without any ASD valve) has been confirmed against experimental data.

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