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 AMPLITUDE 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 osiągów amortyzatora w zakresie małych przemieszczeń oraz średnich i wysokich częstotliwości wy- muszenia. Model zawiera kluczowe zmienne geometryczne i materiałowe uwzględniające podstawowe osiągi zawo- ru w szerokim pasmie przemieszczeń i częstotliwości. Wyniki obliczeń zaprezentowano na przykładzie fuzowej 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, amortyza- tor 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 (Goldasz 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 (byass) 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 tolerate 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) (Ślusarczyk 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, $V_{rez}$ – 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
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 (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 $P_2$

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

(1)

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

(2)

where

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

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

$V_{c,0} (V_{c,0})$ 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 $\beta$ and the density $\rho$. Finally, the pressure of gas $P_g$ in the reservoir can be expressed as follows

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

(3)

where $\gamma$ is the gas constant, $Q_{bv}$ is the bave valve flow rate incorporating both $Q_{bc,c}$ and $Q_{bc,v}$. $P_{g,0}$ 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

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

(4)

2.2. Amplitude-selective-damping valve model

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

\[
\begin{align*}
\dot{Q}_2 &= A_2 \frac{d}{dt} (P_3 - P_4) \quad \text{if } (P_3 - P_4) > 0 \quad \text{and } |x_2| < L_2 - L_3 \\
\dot{Q}_2 &= A_2 \frac{d}{dt} (P_3 - P_4) \quad \text{if } (P_3 - P_4) \leq 0 \quad \text{and } |x_2| < L_2 - L_3 \\
\dot{Q}_2 &= A_2 \frac{d}{dt} (P_3 - P_4) \quad \text{if } (P_3 - P_4) > 0 \quad \text{and } |x_2| \geq L_2 - L_3 \\
\dot{Q}_2 &= A_2 \frac{d}{dt} (P_3 - P_4) \quad \text{if } (P_3 - P_4) \leq 0 \quad \text{and } |x_2| \geq L_2 - L_3
\end{align*}
\]

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

\[
V_2 = A_2 x_2
\]

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

\[
Q_2 = C_4 A_4 \sqrt{2 \frac{P_1 - P_3}{\rho} sgn (P_1 - P_3) = R_{h,4} \sqrt{|P_1 - P_3| sgn (P_1 - P_3)}}
\]

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 \(\tau k4\) 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.
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.
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 characteristics. 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.

References


Goldasz J., Knapczyn M. 2010, Dynamics of a quarter car system with amplitude selective damping, Modelowanie Inżynierskie, 8, 39, pp. 89–96.


