Modelling the dynamic behaviour of car hydraulic dampers

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Abstract. When the vehicle travels on a road, that usually isn’t perfectly smooth, this having a series of irregularities, occur shocks and vibrations due to the interaction of the wheels with the irregularities of the road, as well oscillations of sprung and unsprung masses that characterize comfort. This paper presents the theoretical and experimental researches undertaken by the author on the dynamic behaviour of hydraulic shock absorbers, in order to improve the comfort and dynamics of vehicles. The main objective of the paper is to analyse the damping characteristics of hydraulic dampers using simulation with AMESim specialized program. To validate the model, a set of experimental determinations was used with the help of a hydraulic dampers test stand. The experimental researches undertaken allowed to determine the real dynamic characteristics of the damper and to identify the influence of the parameters on these characteristics.

1. Introduction

The existence of the vibrations and oscillations at cars influences negatively the comfort of passengers, causing them great fatigue and unpleasant physiological sensations. Due to the vibrations and oscillations, the wear of some components of the car's transmission increases, but also the possibility of failure by breaking some of them due to the dynamic loads that appeared in the transmission. They also affect the integrity of the commodities transported. They negatively affect the comfort, stability, manoeuvrability, but also the reliability of the roads.

The appearance of vibrations and oscillations leads to an increase in fuel consumption, because the additional resistances in the elastic elements of the suspension must be overcome, but also to the increase of the additional energy losses in the tires [1], [2].

The dynamic behaviour of the hydraulic shock absorber and its damping characteristics greatly influence the overall response of the suspension system to input signals. To achieve a satisfactory level of comfort and manoeuvrability, the shock absorbers must be designed and adjusted according to various criteria, such as force, maximum displacement, speed etc. [3], [4].

The vibrations and oscillations in cars depend on the type of road they travel on and the type of vehicle. Thus they fall into two broad categories:

- Vibrations and oscillations produced by external sources – occur especially due to the road irregularities, but also to the wind gusts, braking etc.
- Vibrations and oscillations whose origin is located inside the car, being caused by the powertrain and steering system.

According to the causes that produce them, the oscillations are divided into:

- Free (or own) oscillations – occurs when a body or oscillating system is removed from its equilibrium position and then left free, oscillating around the equilibrium position.
Forced (or maintained) oscillations – manifests itself when on a body or an oscillating system acts in addition to the tensile or compressive force of the suspension spring an external periodic force, which is called disturbing force (or excitation force).

Free oscillations can be:
- Unamortized – when theoretically it continues to infinity.
- Amortized – when after a certain time it stops by itself.

In conclusion, in the suspension system, the damper has the following roles:
- Dissipates quickly the energy of the vertical oscillations of the sprung mass (body car, chassis etc.), resulted from the deformation of the suspension;
- Determines the fast decrease of the oscillations of the unsprung masses (wheels, axles etc.) in order to ensure the continuous contact of the wheels with the road, by limiting the amplitude of the forced oscillations at the resonant frequency.

The shock absorbers are of two categories, friction (solid elements) and hydraulic (fluid elements). The friction shock absorber are distinct due to associated problems, such as sensitivity to water or oil contamination. Other problems of the friction shock absorber are due to the Coulomb friction characteristics of the shock absorbers [5]. There have been numerous studies on different aspects of hydraulic shock absorbers characteristics and performance [6], [7], [8], [9], [10], [11].

The influence of the damper on the vibrations of the sprung masses ($m_1$) and unsprung ($m_2$) is shown schematically in Figure 1, where $A$ represents the amplitude of the vibrations, the blue line represents the non-damped vibration, and the magenta line the damped one.

The mass-spring and the spring-damper systems without the source term were studied in [12], [13]. Other applications of fractional calculus to mechanical oscillators are given in [14], [15], [16].

The influence of the shock absorber on the car.

Excitations are directly responsible for the oscillations of the cars. The main excitations that have major implications on the comfort are given by the irregularities of the road, to which are added the excitations due to the non-uniformity and unbalance of the wheels. The excitations can also be given by the change of the movement regime (acceleration, braking, turning), but these have low implications on the comfort.

2. Material and Methods
In this chapter I will present some the experimental results of the tests of hydraulic dampers, in order to analyse their dynamic behaviour. An example of a typical and practical simulation of a suspension equipped with a hydraulic damper is also presented in this chapter.

The test bench consists of the main components: electrohydraulic servomechanism, position transducer, force transducer, velocity transducer, accelerometer, hydro pneumatics accumulator, hydraulic damper, servo valve, servo cylinder etc. For the control of the electrohydraulic servomechanism and for the acquisition of the measured data, a PXI type industrial electronic computer is used, provided with a data acquisition board produced by the National Instruments Corporation, assisted by the LabVIEW program produced by the same corporation (Figure 2).
To determine the regulation characteristic, the reference signal used was a triangular signal with an amplitude of 9 V and a frequency of 0.02 Hz, which provided the servomechanism with a quasi-stationary operating mode (piston speed was 3.6 mm/s). The sampling frequency was in this case 10 Hz. The registered characteristic is presented in Figure 3.

**Figure 2.** A part of the installation for testing hydraulic dampers.

**Figure 3.** Schematic regulation characteristic of the electrohydraulic servomechanism.
The principle scheme of the tested electrohydraulic servomechanism is presented in Figure 4. The servomechanism electrohydraulic is compound from: 1 - assembled body; 2 - piston; 3 - composite sealing; 4 - anti-friction ring; 5 - anti-friction ring; 6, 7, 10 - "O" rings; 8 - compound seal; 9 - lid; 11 - felt ring; 12 - lid; 13 - grower washer; 14 - screw; 15 - position transducer; 16 - nut; 17 - lid; 18 - transducer support; 19 - screw; 20 - grower washer; 21 - limiter; 22 - screw; 23 - grower washer; 24 - support ring; 25 - lid; 26 - gasket; 27 - screw; 28 - servo valve.

![Figure 4. The principle scheme of the tested electrohydraulic servomechanism.](image)

Figure 4. The principle scheme of the tested electrohydraulic servomechanism.

Figure 5 shows the interface of the LabVIEW work program, with the aid of which the main data acquisition program was developed.

![Figure 5. The data acquisition model in LabVIEW: a - The main data acquisition program; b - The data acquisition subprogram.](image)
3. Result analysis
In recent years, a new language for modelling and numerical simulation of technical systems has emerged, called AMESim, developed by professors Lebrun and Richards in 1997. The language has been used successfully in both industrial and academic settings. The language allows the assembly of the mathematical models of the studied processes from correct models of technical components stored in the libraries written in the C programming language [17]. The simulation was performed using the AMESim program. The shock absorber was modelled according to the closed loop control law. Using a closed loop control law, we can consider a nonlinear compensation law. This compensation law imposes that we need to measure the pressure in the actuator cylinder (or the force) and the actuator velocity. Indeed, the aim is to adapt the gain of the system to the time constant due to hydraulic volumes of the cylinder and the restriction. The AMESim model of the actuator with the variable gain and the nonlinear compensation law is presented in Figure 6 and response damper in Figure 7 and Figure 8.

**Figure 6.** Model of a force control actuator based on a non-linear close loop control law.

**Figure 7.** The damping characteristic simulated in force-displacement coordinates for a hydraulic damper.

The road excitation signal (input signal) was approximated with a sinusoidal signal with a frequency of 4 Hz and 6 Hz. At low frequencies (up to about 4 Hz) the reference is very well. At high frequencies (starting at about 6 Hz) the response may be better, but instead oscillations occur, especially when changing the slope. The force/velocity characteristic generated by the closed loop actuator has been improved, compared to the open loop one. The modelling of the vehicle suspension is of great interest for road vehicle engineers, but also for vibration engineers.
Figure 8. The damping characteristic simulated in force-velocity coordinates for a hydraulic shock absorber.

During the shock absorber testing, the two main characteristics were determined: damping characteristic in force-velocity coordinates and damping characteristic in force-displacement coordinates; which define the behaviour of the shock absorber. The characteristics were determined using the three transducers: speed, force and position. The damper simulation model was validated by experimental test. Some experimental results obtained for a hydraulic damper are presented in the Figure 9 and Figure 10. The maximum force sites in the range (-1100 N ... +1000 N) for a velocity of the piston sited in the range (-200 mm/s...+200 mm/s). The hydraulic damper tested is new and it is equipping a Chevrolet Corvette car.

Figure 9. The damping characteristic determined experimentally in force-displacement coordinates for a classic shock absorber.

The energy dissipated by the hydraulic damper in a compression-rebound cycle is equal to the surface area delimited by the force-displacement characteristic (Figure 9). The characteristics presented above define the shock absorber itself and according to them the shock absorber produced by a manufacturer specialized in this field is chosen. The chosen shock absorber must provide a suspension damping characteristic, respective a damping characteristic required on the wheel. To determine the damping characteristic, the required damping range for the car body and wheels must first be determined.
A sinusoidal signal were chosen as excitation signal with the excitation frequency is 4 Hz, respectively 6 Hz (Figure 11 and Figure 12). The variation in time of the damping force, respectively of the velocity of movement of the piston for a period of 0.25 s, respectively 0.16 s is observed.

**Figure 10.** The damping characteristic determined experimentally in force-velocity coordinates for a classic shock absorber.

**Figure 11.** The variation of damping force and velocity in time $f = 4.0$ Hz.

**Figure 12.** The variation of damping force and velocity in time $f = 6.0$ Hz.
The excitation of the hydraulic damper is done with a position signal, similar to the real operating situation, a signal which depending on the type of test performed can be sinusoidal, triangular or compound. The output parameter is the damping force developed by the hydraulic damper corresponding to the different load speeds.

4. Conclusions

Modern hydraulic mono-tubular dampers have two-disc valves with different diameters, which are insensitive to temperature variation and thus to viscosity variation. At the moving the vehicle on roads with long length bumps, when the damping force decreases, the comfort also decreases. The change in stiffness of the suspension associated with the change in damping (so that the relative damping remains unchanged) leads to small variations in the car's behaviour in terms of comfort and safety of wheel contact with the road. When moving the car, the dynamic forces taken over by the shock absorbers are quite high. As the load decreases, there is an increase in relative damping, which is essential for low frequency oscillations. When the sprung mass decreases, the natural frequency increases, favouring the appearance of the resonance phenomenon at the usual travel speeds. If the damping factor is maintained when the load decreases, the comfort of oscillation at resonance regimes decreases.

References

[1] Savaresi M S, Poussot-Vassal C, Spelta C, Senname O and Dugard L 2010 Semi-active Suspension Control Design for Vehicles (Elsevier Ltd.)

[2] Guglielmino E, Sireteanu T, Stammers C W, Ghita G and Giulea M 2008 Semi-active Suspension Control – Improved Vehicle Ride and Road Friendliness (London: Springer-Verlag)

[3] Cao D, Song X and Ahmadian M 2011 Editors’ Perspectives: Road Vehicle Suspension Design, Dynamics, and Control. Veh. Sys. Dyn. 49 1–2 pp 3–28

[4] Farjoud A, Ahmadian M, Craft M and Burke W 2012 Nonlinear Modeling and Experimental Characterization of Hydraulic Dampers: Effects of Shim Stack and Orifice Parameters on Damper Performance. Non. Dyn. 67 2 pp 1437–1456

[5] Dixon J C 2007 The shock absorber handbook 2nd edition (London: John Wiley & Sons Ltd.)

[6] Duym S W R 2000 Simulation Tools, Modelling and Identification, for an Automotive Shock Absorber in the Context of Vehicle Dynamics Veh. Sys. Dyn. 33 4 pp 261–285

[7] Eyres R D, Champneys A R and Lieven N A J 2005 Modelling and Dynamic Response of a Damper with Relief Valve Non. Dyn. 40 pp 119–147

[8] Simms A and Crolla D 2002 The Influence of Damper Properties on Vehicle Dynamic Behaviour SAE Technical Paper 2002-01-0319

[9] Talbott M S and Starkey J 2002 An Experimentally Validated Physical Model of a High-Performance Mono-Tube Damper SAE Technical Paper 2002-01-3337

[10] Van Kasteel R, Cheng-Guo W, Lixin Q, Jin-Zhao L and Guo-Hong Y 2005 A New Shock Absorber Model for Use in Vehicle Dynamics Studies Veh. Sys. Dyn. 43 9 pp 613–631

[11] Lu Y, Li S and Chen N 2013 Research on Damping Characteristics of Shock Absorber for Heavy Vehicle Res. J. Appl. Sci. Eng. Technol. 5 3 pp 842-847

[12] Gomez-Aguilar J F, Yepez-Martinez H, Calderon-Ramon C, Cruz-Orduna I, Escobar-Jimenez R F and Olivares-Peregrino V H 2015 Modeling of a Mass-Spring-Damper System by Fractional Derivatives with and Without a Singular Kernel Entropy 17 9 pp 6289-6303

[13] Gomez-Aguilar J F, Rosales-Garcia J J, Bernal-Alvarado J J, Cordova-Fraga T and Guzman-Cabrera R 2012 Fractional Mechanical Oscillators Rev. Mex. Fis. 58 pp 348-352

[14] Li Z H and Au F T K 2014 Damage Detection of a Continuous Bridge from Response of a Moving Vehicle Sh. and Vib. vol. 2014 Article ID 146802

[15] Naranjani Y, Sardahi Y, Chen Y Q and Sun J Q 2015 Multi-Objective Optimization of Distributed-Order Fractional Damping Commun. Non. Sci. Numer. Simul. 24 1-3 pp 159–168

[16] Tavazoie M S 2015 Reduction of Oscillations via Fractional Order Pre-filtering Sign. Proc. 107 pp 407–414

[17] Vasiliu N, Vasiliu D, Calinou C and Puhalschi R 2018 Simulation of Fluid Power Sistem with Simcenter Amesim (CRC Press Taylor & Francis Group)