DAMPING CONTROL IN HYDROPNEUMATIC SUSPENSION

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Abstract. *Since the apparition of the model GS19 in 1955, Citroën has continuously used hydropneumatic suspensions in their cars. Other manufacturers offer hydropneumatic suspensions only in a few sophisticated and expensive models. However, hydropneumatic suspensions are largely used in military chars, off-road and railway vehicles. A drawback of hydropneumatic suspension, despite their excellent performance and flexibility of configuration, is that the damping factor is very sensible to load changes, making necessary to have damping control when applied in vehicles with large mass variations (relation between full loaded and empty mass vehicle). However, in some applications conventional control using servo-valves and electric (analogic or digital) signal processing are not suitable, due to particular characteristics of the environment and cost restrictions. In this work, a new concept of self tuning, robust, low cost valve to control damping is presented. This valve uses pressure of the system as a feed-back signal. An intuitive control strategy is implemented to control damping characteristics. The target values are obtained using the non linear model of the hydropneumatic spring behavior. Two different configurations are simulated in a half car model, and numerical results of each one are presented and discussed.*

Keywords*: Damping Control, Hydropneumatic Suspension, Suspension of Vehicles, Semi-active System*.

1. Introduction

In the first decades of the past century, the largest attention in the automotive industry was gone back to the development of faster, safe and comfortable vehicles. In a general way, the vehicles became fast before the highways presented reasonable conditions for the traffic. In this context, suspensions became a very important element for comfort and security. Several models and optimizations were proposed since the beginning of 20's century, Hrovat (1997). Systems of automotive suspensions have been object of countless analytical and experimental studies. In the 70's several publications were developed about this subject, with emphasis in their theoretical base, due to technological limitations, Alves (1997). One of the most important turn-key in suspension and vehicle dynamics was introduced in the 1955's Paris car saloon by *Citroën*, with their revolutionary model GS19 using hydropneumatic suspension. *Citroën* continues using this technology until now. Some commercial and a lot of special cars uses today this concept of suspension, Crolla (1991). This type of suspension offer excellent performance, high flexibility of configuration and facility to introduce some controls. However, hydropneumatic suspension is not widely used in conventional passenger cars due to their cost and more sophisticated technology, doing their maintenance more expensive. Then, hydropneumatic suspensions are more frequently used in special vehicles.

Recently in Baldi (2004), a hydropneumatic suspension was designed to be applied in a trailer used for spray crop protections. Other than the advantages mentioned, this choice was motived by the facility to introduce height control even when large changes in mass happened. However, in Meirelles and Baldi (2003) it is clearly showed that vehicles with large mass relations using hydropneumatic suspension needs damping control.

Agricultural tools and machinery may present simplicity and robustness characteristics, avoiding use of sophisticated electronic systems. Then, a new heuristic control strategy using an innovative valve design is proposed in this work. The valve designed for this application uses line pressure as a feedback to do the control, and target values are obtained from analytical model of the trailer with hydropneumatic suspension. Two different configurations are numerically tested, and results are presented and discussed.

2. Theoretical analysis

Hydropneumatic suspensions uses gas compressibility as a spring. Several models were proposed to mathematically represent their behavior. Different works proposed a few models for the compression process, assuming in genera ideal gas behavior: isotermic, adiabatic, or polytropic, Crolla (1991). In a quantitative way, the results do not present an important difference, considering the characteristics of the application in study. Then, ideal gas in isotermic process is assumed in Baldi (2004), considering the low frequency of the process, doing the nonlinear spring model used in the present work.

2.1 Description of the hidropneumatic suspension

The system consists of an amount of gas confined in a hermetic camera that is connected directly to the hydraulic cylinder, as shown in figure 1a. The difference comparing with conventional suspension erases in the substitution of the mechanical spring by gas compressed in the camera.

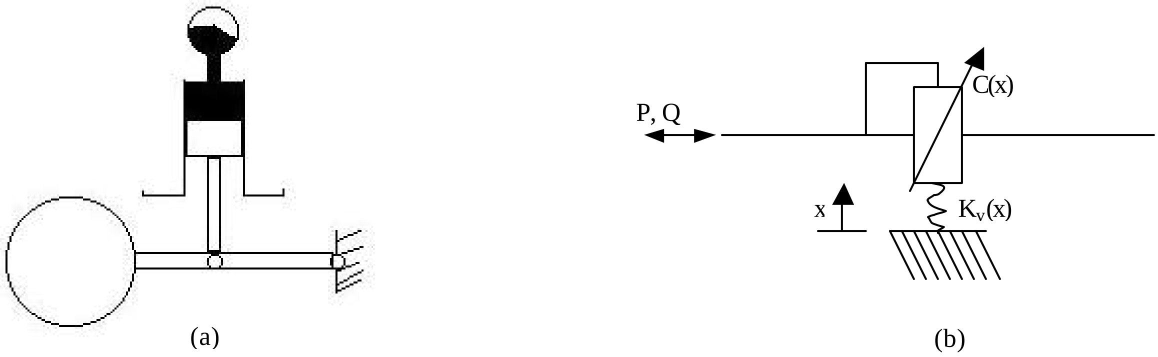


Figure 1 –(a) Schematic represented of Hydropneumatic Suspension, and (b) schematic representation of the valve.

When the wheel moves relatively to the chassis, an amount of hydraulic fluid is pumped to or from the camera, changing gas pressure. Pressure changes modify the restoring forces, given the system again to their equilibrium configuration. In this process, hydraulic fluid is pumped along the pipes, dissipating energy. This flow of energy can be controlled using a servo-valve to modify the load loss in the line, making the damping of the system controllable.

2.2. Damping adjust

Considering that the tests to be accomplished are of low frequencies, the isotermic model of ideal gas will be assumed for the simulations. Then.

$$PV = nRT = \text{cte.} \tag{1}$$

With this hypothesis it is possible to obtain the nonlinear rigidity function of the hydropneumatic suspension, as in Baldi (2004).

$$k(z) = \frac{P_0 A_c^{\ 2}}{(V_0 - A_c z)}\ , \tag{2}$$

where P_0 is atmosphere pressure, V_0 is the volume of the confined gas in the atmospheric pressure, A_c is the area of the hydraulic cylinder, and z is the piston displacement from V_0. Known the geometric characteristics of the system, the range of load to be supported, the waited dynamic loads and the suspension work span z_m, it is possible to determine the correct gas initial volume V_0 for such application, Baldi (2004).

$$V_0 = \frac{A_c P_m P_s z_m}{P_0 (P_m - P_s)} \tag{3}$$

Here P_s is the pressure corresponding to a full loaded vehicle in their natural height, and P_m is the pressure with the same load when suspension is full deflected.

With the spring characteristics and known the values of inertia and viscous damping, it is possible to calculate the damping factor for a system of one degree of freedom using:

$$x = \frac{C}{Cc}\ , \tag{4}$$

where $Cc = 2\sqrt{k.m}$ is the critical damping, and k and m represents the rigidity and inertia of the system.

Then, it is also possible to calculate the damping coefficient C needed to do a specified damping factor x :

$$C = 2x\sqrt{k.m} \tag{5}$$

Recently Meirelles et al. (2003) and Baldi et al. (2003), shown that maintain the damping factor always near a convenient value, for all load and defection conditions, is a very simple damping control strategy that gives satisfactory results.

To attain this objective without use signal processing, a totally mechanical valve using pressure feedback was designed (see figure 1b). The setup of this valve is done through the spring of convenient rigidity $K_v(x)$. Displacement x of the embol of the valve is function of the pressure P, that depends of load, suspension deflection and flow conditions.

Knowing the range of mass to be transported, the working span of the suspension and the rigidity of the suspension in each situation, it is possible to obtain the desired damping C using equation (5) (figure 2). Correlating this surface with the pressure corresponding to each point in the same m x z domain and the dissipative characteristics of the valve, it is possible to obtain the ideal rigidity K_v (x), as will be show in the next section. To do this setup, the valve developed for this application may be characterized.

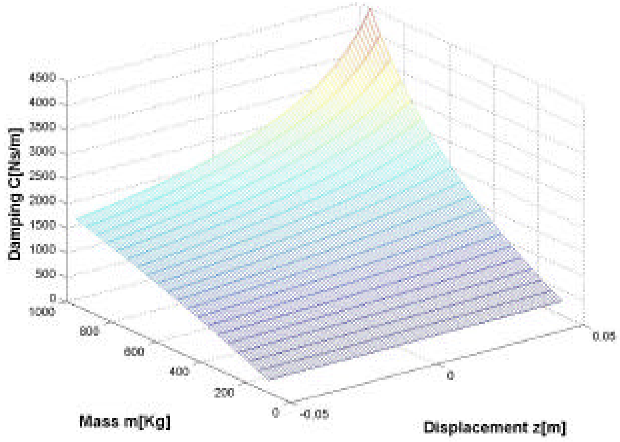


Figure 2. Damping wanted in function of the mass and displacement.

2.3. Characterization and set up of the valve

Besides the plumbing, valves are also responsible for losses of energy. Those losses are called local losses, Macintyre (1997). The general formula of the located losses J, in *mcf* (meters colon of fluid), can be described by:

$$J = K\frac{v^2}{2g} , \quad (6)$$

where, v and g are flow speed and gravity acceleration respectively, and K is the load loss coefficient of the valve, which is dependent of the geometry, and is usually determined experimentally. Values of K are tabulated for a large set of typical valves and localized restrictions. In this work the valve was suposed similar to a gate valve.

The gate valve can offer a great resistance to the flow. Figure 3 presents the experimental results obtained by Weisbach for the load loss K as a function of the opening of one gate valve, that has dimensions similar than the valve used in this work, Azevedo Netto (1970).

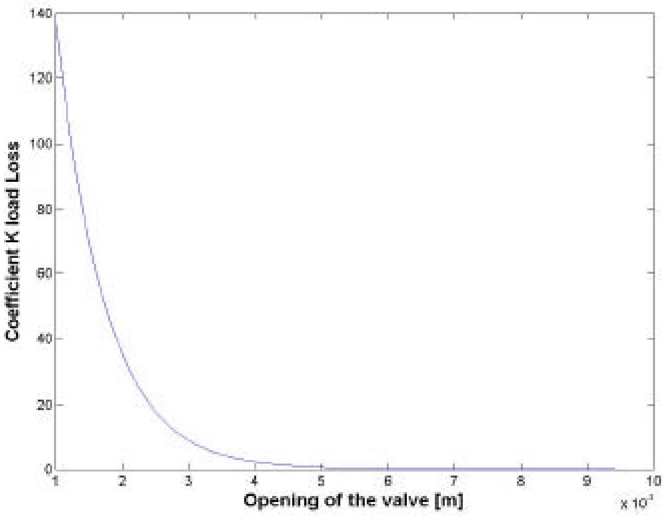


Figure 3. Coefficient K in function of the opening of valve.

Pressure difference between booth sides of the valve is done by:

$$P_a = r\ \ gJ\ , \tag{7}$$

where J is obtained from eq.(6). The force F_a opposite to the movement created by this pressure is $F_a = -A_c P_a$, where A_c is the area of the piston of the suspension. The flow speed v is done by $v = \frac{A_c}{A_t}\dot{z}$, where z is the suspension speed and A_t is the area of the pipeline. In the other side, it is well know that the damping force is done by $F_d = -C.\dot{z}$, and equaling forces $F_a = F_d$, results for the damping C:

$$C = \frac{A_c P_a}{\dot{z}}\ , \tag{8}$$

Making appropriate substitutions we obtain;

$$C = \frac{A_c\, rgK(\frac{A_c}{A_t}\dot{z})^2}{2g\,\dot{z}} = \frac{A_c^{\ 3}\, rK\,\dot{z}}{2A_t^{\ 2}} \tag{9}$$

Providing that the desired damping C be know at any time, the loss load coefficient K can be obtained:

$$K = \frac{2A_t^{\ 2}C}{A_c^{\ 3}r\ \dot{z}}\ , \tag{10}$$

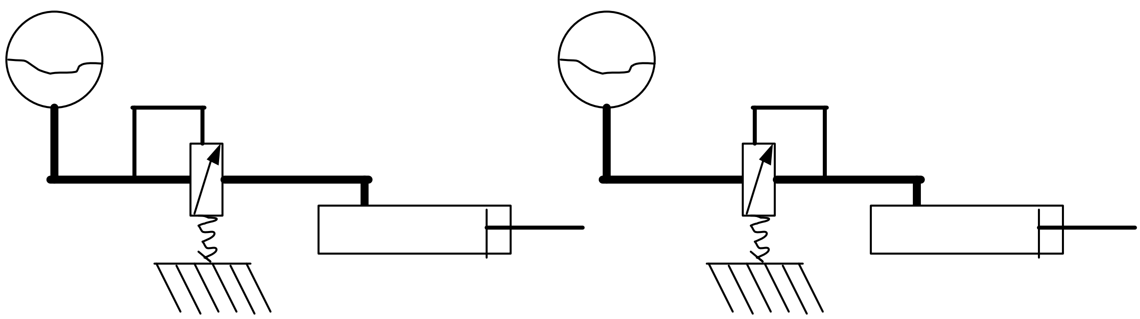
Loss load coefficient K tends to infinity when speed z tends to zero. To avoid speed dependence, a estimated mean value of z is used. After calculating K, the corresponding displacement x of the embol is obtained from interpolated function $K(x)$ of figure 3. Then, the ideal rigidity of the spring of the valve may be calculated by:

$$k_v(x) = \frac{PA_c}{x}\ , \tag{11}$$

for configuration in figure 4(a), and;

$$k_v(x) = \frac{(P \pm P_a)A_c}{x}\ , \tag{12}$$

for configuration in figure 4(b).



(a) (b)

Figure 4. Schematic representation of pressure feed-back in the valve.

$P=P(m,z)$ is the pressure in the chamber, and P_a is the increasing (or decreasing) of pressure caused by the valve, given by equation 7. The plus or minus signal applies when the trailer is lowered and when suspension expands, respectively. In second case the ideal rigidity kv is a function of P_a, that is function of speed z, making equation 12 not applicable in practice, because $kv(x,z)$ does not a unique value for each x. Then, rigidity calculated from equation 11, that is a mean value of that done by equation 12, will be used for both cases. This near optimal curve $k_v\ (x)$ can be shown in figure 5.

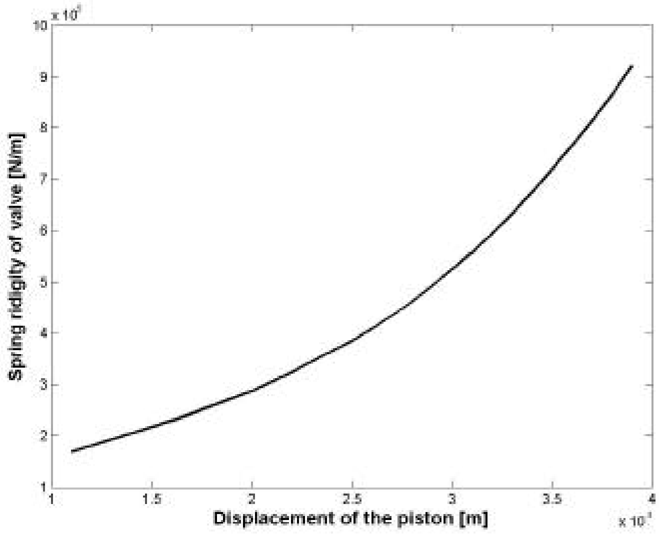


Figure 5. Valve spring rigidity.

3. Simulations

In systems with several degree of freedom, different damping factor happens, in general, for each mode. This problem may be overcame making a convenient choice of geometric parameters and mass distribution. However, in this work, even if the system has four degrees of freedom (see figure 6), only vertical displacement of all masses will be considered, making the system similar to a quarter car model.

With rotational DOF x_2 always null, deflection of suspension is $z = x_1 - x_3 = x_1 - x_4$ (see figure 6).

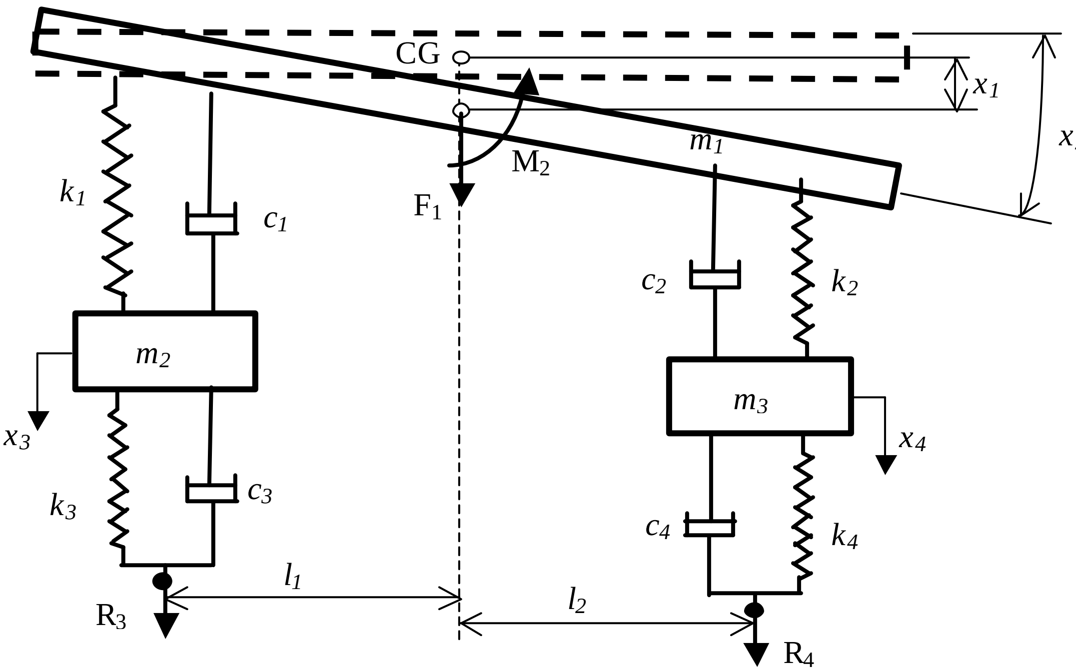


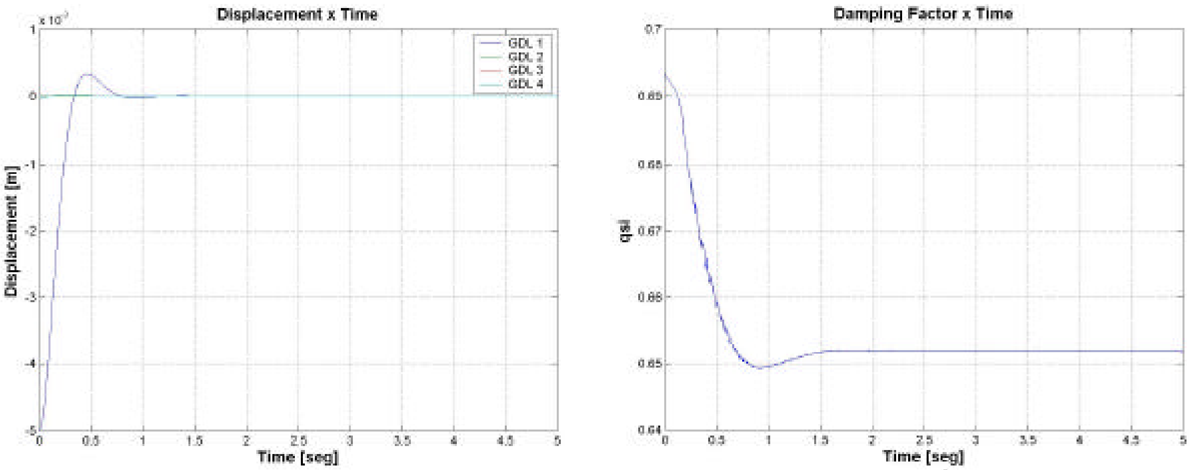
Figure 6. Half-car model of four degrees of freedom

Two kinds of tests will be presented: free vibration from non-null initial position, and response to a frequency limited band (in face in two wheels) excitation wich has similar power spectrum density (PSD) to a real signal registered in field. This tests was done with two different configurations, shown in figure 4a,b. The difference resides in the way that the feedback is implemented. In the first case the feedback is done taking pressure between the valve and pneumatic chamber (figure 4a), and in second case pressure feedback is taking between valve and the hydraulic cylinder (figure 4b).

3.1. Simulation First Case : Free oscillation.

In this test the system is released from this higherest position (z= -0,05 m), with feedback configuration of figure 4a. c x (t) are shown in figure 7a, and 7b, solid line, respectively.

Similar test was done with configuration shown in figure 4b. Results z (t) and damping factor x (t) are shown in figure 7a, and 7b, dashed line, respectively.



(a) (b)

Figure 7. (a)Displacement z and (b) damping factor x in function of time, free oscillation.

This simulation uses the same configurations describe here, but now, considering the system without damping control and the system with damping control related this work. In this test the system is released from this higherest position (z= -0,05 m) considering a load of 1000 [Kg] in vehicle, with feedback configuration of figure 4a. The time evolution of displacement z(t) with damping control and without damping control are shown in figure 8, in solid line and dashed line, respectively.

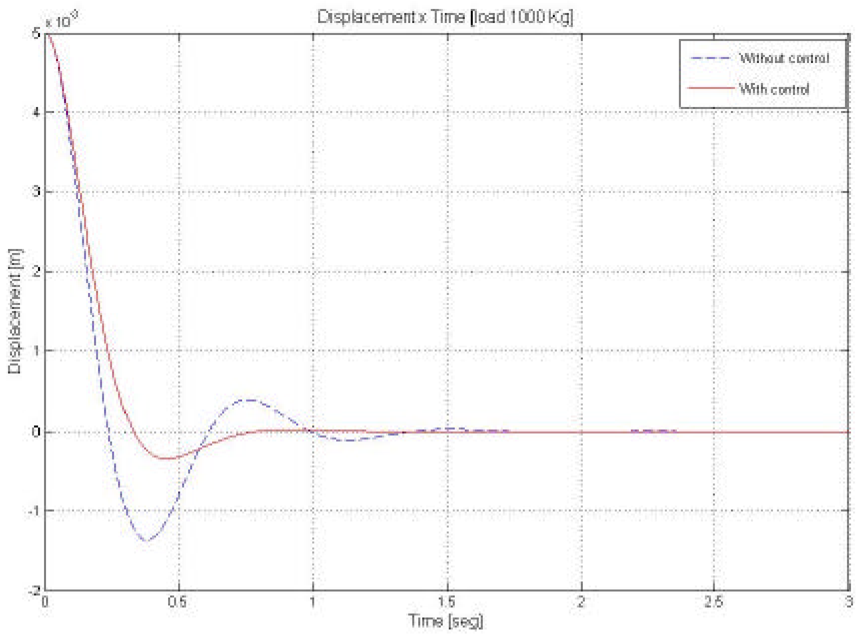


Figure 8. Displacement z in function of time, with load 1000 [Kg] and free oscillation.

3.2. Simulation Second Case: Forced response.

This simulation uses the same configurations described in section 3.1, but the system, initially at rest, is excited with a sintetised signal with frequency band limited between 1 and 15 [Hz], signal that has the same *PSD* of a read signal registered in field. The excitation was applied simultaneously in booth wheels, to excite only vertical displacements.

Time response z(t) and time evolution of damping factor x (t) are shown in figure 9a, and 9b respectively, in continues line for configuration in figure 4(a) and dashed line for configuration in figure 4(b).

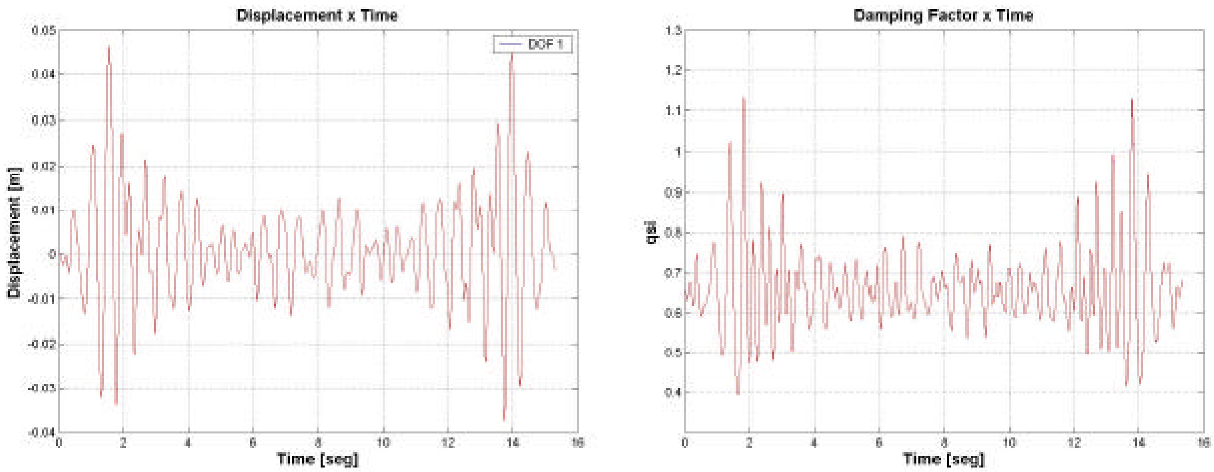


Figure 9. (a)Displacement z and (b) damping factor x in function of time, forced response.

4. Results

Results of the precedent tests shown that the control strategy adopted is very efficient, and can be implemented using the concept of the proposed valve. It is also possible to see that no significative difference exist between the two configurations tested. This is easily understood considering that the variation of pressure introduced by the valve is very week when compared with the static pressure. The variation of pressure produced by the valve, that affect only configuration in figure 4(b), is not greater than 0,015% of pressure in the chamber, that controls the valve in booth configurations.

5. Conclusion

In this work, a new valve design, related with a particular control objective, was presented. The model of a hydropneumatic spring is used. Tests was realized using a half-car model in two different configurations of the feedback signal.

Results have shown that no difference exist in practice between the responses obtained with this two options. It is also possible to evaluate that the proposed control done good results, solving the problem of excessive damping fluctuation in vehicles with hydropneumatic suspension.

6. Acknowledgements

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