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# IMPEDANCE CONTROL APPLIED TO THE SELECTION OF A PASSIVE SHOCK ABSORBER.

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***Abstract.*** *This paper studies the application of the impedance control as a way to define the best curve of a shock absorber during practical tests with the vehicle. The impedance control is performed by means of an active suspension system and, after testing, an equivalent passive shock absorber is selected. The dynamic model used is a quarter car suspension system and for simplicity, the dynamics of the hydraulic actuator was neglected in the design. The aim of the control system is to provide the desired impedance for the shock absorber. The measured variables are the relative position between the sprung and the unsprung mass and the force that the suspension system is submitted. The coefficients of the transfer function of the reference model are set during the procedure and represent changes in the characteristics of the shock absorber. It was used a* $H_2$ *compensator to control the system. The results of a numerical simulation suggest that the proposed method is feasible.*

***Keywords:*** *suspension system; impedance control; shock absorber; ride test;* $H_2$ *control.*

## 1. INTRODUCTION

In order to minimize the transmission of the irregularities from ground, suspension systems have been developed to provide comfort to vehicle occupants. In general, a suspension system can be characterized as a mechanical assembly that performs the interface between the ground and the remainder of the vehicle (Guillespie, 1992). With the increase in technology and in the speed at which vehicles do travel, one new task has been addressed to suspension systems: maintaining handling of the vehicle and therefore safety. Thus, these two criteria, comfort and handling are the main parameters defining the performance of a suspension system.

However, the criteria for comfort and handling are conflicting design features, which makes the suspension design a compromise concerning the application of each vehicle designed (Vargas, 2012).

The force that the tire exerts on the ground affects its adherence since the friction is proportional to the normal force between the surfaces (Pacejka, 2006), so, any factor that alters the load distribution, for instance a curve, compromises adherence (Guillespie, 1992) and handling. Thus it is appropriate that the suspension system absorb all the excitements and maintain the contact force between tire and ground.

In order to achieve this convenience, different suspension systems were developed. It is possible to find currently passive, semi-active and active suspension systems.

Passive suspension were the first systems developed and it is the basis for the developing of other types of suspension system and they are widely present on the literature (Guillespie, 1992), (Milliken and Milliken, 1995). These systems are basically consisting of conventional springs and dampers, with fixed dynamic and without the capability to insert energy into the system.

Semi-active suspension systems are those that have variable and controlled dynamic without inserting energy into the system. It is possible to achieve superior levels of comfort and safety than passive suspension systems at the cost of increased complexity. The required control effort of a semi-active suspension system is typically small, since this is only associated to the energy required to change the dynamics of the suspension system. Examples of elements of semi-active suspension systems are magneto-rheological dampers and gas springs with variable constraint.

Active suspension system is one that has variable and controlled dynamic by means of the insertion of energy to the system. Thus it offers more features for its purpose, but with the cost of greater complexity (Karnopp, 1986).

The different suspension systems were compared by Miller (1988) and Karnop (1986) in order to determine which is the most feasible for a particular application. However, we could not find in the literature an active suspension system as a tool in the development of a passive suspension system.

During the development of a passive suspension system many practical tests are required. One of the most important one is the so called Ride Test. This is a subjective evaluation of the vehicle with several types of shock absorbers (different curves) and roads, in order to select the shock absorber that satisfies the required conditions for comfort and safety. After the Ride Test the vehicles are instrumented in order to measure its acceleration, and then to analyze the performance of the suspension system according to regulated parameters, (Griffin, 1990), (ISO 2631) and (ISO 5982).

This article proposes the use of the impedance control technique to control an active suspension system so as to cause it to behave like a certain passive suspension system instead of replacing it during testing. Through a database of

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different shock absorbers available and changing the parameters of the shock absorbers through the control system, we argue that it is possible to perform the tests without really changing the suspension.

Using a quarter car models, it was developed an impedance control system for an active suspension system as a tool to the development of a passive suspension system.

## 2. ACTIVE SUSPENSION SYSTEM

A good suspension system should minimize the effect of the ground irregularities in safety and comfort issues for a car, so it must be designed according to the type of driving and terrain that are prescribed for the vehicle.

Active suspension system has a variable and controlled dynamics. It is also capable of inserting energy into the system. Active suspension systems offers more features with the cost of greater complexity (Karnopp, 1986) if it was compared to passive and semi-active suspension systems.

This kind of suspension consists of spring, shock absorbers and actuators (normally hydraulic), and has sensors to measure the displacement of the suspension. Its control system produces the force required to get the designed suspension performance. This control may require a large amount of energy, which is directly linked to energy entered into the system, which for a vehicle consists of motor power consumption and consequently fuel consumption.

To improve the performance of the suspension and/or to minimize its consumption, a good controller is necessary. Several control techniques have been developed and applied to active suspension systems such as $H\infty$ control (Du and Zhang, 2007), sliding mode control (Chen and Huang, 2005), optimal control (Corona *et al.*, 2004), fuzzy control (Sharkawy, 2005), linear quadratic control (Sohn and Hong, 2004), impedance control (Fateh and Alavi, 2008), Robust impedance control (Fateh, 2009), Adaptive impedance control (Fateh and Zirkohi, 2011) among others. Although we have found a series of three papers about impedance control applied to a suspension system they appear to be not realistic for a practical application since they use the absolute sprung mass position as a required measure. Besides, they have included the wheel dynamics in the model, which is normally poorly known.

Through the typical features of an active suspension system, modifying the system dynamics in a controlled manner, it is possible to obtain the desired behavior, even if it is to match a certain passive suspension behavior. In this scenario the control technique and the controller design are the key for it.

## 3. IMPEDANCE CONTROL

The impedance control was introduced by Hogan (1985) in robotics to regulate the dynamics of the robots in interaction with the environment. The impedance control objective is to control neither force nor position but their dynamic relation. (Almeida *et* al. 2000).

The objective of the impedance control differs from the conventional control systems in the sense that the main goal is to match a target model rather than to ensure the tracking of a reference signal. The key issue in this approach is to control the relationship of the force acting and the effector position. However, for the sake of simplicity this relationship is commonly adopted in the linear second-order differential equation form, describing simple and well understood six dimensional decoupled mass-spring-damper mechanical system. (Surdilovic, 1998).

In the field of vehicle dynamics, the impedance control was introduced by Fateh and Alavi (2009). As in all active suspension systems, the suspension control was achieved by applying a force generated by the actuator which is installed between the sprung and unsprung masses of the vehicle. The strength of the tire and the desired acceleration of the body (sprung mass) were used as input and output impedance of the law, respectively. In the implementation of impedance law, it was designed a control system such the active suspension system can follow the output impedance of the law.

## 4. METHOD

In the development of a passive suspension, after the design and assembly of the prototype, tests are performed to verify the functionality and durability of the system.

Just during the Ride Test a shock absorber is indeed defined. One reason is based on the simplifications adopted during the design that still need to be validated. Another and probably the main reason is to have a chance to choose a shock absorber from the current available in the market, avoiding costs of developing a new one.

The Test Ride consists of a subjective evaluation on the vehicle in several types of road and shock absorbers of different curves, in order to obtain the shock absorber that satisfies the conditions of the required comfort and safety. After the Ride Test, the vehicle is instrumented in order to measure its acceleration, and thus to analyze the performance of the suspension system according to regulated parameters, (Griffin, 1990), (ISO 2631) and (ISO 5982)

In order to minimize the testing time by avoiding the exchange of shock absorbers it is suggested placing an active suspension so as to emulate the various shock absorbers available.

For this purpose it will be necessary to develop a controller able to cause the actuator of the suspension system to inject the difference between the dissipation energy of the shock absorber installed and the dissipation energy of the

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shock absorber to be simulated. Therefore, knowing the characteristic curves of the shock absorber is so important. Thus, a database of all available shock absorbers is needed.

To design the controller and to validate the method, it was used a quarter car model with an active suspension system.

### 4.1 Model

In the suspension system design, it is common to use simplified models such as a quarter car model to study the dynamic behavior of the vehicle. Fig. 1 depicts a quarter car model including an active actuator.

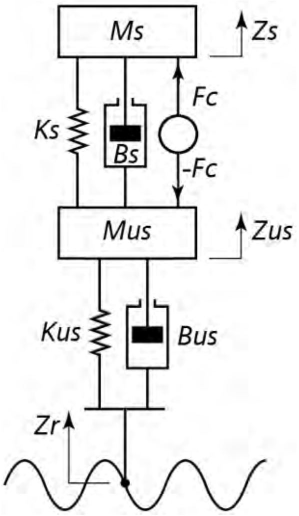


Figure 1. A quarter car model of an active suspension system.

From Fig. 1 it is possible to get the associated dynamic equations

$$M_s \ddot{Z}_s = -F_c - K_s \left(Z_s - Z_{us}\right) - B_s \left(\dot{Z}_s - \dot{Z}_{us}\right) \tag{1}$$

$$M_{us} \ddot{Z}_{us} = F_c + K_s \left(Z_s - Z_{us}\right) + B_s \left(\dot{Z}_s - \dot{Z}_{us}\right) - K_{us} \left(Z_{us} - Z_r\right) - B_{us} \left(\dot{Z}_{us} - \dot{Z}_r\right) \tag{2}$$

Where $M_s$, $M_{us}$, $K_s$, $K_{us}$, $B_s$, $B_{us}$ are the mass, stiffness and damping ratio of sprung and unsprung elements respectively. The variables $Z_s$, $Z_{us}$ and $Z_r$ are the displacements of the body, wheels and track, respectively, measured from the point of static equilibrium generated by the gravitational force. $F_c$ is the force applied by the actuator system. Note that the actuator is installed between the sprung and unsprung masses.

The variable $Z_r$ and $F_c$ are the inputs of the system and they make the system get out of the equilibrium.

We can write the equations (1) and (2) in the vetorial form

$$M\,\ddot{x}(t) + B\,\dot{x}(t) + K\,x(t) = L\,u(t) + N\,\dot{u}(t)\,, \tag{3}$$

where

$$M = \begin{bmatrix} M_s & 0 \\ 0 & M_{us} \end{bmatrix}, \tag{4}$$

$$B = \begin{bmatrix} B_s & -B_s \\ -B_s & B_s + B_{us} \end{bmatrix}, \tag{5}$$

$$K = \begin{bmatrix} K_s & -K_s \\ -K_s & K_s + K_{us} \end{bmatrix}, \tag{6}$$

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$$L = \begin{bmatrix} 0 & -1 \\ K_{us} & 1 \end{bmatrix}, \tag{7}$$

$$N = \begin{bmatrix} 0 & 0 \\ B_{us} & 0 \end{bmatrix}, \tag{8}$$

$$X = \begin{bmatrix} X_1 \\ X_2 \end{bmatrix} = \begin{bmatrix} Z_s \\ Z_{us} \end{bmatrix}, \tag{9}$$

$$U = \begin{bmatrix} U_1 \\ U_2 \end{bmatrix} = \begin{bmatrix} Z_r \\ F_c \end{bmatrix}. \tag{10}$$

Applying the Laplace transform, the Eq. (3) become

$$\left(s^2 M + sB + K\right) X(s) = \left(L + sN\right) U(s). \tag{11}$$

That is,

$$X(s) = G(s)U(s), \tag{12}$$

where

$$G(s) = \left(s^2 M + sB + K\right)^{-1} \left(L + sN\right). \tag{13}$$

Defining the outputs of the system as

$$Y = \begin{bmatrix} Y_1 \\ Y_2 \end{bmatrix} = \begin{bmatrix} Z_s - Z_{us} \\ f_r \end{bmatrix}, \tag{14}$$

where

$$f_r = K_s\left(Z_s - Z_{us}\right) + B_s\left(\dot{Z}_s - \dot{Z}_{us}\right), \tag{15}$$

it can be described in the Laplace domain as

$$f_r(s) = \left(sB_s + K_s\right)\left(Z_s(s) - Z_{us}(s)\right). \tag{16}$$

Therefore

$$Y = \begin{bmatrix} Y_1 \\ Y_2 \end{bmatrix} = \begin{bmatrix} Z_s - Z_{us} \\ f_r \end{bmatrix} = \begin{bmatrix} 1 \\ sB_s + K_s \end{bmatrix} \left(Z_s - Z_{us}\right) \tag{17}$$

$$Y = \begin{bmatrix} 1 \\ sB_s + K_s \end{bmatrix} \begin{bmatrix} 1 & -1 \end{bmatrix} X \tag{18}$$

Thus

$$Y(s) = H(s)U(s), \tag{19}$$

where

$$H(s)=\begin{bmatrix}1\\ sB_s+K_s\end{bmatrix}\begin{bmatrix}1 & -1\end{bmatrix}\left(s^2M+sB+K\right)^{-1}\left(L+sN\right)\cdot \tag{20}$$

In other words,

$$H(s)=\begin{bmatrix}R_1(s) & P_1(s)\\ R_2(s) & P_2(s)\end{bmatrix}. \tag{21}$$

and it is illustrated by the block diagram of Fig. 2.

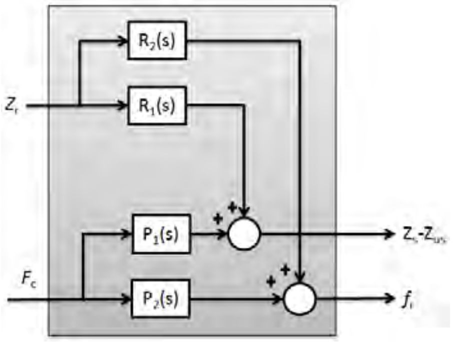


Figure 2. A quarter car model tranfer matrix.

## 4.2 Controller Design.

The goal of the impedance control of an active suspension system can be viewed as the way to impose, by the force $F_c$, the impedance

$$I_1(s)=\frac{Z_s(s)-Z_{us}(s)}{f_r(s)+F_c(s)} \tag{22}$$

to be

$$I_1(s)=\frac{1}{s\left(B_s+B_c\right)+\left(K_s+K_c\right)}=\frac{1}{s\overline{B}+\overline{K}}, \tag{23}$$

where $B_c$ and $K_c$ are the damping ratio and the stiffness generated by the actuator, respectively.

The closed-loop implementation is illustrated by the diagram in Fig. 3, where it is clear the need for sensors of the relative position ($Z_s$ - $Z_{us}$) and for the force $f_r$ so that the controller F(s) may keep the error e(t) small, what ensures that the impedance of the shock absorber is approximately equal to $I_1(s)$.

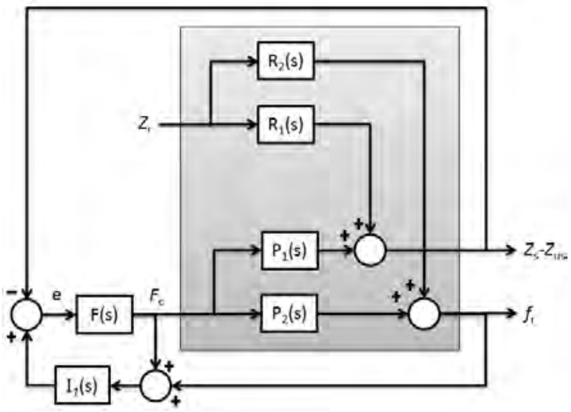


Figure 3. Closed loop control system.

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The two feedbacks present in the control system suggest that this problem has two controlled variables and one manipulated variable. However, for design purposes it is sufficient to have the error e(t) small in the range of frequencies that matters to the problem, featuring an equivalent SISO problem. To further examine this question, one can use the equivalent diagram of Fig. 4.

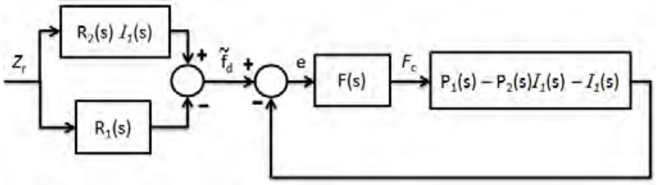


Figure 4. Equivalent diagram of the control system.

The control problem is to find F(s) such that $e \approx 0$ in the frequency range where $f_d$ has most of its energy. Additionally, the control effort $F_c$ must be compatible with the power available in the system.

Defining:

$$PP = P_1 - P_2 I_1 - I_1 \tag{24}$$

and

$$RR = R_2 I_1 - R_1 \text{ ,} \tag{25}$$

and considering that a step change of $Z_r$ is drastic enough to evaluate the performance of the system, the spectrum of $f_d$ is given by

$$\tilde{f}_d(j\omega) = \frac{R_2(j\omega) I_1(j\omega) - R_1(j\omega)}{j\omega} . \tag{26}$$

In what follow we adopt the numerical values suggested by Fateh (2009), i.e.,

- $M_s$ = 243 kg;
- $M_{us}$ = 40 kg;
- $B_s$ = 370 Ns/m;
- $B_{us}$ = 414 Ns/m;
- $K_s$ = 14671 N/m;
- $K_{us}$ = 124660 N/m;

The Bode diagram $|PP(j\omega)|$ and the spectrum $|f_d(j\omega)|$ are shown in dB in Fig.6. Note that the frequency range that goes to approximately 100 rad/s contains the most energy of $f_d$. One can also see that the band pass of $|PP(j\omega)|$ extends up to about 100 rad/s. Therefore, for the design of F(s) we simply ensure that the closed-loop sensitivity function S(s) is small enough up to approximately 100 rad/s.

To enforce this performance specification it was used the Mixed Sensitivity Technique of the $H_2$ control theory. Figure 5 shows the diagram used in the project where $W_1(s)$ is the penalty function for the sensitivity function and $W_3(s)$ is the penalty used to mitigate the gains of the control loop at high frequencies to avoid amplification of measurement noise and modeling errors.

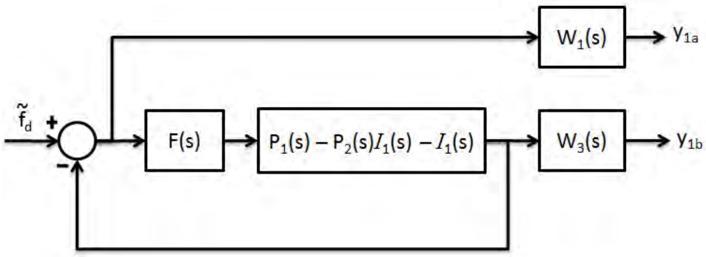


Figure 5. Diagram of the $H_2$ mixed sensitivity design.

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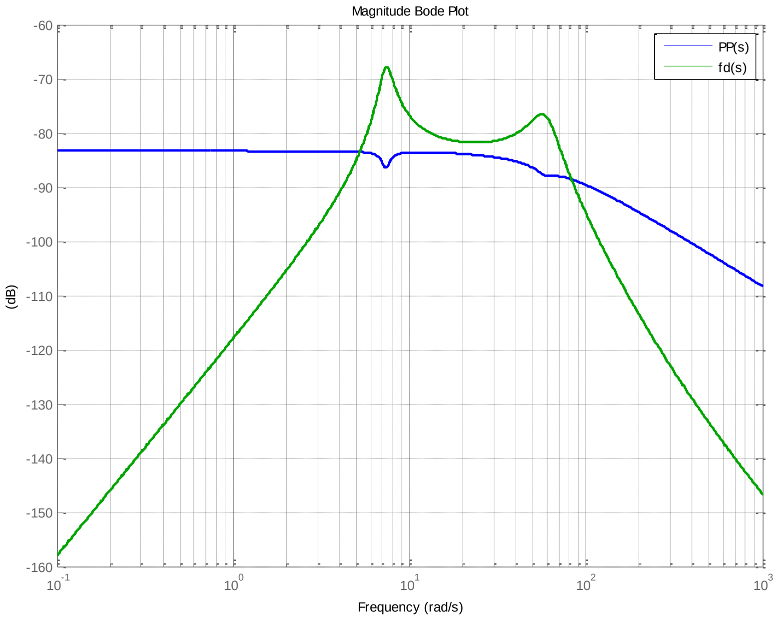


Figure 6. Bode plots.

We have chosen the following functions for both. Note that the $W_1(s)$ tries to impose a high gain up to about 100 rad/s and $W_3(s)$ tries to impose gain attenuation from that frequency.

$$W_1(s) = \frac{10^3}{0,01s+1} \tag{27}$$

$$W_3^{-1}(s) = \frac{1}{0,01s+1} \tag{28}$$

The obtained controller F(s) has two real poles *-4374,4 +0j* and *-100 + 0j* and one real zero *-56,6 + 0j*. It is noted from the plot in Fig 7 that the sensitivity function of the closed loop has gains below -20dB to approximately 200 rad/s, ensuring a small

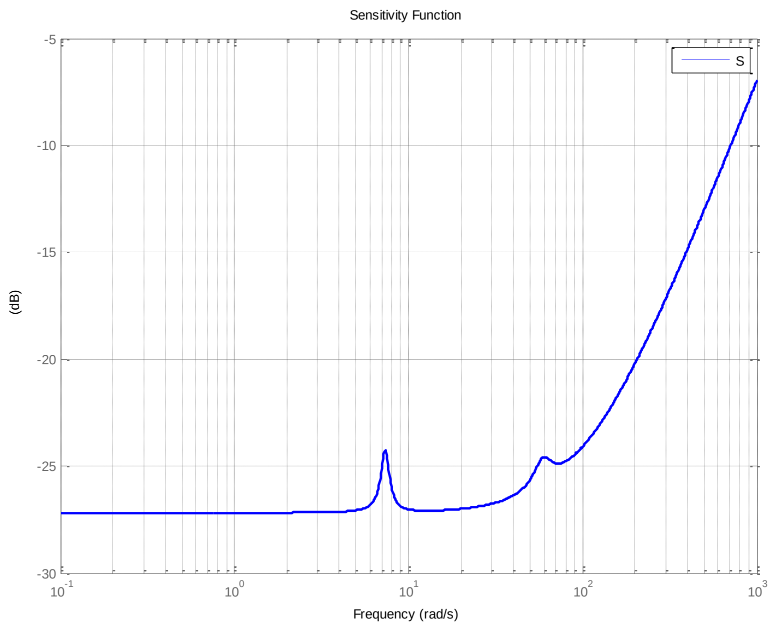


Figure 7. Sensitivity function.

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## 5. RESULTS AND ANALYSIS

To evaluate the performance of the controller F(s) the impedance of the suspension is actively changed and the system behavior is compared to an equivalent passive suspension. Comparisons are made by means of time domain responses to a disturbance in the runway in the form of a step change. It is also evaluated the control effort needed to keep small the matching error between the reference model $I_1(s)$ and the actual impedance. It is expected that reference models that are far from the actual passive impedance implies in a greater effort control and reference models that are close, exert less control.

The first test refers to a -50% change in damping of the reference model. Fig. 8 illustrates the performance achieved. The plots in (a) suggest that the control system can make the suspension mimic the behavior of the equivalent passive suspension with good approximation. Notice that visually the Real and Reference plots match each other. In plot (b) we note that the control effort is higher at the beginning of the disturbance and reaches about 75N.

The second test refers to a +50% change in damping of the reference model. Fig. 9 illustrates the performance achieved. The plots in (a) suggest that the control system can make the suspension mimic the behavior of the equivalent passive suspension with good approximation. Notice that visually the Real and Reference plots match each other. In plot (b) we note that the control effort is higher at the beginning of the disturbance and reaches about 70N.

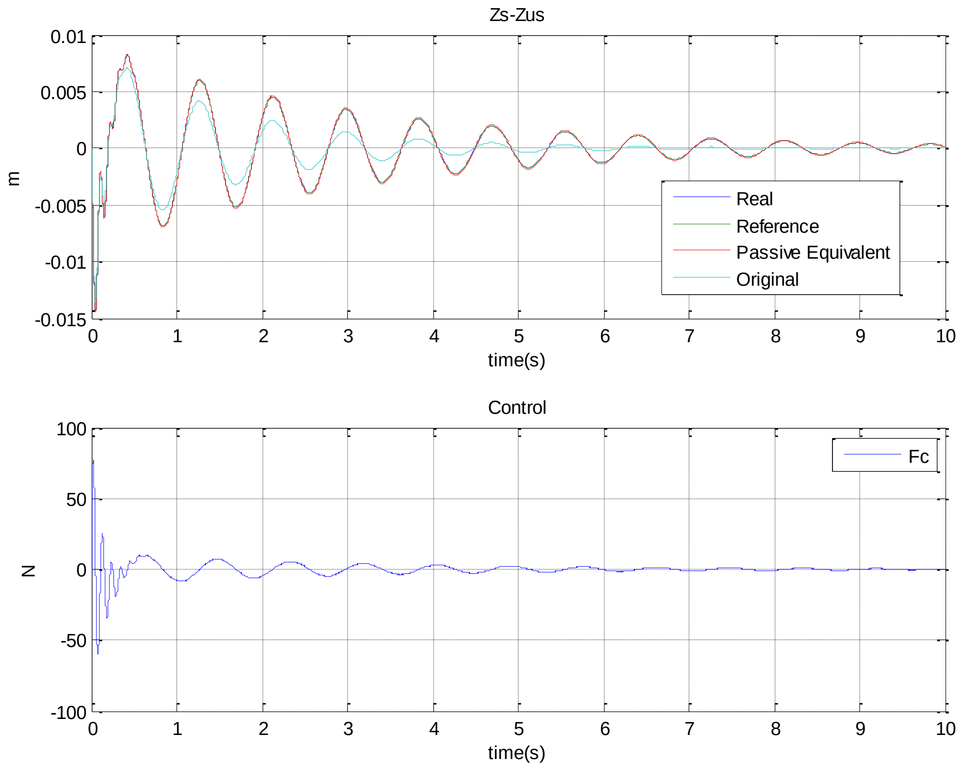


Figure 8. (a) Performance for a variation of -50% in the damping. (b) Control effort required for the model matching.

The third test, although not related to the purpose of this work that aims only to change the damping, illustrates that the active change of the stiffness is more difficult to be achieved because it represents a larger modeling error to be compensated by the controller that should change actively oscillation frequency of the system.

In this test the stiffness and the damping reference were changed in 50%. Fig. 10 illustrates the performance achieved. The plots in (a) suggest that the control system can make the suspension mimic the behavior of the equivalent passive suspension with good approximation. Notice that visually the Real and Reference plots match each other. In plot (b) we note that the control effort is higher at the beginning of the disturbance and reaches about 120N. Note that the maximum control effort is greater in this case.

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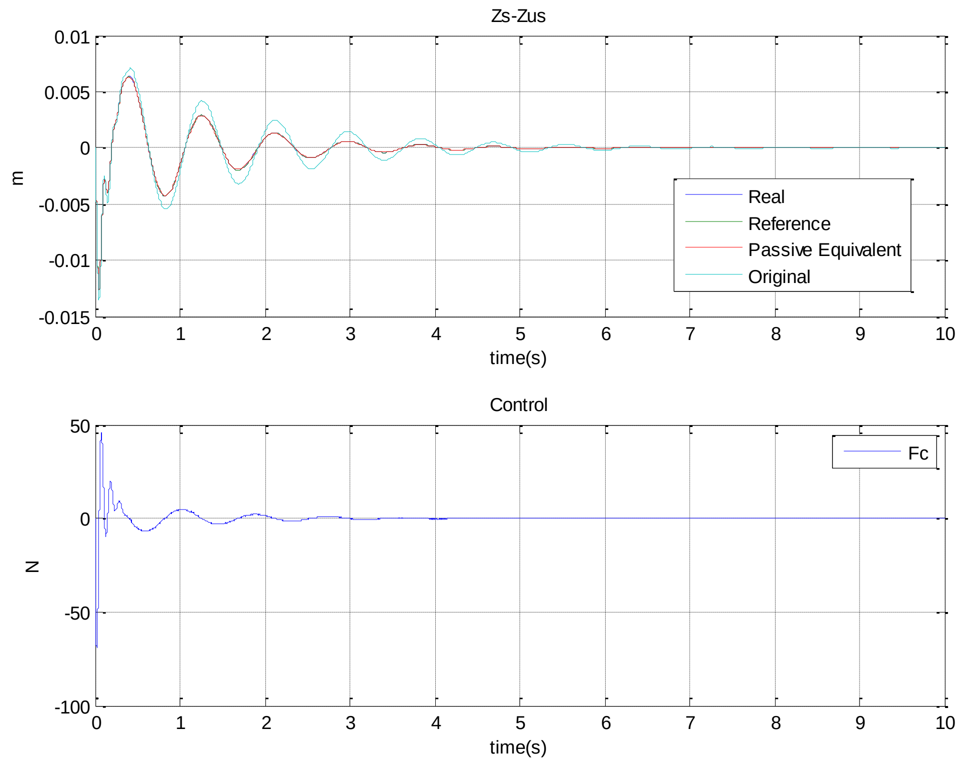


Figure 9. (a) Performance for a variation of +50% in the damping. (b) Control effort required for the model matching.

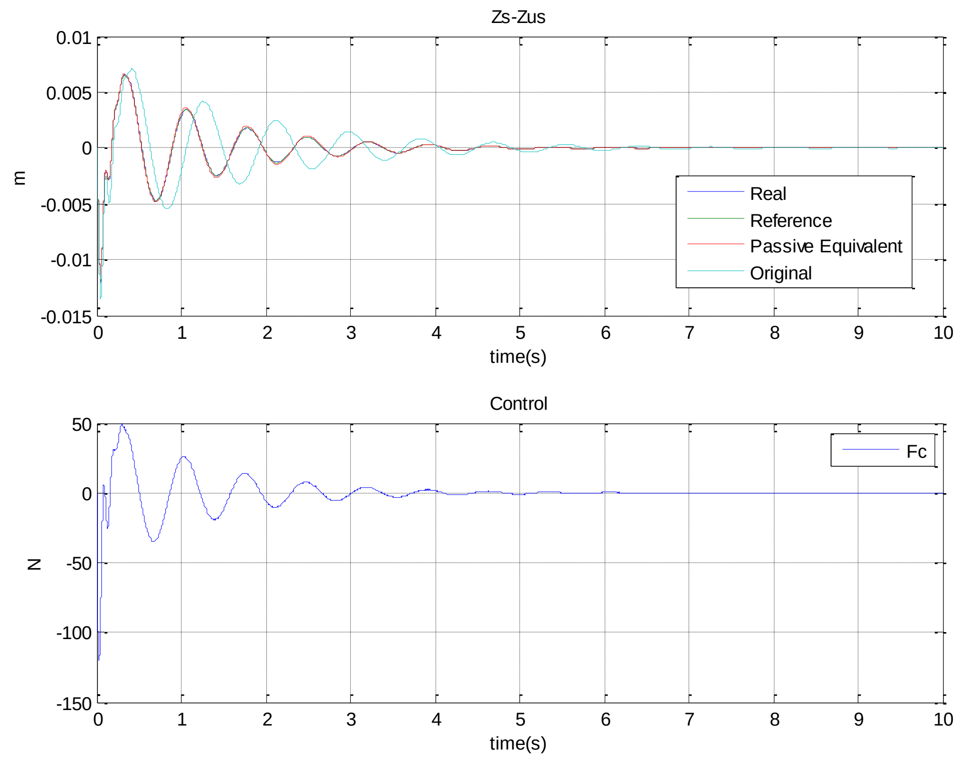


Figure 10. (a) Performance for a variation of +50% in stiffness and damping. (b) Control effort required for the model matching.

## 6. CONCLUSIONS

During the development of a passive suspension system practical tests are required, such as the ride test. That test is an evaluation of the vehicle with several kinds of shock absorbers in order to select the one that better satisfies comfort and safety criteria. This work proposed the use of the impedance control technique to drive an active suspension system to mimic the behavior of a certain passive shock absorber instead of sequentially replacing it during testing. Through a database of different shock absorbers available and changing the parameters of the shock absorbers through the control system, we argue that it is possible to perform the test without really changing the suspension.

To do so an impedance control system was designed in order to apply an active suspension system as a tool to develop of a passive shock absorber.

The application of impedance control was effective in order to change the system dynamics by changing the damping coefficient, as suggested by the plots in all tests. To change the suspension dynamics it is necessary enough energy to generate force to change the dynamics accordingly. The tests also suggested that the energy is directly related to how far the passive components installed in the active suspension system are from the desired system.

Based on the performed tests it can be concluded that the impedance control allows changing the parameters of the suspension, thus making it technically feasible to develop an active suspension for chosen the equivalent passive shock absorber.

As continuity of this work it is proposed to carry out practical tests with actual equipment. It is also recommended to use more realistic models of vehicular shock absorbers such as those including the typical asymmetry.

## 7. ACKNOWLEDGEMENTS

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