EXPERIMENTAL IMPLEMENTATION OF AN ELECTROMAGNETIC AUTOMOBILE SUSPENSION SYSTEM

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Abstract: The suspension systems currently in use can be classified as passive, semi-active and active. The passive suspension systems are the most commonly used due their low price and high reliability. However, this system cannot assure the desired performance from a modern suspension system. An important improvement of suspension performance is achieved by the active systems. Nevertheless, they are only used in a very reduced number of automobile models because they are expensive and complex. Another disadvantage of active systems is relatively high energy consumption.
The use of electromagnetic linear actuators is an alternative for the implementation of active automobile suspensions systems. Moreover, this solution has the advantage of the suspension energy recovery. The author's work group have proposed an electromagnetic actuator that is already implemented in a workbench. With the materials development the prototype can be improved in order to obtain higher power with less volume.
In this paper it is analysed a pseudo-levitating suspension system. Simulation and experimental results are also presented.

Keywords: Automobile suspension, electromagnetic, pseudo-levitating, control laws, test bench.

I. INTRODUCTION

The main function of vehicle suspension system is to isolate the vehicle body and passengers from the oscillations created by the road irregularities and to produce a continuous road-wheel contact.
Three types of vehicle suspensions are used: passive, semi-active and active. All the systems known as implemented in automobiles are based in hydraulic or pneumatic operation. However, it is verified that these solutions cannot solve satisfactory the vehicles oscillations problem or they are very expensive and contribute to the increasing of the energy vehicle consumption. The figure 1 presents the tree types of suspension systems that are used, where m_s represents the sprung mass, m_u the unsprung mass, K_s the spring stiffness, B_S the damping coefficient, F_a the actuator force, K_t the tyre stiffness, Z_r the road position reference, Z_u the unsprung mass position reference and Z_s the .sprung mass position reference.

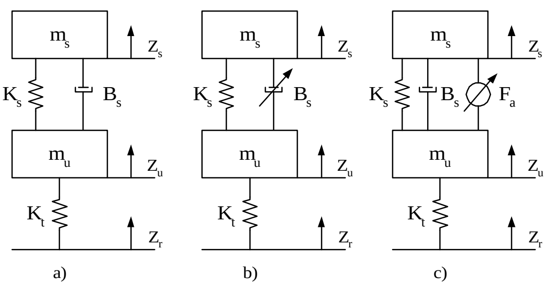

Figure 1: Used suspension systems: a) Passive; b) Semi-active; c) Active.

One possible solution, that has been proposed [1] to solve these difficulties, is to use electromagnetic actuators in order to improve the performance of suspension systems without increasing the energy consumption and costs. All referred types of suspension systems could be built using electromagnetic actuators. Other advantage of electromagnetic suspensions is the possibility to work in the generator mode operation. This characteristic allows recovering energy from the suspension.
In order to experimental test the mathematical models a workbench was implemented in the laboratory. A sprung mass, an electromagnetic actuator, a spring, a wheel, and a road perturbation generator compose the suspension system prototype.
The analysed suspension system was introduced by [1] as a pseudo-levitating suspension. This

suspension system uses an electromagnetic actuator and a spring between the sprung and the unsprung masses. The spring supports the body weight and the electromagnetic actuator is controlled in order to compensate the spring deformation force. With this topology smaller power electromagnetic actuators can be used.

II. PSEUDO-LEVITATING SUSPENSION SYSTEM ANALYSIS AND CONTROL

2.1. Pseudo-Levitating Suspension Analysis

The suspension system analysis is made considering the representation of a quarter of the sprung mass, in vehicles with independent suspensions. The model of ¼ of vehicle is presented in figure 2. m_s represents the sprung mass, m_u the unsprung mass, K_s the spring stiffness, F_a the actuator force, K_t the tyre stiffness, the Zr, Zu, and Zs represents the reference position of the road, unsprung and sprung masses, respectively.

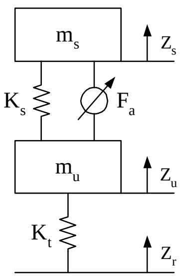

Figure 2: Pseudo-Levitating suspension system.

In this system, the spring supports the body weight and the actuator is controlled in order to compensate the spring deformation force

From the model in figure 2 we can write the system's equations (1).

$$\begin{cases} m_s \ddot{z}_s = -K_s\left(z_s^{'} - z_u^{'}\right) + F_a - m_s g \\ m_u \ddot{z}_u = K_s\left(z_s^{'} - z_u^{'}\right) - F_a - K_t\left(z_u^{'} - z_r^{'}\right) - m_u g \end{cases} \quad (1)$$

It is possible to make the following variable change:

$$\begin{cases} z_s^{'} = z_s^0 + z_s \\ z_u^{'} = z_u^0 + z_u \\ z_r^{'} = z_r^0 + z_r \end{cases} \quad (2)$$

$$\begin{cases} K_s\left(z_s^0 - z_u^0\right) = -m_s g \\ K_t\left(z_u^0 - z_r^0\right) = -\left(m_s + m_u\right) g \end{cases} \quad (3)$$

Rewriting the system (1) we obtain the suspension equation as (4).

$$\begin{cases} m_s \ddot{z}_s = -K_s\left(z_s - z_u\right) + F_a \\ m_u \ddot{z}_u = K_s\left(z_s - z_u\right) - F_a - K_t\left(z_u - z_r\right) \end{cases} \quad (4)$$

Choosing the state space and the output vector as:

$$x^T = \begin{bmatrix} z_s & \dot{z}_s & z_u & \dot{z}_u \end{bmatrix} \quad (5)$$

$$y^T = \begin{bmatrix} \ddot{z}_s & z_u - z_r & z_s - z_u \end{bmatrix} \quad (6)$$

The input vector is:

$$u^T = \begin{bmatrix} F_a & z_r \end{bmatrix} \quad (7)$$

The state space system will be written as:

$$\begin{cases} \dot{x} = Ax + Bu \\ y = Cx + Du \end{cases} \quad (8)$$

$$\begin{bmatrix} \dot{z}_s \\ \ddot{z}_s \\ \dot{z}_u \\ \ddot{z}_u \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -\frac{K_s}{m_s} & 0 & \frac{K_s}{m_s} & 0 \\ 0 & 0 & 0 & 1 \\ \frac{K_s}{m_u} & 0 & -\frac{\left(K_s + K_t\right)}{m_u} & 0 \end{bmatrix} \cdot \begin{bmatrix} z_s \\ \dot{z}_s \\ z_u \\ \dot{z}_u \end{bmatrix} + \begin{bmatrix} 0 & 0 \\ \frac{1}{m_s} & 0 \\ 0 & 0 \\ -\frac{1}{m_u} & \frac{K_t}{m_u} \end{bmatrix} \cdot \begin{bmatrix} F_a \\ z_r \end{bmatrix} \quad (9)$$

$$\begin{bmatrix} \ddot{z}_s \\ \left(z_u - z_r\right) \\ \left(z_s - z_u\right) \end{bmatrix} = \begin{bmatrix} -\frac{K_s}{m_s} & 0 & \frac{K_s}{m_s} & 0 \\ 0 & 0 & 1 & 0 \\ 1 & 0 & -1 & 0 \end{bmatrix} \cdot \begin{bmatrix} z_s \\ \dot{z}_s \\ z_u \\ \dot{z}_u \end{bmatrix} + \begin{bmatrix} \frac{1}{m_s} & 0 \\ 0 & -1 \\ 0 & 0 \end{bmatrix} \cdot \begin{bmatrix} F_a \\ z_r \end{bmatrix} \quad (10)$$

The objective of a system would be a suspension with a zero sprung mass acceleration and a constant tyre deflection:

$$\ddot{z}_s = 0 \quad (11)$$

$$\left(z_u - z_r\right) = const. \quad (12)$$

However these two objectives cannot be reached together for all the frequency values of the road disturbance.

From the system 9 it's possible to obtain the open loop transfer functions and then compute the poles of the system. In this case the frequencies obtained are:

f_1= 1,4Hz and f_2= 10Hz

The low frequency can be associated to the interaction between the sprung mass and the spring, and the second critical frequency is resonance referred to the tyre.

2.2. Pseudo-Levitating Suspension Control Law

Forcing to zero the sprung mass acceleration in the equation 1 of the system (4) we obtain the ideal actuator force given by (14) that totally compensates the spring deformation.

$$\ddot{z}_s = 0 \quad (13)$$

$$F_a = K_s(Z_s - Z_u) \quad (14)$$

However with a force given by (14) the unsprung mass subsystem will be unstable, as it can be seen in the second equation of the system (4), because it will not have attenuation. In order to damp the unsprung mass oscillations near to its resonance frequency a gain k was introduced. The force equation becomes:

$$F_a = K_s(Z_s - Z_u) + k\dot{Z}_u \quad (15)$$

This control law provides a minimal sprung mass acceleration and with a safe tyre deflection. The value of k should be chosen according to the oscillations frequency.
Rewriting the system (4) with the equation (15) we obtain the close loop system (16).

$$\begin{cases} m_s \ddot{z}_s = k\dot{z}_u \\ m_u \ddot{z}_u = -K_t(z_u - z_r) - k\dot{z}_u \end{cases} \quad (16)$$

From the previous system its possible to obtain the close loop transfer functions, that are:

$$\begin{cases} \dfrac{\ddot{z}_s}{z_r} = \dfrac{kK_t s}{m_s m_u s^2 + km_s s + m_s K_t} \\[2ex] \dfrac{z_u - z_r}{z_r} = \dfrac{kK_t s}{m_u s^2 + ks + K_t} \\[2ex] \dfrac{z_s - z_u}{z_r} = \dfrac{m_s K_t s - kK_t}{m_s m_u s^2 + km_s s + m_s K_t} \end{cases} \quad (17)$$

The frequency response obtained is presented in figure 3, which shows the differences between the tree control laws.

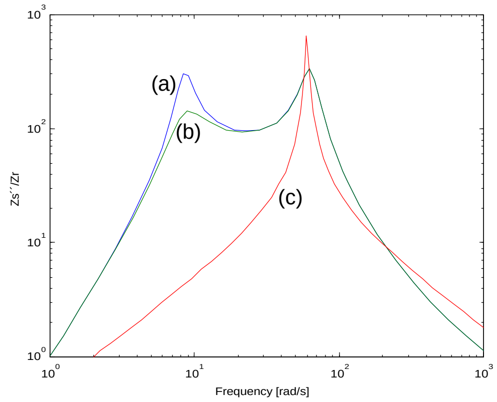

Figure 3: Frequency response: a) Passive b) Active c) Pseudo-Levitating.

III. PSEUDO-LEVITATING SUSPENSION SYSTEM IMPLEMENTATION

3.1. Description

In the figure 4 is presented the pseudo-levitating suspension system workbench that was implemented.

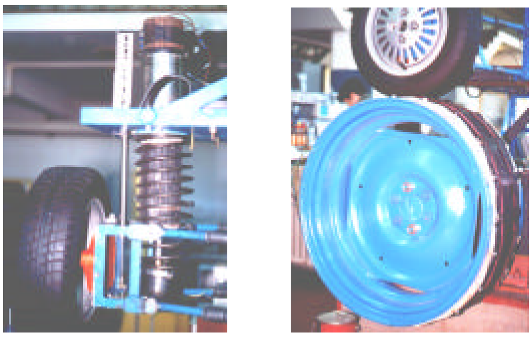

Figure 4: Pseudo-Levitating suspension system.

The pseudo-levitating suspension system workbench is composed by seven elements, which are:
1. The road perturbation generator, where the mechanical torque is provided by a *7,5kW* ac motor coupled to a *1/6* gearbox. An inverter is used to power the motor.
2. The unsprung mass that includes the wheel and the respective support. The unsprung mass weight (m_u) is *36,5kg*.
3. The spring, installed between the sprung and the unsprung masses in order to support the sprung mass. The spring was dimensioned to have a stiffness (K_S) of $16000Nm^{-1}$.
4. The electromagnetic linear actuator, which is a double phase, with permanent magnets, linear motor.

A power electronic current controller converter feeds the linear motor that has a maximum force of *980N*.
5. The sprung mass that represents a quarter of the vehicle body and passengers weight. It's value is *200kg*.
6. The sensors, which are two accelerometers installed in the sprung (Z_s'') and unsprung masses (Z_u'') and one *lvdt* between the two masses in order to measure the (Z_s-Z_u) value.
7. The suspension system is controlled using a DSP, *dSpace ACE1102* with the processor *TMS320C31/60MHz*. This controller receives the sensors signals, processes the information according to the specified control law and generates the actuator reference force.
The workbench has also the facility of enabling the implementation of several control laws, such as the passive and active suspensions control laws. The figure 5 shows the suspension system workbench diagram.

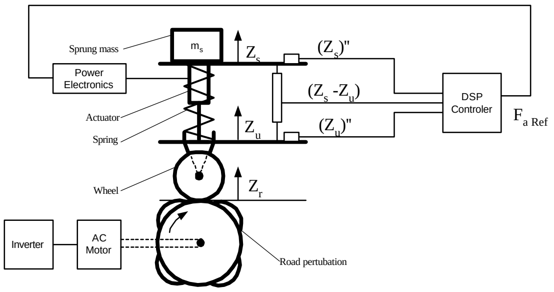

Figure 5: Suspension system workbench diagram.

3.2. Reference Result

As a first approach it was implemented in the suspension system a passive control law. The passive control law results stays as an improvement reference for the further next control laws that can be implemented in the suspension system workbench.
In order to obtain a passive suspension response, [3], in the suspension system workbench we need an actuator control law as (15).

$$F_a = -B_s\left(\dot{Z}_s - \dot{Z}_u\right) \quad (18)$$

Where B_S is the damper coefficient. With this control law the electromagnetic actuator will be working as damper with adjustable coefficient B_S.

IV. SIMULATION AND EXPERIMENTAL RESULTS

The workbench parameters values were experimentally obtained, and were used in the numerical simulations. The table 1 presents the parameters values.

m_s	K_s	m_u	K_t	$F_{a\ max}$
200 Kg	16000 Nm^{-1}	36,5 Kg	130000 Nm^{-1}	1000 N

Table 1: Workbench parameters values.

The simulations and experimental results were obtained for a road oscillation frequency equal to *Fp = 2Hz.*

4.1. Actuator frequency response

The figures 6 and 7 show the actuator response under different reference forces. *Fa* represents the reference force and *(Zs-Zu)* the differential position between the sprung and the unsprung mass.

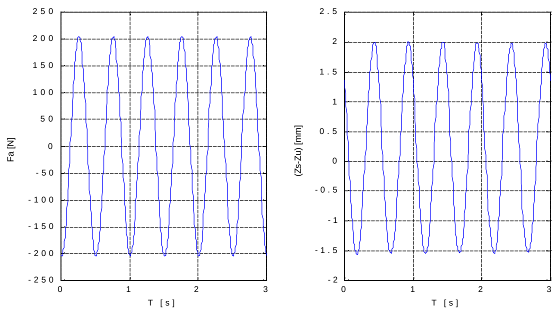

Figure 6: Experimental actuator frequency response, Freq = 2Hz

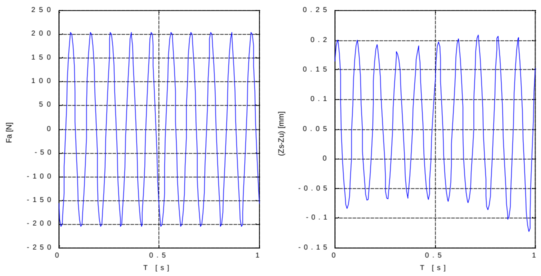

Figure 7: Experimental actuator frequency response, Freq = 10Hz

4.2. Passive control

The figures 8 and 9 present the sprung mass acceleration response to a step in the damping coefficient: *Bs=2400[N/m/s]* to *Bs=0[N/m/s]*, with a frequency of 2Hz. It is possible to observe the acceleration increase in the lack of the damping coefficient.

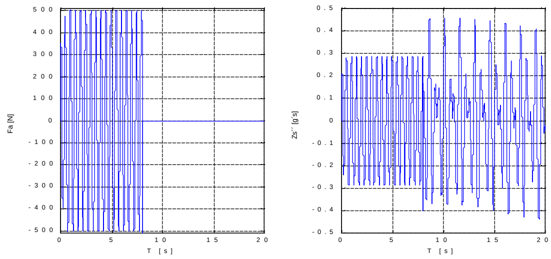

Figure 8: Simulation passive control response to a step in the damping coefficient.

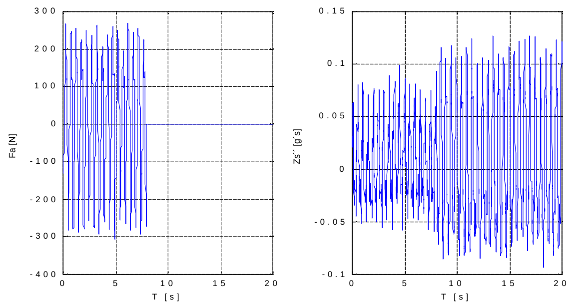

Figure 9: Experimental passive control response to a step in the damping coefficient.

4.3. Pseudo-Levitating control

The figure 10 presents the pseudo-levitating control simulation that shows a total sprung mass acceleration compensation for low frequency.

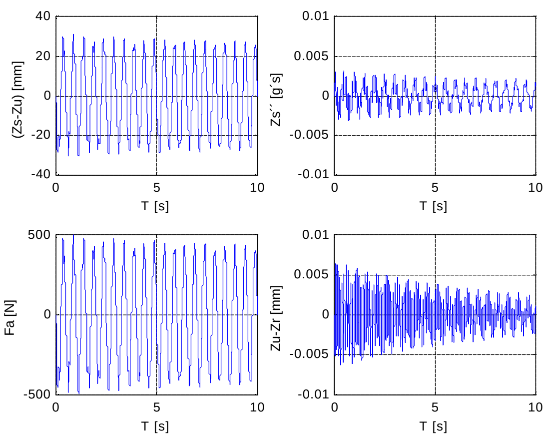

Figure 10: Pseudo-levitating control simulation, Fp=2Hz.

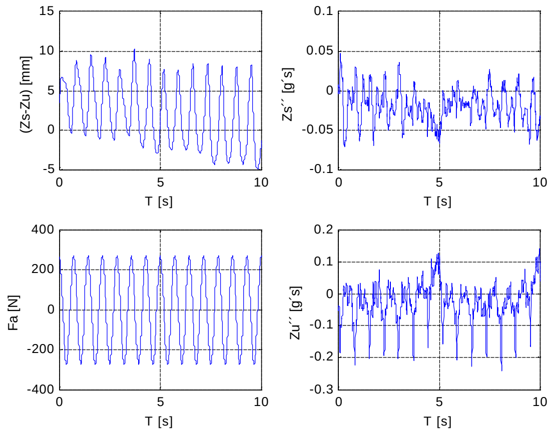

Figure 11: Experimental pseudo-levitating control. Fp=1,4Hz.

V. CONCLUSIONS

In this paper was presented and analysed a pseudo-levitating suspension system, which was implemented in a workbench. This suspension system uses an electromagnetic actuator and a spring between the sprung and the unsprung masses. The spring supports the body weight and the electromagnetic actuator is controlled in order to compensate the spring deformation force.
In the system there are two critical frequencies, the first one is associated with sprung mass and the other with the unsprung mass. The simulations and the frequency response diagram shows that for low frequencies the pseudo-levitating control system can bring more benefits in the passenger's comfort when comparing with other suspension systems solutions.

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