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MODELLING, SIMULATION AND CONTROL FOR A SKYHOOK SUSPENSION

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Abstract

This project aims to give a general overview of the world suspensions. In particular, it is shown how the modern control systems allow to the driver high results in terms of performance and comfort.

Are first introduced the suspension of the passive type by analyzing the individual components and the various dynamic models which constitute a generic vehicle starting from the quarter car model up to the half model car. It also introduced the concept of Skyhook from which derives the homonymous control system, which will be implemented to improve driving conditions posed particular conditions of the road profile.

The whole is developed in simulation environment with the aid of the MATLAB software that allows to put in evidence the improvements which are that a system of control type Skyhook offers the driver.

Keywords: Modeling; Simulation; Suspension; Control; Vehicle

Modelagem, simulação e controle de suspensão de tipo Skyhook

Resumo

Este projeto tem como objetivo dar uma visão geral do mundo suspensões. Em particular, é mostrado como os modernos sistemas de controle permitem alta perfomance dando ao motorista alto desempenho em termos de desempenho e conforto.

São introduzidos primeiro a suspensão do tipo passivo através da análise dos componentes individuais e das diferentes modelos dinâmicos, que constituem um veículo genérico a partir do modelo de quarto de carro até o modelo de metade de carro. Também é introduzido o conceito de Skyhook da qual deriva o sistema de controle homônimo que será implementado para melhorar as condições de condução colocadas condições específicas do perfil da estrada.

O conjunto é desenvolvido em ambiente de simulação, com auxílio do software Matlab que permite realçar as melhorias, que são de que um sistema de controlo do tipo Skyhook oferece ao condutor.

Palavras-chave: Modelagem, Simulaçao Suspensões; Controle; Sky-Hook; Veiculo

Summary

1. Introduction

In the context of automotive suspensions play a crucial role. The technological innovations have brought significant improvements in recent years allowing high performance from the point of view of comfort and from the point of view of performance. Through sensors and advanced control systems, nowadays the major car manufacturers introduce suspension systems that provide the driver with a perfect driving position. From the technological point of view in recent years what is done is to try to get the same results in terms of performance with the least cost which depends in part on the technology and materials used, as well as by the types of sensors that are mounted. In this thesis will be introduced in a first phase, the passive suspension and after that will be introduced dynamic models on which will be assumed control systems for active suspension with particular interest in the Skyhook suspension type.

a) Objective of thesis

The main purpose of this thesis is to give a general overview of the world of the suspension. Initially passive suspension are introduced and analyzed the components that belong to it. Following are introduced dynamic models that make up a vehicle and are made of the simulations in order to understand the behavior. Following are introduced electronically controlled suspension with particular interest to the active suspension that allow you to vary in a manner virtually instantaneous, the damping coefficient. In phase control is implemented the control type Skyhook going to see in detail the actual benefits that this type of control can offer.

b) Organization of chapters

The thesis is divided into chapters listed below:

- **Chapter 2:** It presents a brief description of the characteristic of the suspension and the components that characterize it. Then are introduce the dynamic models that characterize a vehicle starting from the quarter car model up to the half car model. More of this are describes the characteristics of the Skyhook suspension from theoretical point of view.

- **Chapter 3:** They made a series of simulations, carried out in MATLAB environment, to understand the behavior of the vehicle. Initially, an analysis is made in frequency through the Bode diagrams, and thereafter are placed under certain conditions in which the vehicle could find himself in and are analyzed in the step responses and the response to entry a sinusoid of constant amplitude at different frequencies. Are highlighted improvements to the system in case it increases the value of the damping coefficient with respect to the nominal value in particular in the case of the models Skyhook.

- **Chapter 4:** Are introduced electronically controlled suspension. Depending on the range of operating frequencies are divided in order of increasing frequency of type load leveling suspension, adaptive, semi-active and active. In the case of semi-active suspension, are also made considerations on the technological aspects, in particular the sensors and the architecture of three-tier system.

- **Chapter 5:** Is made the control for a type Skyhook suspension semi active on the quarter-vehicle models and is compared with the passive suspension, through various simulations.

- **Chapter 6:** Last chapter summarizing remarks and comments on the work done with the results.

2. The Suspension

The suspension systems represent key points in any means of transport, of any type. Traditionally they are constituted by a pair spring-damper.

During the running of the vehicle arising impact of forces, due to the irregularity of the road bottom, that in the absence of suspension are transmitted to the vehicle body. If the wheels and the chassis there are elements capable of storing energy, only a portion of the vertical forces is transmitted to the vehicle.

The suspension system allows the relative vertical motion between the wheels and chassis in order to compensate for irregularities in the road surface and filter and absorb the motion of the suspended body. In practice the suspension behaves like a real filter: cuts the high-frequency components and amplifies those found in a narrow band around the resonance condition.

They do I deal with the motions of vertical displacement (heave), roll and pitch, and also has influence on the lateral grip. In theory, the ideal suspension would require excursion infinite thing that is certainly impossible and infinite speed, taking into account that until now the full-active are those with the best performance. In reality the suspensions let pass the disturbances of low frequency and filter the disturbances of high and therefore the line of chassis undergoes the action of a low pass filter.

To understand how the suspensions affect the dynamics of the vehicle, will analyze the quarter-car models of various types up to the half car model that we are going to use later in the control phase. The first is to understand how to intervene to alleviate the vertical accelerations of order (comfort) and limit variations and dynamics of the tire road contact force (handing / performance) while preserving the maximum travel of the suspension. In the model of the middle vehicle, which studies the coordination of the front and rear, are examined in detail the measures necessary to reduce the roll motion. The suspension that we analyze is typical composed by a spring and a damper.
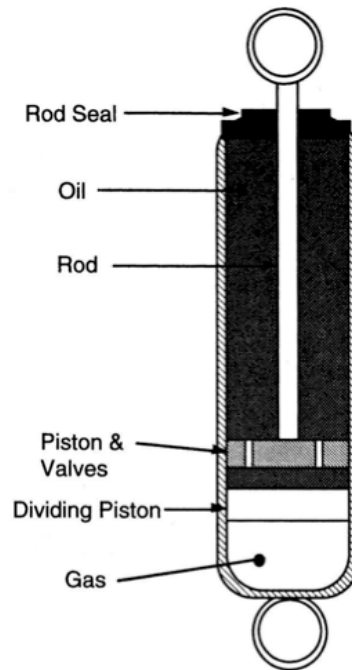
Figure 1 The Dumper

The damper develops a force proportional and opposite to the velocity of elongation according to the known law:

$$F = -c\frac{dx}{dt} \tag{2.1}$$

It is formed by a piston with passages where flows the oil, which moves between the upper chamber and the lower one. The force is developed by the friction of the fluid that passes over the orifices, and is dissipated as heat. The sealing member at the top, has the task to not release the fluid. Moving very slowly a damper we hear only static friction forces and thus essentially zero, keeping in mind that it does not support static load. If widen the passage holes diminish the damping coefficient, vice versa limiting them increase its value. The damper typically also includes a part loaded with pressurized gas that generates an elastic component that is not modeled as much stiffer than the effective spring which is in parallel with it. The use of the gas spring serves only for shock absorbers from competition where, especially in motorcycle, generates a filtering effect important for the very high frequencies. If there was this element of compensation would have excess or uplift pressure for when the damper is all stretched or compressed. We can also say that the characteristic, ideally linear, actually present a different slope between the compression with less damping than that of the extension. In addition I can decide independently how to move compression and rebound at high and low speeds, and in practice it is as if I had 4 degrees of freedom that allow me to change the form of characteristic x-F as I prefer to generate a desired curve. We can therefore conclude that the shock is a refined element with high capacity for regularization.

**Air-suspension level-control system
(partially loaded system)**
1 Air connection, 2 Steel spring,
3 Auxiliary air-spring element,
4 Gas volume, 5 Shock absorber.
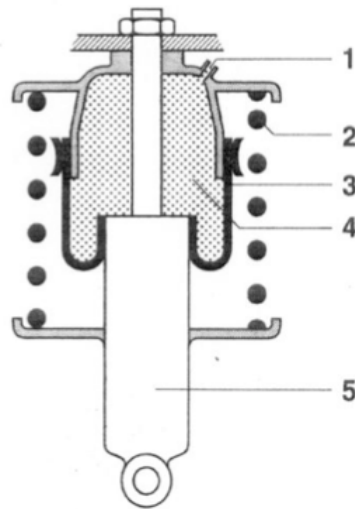
**Hydropneumatic level-control system
(partially loaded system)**
1 Fluid supply, 2 Steel spring, 3 Accumulator,
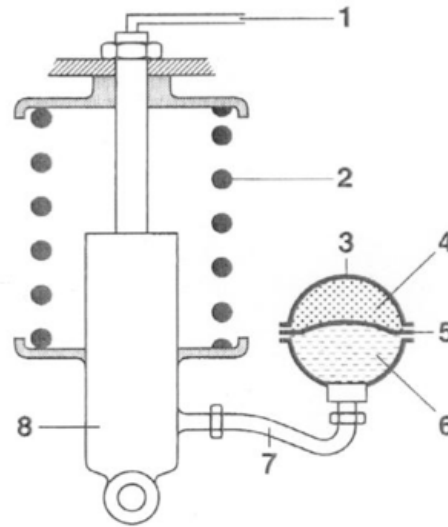4 Gas volume, 5 Rubber diaphragm, 6 Fluid,
7 Hose, 8 Shock absorber.

Figure 2 Gas and hydro-pneumatic springs

The spring instead develops a force proportional and opposite to the elongation:

$$F = -Kx \qquad (2.2)$$

It supports the static load and draws the wheel to the rest condition. There are various types, although the most common is the steel helical spring, usually coaxial to the damper. In alternative to the coil springs are pneumatic springs (gas) and hydro-pneumatic compress and return the force with the configuration of a compressible fluid. The advantages of gas springs are the disadvantages of hydro-pneumatic. In fact, the air does not require a dedicated hydraulic system and allow to clearly separate the part from the elastic damping. The damper is placed coaxially as in coil springs so as to make them decoupled.

The hydro-pneumatic present a hydraulic cylinder which is an element that carries the force towards the accumulator, and then, in essence, we can consider it as a gas spring delocalized. Even those hydro-pneumatic can be controlled through the valves, which unlike the gas springs which act on the gas in the chamber by increasing the pressure, acting on the fluid flow that interacts with the room, transferred or not. Since the spring a bulky object, it can often be useful to place part of it in another area. The part of the damping is carried out by the narrowing of the tube that generates more friction and therefore dissipations significant. The hydro-pneumatic spring is initially more used even if today is quite into disuse as it has the decoupling between the spring and damper and needs a system of hydraulic circuit. Precisely for this reason is often used in tractors which provide in each case a hydraulic system. The gas spring, in addition, has no problems of sealing organs as has almost no static friction.

a) Quarter car model

The first model that we consider about a quarter of the car in case it is a four-wheeled vehicle, or half in case it is analyzed a motorcycle. The model predicts a suspended mass M (sprung) connected through the damper to the mass m of the wheel (unsprung). Finally we have a spring kt that models the elasticity of the tire.
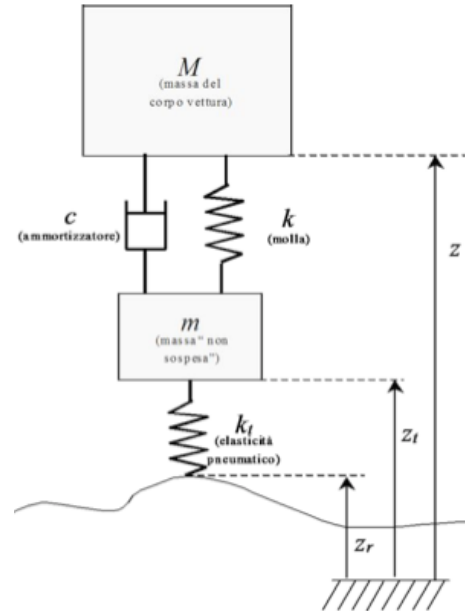

Figure 3 Quarter car model

Although in the case of semi-active suspension, the damper will certainly be time-variant, in this case we can consider constant to represent the simplified system as we turn to study the equilibrium. In any case, the damping will not to affect the balance as not holding the static load. Analysis of the system shown in the figure can be derived differential equations that determine the system:

$$\begin{cases} M\ddot{z}(t) & = -c(\dot{z}(t) - \dot{z}_t(t)) - k(z(t) - z_t(t)) - Mg \\ m\dot{z}_t(t) & = c(\dot{z}(t) - \dot{z}_t(t)) + k(z(t) - z_t(t)) - K_t(z_t - z_r) - mg \end{cases} \quad (2.3)$$

From the model above is just possible find its representation in matrix form:

$$\begin{bmatrix} \dot{z} \\ \ddot{z} \\ \dot{z}_t \\ \ddot{z}_t \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -\frac{k}{M} & -\frac{c}{M} & \frac{k}{M} & \frac{c}{M} \\ 0 & 0 & 0 & 1 \\ \frac{k}{m} & \frac{c}{m} & -\frac{k+k_t}{m} & -\frac{c}{m} \end{bmatrix} \begin{bmatrix} z \\ \dot{z} \\ z_t \\ \dot{z}_t \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ \frac{k_t}{m} \end{bmatrix} z_r \quad (2.4)$$

$$\begin{bmatrix} z \\ z_t \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \end{bmatrix} \begin{bmatrix} z \\ \dot{z} \\ z_t \\ \dot{z}_t \end{bmatrix} \tag{2.5}$$

From these matrices will be useful to find the transfer functions we're going to use to understand the trend as a function of frequency. In our case we only concerned that between the road profile and the position of the chassis that will call Fz (s) and will be equal to:

$$F_z(s) = \frac{k_t(sc + k)}{s^4 Mm + s^3 c(m + M) + s^2 (Mk + mk + Mk_t) + sck_t + kk_t} \tag{2.6}$$

From a unity gain at zero frequency, we find a first resonance frequency, called "body resonance", between 1-2 Hz that typically perceive it and it is vital for all cars and motorcycles. After it drops the curve with slope -1 up to the "wheel resonance" frequency, between 10-15 Hz, fundamental for the road holding of the vehicle, although we have no perception. The complex conjugate poles cause both resonances. After the second resonance, the curve drops with slope -3.
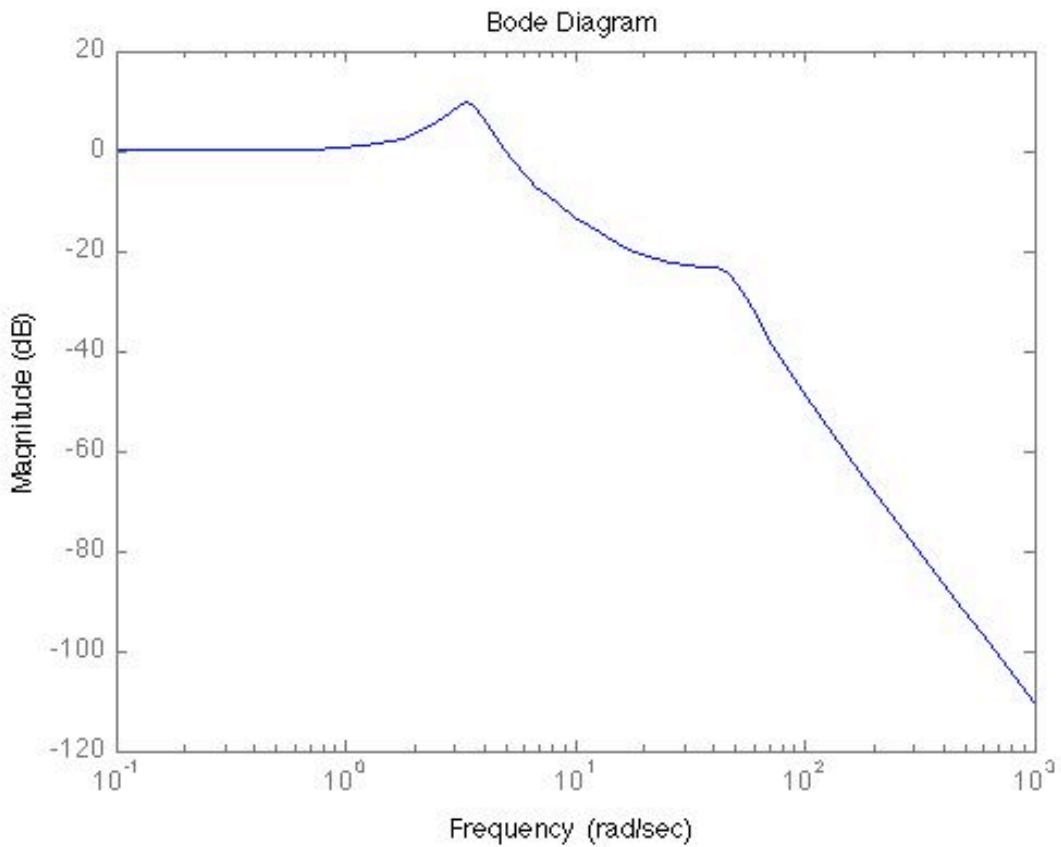
Figure 4 Bode diagram of Fz(s)

b) Half car model
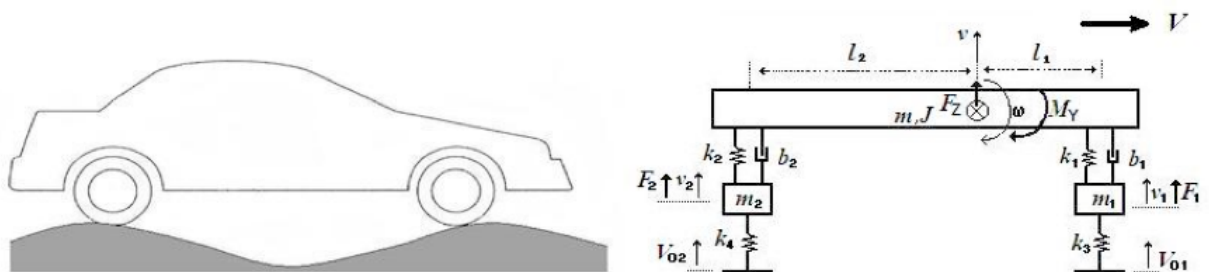
Figure 5 Half car model

Evolution of the quarter car model is half car where it is analyzed only half part of a four-wheeled vehicle that is the same of the totality of a motorcycle. After analyzing the quarter-car, see what comes from the union of a front and rear wheel assembly. In this case we have masses 3 and 4 degrees of freedom, with 4 second-order equations. In addition to the translation of the three masses should be

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taken into account for the rotation of the body that introduces me to a movement of pitch. The state model of this system in the state variables Lagrangian, relating to the inputs with its degrees of freedom and its derivatives, is given by:

$$
\begin{bmatrix} \dot{z} \\ \dot{\theta} \\ \dot{z}_{t1} \\ \dot{z}_{t2} \\ \ddot{z} \\ \ddot{\theta} \\ \ddot{z}_{t1} \\ \ddot{z}_{t2} \end{bmatrix} =
\begin{bmatrix}
0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 \\
-\frac{k_1+k_2}{M} & -\frac{l_2 k_2 - l_1 k_1}{M} & \frac{k_1}{M} & \frac{k_2}{M} & -\frac{c_1+c_2}{M} & -\frac{l_2 c_2 - l_1 c_2}{M} & \frac{c_1}{M} & \frac{c_2}{M} \\
-\frac{l_2 k_2 - l_1 k_1}{J} & -\frac{l_2^2 k_2 + l_1^2 k_1}{J} & \frac{l_1 k_1}{J} & \frac{l_2 k_2}{J} & -\frac{l_2 c_2 - l_1 c_2}{J} & -\frac{l_2^2 c_2 + l_1^2 c_1}{J} & -\frac{l_1 c_1}{J} & \frac{l_2 c_2}{J} \\
\frac{k_1}{m_1} & -\frac{l_1 k_1}{m_1} & -\frac{k_1+k_{t1}}{m_1} & 0 & \frac{c_1}{m_1} & -\frac{l_1 c_1}{m_1} & -\frac{c_1}{m_1} & 0 \\
\frac{k_2}{m_2} & \frac{l_2 k_2}{m_2} & 0 & -\frac{k_2+k_{t2}}{m_2} & \frac{c2}{m_2} & \frac{l_2 c_2}{m_2} & 0 & -\frac{c_2}{m_2}
\end{bmatrix}
\begin{bmatrix} z \\ \theta \\ z_{t1} \\ z_{t2} \\ \dot{z} \\ \dot{\theta} \\ \dot{z}_{t1} \\ \dot{z}_{t2} \end{bmatrix} +
$$

$$
\begin{bmatrix}
0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 \\
\frac{1}{M} & 0 & 0 & 0 & 0 & 0 \\
0 & \frac{1}{J} & 0 & 0 & 0 & 0 \\
0 & 0 & \frac{1}{m_1} & 0 & \frac{k_1}{m_1} & 0 \\
0 & 0 & 0 & \frac{1}{m_2} & 0 & \frac{k_2}{m_2}
\end{bmatrix}
\begin{bmatrix} F_z \\ M_Y \\ F_1 \\ F_2 \\ Z_{01} \\ Z_{02} \end{bmatrix}
$$

c) Simplified Model

Only a preliminary analysis of the half car model is easily seen, as the system remains considerably complicate adding degrees of freedom. Before making considerations on the latest pattern introduced for the type of study we want to do, it is reasonable to make the simplifications by introducing the simplified model. This type of model is useful as it simplifies all while well approximating the low-frequency and the first resonance. To simplify is only possible to consider the mass of the body without taking into account the mass of the wheel and its relative elasticity as shown in the figure.
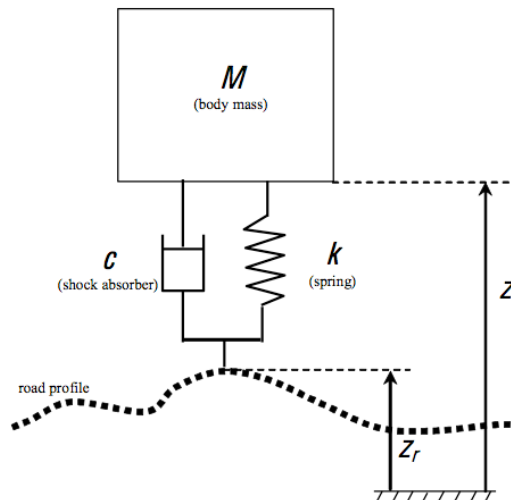

Figure 6 Simplified model

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What comes out is a first-order model with the necessary steps that generates a transfer function:

$$F_z(s) = \frac{s\frac{c}{M} + \frac{k}{M}}{s^2 + s\frac{c}{M} + \frac{k}{M}} \qquad (2.7)$$

Which is of the second order and then presents the resonance frequency and damping, respectively:

$$\omega_n = \sqrt{\frac{k}{M}} \qquad \xi = \frac{c}{2}\sqrt{\frac{1}{kM}} \qquad (2.8)$$

Compared to the quarter car classic, the resonance of the wheel is no longer modeled and disappears the filtering effect of the wheel.

By switching to half car instead we note that in this case there will remain only inputs My and Fz while the degrees of liberty will remain only the first two of the original model. Let's see in detail the representation of state:

$$\begin{bmatrix} \dot{z} \\ \dot{\theta} \\ \ddot{z} \\ \ddot{\theta} \end{bmatrix} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -\frac{k_1+k_2}{M} & -\frac{l_2 k_2 - l_1 k_1}{M} & -\frac{c_1+c_2}{M} & -\frac{l_2 c_2 - l_1 c_2}{M} \\ -\frac{l_2 k_2 - l_1 k_1}{J} & -\frac{l_2^2 k_2 + l_1^2 k_1}{J} & -\frac{l_2 c_2 - l_1 c_2}{J} & -\frac{l_2^2 c_2 + l_1^2 c_1}{J} \end{bmatrix} \begin{bmatrix} z \\ \theta \\ \dot{z} \\ \dot{\theta} \end{bmatrix} + \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ \frac{1}{M} & 0 \\ 0 & \frac{1}{J} \end{bmatrix} \begin{bmatrix} F_z \\ M_Y \end{bmatrix} \qquad (2.9)$$
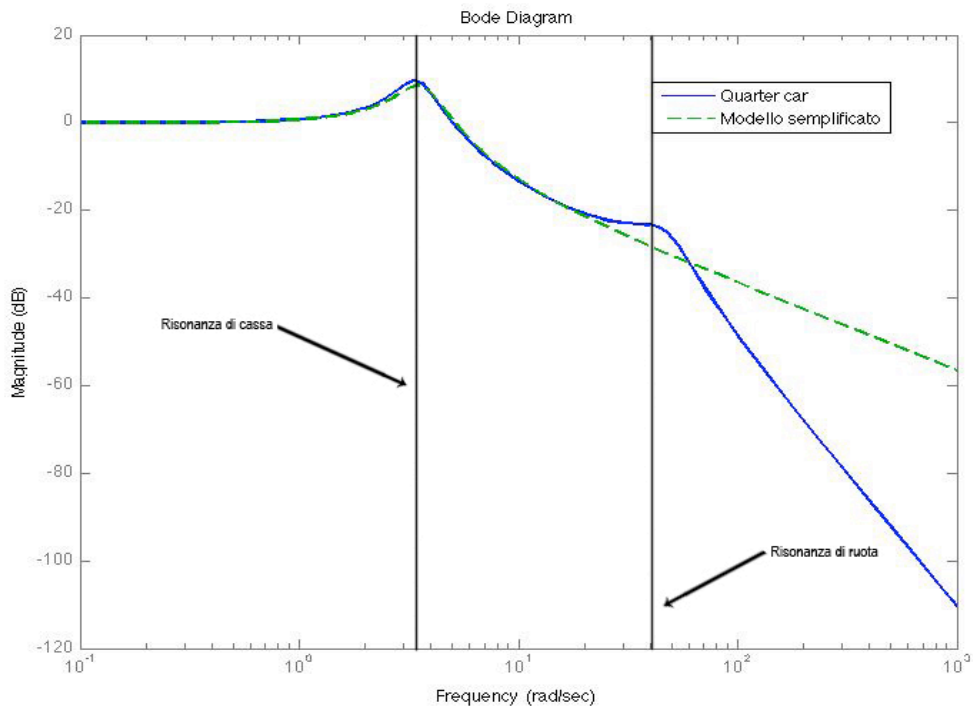

Figure 7 Bode diagram quarter car model and simplified

d) Skyhook Suspension

The concept of control "Skyhook" includes the presence of an ideal damper connected between the sprung mass and an inertial reference.

The active control of the suspension called Skyhook is obtained by working a linear viscous damper to the absolute velocity of the mass to which it is connected instead to the relative speed. In this way the mass is conceptually "docked" to the sky. This phenomenon gives its name to this particular control.

As opposed to the conventional damping, through this technique, the module of the transfer function between the position of the wheel and that of the suspended mass, with increasing damping, decreases for all frequencies and not only for low ones. In this way the people inside the car will not be disturbed by stress resulting for example from navigating a particularly bumpy road. The limit of this technology is given by the performance of the electromagnetic actuator that will never be sufficient to completely cancel the transmissibility. Moreover, having to employ an actuator for each of the four suspensions, makes this control proves quite expensive.
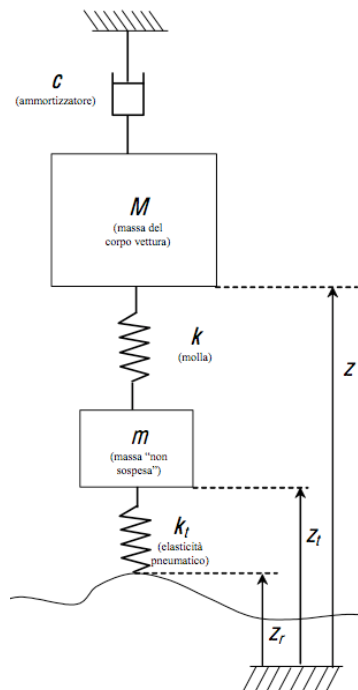
Figure 8 Skyhook model

SH brings general benefits to low frequencies and trajectory tracking, but the absence of damping wheel side inevitably entails significant resonances.

Let's see how to change the basic model analyzed first quarter of the car and that it goes to affect half the simplified model car doing the same simplifying assumptions made earlier.

$$\begin{cases} M\ddot{z}(t) & = -c\dot{z}(t) - k(z(t) - z_t(t)) - Mg \\ m\dot{z}_t(t) & = k(z(t) - z_t(t)) - K_t(z_t - z_r) - mg \end{cases} \quad (2.10)$$

The model in form of the states:

$$\begin{bmatrix} \dot{z} \\ \ddot{z} \\ \dot{z}_t \\ \ddot{z}_t \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -\frac{k}{M} & -\frac{c}{M} & \frac{k}{M} & 0 \\ 0 & 0 & 0 & 1 \\ \frac{k}{m} & 0 & -\frac{k+k_t}{m} & 0 \end{bmatrix} \begin{bmatrix} z \\ \dot{z} \\ z_t \\ \dot{z}_t \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ \frac{k_t}{m} \end{bmatrix} z_r \qquad (2.11)$$

$$\begin{bmatrix} z \\ z_t \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \end{bmatrix} \begin{bmatrix} z \\ \dot{z} \\ z_t \\ \dot{z}_t \end{bmatrix} \qquad (2.12)$$

From here we can make the same simplifying assumptions that we had done previously and derive the transfer function between our road profile and chassis previously introduced:

$$F_z(s) = \frac{k}{Ms^2 + sc + k} \qquad (2.13)$$

Note how the transfer function differs from that of classical simplified system only where the numerator disappears the zero. At high frequencies will be reasonable to consider:

$$\frac{F_z(s)_{skyhook}}{F_z(s)_{conventional}} \approx \frac{k}{sc} \qquad (2.14)$$

It may, at this point, lead us to the half car model simplified where the two unsprung masses and the two resistors wheel are not considered. The equivalent model in the form of state is the following:

$$\begin{bmatrix} \dot{z} \\ \dot{\theta} \\ \ddot{z} \\ \ddot{\theta} \end{bmatrix} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -\frac{k_1+k_2}{M} & -\frac{l_2k_2-l_1k_1}{M} & \frac{c_1+c_2}{M} & \frac{l_2c_2-l_1c_2}{M} \\ -\frac{l_2k_2-l_1k_1}{J} & -\frac{l_2^2k_2+l_1^2k_1}{J} & \frac{l_2c_2-l_1c_2}{J} & \frac{l_2^2c_2+l_1^2c_1}{J} \end{bmatrix} \begin{bmatrix} z \\ \theta \\ \dot{z} \\ \dot{\theta} \end{bmatrix} + \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ \frac{1}{M} & 0 \\ 0 & \frac{1}{J} \end{bmatrix} \begin{bmatrix} F_z \\ M_Y \end{bmatrix} \qquad (2.15)$$

3. The Simulation

Having described the quarter car and half car models in their variants we are going to get into simulation environment to see the behavior of these models under certain conditions. All this will be useful to be able to draw some qualitative considerations on the performance of the vehicle, depending on certain conditions of the asphalt, and understand for how to intervene in the control phase to attenuate the dynamic effects. All simulations were carried out with the help of MATLAB which we'll discuss some of the graphics in this chapter.

Considering the generic irregularities of the road, have been taken into account situations in which the vehicle is located in front of a sharp change of the road surface and stretches where the asphalt road presents irregularities continuous. In the first case, the simulation will be modeled with a step, while for continuous irregularities will be modeled with a sinusoid that will be of different frequencies depending on the road irregularities. The whole was made considering the first quarter car model and then the model half car where it is also analyzed the trend of the angle of pitch that is introduced by adding a degree of freedom.

a) Quarter car model simulation

Resuming the quarter car model where we had seen various possibilities with the simplified model and the model with Skyhook. In the case of the simplified model was decided not model the second resonance frequency as it was of greater interest that were reproduced the trend around the first resonance frequency for the benefit of a significant simplification of the model. In the case of model Skyhook is, however, virtually placed the damper "hooked" to the sky as a result of the type of control, which we have already spoken extensively and of which we will make use in control step. To be clear about the behavior of these possible combinations let's consider the Bode graph of the various models.
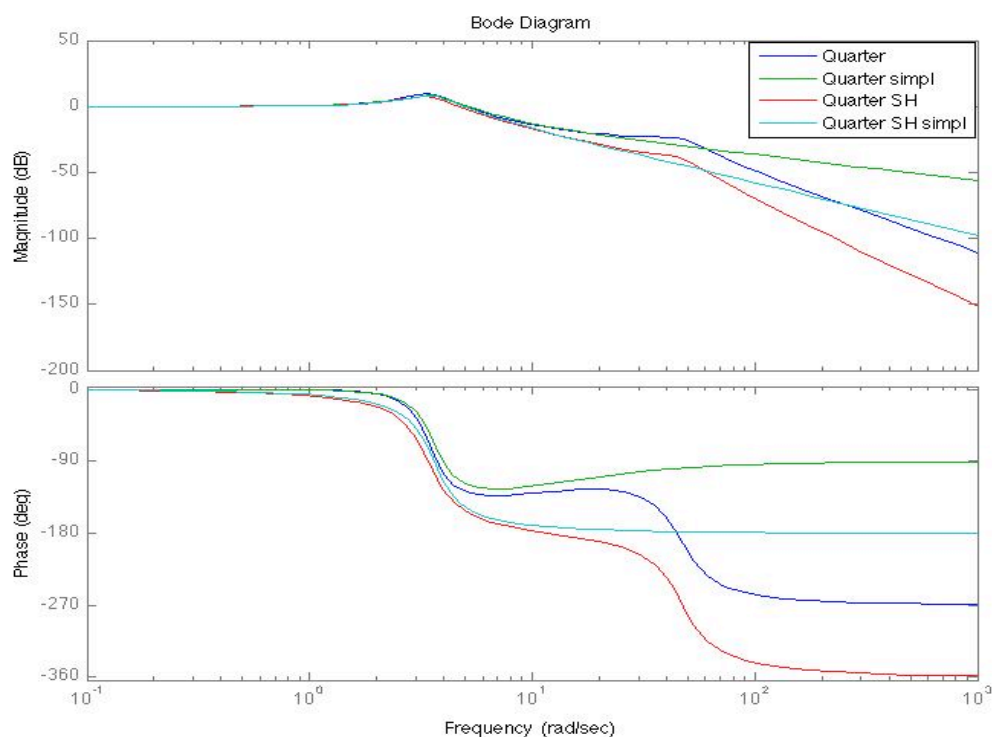

Figure 9 Bode diagrams for quarter car models

Name	Symbol	Value
Sprung mass	M	$800Kg$
Unprung mass	m	$50Kg$
Damper	c	$1200Ns/m$
Spring	k	$10500N/m$
Wheel spring	k_t	$100000N/m$
Amplitude	A	$0.02m$
1st frequency	f_1	$0.5Hz$
2nd frequency	f_2	$3Hz$
3rd frequency	f_3	$10Hz$

Figure 10 Table of values

The models taken into account as can be seen from the figure are quarter car, quarter car simplified, quarter car Skyhook and quarter car Skyhook simplified. We note that, in the diagram of the module, the simplified models are an excellent reproduction of the classic models to the first resonance frequency. Then disappears the effect of the wheel resonance and we have also confirmed in the graph of the phase where the gradient after being dropped to 180 ° C in the case resonance continues without other alterations differently from simplified models which, instead, subjected to a second attenuation of other 180 ° C in conjunction with the resonance of the wheel. Among the models with and without Skyhook instead we can observe how the trend is rather similar with a slight higher attenuation for the model Skyhook that will bring benefits in terms of attenuation as will be shown below.
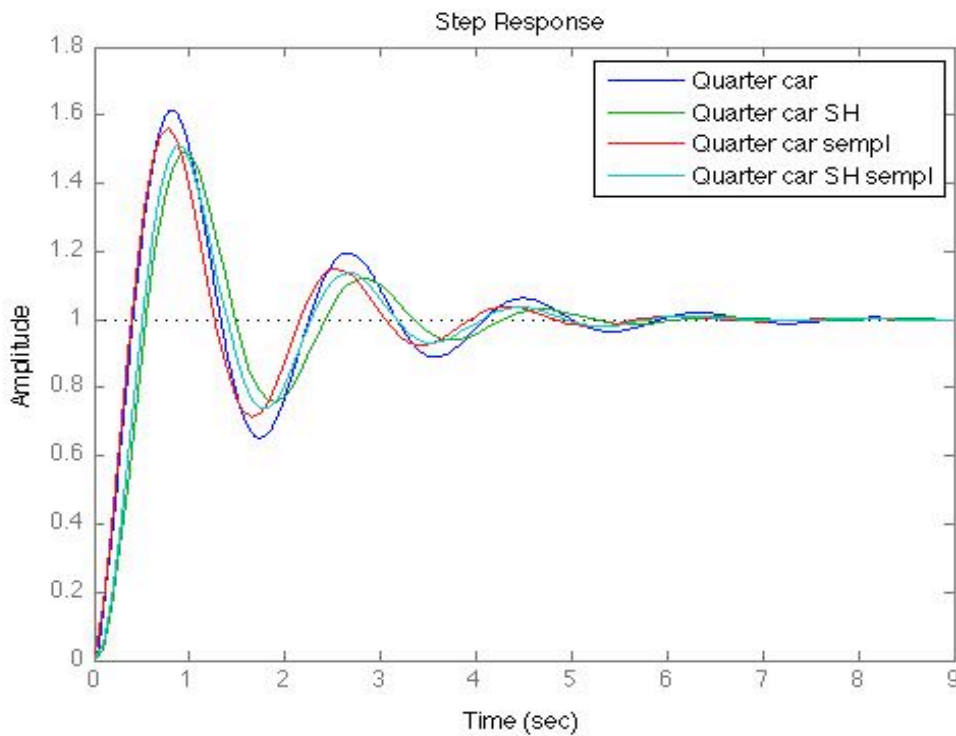

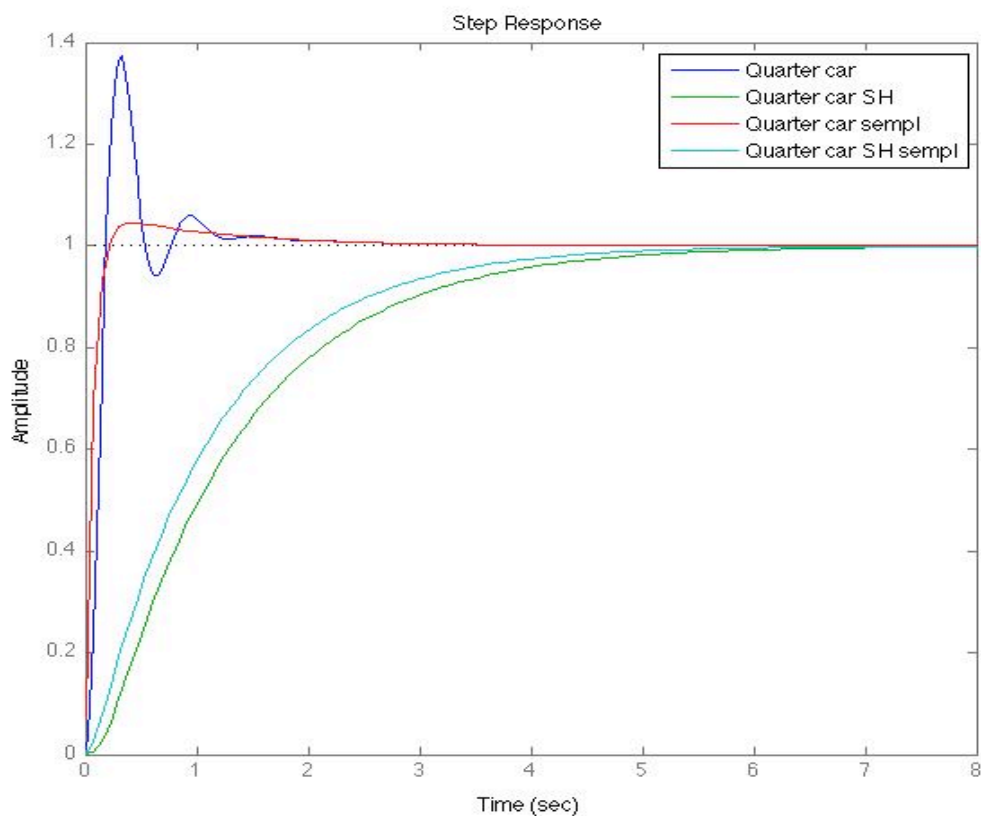**Figure 11 Step response for quarter car models with nominal dumper**

Figure 12 Step response for quarter car models with tenfold dumper

Let us at this point analysis of the response to a step input. In our case we have chosen a unit step but would not change anything if we had chosen a different value. The simulations were carried out for quarter car models and models simplified quarter car and the values considered are those shown in the table. The resulting behavior in the case of damper with real value is typical with an initial overshoot and progressive attenuation with sinusoidal pattern that fades around 7-8s.

In all four cases we note how the overshoot various between 40% and 60% which is a considerable value and that we will try to lower in control step going to operate on a damping value adjusted. Assuming to vary the damping coefficient, it is interesting to observe the effects if the value is multiplied by ten. We observe in the figure below as the overshoot suffer a considerable decrease in the case of the quarter car model and is rapidly attenuated. In this simplified case is instead a value of almost negligible overshoot and mitigating the effect disappears harmonious. Even more surprising the dynamic models Skyhook where we no longer have peak overshoot and the trajectory tracking is done with a curve of the second order. We also note that the two trends are quite similar with a slight advance of the simplified model compared to the classical one.

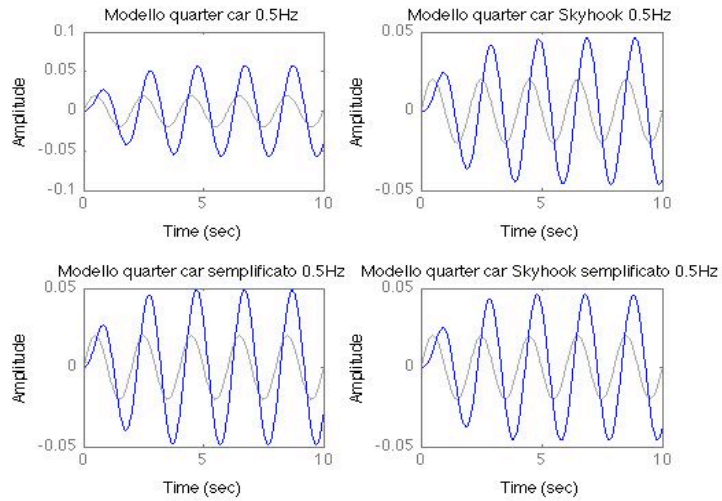**Figure 13 Quarter car simulations for nominal dumper at 0.5Hz**
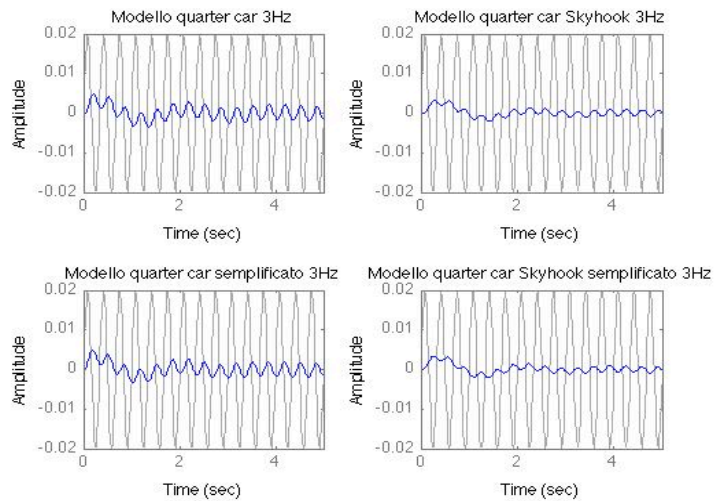

Figure 14 Quarter car simulations for nominal dumper at 3Hz
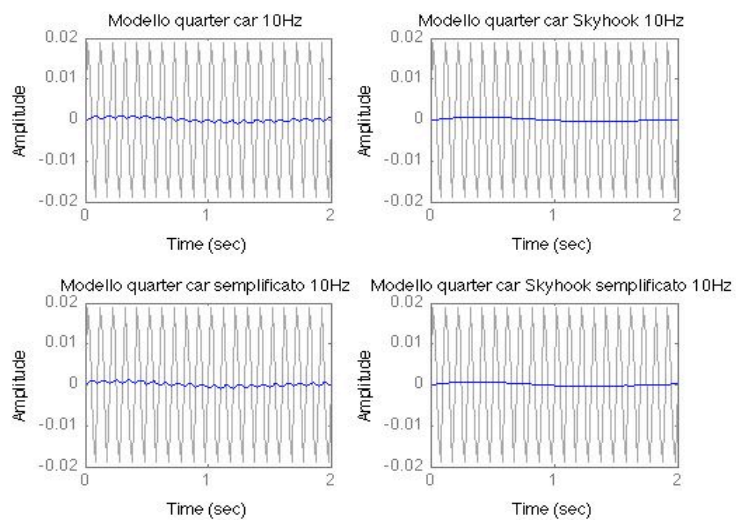

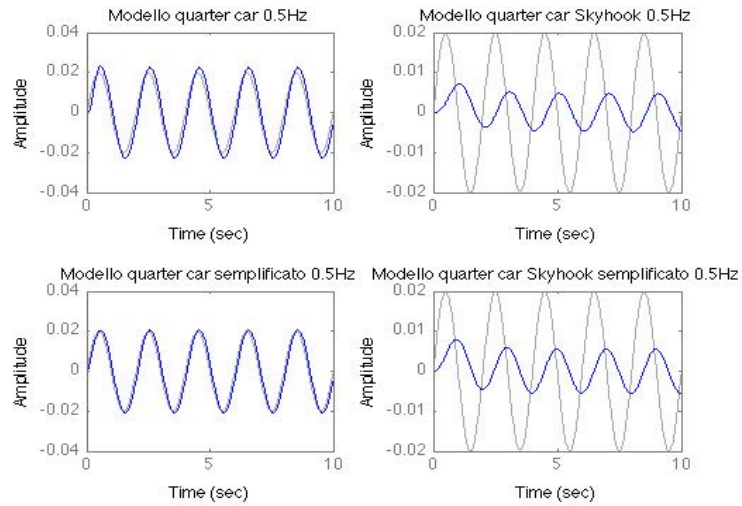**Figure 15 Quarter car simulations for nominal dumper at 10Hz**

15

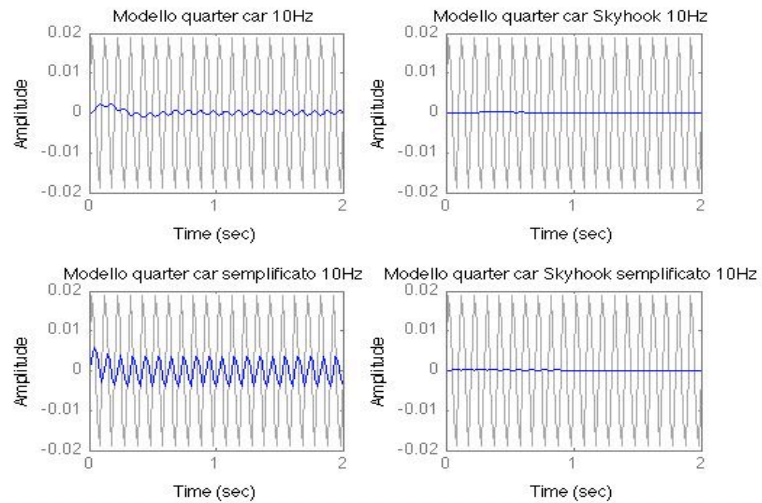**Figure 16 Quarter car simulations for tenfold dumper at 0.5Hz**

Figure 17 Quarter car simulations for tenfold dumper at 3Hz
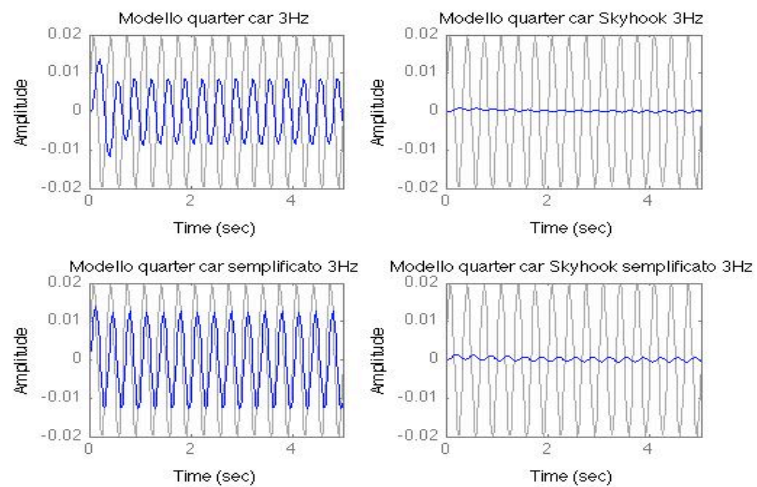

Figure 18 Quarter car simulations for tenfold dumper at 10Hz

16

Let us at this point analysis of the response to sinusoidal input. To get a match with the reality was chosen a fixed width of 2cm and simulated different conditions of frequency which in this case were selected by 0.5Hz, 3Hz and 10Hz. The reason for this decision was to be able to get an idea of what is happening around the frequency of case, something between the two resonant frequencies and what's at high frequency. From the analysis of the graphs with the nominal value of damping is evident as low frequency has greater accentuation of harmonic effects with amplitude responses in more that incoming and then with obvious vehicle vibration. With increasing frequency we note that the effects diminish, but are however obvious and produce vibrations which we want to avoid. We also note as in the case of the models Skyhook vibration is less significant although not produce surprising effects.

Changing also in this case the damping coefficient and also in this case by choosing a value 10 times greater than the nominal we see significant improvements. In the case of low frequency, we have the same wave incoming reproduced in output in the cases of the models without Skyhook which is already an improvement compared to models with nominal c where the output amplitude was greater than that of entry. However, at all frequencies of simulation, since most worth noting is how you have the obvious improvements between models using Skyhook than those who do not use it. In particular, we note that the effects of medium and high frequency harmonics are almost non-existent. It is in this direction that we orient ourselves to ensure dynamic effects in the least significant output possible.

4. Suspension With Electronic Control

Let us at this point to the electronically controlled suspension. It 'important first of all bring to the attention that the active suspension are a particular type among those with electronic control. To introduce the most of this new class is worth dwelling on is classified as the world of the suspension. Up to now have been introduced suspensions, which fall into the category of passive whose characteristics are time invariant. Recall how the characteristics spring-force and damper-strength are of the linear type, possibly with different inclination between positive and negative half-plane and with eventual changes of slope for the stroke end. The remaining types of electronically controlled suspensions are those that are classified under various levels. A first distinction within them can be made between those who work in the first and third quadrant and those which cover all four. What in practice the distinction lies in the fact that the second introduce energy into the system and thus have a range of forces certainly more extended. In the first category are instead part of the adaptive suspension and semi-active suspensions that have the possibility to vary over time the characteristic of damping and elasticity even if in practice one on which going to intervene is the characteristic of the damper.

Classification of suspension systems. Natural frequencies: f_B body and f_W wheel

System	System representation	Force range	Operation range	Max. energy demand	Improvements compared to passive system	
					Comfort	Safety
Passive			—	—	—	—
Slowly variable/adaptive			$< f_B$	ca. 50 W	15–20%	10–25%
Semi-active			$f_B - f_W$	ca. 50 W	20–30%	10–25%
Active partially loaded			$0 - f_B$	1–2 kW	> 30%	—
Active fully loaded			$0 - f_W$	1.5–7 kW	> 30%	25%

Figure 19 Classification of electronically controlled suspension

The adaptive varying more slowly the damping characteristic, compared to those that have semi-active evidently a bandwidth greater. The adaptive, in fact working under the frequency of body, while the semi-active working between the frequency of the body and that of the wheel. The remaining class of suspensions are those that introduce energy of the system: they can in turn be classified according to the frequency in which they work between load-leveling suspension, suspensions slow-active and active suspensions full-active. The load leveling have a bandwidth below 1Hz and can be considered active very slow, the slow-active suspensions have the characteristic of classic with the addition of an active actuator, electrohydraulic or electromechanical, while the full-active present the only actuator. Having an active device may enter the second and fourth quadrant introducing energy at the expense of a considerable power demand. The slow-active suspension pull within himself the first resonance frequency, while the full-active have a higher bandwidth to 20Hz. Currently suspension with high performance are the combination of the load-leveling and semi-active.
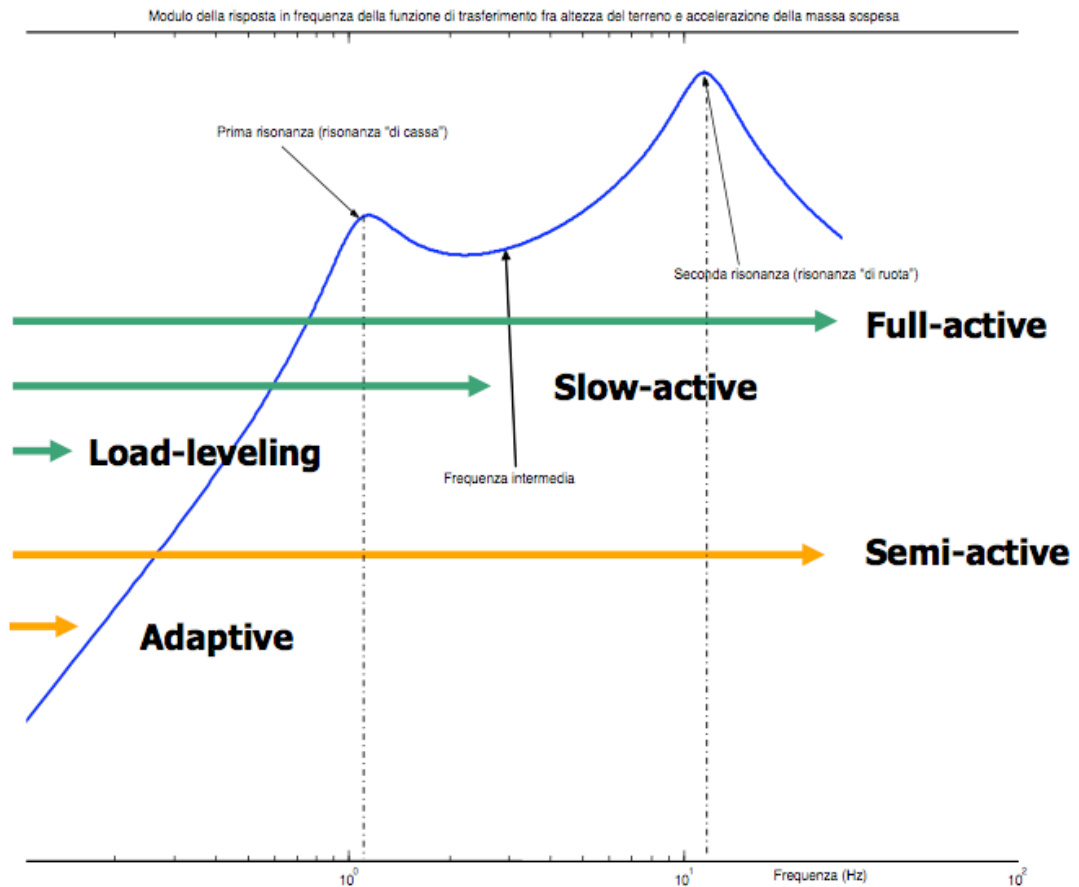
Figure 20 Bandwidth for electronically controlled suspension

a) Load-leveling Suspension

Their basic principle is to vary the static force developed very slowly, in the equilibrium position of the center or nominal stroke. Work at very low frequencies and in fact takes about ten seconds to level the vehicle in the desired position. They can be of two types: for vehicles designed at nominal loads, where the variation of it, is smoothed through a gaming leveling, or in off-road and sports cars are designed to vary in height in a continuous manner. In both cases have the advantage of avoiding the asymmetry of the suspension to ensure good performance both in expansion and compression. The technologies used are those of the hydro pneumatic springs, and especially those of the gas springs. To better understand how they work, we start from the analysis of the equation of static equilibrium:
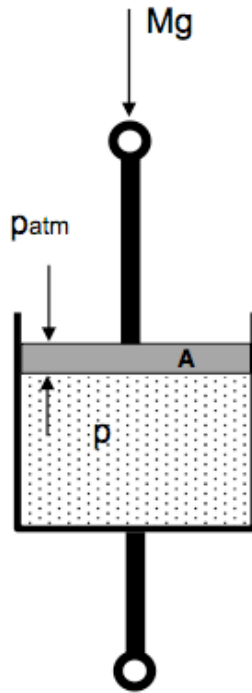
$$Mg = (p - p_{atm})A \qquad (4.1)$$

Figure 21 Gas spring

With A section of the gas spring and p pressure on it. What happens if the load increases is a lowering of the height from the ground that I can compensate by increasing the pressure value. This can happen in the case of gas springs in volume of gas constant, while in the hydro-pneumatic springs to constant mass of gas. In order to fully understand the operation of the gas spring is also necessary to understand its spring constant to vary the position of equilibrium. To get to it hypothesize that compression and extension are of type adiabatic or without heat transfer for which:

$$pV^\gamma = const. \tag{4.2}$$

From it we derive the total differential:

$$\partial p V^\gamma + p V^{\gamma-1}\partial V = 0 \Longrightarrow \partial p = -\frac{p\gamma\partial V}{V} \tag{4.3}$$

Given that

$$V = (z - z_t)A \Longrightarrow \partial V = A\partial(z - z_t) \tag{4.4}$$

So by combining the formulas and considering that the change in atmospheric pressure is null we find that:

$$\partial F = A\left(-\frac{p\gamma A\partial(z - z_t)}{V}\right) \Longrightarrow k = -\frac{\partial F}{\partial(z - z_t)} = \frac{p\gamma A^2}{V} \tag{4.5}$$

That is, by definition, the variation of stiffness of an elastic element. It 'important this relation to understand the link, which a gas spring, has with a series of parameters: it is linearly proportional to the pressure and in a quadratic manner with the section. Is instead inversely proportional with the volume which leaves us understand how to make soft springs should expand the spring length and can lead to problems of space. Interesting can also be a brief analysis of the dynamic effects of a system load leveling depending on whether it uses gas springs or hydro-pneumatic. In both cases, the spring pressure is increased about 10 times compared to the atmospheric pressure to which it is reasonable approximation:

$$p = \frac{Mg}{A} + p_{atm} \approx \frac{Mg}{A} \qquad (4.6)$$

In the case of gas springs at a constant volume, we have resonance frequency and damping:

$$\omega = \sqrt{\frac{k}{M}} = \sqrt{\frac{1}{M}\left(\frac{p\gamma A^2}{V}\right)} \approx \sqrt{\frac{1}{M}\frac{\gamma A^2}{V}\frac{Mg}{A}} = \sqrt{\frac{\gamma Ag}{V}} \qquad (4.7)$$

$$\xi = \frac{c}{2}\sqrt{\frac{1}{kM}} = \frac{c}{2M}\sqrt{\frac{M}{k}} = \frac{c}{2M\omega} \qquad (4.8)$$

Where ω is invariant under varying load if V is constant that is if the car is leveled, while the damping decreases with the load in inverse proportion. The first consideration is a considerable advantage for this type of suspensions using gas springs, while the fact that the damping worst with the load is negative and forces me to vary in time the damper to not miss results in dynamic performance. Regarding instead hydro pneumatic, with constant mass, we find pulsation and damping taking into account the relationship of the pulse Vp = Vn*pn:

$$\omega = \sqrt{\frac{1}{M}\frac{p\gamma A^2}{V}} = \sqrt{\frac{1}{M}p\gamma A^2\frac{p}{V_n p_n}} = \sqrt{\frac{1}{M}\gamma A^2\frac{1}{V_n p_n}\left(\frac{M^2 g^2}{A^2}\right)} = \sqrt{\gamma\frac{Mg^2}{V_n p_n}} \qquad (4.9)$$

$$\xi = \frac{c}{2M\omega} \qquad (4.10)$$

We note that the resonance frequency increases with the load and the damping decreases rapidly with the load in a more significant manner than the gas springs. Considering what has been said it is confirmed that the gas springs are better than those hydro-pneumatic.
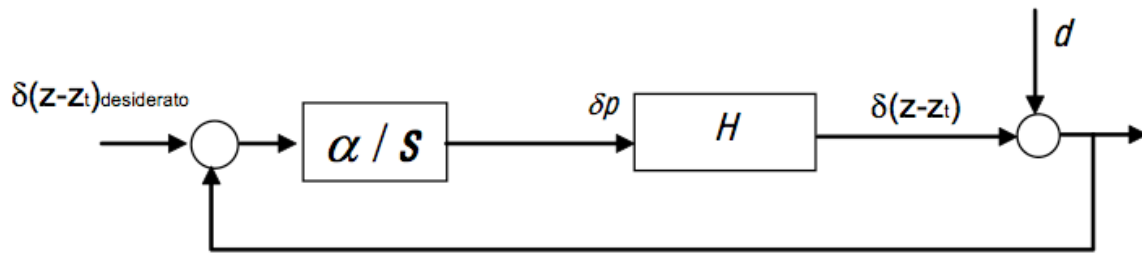
Figure 22 Control system of the suspension load-leveling

The control system for the load-leveling suspension is extremely simple. We take the dynamic relationship between spring pressure p and z-zt at very low frequency as a simple gain:

$$\delta(z - z_t) = H\delta p \qquad (4.11)$$

With H factor of proportionality. In this way we arrive at a integral controller, with the possibility of adjusting a generic gain α as shown in the figure. So deletes all the noise at very low frequencies due to the presence of the zero in the origin in the transfer function of noise. In most does not interfere with the dynamics of the suspension and choose a value of α so that the response is less than 10-20s. We must be careful not to raise too much the value of this constant as it would have unpredictable interactions with the dynamic of the suspension. E 'therefore thought to act on a bandwidth of the fraction of Hertz with a subsequent response time in the order of ten seconds. The trend curve of the elastic force in function of its elongation, until now supposed linear, we note how in reality present nonlinearities. In particular, we will say that beyond a certain elongation will have a progressive course, if the curve will tend to go above the linear trend and a regressive if it will tend to bend downwards. The next image shows this particular type of trend that in the case of the gas spring is of the progressive type in compression and regressive type in extension. The behaviors are reported for tires inflated in different ways and all have a small hysteresis. What varies is the slope of the curve at the center stoke, which increases with the pressure so the curve above in the figure has the slope 4 times greater than that the down curve at 2bar. The optimal solution would be to always have a progressive trend in both directions so that does not have problems with end of stroke, which would be impossible to get. An alternative to the mechanical load leveling is that the preload where, through the use of a threaded ring that does change the resting point of the spring with respect to the shock absorber, it is possible to choose the optimal calibration. This type of solution, generally only used on motorcycles, has been improved with the addition of an electronic controller.
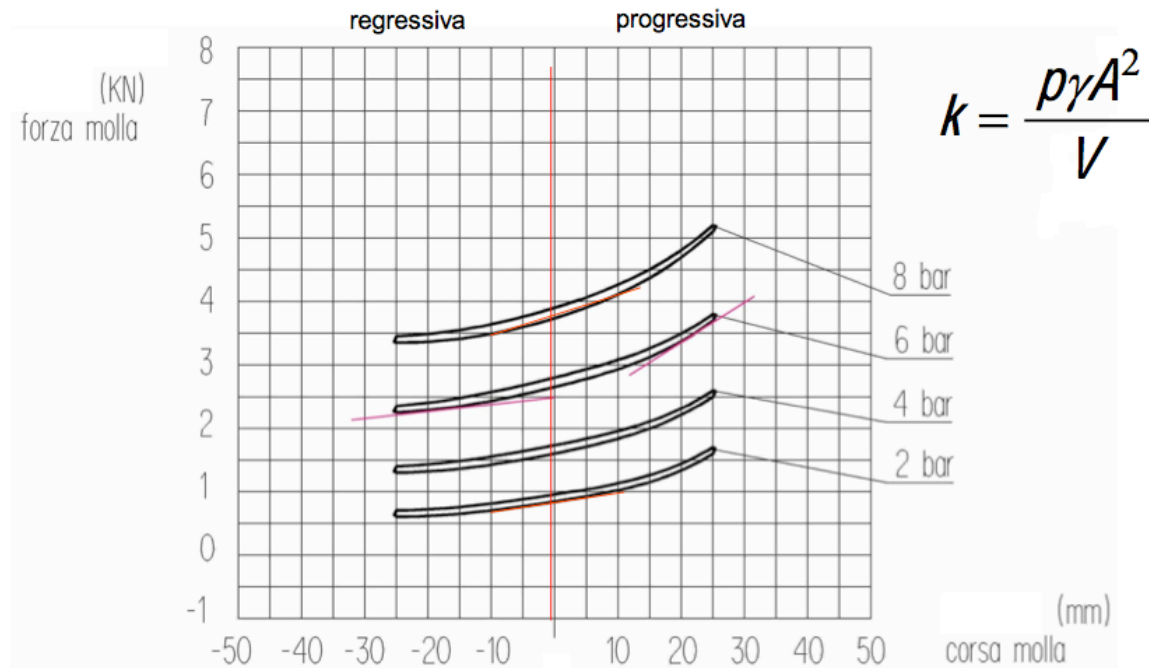
**Figure 23 Progressive-regressive behaviors for gas springs**

b) Adaptive Suspension

The adaptive suspension allows you to vary slowly in the damping coefficient. Early versions allowed the shock absorber, with two or three levels of damping, which were chosen in a reasonably appropriate by the driver. With the passing of time have been developed more sophisticated versions with a supervision system that extrapolates the driving style of the driver, the type of use, the type of soil and the type of calibration. Than the classification can be made between the variation in step or continuous and automatic or adaptive deployment. It's obvious that the combination auto adaptive and continues present the more refined version which lately has fallen into disuse because with a small sacrifice in economic terms allows to use the semi-active that produce a significant increase in performance. Not require high speed switching, as 500m are adequate.

c) Semi-active Suspension

This type of suspension has a high switching speed. Compared to the adaptive you pass by switching half second to 10-30ms. The second aspect that must be such as to influence in terms of cost, in addition to the switching speed, concerns the fact that sensors are used to close the control loop. The semi-active suspensions are characterized by a high frequency modulation in the feedback of 5ms of the damping coefficient of the shock absorber, with closed loop logic always inserted. The main differences within the family of semi-active suspension are based on the different control logic that can be comfort-oriented or performance. In addition to this they differ according to the different types of shock absorber which can be at levels or continuous, electro-hydraulic (EH), magnetic-rheological (MR)

and electro-rheological (ER). Most cars use algorithms oriented comfort with electro-hydraulic technology.
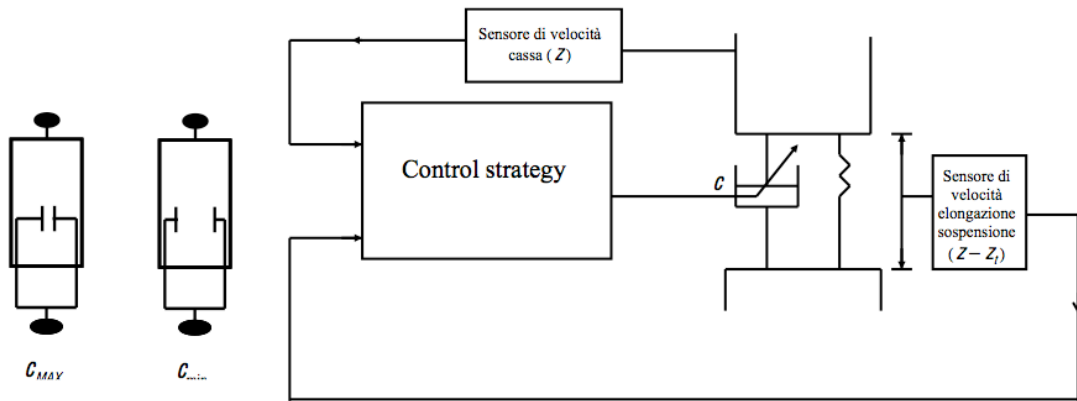
$$\begin{cases} c(t) = c_{MAX} & if \quad \dot{z}(\dot{z} - \dot{z}_t) \geq 0 \\ c(t) = c_{min} & if \quad \dot{z}(\dot{z} - \dot{z}_t) < 0 \end{cases}$$

Figure 24 Skyhook Control

In the semi-active suspension that is most important is that the control algorithm becomes an aspect to be considered in the design phase. As we already mentioned, the control is implemented Skyhook, which can be a two-state or modulating. Regarding the control SH in two states, certainly more simple, we note that if the chest is rising (z> 0) and chest-wheel recede (z-zt> 0), the damper counteracts correctly the movement of the chest. The opposite case, when they are approaching, when they are making a motion opposite to what we want. In this case, the best use I can do is switch off the shock or at least bring it to its minimum value.

The algorithm SH modulating, that is more refined, gives us a more detailed description. What we're going to do is introduce variation intermediate between Cmin and Cmax so to use an exact value if we fall within these two values. The control scheme is shown in the figure and we can notice how the SH in two states can be imagined as an extreme version of this type of control. Note that there is a constant α in the control algorithm that determines the weight of the shock distributed between the case and the case-wheel. Generally, it make sense to think that it is distributed in a fair manner and therefore α = 0.5.

Generalizing the control algorithms for semi-active suspension until now seen, we can summarize that it is always a variable to be controlled Z subject to non-measurable interferences Zr and a control variable C. The latter is out by a controller that has in input the measurements of the sensors that are present in the suspension. The analysis that follows is of a complex type as both the controller and the system to be controlled are not linear in the controller that are due to a computational logic while in the system are given by the presence of the stroke end. We also saw how difficult it is to manage the process through a linearization system, which in fact has not been done.
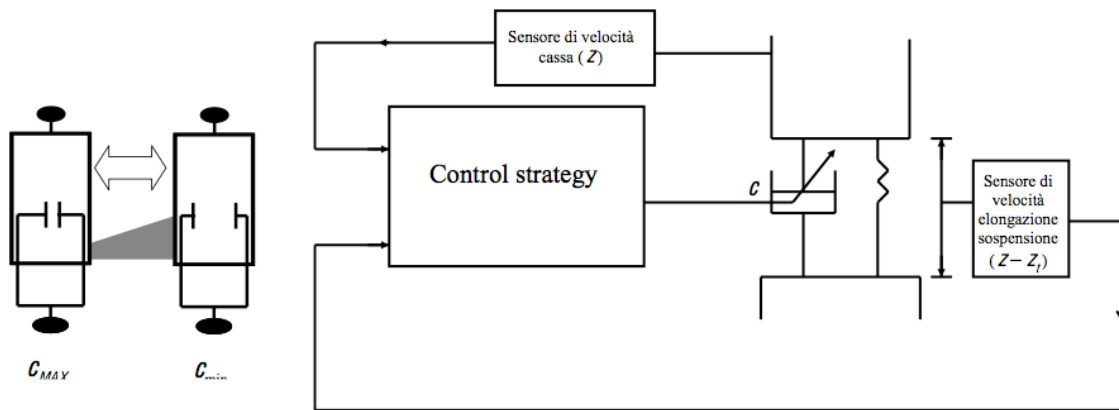
$$c(t) = \underset{[c_{\min}, c_{MAX}]}{sat} \left[\frac{\alpha c_{SH}(\dot{z} - \dot{z}_t) + (1-\alpha)c_{SH}\dot{z}}{(\dot{z} - \dot{z}_t)} \right]$$

Figure 25 Modulating Skyhook control

d) Technologies

We can also make some considerations on the technologies used up to now. Regarding the possibility to vary the damping, we can operate mechanically widening or narrowing the passage channels. Even more sophisticated is the magnetic-rheological technology where the piston of the magnetic fields are present and the variation of the field produced fluid changes its viscosity. With the magnetic field becomes a solution similar to the gel-type semi-solid. The advantage of this technology is in the high range of modulation rates of wear in the time of sealing organs, which can lead to leakage of fluid. Similar to this technology is the electro-rheological where the reaction occurs for electric fields instead of magnetic ones. From the point of view of the pilot switching from variations in voltage of 12V and 3-4A in current for the MR to variations from 2-550V and currents of the order of 1mA for the ER. For the moment, high voltages for the latter technology described constitute a barrier for its development.

The sensors

Regarding the sensors is important to observe that in general there is an accelerometer placed on the chest and a potentiometer placed along the suspension. What we are interested in measuring the speed of body and that of stroke. It follows that we are going to use in the first case an integrator and a differentiator in the second. In the treatment of the reconstruction of the speed must be taken account of the noise inevitably present on the signals that we measure, it will be in the white noise and in part a constant signal which is the most delicate and is typically due to problems of alignment or incorrect position of the same sensor. Given the inevitable presence of noise, regarding the use of the integrator, will have problems of infinite amplification of it at low frequency. Instead of using a true integrator we put a pole height of the first resonance frequency, whereas above 0.1Hz speed is well approximated, as there is enough in shape from that frequency and then at the same time solve the problem of amplification noise. Regarding the derivative action will dually the problem of amplification of high frequency noise. As a solution to this problem is added a low pass filter of the second order around 10Hz above the resonance frequency of the wheel. Affecting economically significantly tends to mount the lower number of sensors possible. What is typically done is accurate modeling of the front because you think that the rear incorporate the dynamics of it.

Semi-active architecture system

From the point of view of controlling the real control variable is the driving voltage of the circuit that goes to the valve piloting. The real architecture of a system of semi-active control should however be seen as a system of three levels in cascade with three loops one inside the other. The inner one is very fast compared to the next, up to the third slower than the others in such a way as to have decoupling frequency. The intermediate level may in certain cases not be present.
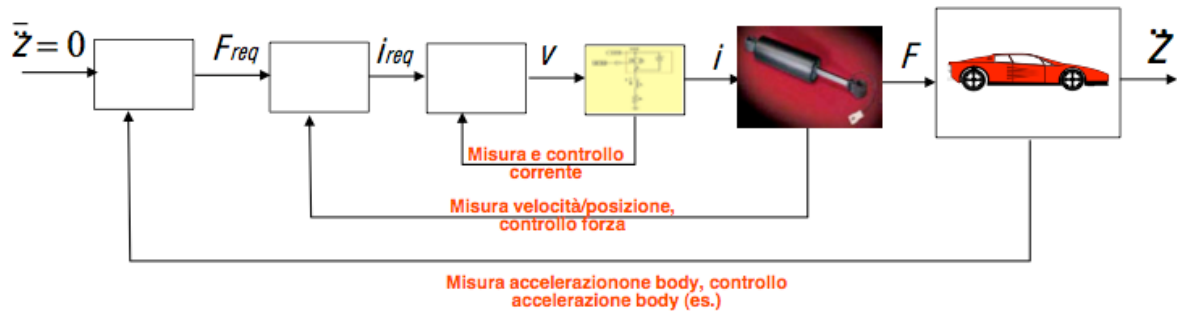

Figure 26 Three levels architecture

The first level is that of the excitation circuit where normally a check is done in the current to compensate for thermal drift. From the moment the current tends to descend to the variation of resistance with increasing temperature becomes uncertain the relationship between current and damping of the shock. For this reason it is inserted this first feedback; also used to treat non-linear relationship present in some places going to remove uncertainty and it is especially convenient to decrease the transient. In this way, you pass by a response in open loop around 30ms to 7ms in closed loop. Conceptually what happens is that depending on the current that insert get a specific damping value. The reports that there are between current and damping may be direct if the valve closes by increasing the current and reverse when the valve is opened instead as the current increases. From the energy point of view, the direct relationship is better since we averagely low values of damping. The problems may be the case is not able to supply more current to break the unit or battery pack, as I would be in direct relation to power off and therefore on minimum damping with complications from the standpoint of safety. Ohlims has solved this problem with a safety-related mechanical changes to null current that moves the damping to mean value but in general there is a trade-off energy-safety.

The second level concerns the shock absorber itself and offers no advantage in dynamic level, it simplifies the design of the outer ring. To fully understand this level you need to keep in mind the slope of the force depends on the speed of elongation. For electro-hydraulic suspension, the characteristic is sufficiently linear and for this reason may not be necessary at this level. The characteristic features of the forces in the MR instead of the obvious non-linearity with current constant: steep in the central and almost constant thereafter. From the moment the algorithm reasons in terms of Cmin-Cmax is required reversal of the non-linearity to drive precise values of damping to the outermost level. The fact that the MR should always have to vary the current variation of the damping is in itself a disadvantage, however, but it allows me to develop high forces at low speeds as it covers virtually the whole of the I and IV quadrant.

The third level is the dynamics of the vehicle which expects a nearly instantaneous reaction to the change in damping. As the dynamics of subsequent levels are fast will never be reached in an ideal characteristic since the behavior will normally not be linear and the switching speed will certainly not be infinite: for these reasons we will have a loss of performance.

e) Active suspension

These special suspension introduce energy into the system: they can induce any movement of the vehicle (even a counter-roll). They typically use electro-hydraulic actuators and they differ depending on the frequency band within which they can be implemented (slow / full active). The slow active have

a band with control flows (up to 3-5Hz) and provide for the aid of the spring and damper classic addition to the actuator while the full active control band also have a wheel (up to 30Hz) and provide the use of only one actuator of force.

The basic operation of the actuator for the slow-active technology is the use of electro-hydrostatic with an electric motor that rotates in one direction or the other has the ability to enter or remove fluid to the actuator through a mechanical transmission that governs the flow. Recently the Bose is developing a linear electric motor that acts directly without the hydraulic transmission through that face so as to eliminate the losses for transmissions and can also make energy recovery. The dynamic model for this type of suspensions, in particular for those full-active, it is more simple. The control law that results is a time-invariant algebraic linear feedback of the state of the type:

$$\delta F = K \delta x \tag{4.12}$$

Where we apply a linear quadratic optimal control in which it is set as a target to minimize the figure of merit:

$$J = \frac{1}{2} \int_0^\infty [\Gamma_a(\delta F)^2 + \Gamma_e(\delta z - \delta z_t)^2 + \Gamma_c(\delta z_t - \delta z_r)^2]dt \tag{4.13}$$

Where Γ_a Γ_e and Γ_c are the weights with which the respective objectives of comfort, minimization of elongation of the suspended particles, and road holding are balanced. I could also pose as additional targets within the figure of merit as to optimize the roll and pitch motions. And 'interesting to observe how the benefits of this type of suspension are significant. Next image shows the comparison between a full-active suspension and a common passive suspension. Note how slow the produce obvious results up to frequencies of 4-5Hz going to remove all the resonance component of body.
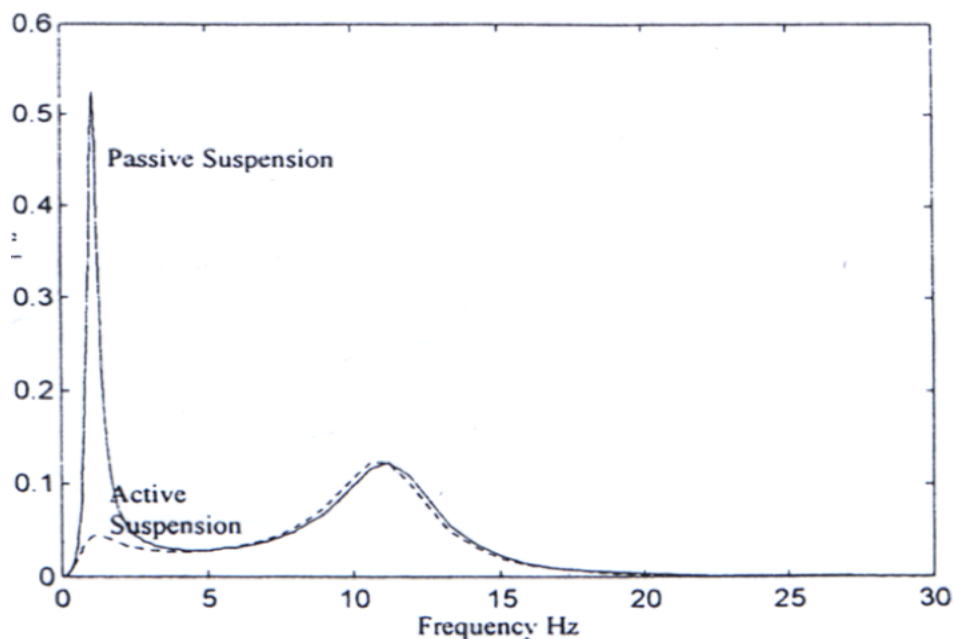

Figure 27 Different between passive suspension and active suspension

5. The control

Now that the various models have been introduced that shape the suspension systems and, after having described and simulated the Skyhook suspension type, we can move to simulate our control system.

First, in reference to the results obtained up to now, we have seen how the various car models quarter were to offer an optimum representation of the condition of the vehicle. In the case of half car model representation is certainly more complete because it takes account of the pitching movement, but in our case we are interested in seeing the behavior of the vertical displacement which is the most crucial part and for this the quarter car model is more than sufficient and easier to mold.

Between quarter car models, it was decided to use the classic model that models also the mass of the wheel, as in the references used this model was used. However, the simulations were done in the third chapter suggest that similar results could be achieved with the simplified model.

All simulations were carried out with the toolbox of Matlab Simulink which recreate the system for a quarter car semi-active suspension using a Matlab function. The information that this function has input are those relating to the characteristics of the vehicle or the masses, the elasticity and so 'on. In addition to this comes the value of the force that the actuator to replace the damper must provide. Will own the Skyhook controller to provide this value precisely that depend on the characteristics of the vehicle and the condition of the road surface. In reference to the latter have been made various simulations assuming a case in the road surface with a sinusoidal shape and in another case were considered a series of steps of variable amplitude. To create a series of steps was used a block of white noise with noise power of the order of 10 ^ -6 and a sampling time of one second.

At the center of it all is as we said, our control system which we have already extensively discussed qualitatively. The type that we are going to use is that of modulating Skyhook control that uses values between the minimum value of 1000 and the maximum value of 3000. From the logical point of view has been implemented a feature if-else where if the ideal value at that time is between the values of maximum and minimum is used then the exact value; case this value is below the minimum value is the minimum value of the same to be used and always in a similar manner in the case the value exceeds the maximum, it will be just the maximum value to be used. In entry are present case and the speed of the speed of the wheel, while the output as we have already said, this is the value of the force that will be given to the actuator to replace the damper.
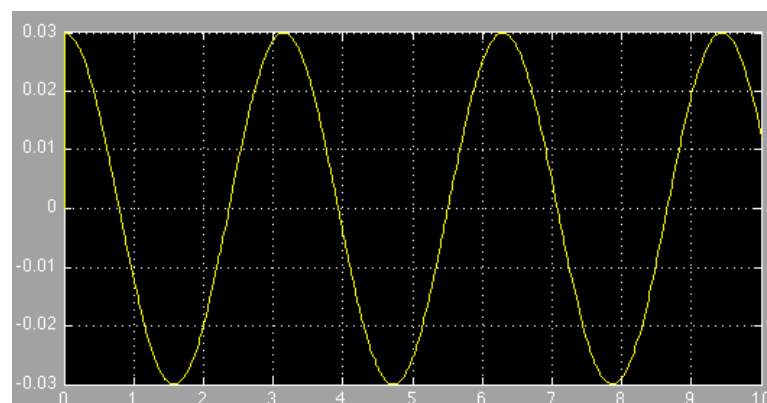

Figure 28 Sinusoidal road profile

Let's go at this point to see in detail the results of our simulation. We start from the case of the sine wave of the figure represented in the figure.

The control scheme has been designed so as to be able to view some information that may be useful for an analysis of the vehicle. In particular, in each simulation are compared to the responses between the semi active suspension system with Skyhook control and the system with the simple passive suspension.

Let's see in detail in the figure the different distortions of the suspension system and graphics acceleration flows.
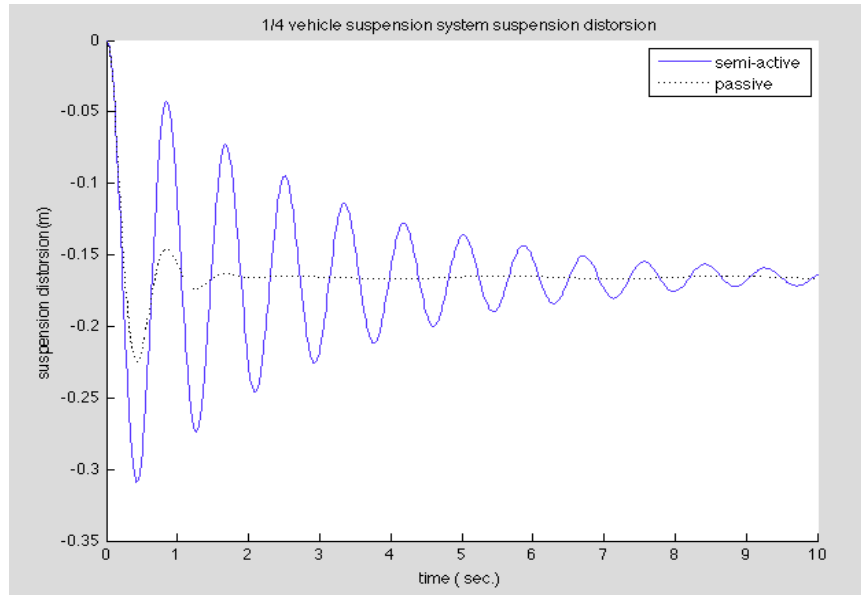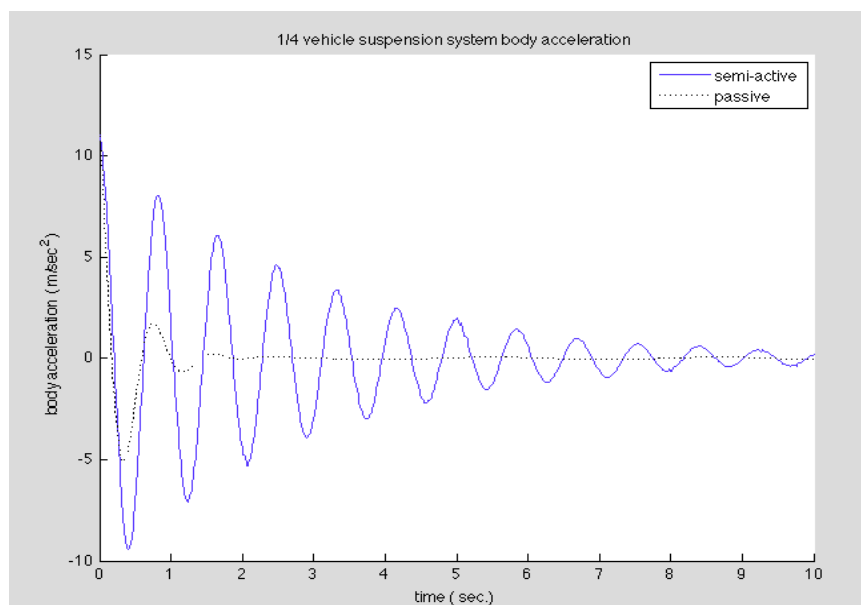

Figure 29 Suspension distortion

Figure 30 Body acceleration

We note that in the case of passive suspension, not having the opportunity for the input energy into the system, converts quickly to the required value and the acceleration of cash rapidly decrease. In the case of semi-active suspension we can see how the accelerations Flows both the distortion of the suspension are characterized by a sinusoidal damped.

29

Finally we go to observe output from this controller Skyhook values that are passed to the actuator that modulates the damping coefficient.
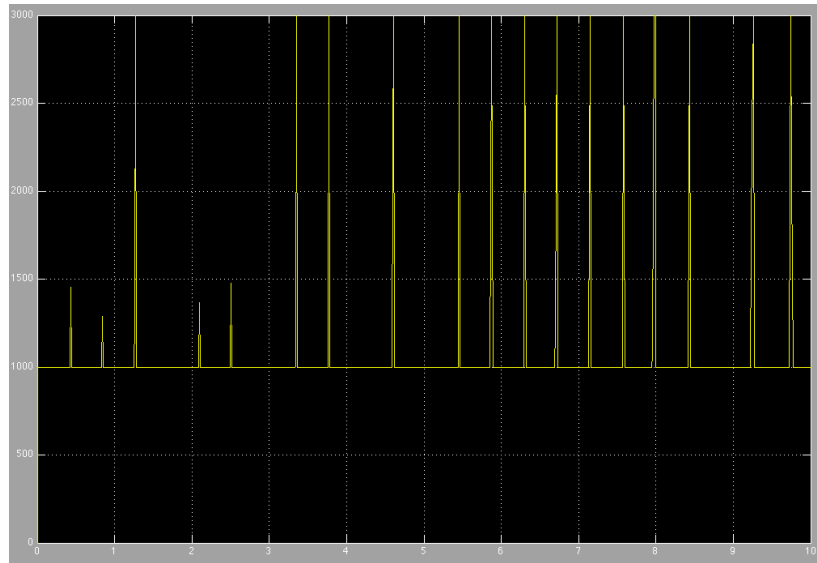

Figure 31 Damping values

As a second simulation is instead introduced a system of steps of varying amplitude generated as if it were a white noise shown in the image.
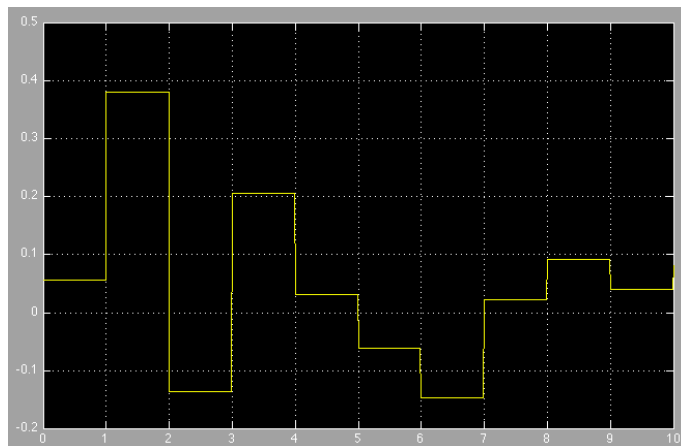

Figure 32 Steps road profile

As expected, since the input signal is characterized by a pattern of random type well accelerations Flows and distortions of the suspensions will resume the performance characteristic of damped sinusoidal type with minor distortions in the features of fields sudden slope of the road profile .
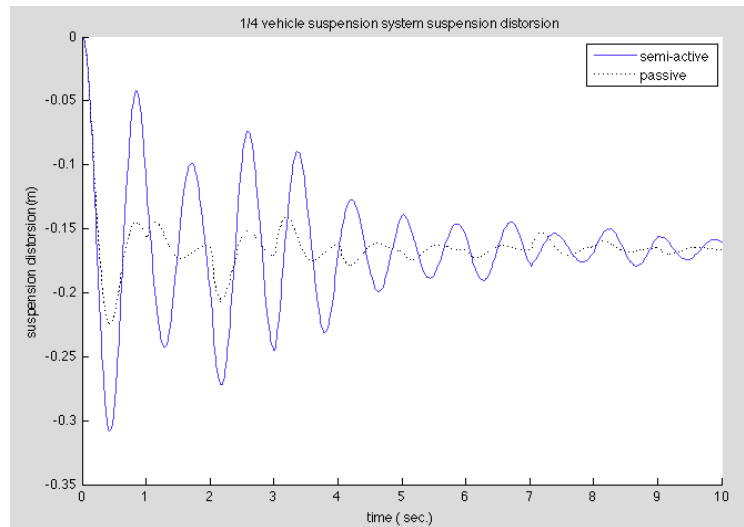
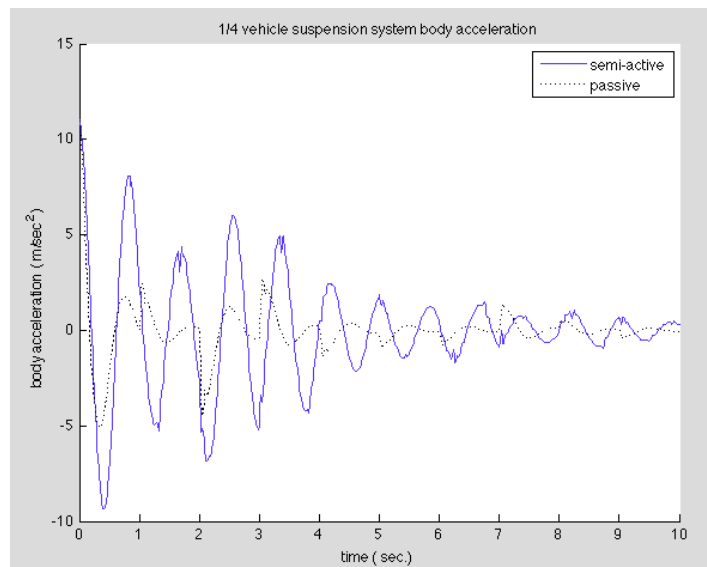**Figure 33 Suspension distortion**

Figure 34 Body acceleration

To conclude, consistently with what already said, we can see how in this case the actuator works on values greater than the previous case.
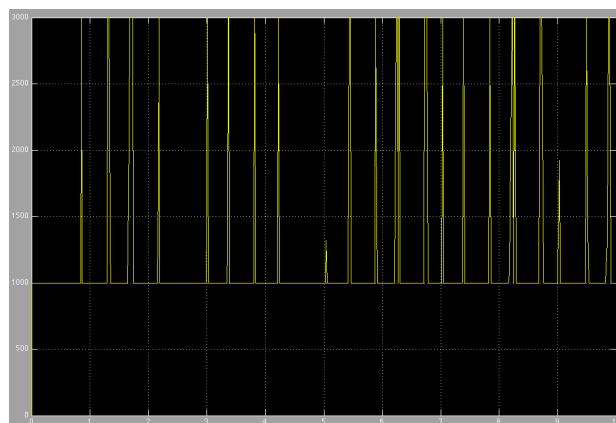

Figure 35 Damping values

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6. Conclusion

As we suggested at the beginning, suspension systems allow a significant improvement in driving conditions. The addition of the control systems has contributed in a clear manner to improve even more the conditions of safety and comfort.

In our case, we have seen how studied by varying the value of the damping coefficient can be achieved best performance visibly already in open loop.

The addition of a feedback loop in control systems applied to electronic suspension control has made it even more optimal the result and the performance of the suspension.

Although the results are only theoretical and were not taken into account the saturation of the components or switching limits which in reality will be present and will lead to significant non-linearity in the suspension systems, it has been shown that control systems are fundamental to the means of transport to the day today and are increasingly being used.

7. References

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