Modelling and Control of a Semi-active Suspension System

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Abstract. The suspension system is composed of several deformable elements such as springs and dampers, which connect the car body to wheels and absorb vibrations generated by road irregularities. The main purpose of suspension system is to isolate the vehicle body from disturbances in order to keep wheels in contact with the road surface to contribute to road holding, and in order to maximize passenger ride comfort. This paper describes a semi-active suspension system of 2 degrees of freedom (2DOF), typically referred to as a quarter car model. To design a suspension control system that improve ride comfort, dynamic modelling of semi-active suspension was developed. Control strategies were implemented for these semi-active suspension systems using MATLAB® and Simulink® software. The results show that the semi-active suspension system controlled by a logical strategy minimizes vertical acceleration experienced by passengers, compared to passive suspension system.

Keywords: semi-active suspension, quarter car model, ride comfort, classical control.

1 Introduction

In recent years, researches in the field of automotive vehicles are focused on improving driving safety and passenger comfort. These two parameters are mainly influenced by the design of the suspension system. The suspension system works between vehicle chassis and wheels, and its main goal is to reduce motion of the vehicle body called sprung mass. The design of a good suspension system is focused on the isolation of the disturbances coming from road irregularities, cornering and breaking, in order to maximize passenger ride comfort and keep wheels in contact with the road surface to contribute to road holding.

A good ride comfort requires a soft suspension, however, this suspension system produces excessive roll during cornering and pitch during breaking, and

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therefore can be uncomfortable for passengers. While a stiffer suspension improves the phenomena of roll and pitch, ride comfort is reduced. As suspension design is a compromise between these two goals, this problem can be solved with an adaptive suspension system that changes the suspension parameters depending of the features of the terrain or the vehicle driving manoeuvres.

Suspension systems are classified by the control system in passive, semi-active and active. Passive suspension systems consist of conventional springs and dampers, whose properties are fixed, and there is no external energy source in the system. Semi-active suspension systems, generally, consist of controllable dampers and passive springs without requiring large power sources, so the control system is not destabilized [1]. Different types of semi-active dampers have been investigated, the most representative are: magnetorheological dampers, whose response varies with the magnetic field applied [1], electrorheological dampers, whose response varies with the electric field applied [11], pneumatics dampers, generally used in buses and lorries [3], and dry friction dampers, highly non-linear and based on the friction between surfaces in contact [8]. Finally, active suspension systems have the capabilities to adjust themselves continuously to changing road conditions, so that mechanical elements (springs and dampers) are replaced by actuators that can generate forces according to a control algorithm [7, 10, 16, 18]. Moreover, semi-active and active suspensions can reduce the resonance peak and the amplitude of movement of the sprung mass in most of the frequency range [19].

To study the behaviour of suspension systems, different vehicle models have been used. The most used model is the quarter car model [1, 16], because it takes into account the most important features of suspension system preserving the simplicity of the model. It is a model of two degrees of freedom (2DOF) which considers the vertical dynamics of a single wheel. The half vehicle model is four degrees of freedom model (4DOF) which generally represents the pitch motion [4]. And the full vehicle model is a model of seven degrees of freedom (7DOF) that consists of a sprung mass that is connected to four unsprung masses, and represents the pitch, roll and yaw movements of the vehicle body [7, 10, 18].

Regarding the control, it is a multi-objective and non-linear problem for which classical and modern methodologies, based on a mathematical model of the system, have been developed [9, 15].

Due to its simplicity, classical control strategies for semi-active suspension have been implemented: PID [16] and Logical Control Strategies [15]. These last Control Strategies are oriented to improve in a simple way the comfort, so its goal is to minimize the vertical acceleration of the sprung mass. The most common logical control strategies oriented to comfort are the 2-States Skyhook Control, Skyhook Linear Control, Acceleration Driven Damper Control (ADD), Power Driven Damper Control (PDD), Skyhook-Add Control and Skyhook-PDD Control.

Modern control strategies for semi-active suspension are more difficult to implement with higher computational cost: Adaptive Control [17], Optimal Control [6], Predictive Control [5] and Robust Control [7]. Adaptive Control automatically adjusts its characteristics to operate optimally in a changing environment, reducing disturbances and vibration of the vehicle at specified levels. The use of

Optimal Control is recommended when the system behaviour has uncertainties. Predictive Control is used in complex, multivariable or unstable dynamic system; it is based on the use of an optimized model for predicting system behaviour and future control signal. The Robust Control considers the uncertainties in the mathematical modelling in order to become independent the system of disturbances.

In this paper a quarter car model is used to analyse the ride performance behaviour. The purpose of this paper is to compare the different logical control strategies for semi-active suspension system based on a quarter car model, due to its simplicity, in order to maximize passenger ride comfort. In order to detail this approach, the rest of the paper is structured as follows. In section 2, the suspension system modelling is detailed; section 3 presents the control strategies; in section 4 a discussion of simulation results is presented. Finally, the most important ideas are summarized.

2 Suspension System Modelling

In this paper, a quarter car model with two degrees of freedom (2DOF) is considered (Fig.1), because this model represents most of the features of the full vehicle model while preserving the simplicity of the model.

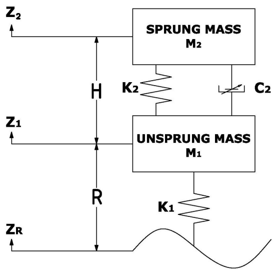

Fig. 1 Quarter car model with semi-active suspension

The vehicle chassis is modelled as a rigid body of mass M_2, and unsprung mass, represented by M_1. The suspension system consists of a spring of stiffness K_2, and a damper with a variable damping C_2. The wheel is modelled as a single spring with a stiffness constant K_1.

This model is described by the following system of second order ordinary differential equations:

$$\ddot{Z}_2 M_2 = -M_2 g - K_2(Z_2 - Z_1 - H) - C_2(\dot{Z}_2 - \dot{Z}_1) \quad (1)$$
$$\ddot{Z}_1 M_1 = -M_1 g - K_1(Z_1 - Z_R - R) - K_2(Z_1 - Z_2 + H) - C_2(\dot{Z}_1 - \dot{Z}_2)$$

Where Z_2 is the sprung mass vertical displacement, Z_1 is the unsprung mass vertical displacement and Z_R is road profile acting as the disturbance. Gravity is represented by g. H and R are two parameters that represent initial conditions.

Furthermore, in order to make sure that the suspension system is working properly, certain functional limit values must be defined, so that it cannot be extended or compressed more than a certain predetermined amount. For this purpose, two bump stop limits have been modelled. Moreover, the system distinguishes the motion of the damper between rebound travel and jounce travel.

3 Control Strategies

The control objective of this paper is oriented to comfort performance. To evaluate passenger comfort, the approach considered to determine the performance of the suspension system is the acceleration of the sprung mass. So the control objective consists in minimizing the vertical acceleration of sprung mass with the most simple control strategies. The different logical control strategies mentioned above are implemented in a quarter car model. Then, analysing the results, the best control strategy will be determined.

The following logical control strategies have in common that change the damping factor of the damper (C_2) according to the chassis velocity ($\dot{Z}_2$), the chassis acceleration ($\ddot{Z}_2$), the suspension deflection position ($Z_{def} = Z_2 - Z_1$) and/or the suspension deflection velocity ($\dot{Z}_{def} = \dot{Z}_2 - \dot{Z}_1$), depending on the algorithm. Moreover, all of them use only two sensors. Control strategies that will be implemented are shown in Table 1.

Table 1 Logical Control Strategies

Controller	*Control Law*	*Other Authors*
SH2-States	$C_2 = \begin{cases} C_{min} & if\ \dot{Z}_2\dot{Z}_{def} \leq 0 \\ C_{max} & if\ \dot{Z}_2\dot{Z}_{def} > 0 \end{cases}$	[2]
SH Linear	$C_2 = \begin{cases} C_{min} & if\ \dot{Z}_2\dot{Z}_{def} \leq 0 \\ sat_{C_2\in[C_{min};C_{max}]}\left(\frac{\alpha C_{max}\dot{Z}_{def} + (1-\alpha)C_{max}\dot{Z}_2}{\dot{Z}_{def}}\right) & if\ \dot{Z}_2\dot{Z}_{def} > 0 \end{cases}$	[13]
ADD	$C_2 = \begin{cases} C_{min} & if\ \ddot{Z}_2\dot{Z}_{def} \leq 0 \\ C_{max} & if\ \ddot{Z}_2\dot{Z}_{def} > 0 \end{cases}$	[14]
PDD	$C_2 = \begin{cases} C_{min} & if\ kZ_{def}\dot{Z}_{def} + C_{min}\dot{Z}_{def} \geq 0 \\ C_{max} & if\ kZ_{def}\dot{Z}_{def} + C_{max}\dot{Z}_{def} < 0 \\ \frac{C_{min} + C_{max}}{2} & if\ Z_{def} \neq 0\ and\ \dot{Z}_{def} = 0 \\ -\frac{kZ_{def}}{\dot{Z}_{def}} & otherwise \end{cases}$	[12]

Table 1 (*continued*)

SH-ADD	$\begin{cases} if \;\; f > f_c \;\; then \;\; ADD \\ if \;\; f < f_c \;\; then \;\; SH \end{cases}$
SH-PDD	$\begin{cases} if \;\; f > f_c \;\; then \;\; PDD \\ if \;\; f < f_c \;\; then \;\; SH \end{cases}$

Where f_c is the cut-off frequency. In this paper, it is considered that f_c has a value of 3 Hz.

4 Simulation and Results

In order to simulate the control strategies mentioned above, the general Sine Sweep test (ISO7401) is considered. This input represents variations of the road irregularities amplitude at high and low frequencies, from 0.1 to 12 Hz, allowing to analyse how the controller responds to both effects. Moreover, this test covers different types of roads with different frequency disturbances from typical low frequency speed bumps, to high frequency Belgian blocks pavement.

Table 2 shows the parameters used for simulation tests, which are based on a Fiat Punto.

Table 2 Parameter Values for Simulation tests

Parameters	***Value***	***Parameters***	***Value***
Sprung mass (M_2)	284.325 kg	Suspension Stiffness (K_2)	58,860 N/m
Unsprung mass (M_1)	41.8125 kg	Extended Damping (C_2)	8,181.8 Ns/m
Tire Stiffness (K_1)	210,000 N/m	Compression Damping (C_2)	4,090.88 Ns/m
Tire Radius (R)	0.33 m	Jounce Travel	0.06 m
		Rebound Travel	0.075 m

Fig. 2 shows road profile and the simulations results of the vertical acceleration of sprung mass for passive suspension system and for each Control strategies: SH 2-States, SH Linear, ADD, PDD, Mixed SH-ADD and Mixed SH-PDD, respectively.

Fig. 3 shows a comparison between the above strategies (Table 1).

The problem of the on/off control algorithms is the chattering phenomenon, which consist in a very fast change of the controlled variable in the nearest of the change position. In the simulations, this phenomenon appeared in the SH, ADD and SH-PDD strategies. In the ADD this problem is more relevant because the oscillations are bigger and faster, but in the SH-PDD is an allowable phenomenon because it only appeared in a small range of frequencies, more specifically at low frequencies.

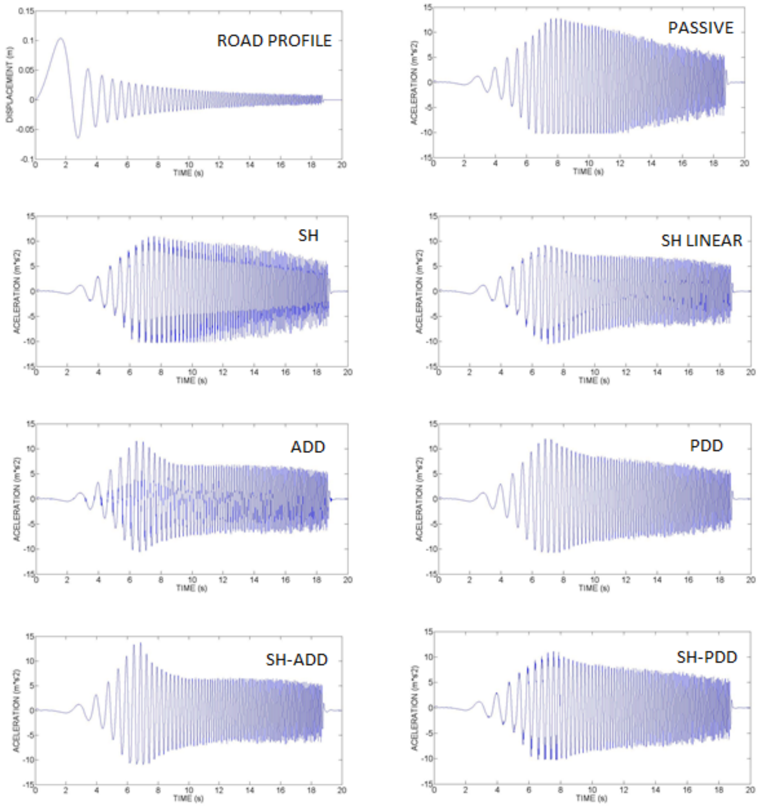

Fig. 2 Road profile and vertical acceleration of sprung mass for each control strategy

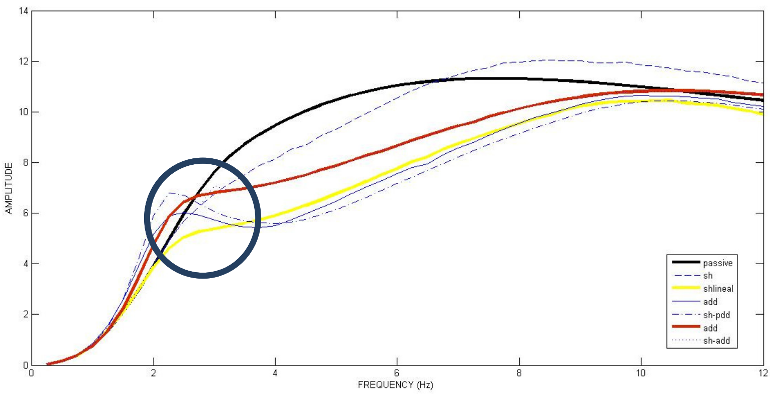

Fig. 3 Comparison between all strategies

On the other hand, in Fig.3 it can be emphasized that at 3 Hz frequency the SH algorithm is intersected by the ADD algorithm, so that is the reason why 3Hz is the cut off frequency (f_c) in the mixed algorithm, SH-ADD. As a result, the best control strategy in order to improve ride comfort is the Skyhook Linear Control. However, this control has the disadvantage that it is difficult to control a linear damper because it is necessary to have a continuous damper. Another control strategy that provides a good result is the Mixed SH-PDD strategy, where a frequency selector has been developed to avoid the problems of chattering from the Savaresi's frequency selector. The problem is that this control needs also a linear damper and it does not improve the Skyhook Linear Control results. In order to implement an on/off control algorithm that can be implemented with discontinues damper, the Mixed SH-ADD Control has been adjusted to avoid the chattering problem that this model showed introducing a relay in the logical algorithm conditions. The relay deteriorates the attenuation but avoid the high frequency accelerations.

5 Conclusions

Different logical control strategies have been implemented in order to improve ride comfort with a semi-active suspension system. The results indicate that semi-active suspension with Mixed Skyhook-ADD Control is the best on/off suspension control. It is a simple control that avoids the problem of chattering by the use of relays. This control uses the good behaviour of Skyhook Control at low frequencies, and good dynamic response of the algorithm ADD at mid and high frequencies, where shows a better response that a continues algorithm like Skyhook Linear Control. If the damper is continuous, the best suspension control is the Skyhook Linear Control, although at high frequencies it is exceeded by the ADD strategy.

Future work will be performed to improve passenger ride comfort by implementing an active suspension control system. This improvement will be accomplished through the utilization of force actuators.

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