A comparison of optimal semi-active suspension systems regarding vehicle ride comfort

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Abstract. The aim of this work is to present a comparison of the main semi active suspension systems used in a passenger car, after having optimized the suspension systems of the vehicle model in respect with ride comfort and road holding. Thus, a half car model, equipped with controllable dampers, along with a seat and a driver was implemented. Semi-active suspensions have received a lot of attention since they seem to provide the best compromise between cost (energy consumption, actuators/sensors hardware) and performance in comparison with active and passive suspensions. In this work, the semi active suspension systems studied are comfort oriented and consist of (a) the two version of Skyhook control (two states skyhook and skyhook linear approximation damper), (b) the acceleration driven damper (ADD), (c) the power driven damper (PDD), (d) the combination of Skyhook and ADD (Mixed Skyhook-ADD) and (e) the combination of the two with the use of a sensor. The half car model equipped with the above suspension systems was excited by a road bump, and was optimized using genetic algorithms (GA) in respect with ride comfort and road holding. This study aims to highlight how the optimization of the vehicle model could lead to the best compromise between ride comfort and road holding, overcoming their well-known trade-off. The optimum results were compared with important performance metrics regarding the vehicle’s dynamic behaviour in general.

1. Introduction
The primary purpose of a vehicle’s suspension system is to provide stability, support, and comfort during the drive. Ride comfort and isolation from road disturbances are greatly influenced by the suspension system. Therefore, suspension design is a trade-off between the three conflicting criteria of road holding, load carrying and passenger comfort. In order to study the suspension system in vehicles, various mathematical models have been developed such as the quarter car, the half car and the full car model. The quarter model is dominant in the research studies due to its simplicity [1], however the half car model demonstrates more accurate results including the effect of either the pitch or the roll phenomena [2, 3]. Although, for greater accuracy the full car model is used including both the roll and the pitch phenomena, but it is more demanding computationally than the others and hence very few studies have been performed based on it. In addition to the vehicle model in recent studies, a seat [4] and a driver [5] have been added, so as to study the ride comfort of the passengers extensively and investigate in depth the effects of the vibrations induced by the road to the passengers.

The main focus of these models is the suspension system modeled with a spring and a damper. Vehicle suspensions are mainly classified into three types depending on their operation: passive, semi-active and active. Passive suspensions are older conventional systems having non-controlled springs and
shock absorbing dampers with fixed parameters. On the contrary, active suspensions can adapt to the system variations based on online changes of the actuating force. Therefore, active suspensions have been extensively studied since 1960s and various approaches have been proposed in order to exploit their advantages. However, active suspensions normally require important amounts of power supply, which is their main drawback, preventing them from being used extensively in practice. From 1970s, semi-active suspensions have received much more attention, since they can achieve greater performance than passive ones and consume less power than active ones. Thus, the dominant line of research lies in the semi-active suspensions since they provide the best trade-off between cost (energy-consumption, actuators/sensors hardware) and performance.

The research in this field follows two mainstreams: the study of new technoloies of semi-active actuation of damping (like electro-hydraulic, electro-rheological and magnetorheological damper), and the design of semi-active control strategies. Bourmistrova et al. (2005) [6] applied evolutionary algorithms to the optimization of the control system parameters of a quarter car model. The multi objective fitness function which was a weighted sum of car body rate-of-change of acceleration and suspension travel was minimized. Poussoit-Vassal et al. (2010) [9] aimed at providing a full analysis of the optimal performances both for the comfort and for the handling objectives. These optimal performances were approximated through a numerical optimization approach, using a Model Predictive Control model. Lajqi et al. (2012) [1] introduced a design and optimization procedure for active and semi-active non-linear suspension systems regarding terrain vehicles using a quarter car model. The excitation of the model was represented by double cosine road bumps. Savaresi et al. (2007) [10] presented a study of the performance limits of a semi-active suspension system mainly focused on the objective of comfort. The complementary characteristics of skyhook and acceleration driven damping (ADD) were combined and the mixed control of the two control laws was investigated. Poussoit-Vassal et al. (2011) [11] evaluated the most classic semi-active control laws in time and frequency domain using a single corner model. Colina et al. (2014) [12] also investigated the basic semi-active control laws compared to the passive ones. The model was a quarter car and the general Sine Sweep test (ISO7401) was used as excitation for the models.

The objective of this paper is to optimize the parameters of passive and semi-active suspension systems using the Pareto Front optimization method in respect with ride comfort and road holding. The results are compared using performance metrics. The organization of this paper is as follows: in Chapter 2 a brief description of the models used for both the passive and semi-active cases is provided, in Chapter 3 the road excitation applied is illustrated, while in Chapter 4 the optimization procedure is outlined, its results are highlighted in Chapter 5 and, finally, in Chapter 6 conclusions are presented.

2. Mathematical models

2.1. Half car model

A half car model, equipped with a seat and a 7-DOF human-body model is considered for analysis. The model includes the front and rear axle allowing the observation of the pitch phenomena. As shown in Figure 1, the vehicle model consists of three basic subsystems: the tires, the suspension systems and the body of the vehicle. The latter is considered as a rigid body of mass \( m \), equal to the half of the total mass of the vehicle. The distance of the front and rear unsprung masses (\( m_F \) and \( m_R \)) from the center of mass is equal to \( a_F \) and \( a_R \), respectively. The front and rear tires are modelled as linear springs, which receive as input the irregularities of the road profile. The governing equations of the model have been described in detail in previous work [2] and the parameters with their values are show in Tables 1 and 2.

Additionally, the seat and the driver of a vehicle are modelled, as shown in Figure 2, in order to investigate in depth the ride comfort of the passenger. Different parts of the human body such as the pelvis, the diaphragm, the thorax etc. are described via several \( m – c – k \) subsystems. The equations of the model as well as the parameters were retrieved from the literature [5], [2].

One of the most important parameters of a vehicle model is the suspension travel, which is described as \( z_{ST,F} = z_F - a_F \theta \) and \( z_{ST,R} = z_R + a_R \theta \) for the front and rear suspension respectively, and it is used
extensively in the control laws of semi active suspension systems as part of their condition values. Moreover, another parameter crucial for the dynamic behavior of the vehicle is the tire deflection, described as 

\[ z_{T,F} = z_F - z_{Road_F} \] and \[ z_{T,R} = z_R - z_{Road_R} \] for the front and rear tire, respectively.

\[
\text{Figure 1. Half Car Model.}
\]

\[
\text{Figure 2. Seat – Driver Model.}
\]

| Table 1. Nomenclature of vehicle parameters. |
|---------------------------------------------|
| **Parameters**                             | **Subscripts** |
| \( z \) Vertical motion coordinate         | S Body         |
| \( \theta \) Pitch motion coordinate       | F Front        |
| \( z_{road} \) Road excitation             | R Rear         |
| \( m \) Mass                               | T Tire         |
| \( C \) Damper’s coefficient               | ST Suspension  |
| \( K \) Spring’s stiffness coefficient     | deflection     |
| \( a \) Distance of the centre of the mass of the vehicle | |

| Table 2. Vehicle design parameters.        |
|---------------------------------------------|
| **Parameter**                              | **Value** |
| \( a_F \) [m]                              | 0.91      |
| \( m_F \) [kg]                              | 25        |
| \( m_S \) [kg]                              | 520       |
| \( K \) [N/m]                               | 2.0 \times 10^5 |
| \( I \) [kg \cdot m^2]                     | 473       |

2.2. Semi active control laws

The different semi active controls applied to the suspension systems of the vehicle model are illustrated in this section. All of them are comfort oriented, mostly improving the comfort of the passengers by adjusting the damping coefficient of the suspension system according to the dynamic behavior of the vehicle and its response to the applied excitation. The same control law will be applied in each case in both the front and rear suspension system. Taking into consideration the results of previous studies, most of the control laws applied in this work were expanded from the ones proposed to the literature so as to minimize the changes in the damping factor and thus, avoid the energy cost because of unnecessary changes of the damper’s state. In order to achieve this, a small positive value of the condition used in each control law, for both the front and rear suspension system (\( A_F \) and \( A_R \)), was set as the limit for the different states, leading the damper’s coefficient to change not only regarding the sign of its condition but also based on this value.

2.2.1. Skyhook two state damper control (SH-2). The 2-states Skyhook control is an on/off strategy that switches between soft and stiff damping coefficient. This control law in literature consists of two states in which the damping factor \( C_i \) of the damper (i.e. its fluid viscosity, air resistance, etc.) changes according to the sign of the product of \( z_{ST,i} \) and \( \dot{z}_S \) (equation 1).
2.2.4. Acceleration driven damper control (ADD). The ADD control approach is described by the equation:

\[
C_i = \begin{cases} 
C_{\text{min},i}, & \text{if } |\dot{z}_{ST,i} \dot{z}_S| \leq A_i \\
C_{\text{min},i}, & \text{if } \dot{z}_{ST,i} \dot{z}_S < -A_i \\
C_{\text{max},i}, & \text{if } \dot{z}_{ST,i} \dot{z}_S > A_i 
\end{cases} \quad i = F, R
\] (4)

where \(\dot{z}_{ST,F}, \dot{z}_{ST,R}\) are the stroke velocities of the front and rear, respectively, and \(C_{\text{min},i}\) and \(C_{\text{max},i}\) are the minimal and maximal damping factors achievable by the considered controlled damper.

2.2.2. Skyhook linear approximation damper control (SH-L). An improved version of Skyhook control has been used to achieve variable damping for additional energy saving. The linear approximation of the Skyhook control includes the change of the damping factor \(C_i\) according to the sign of the product \(\dot{z}_{ST,i} \dot{z}_S\): 

\[
C_i = \begin{cases} 
C_{\text{min},i}, & \text{if } |\dot{z}_{ST,i} \dot{z}_S| \leq A_i \\
C_{\text{min},i}, & \text{if } \dot{z}_{ST,i} \dot{z}_S < -A_i \\
\frac{a c_{\text{max},i} \dot{z}_{ST,i}^2 + \dot{z}_S^2}{\dot{z}_{ST,i}} & \text{if } \dot{z}_{ST,i} \dot{z}_S > A_i 
\end{cases} \quad i = F, R
\] (2)

where \(a \in [0, 1]\) is a tuning parameter that modifies the closed-loop performance and enables the controller to adjust its stiffness \((C_{\text{max},i})\) according to the needs of the application, saving more energy. More specifically, when \(a = 1\), this control law is equivalent to the two-state Skyhook control. The difference relies on the fact that, according to the third expression (when \(\dot{z}_{ST,i} \dot{z}_S > A_i\)), such a control provides an infinite number of damping coefficients, providing the ability of tuning it to be continuous.

2.2.3. Power driven damper control (PDD). The power driven damper (PDD) is a semi-active control law that consists in changing the damping coefficient \(C_i\) as:

\[
C_i = \begin{cases} 
C_{\text{min},i}, & \text{if } |\dot{z}_{ST,i} \dot{z}_S| \leq A_i \\
C_{\text{min},i}, & \text{if } \dot{z}_{ST,i} \dot{z}_S < -A_i \\
\frac{K i \dot{z}_{def,i}^2 + C_{\text{max},i} \dot{z}_{def,i} \dot{z}_S}{\dot{z}_{ST,i}} & \text{if } \dot{z}_{ST,i} \dot{z}_S > A_i 
\end{cases} \quad i = F, R
\] (3)

where \(\dot{z}_{ST,i}\) is the suspension travel divided by the stroke velocity. The damping coefficient may be the minimum, maximum, their mean value or the product of the spring stiffness with the Skyhook control includes the change of the damping factor \(C_i\) according to the sign of the product \(\dot{z}_{ST,i} \dot{z}_S\).

This control law depends on the value of the suspension travel and the stroke velocity. The damping coefficient may be the minimum, maximum, their mean value or the product of the spring stiffness with suspension travel divided by the stroke velocity.

2.2.4. Acceleration driven damper control (ADD). The ADD control approach is described by the equation:

\[
C_i = \begin{cases} 
C_{\text{min},i}, & \text{if } |\dot{z}_{ST,i} \dot{z}_S| \leq A_i \\
C_{\text{min},i}, & \text{if } \dot{z}_{ST,i} \dot{z}_S < -A_i \\
C_{\text{max},i}, & \text{if } \dot{z}_{ST,i} \dot{z}_S > A_i 
\end{cases} \quad i = F, R
\] (4)

Thus, this strategy is shown to be optimal in the sense that it minimizes the vertical body acceleration \((\ddot{z}_b)\). In literature, it is stated that the “switching dynamic” may influence the closed-loop performances. The swift changing of values of the sprung acceleration leads to quick changes of the damping factor and this dynamic change creates a noise.

2.2.5. Mixed Skyhook-Acceleration driven damper control (SH-ADD). SH and ADD present complementary behaviors in terms of performance, with SH providing the best performance at low frequency (around the body resonance), and ADD ensuring optimality at mid and high frequency (beyond the body resonance). Therefore, a mixed control law has been proposed in the past using a very simple but effective frequency range selector \((\ddot{z}_b - a^2 \ddot{z}_S^2)\). According to the sign of the main condition, either the ADD strategy is selected (mid and high frequency dynamics), or the SH strategy is used. The equation describing this control law is:
\[ C_i = \begin{cases} C_{\text{max},i} & \text{if } (\dot{z}_S^2 - a^2 \dot{z}_G^2) \leq 0 \text{ and } \dot{z}_{ST,i} \dot{z}_G > 0 \text{ or } (\dot{z}_S^2 - a^2 \dot{z}_G^2) > 0 \text{ and } \dot{z}_{ST,i} \dot{z}_G > 0 \\ C_{\text{min},i} & \text{otherwise} \end{cases} \]  \tag{5}

The parameter \(a\) represents the frequency limit between the low and the high frequency ranges. Specifically, the value of \(a\) is set at the crossover frequency (in rad/s) between SH and ADD. This law is almost optimal as it provides a mix of the best performance of the SH and ADD.

2.2.6. Mixed Skyhook-Acceleration driven damper control with 1 sensor (SH-ADD-1s). The current law is similar to the mixed Skyhook-ADD, being based upon the complementary performances of soft and hard passive suspensions. Hard suspension is able to damp optimally the body resonance, but without a desiring filtering at high frequency. On the contrary, a soft suspension ensures the best filtering but with the drawback of a poorly damped body resonance. This complementarity resembles the complementarity between the SH and the ADD, described before, however the main difference is that at mid frequency the passive performance is far from the one achievable by the best possible semi-active suspension. The law is described by the following equation:

\[ C_i = \begin{cases} C_{\text{min},i} & \text{if } (\dot{z}_S^2 - a^2 \dot{z}_G^2) \leq A_i \\ C_{\text{min},i} & \text{if } (\dot{z}_S^2 - a^2 \dot{z}_G^2) \geq A_i \\ C_{\text{max},i} & \text{if } (\dot{z}_S^2 - a^2 \dot{z}_G^2) < A_i \end{cases} \]  \tag{6}

According to the current value of \(\dot{z}_S^2 - a^2 \dot{z}_G^2\), the soft damping condition is selected when it is positive, otherwise the hard-damping condition is used. Similarly to the mixed SH-ADD, the amount \(\dot{z}_S^2 - a^2 \dot{z}_G^2\) can be considered as a simple “frequency-range selector”, where the parameter \(a\) represents the limit between the ranges of low and high frequency.

3. Road excitation
In order to study the dynamic behavior of the vehicle an excitation for the model is required. In this study, a sinusoidal function was used as excitation representing a road bump, as it is shown in Figure 3. The geometrical aspects of the bump were chosen as follows: its height was set to 0.05m and its length was set to 5m. The velocity of the vehicle was considered constant, at 8 m/s. Finally, a time lag between the front and rear wheel of the vehicle was applied. Specifically, the two wheels follow the same trajectory with a time delay due to the distance \(a_F + a_R\) between them (\(t_{\text{distance}} = \frac{a_F + a_R}{V}\)).

![Figure 3. Road Bump.](image)

4. Optimization Procedure
A multi objective optimization was applied to the model in respect to ride comfort and road holding using the Pareto front for several constraints. As design variables of the optimization, the controllable dampers and the spring stiffness were selected, as shown in equation 7:

\[ \text{design variables} = [C_{\text{min}, F}, C_{\text{max}, F}, C_{\text{min}, R}, C_{\text{max}, R}, K_F, K_R] \]  \tag{7}

where \(C_{\text{min}, F}, C_{\text{max}, F}, C_{\text{min}, R}\) and \(C_{\text{max}, R}\) were the minimum and maximum achievable damping coefficient of the front and rear damper, respectively, according to the semi active control law that is in use (Equations 1-6). Additionally, in semi active control laws SH-L, SH-ADD and SH-ADD-1s the
tuning coefficient a was added as a design variable. The bounds of the design variables are illustrated in Table 3.

Moreover, the optimization of the passive model was carried out for two different bounds for the damping coefficient of the front and rear suspension. At first, the bounds of the minimum damping coefficient of the semi active laws were used (PS-LP) and then the ones of the maximum damping coefficient were applied to the design variables (PS-UP). The bounds of the springs’ stiffness were the same. The objective functions of the optimization were selected so as to improve both the ride comfort and the road holding. Therefore, two targets were chosen: the root mean square of the seat’s acceleration (f1), representing the ride comfort, and the average of the variances of both the front and rear tire deflections (f2), representing the road holding (Table 3). The constraints used in the optimization problem were selected so as to incorporate the practical considerations into the design process, such as the working space of the suspension system, as well as to enhance the optimization targets (Table 4).

Table 3. Design variables, bounds, objectives and constraints.

| Design Variables | Lower Bounds | Upper Bounds | Objectives | Constraints |
|------------------|--------------|--------------|------------|-------------|
| Cmin, F, Cmin, R (Ns/m) | 500 | 2500 | f1=rms( z), 0.127 m ≤ z | zST, ≤ 0.127 m |
| Cmax, F, Cmax, R (Ns/m) | 2500 | 5000 | f2= 1/2(var(zF)+var(zR)), 0.051 m ≤ z | zT, ≤ 0.051 m |
| KF, KR (N/m) | 15000 | 70000 | | z ≤ 0.07 m |
| a (SH-ADD, SH-ADD-1s) (rad/s) | 10 | 60 | | max( z), ≤ 4.5 m/s² |

5. Results

The Pareto fronts of all the aforementioned cases are illustrated, in Figure 4. It is obvious that the Pareto fronts are located in three different areas. In the first area (A1) lay the Pareto fronts of four cases: SH-2, SH-L, SH-ADD and PS-UB, in the second one (A2) lays the Pareto front of SH-ADD-1s, while the Pareto fronts of ADD, PDD and PS-LB lay in the third area (A3) which is dominated by the previous ones, providing the “less optimal” solutions. The results belonging in A1 reach the lowest values of f1 (objective for ride comfort), whilst, in all the studied cases, regardless area, f2 (objective for road holding) is in same range, apart from the PS-UB which has achieved the highest road holding of all cases (minimum value of f2).

Figure 4. Comparison of the optimal solutions of all the cases

As it can be observed, in Figure 4, all the Pareto fronts of the semi active suspension laws are within the area created by the Pareto Front of PS-UB (black) and PS-LB (cyan), proving the main idea of semi active suspensions, which is the combination of a stiff and a soft damper. Another issue that should be mentioned is the difference between the Pareto of the passive case studies (PS) compared to the one of the semi-active case studies (SA). All the SA Pareto fronts reveal a different behavior during their attempt to increase the road holding of the vehicle. Instead of following a common trajectory like the Pareto of PS, which reveals the well-known conflict between ride comfort and road holding (increase of
the one leads to the decrease of the other), they are demonstrating a vertical “jump off” for $f_2$ values (circled areas in Figure 4) and then, the Pareto front becomes almost parallel to the axis of $f_1$. This behavior illustrates the inability of the SA cases to further improve the road holding after a certain point, which is acceptable considering the fact that the applied control laws are comfort oriented.

The optimal design variables delivered from the Pareto front are illustrated in Figure 5. For each control law (Figures 5a-5f), eight curves of C-K are plotted. The four of them illustrate the optimal solutions of cases PS-LB and PS-UB ($C_F$-$K_F$ and $C_R$-$K_R$ for each case), while the other four demonstrate the optimal solutions of the SA cases ($C_{Fmin}$-$K_F$, $C_{Fmax}$-$K_F$, $C_{Rmin}$-$K_R$ and $C_{Rmax}$-$K_R$). Each plot is divided in two areas with a red line and in the left one $C_{Fmin}$-$K_F$ and $C_{Rmin}$-$K_R$ are compared with the results of PS-LB, whilst in the right one $C_{Fmax}$-$K_F$ and $C_{Rmax}$-$K_R$ with the results of PS-UB. A “jump off” similar to the one of the Pareto fronts is again obvious. All the curves concerning the front suspension system of SA cases, shape an “L” regarding the values of C and K. This L-shape means that the optimization has concluded in various values of K for a specific value of C (vertical part of the L-shape), as well as in various values of C for a specific value of K (the horizontal part of the L-shape). The fact that the L-shape appears in the front suspension system, validates the necessity of using different suspension systems to the front and rear axles. On the other hand, in all the SA cases, the optimal solutions of the rear suspensions system have an almost fixed value for the spring stiffness and many alternatives for $C_{Rmin}$ and $C_{Rmax}$, indicating a convergence to the optimization procedure.

Finally, some important metrics for the vehicle dynamics are illustrated in Table 4. The metrics calculated here are the root mean square value of the seat’s acceleration, the front and rear suspension

| # | Metric | SH-2 | SH-L | ADD | PDD | SH-ADD | SH-ADD-1s | Pass lb | Pass ub |
|---|--------|------|------|-----|-----|--------|-----------|---------|---------|
| 1 | RMS (z_s)[m/s^2] | 0.293 | 0.318 | 0.359 | 0.352 | 0.301 | 0.334 | 0.373 | 0.311 |
| 2 | $z_{ST, F}$[m] | 0.019 | 0.019 | 0.023 | 0.026 | 0.019 | 0.017 | 0.024 | 0.015 |
| 3 | $z_{ST, R}$[m] | 0.019 | 0.020 | 0.017 | 0.023 | 0.022 | 0.011 | 0.021 | 0.013 |
| 4 | VDV head | 5.727 | 6.001 | 6.342 | 6.487 | 5.637 | 6.077 | 6.545 | 6.085 |
| 5 | $C_F$head | 3.691 | 3.715 | 3.544 | 3.580 | 3.607 | 3.540 | 3.456 | 3.340 |
| 6 | $z_{T, F}$[m] | 0.004 | 0.004 | 0.005 | 0.005 | 0.004 | 0.004 | 0.004 | 0.003 |
| 7 | $z_{T, R}$[m] | 0.002 | 0.002 | 0.003 | 0.003 | 0.003 | 0.002 | 0.003 | 0.002 |
travel, the vibration dose value (VDV) and the crest factor of the head [2] and the front and rear tire deflection. The values included in Table 4 are the mean values of the optimal solutions in order to present the mean dynamic characteristics of all the solutions delivered from the Pareto front. The current comparison validates the conclusions stated previously. ADD and PDD do not deliver “good optimal” results, being unable to improve both the passengers’ ride comfort (line 1 of Table 4) or the road holding of the vehicle (lines 6 and 7 of Table 4). Whereas, SH-2, SH-ADD and SH-ADD-1s have successfully converged to solutions either equal or better than the ones of PS-UB, which is the main reference case.

6. Conclusions

To sum up, the main aim of the current work was to present optimized semi-active suspensions along with optimized passive ones in order to compare them both in terms of the optimization results and the dynamic characteristics of the solutions. Moreover, in this work a different approach was proposed regarding the conditions used in the formulas of the control laws, so as to minimize the unnecessary changes in the damper’s state and save energy for the system. Additionally, during the multi-objective optimization applied to the problem, remarks were illustrated regarding the well-known conflict of the suspension system and the attempt of the semi-active suspension systems to overcome it. Future work is in progress not only to expand this work for semi active control laws which are road holding oriented but also to implement decision algorithms to the Pareto solutions, so as to figure out the importance of the “jump” mentioned in the previous sections.

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