

Efficiency of On-Off Semiactive Suspensions in a Pick-up Truck

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ABSTRACT

A comparative analysis between three different semi-active suspension control techniques in a full vehicle model of a pick-up truck is presented. Each independent corner of the vehicle uses a Magneto-Rheological (*MR*) damper model. The *MR* damper model includes nonlinearities, hysteresis, and transient response between the manipulation and actuation. The damper model parameters have been obtained from experimental data provided from a BWI™ *MR* damper. Different tests were used to evaluate the semi-active control strategies by using the CarSim® software. The evaluated controllers, which are based on switching manipulation between the low and high damping force, are: 1) the hybrid controller, 2) Mix 1-sensor (Mix-1) and 3) Frequency Estimation-Based (FEB) controller. These selected controllers share the next features: the use of few sensors, free of models, and they do not require anti-saturation mechanisms. Simulation results showed that the FEB controller had the best performance for passengers comfort in the bounce sine sweep test (9% less of movement in the vehicle body respect to the passive suspension system), and better performance, in conjunction with the Mix-1 controller, for road holding and roll control in the double lane change test. Finally, the FEB and Mix-1 control strategies reduce up to 2 degrees the vehicle slipping in the fishhook test, in comparison to the commercial suspension system of the pick-up truck.

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INTRODUCTION

The main function of an automotive suspension system is to provide comfort to the passengers by isolating the vehicle motion from the road irregularities, and to ensure the road holding and handling in any driving condition. According to the level of damping force, the automotive suspension systems can be classified in passive, semi-active and active. The passive suspension systems are the most used in commercial vehicles, the damping properties are designed mechanically; however, the constant damping force can have many limitations for ensuring the comfort and safety under risk driving conditions.

On the other hand, the semi-active and active suspension systems can regulate the vehicle motions through a control system over the damping force. The framework of this kind of suspensions can be magnetic, hydraulic or pneumatic. The active suspension achieves a better comfort performance but it is necessary to have an external power system for supplying/dissipating energy from the tire and chassis motions; its practical implementation is more expensive than a semi-active suspension system. Thus, a semi-active damper allows to control the force for reducing the relative movement between the sprung and unsprung masses in a

vehicle corner; however, the power requirement is lower than an active damper. Commercially, there are two types of semi-active dampers: mono-tube and double-tube, the mono-tube dampers are the most used; the Magneto-Rheological (*MR*), electro-hydraulic and pneumatic dampers are commercial mono-tube dampers. The main advantages of the *MR* damper versus the other semi-active dampers are: (a) fast time response (20-40 ms) and, (b) long bandwidth of control.

The *MR* damper is a non-linear device with dissipative capability used in an Automotive Suspension Control System (*ASCS*), where the damping coefficient varies according to the applied electric current. This damper allows to achieve good comfort, to emphasize road holding and to keep a safety suspension deflection. In an experimental evaluation, it has proved that the semi-active dampers, with up to 25 operating states, only activates two states for an average of 92% of the driving time [1], concluding that an on-off control strategy offers sufficient potential benefit to evaluate the semi-active property. In the practical side, the model-free on-off controllers Sky-Hook (*SH*) technique [2], hybrid technique [3], Mix-1-sensor (Mix-1) strategy [4], and Frequency Estimation-Based (*FEB*) control [5] have and efficient performance in comfort. The hybrid controller is an improvement of the *SH* control because the Ground-Hook

(GH) control strategy is included to ensure the road holding objective. Although, the Mix-1 and FEB controllers have been evaluated in a Quarter of Vehicle (QoV) model, they offer a good alternative to improve the passengers comfort in a full vehicle.

In semi-active suspension control for vehicles, many other researchers are focused to improve the vehicle comfort, ensuring its safety and stability. The nonlinear control techniques such as Linear Parameter-Varying (LPV) control [6, 7] and sliding mode control [8, 9] offer interesting simulating results where two or more control objectives are included into the control law, but their implementation can be difficult. Robust control solutions are also proposed to improve the vehicle comfort and lateral stability [10, 11, 12], but the performance of the control system depends on a reliable model with the low possible model uncertainties; similarly, the performance of optimum controllers [13] strongly depends on the reliability of the full vehicle model. Recently, heuristic control systems are proposed to control the comfort in the heave, pitch and roll movements [14, 15] by using two control levels, the internal controller is used to regulate the damping force, while the master controller includes the coupling features among the four QoV subsystems; from the practical point of view, the proposal in [14] is more feasible to implement where the internal controllers are on-off. Almost all of these control strategies have been analyzed for small or medium-size vehicles, and only a pick-up truck was used as test-bed in [13].

The aforementioned on-off control strategies (hybrid, Mix-1 and FEB) have shown an efficient performance in comfort under different simulation/experimental tests; however, it is not possible to establish a comparison among these approaches. On the other hand, the direct effect of the damping force in each corner to the vehicle motion is not analyzed. This paper proposes a comparative analysis among the hybrid, Mix-1 and FEB controllers under the same driving conditions by using a pick-up truck model of CarSim® which is created by the parameters of a commercial light truck vehicle. The Root Mean Square (RMS) index is used to quantify the driving performance in each control strategy. The suspension system includes four experimental MR pick-up damper models. Each MR damper is controlled in independent form.

DESIGN OF ON-OFF CONTROLLERS

HYBRID CONTROLLER

This control strategy has been implemented in several test-rigs. It is a strategy of two control objectives: comfort and road holding. For this, the hybrid controller combines the SH control technique for comfort [16] and the GH control law for road holding [17]. The SH controller is focused on minimize the vertical acceleration of the sprung mass, the damping force defined by the SH controller is:

$$F_{sky} = \begin{cases} C_{sky}\dot{z}_s & \text{if } \dot{z}_s(\dot{z}_{def}) \geq 0 \\ C_{min}\dot{z}_{def} & \text{if } \dot{z}_s(\dot{z}_{def}) < 0 \end{cases} \quad (1)$$

where, $\dot{z}_s$ is the vertical velocity of the sprung mass, $\dot{z}_{def}$ is the deflection velocity between the sprung and unsprung masses and C_{sky} and C_{min} are the *sky* and minimum damping coefficients, respectively. Its well known that SH improves the passengers comfort but it has a poor road holding performance. Thus, its counterpart GH is proposed to reduce the dynamic forces between the wheel tire and the road surface. The control law is:

$$F_{ground} = \begin{cases} C_{ground}\dot{z}_{us} & \text{if } -\dot{z}_{us}(\dot{z}_{def}) \geq 0 \\ C_{min}\dot{z}_{def} & \text{if } -\dot{z}_{us}(\dot{z}_{def}) < 0 \end{cases} \quad (2)$$

where, $\dot{z}_{us}$ is the vertical velocity of the unsprung mass and C_{ground} is the *ground* damping coefficient which corresponds to a design parameter. When the SH and GH techniques are combined, the damping force oriented to comfort and road holding are weighted. Equation (3) defines the hybrid control law.

$$F_{hyb} = \lambda F_{ground} + (1 - \lambda)F_{sky} \quad (3)$$

where, λ represents the weighting factor between the SH and GH damping forces. In general, three design parameters are required for each QoV model. As the controller output is force, a look-up table of force versus electric current is required.

MIX-1-SENSOR CONTROLLER

The Mix-1 controller has experimental validation in a vehicle of two wheels [4]. It is a control strategy that only demands one sensor in the control system of a QoV: an accelerometer on the sprung mass. The control law selects, at the end of each sampling, the maximum or minimum damping coefficient according to the frequency of the vertical movement of the sprung mass, i.e. the term $(\ddot{z}_s^2 - \alpha^2 \dot{z}_s^2)$ is used as an instantaneous approximation in the time domain of the frequency range. The control law is defined as,

$$C_{Mix-1} = \begin{cases} C_{max} & \text{if } (\ddot{z}_s^2 - \alpha^2 \dot{z}_s^2) \leq 0 \\ C_{min} & \text{if } (\ddot{z}_s^2 - \alpha^2 \dot{z}_s^2) > 0 \end{cases} \quad (4)$$

where, $\ddot{z}_s$ is the vertical acceleration of the sprung mass and C_{max} is the maximum damping coefficient in the semi-active damper. The value of α is defined by the invariant points of the QoV model, if α is associated to the resonance frequency of the sprung mass the Mix-1 controller is oriented to comfort; while, if α takes the value of the resonance

frequency of the unsprung mass, the control objective will be road holding. Additionally, the α can take the value of the cut frequency between the soft and hard damping force for weighting the comfort and road holding objectives. Since the controller output is a damping coefficient, the damping force is $F_{Mix-1} = C_{Mix-1}\dot{z}_{def}$, and a look-up table (force - electric current) is required. Thus, only one design parameter is required for each QoV model.

FEB CONTROLLER

This control strategy is based on the idea of monitoring the excitation frequency in the suspension for applying the required electric current into the semi-active damper in order to achieve the desired objective control [5]. Since the comfort and road holding conditions depend on the displacement frequency f_{zr} in the suspension of a QoV, the FEB controller is multi-objective. In order to meet the specified performances, for a given f_{zr} corresponds a damping coefficient ζ . Hence, by having a *measurement* or *estimation* of f_{zr} is possible to assign specific values of ζ according to a pre-analysis of the frequency responses in the performances objectives.

In a QoV system with an MR suspension, the damping varies according to the applied electric current $\zeta_{MR}(I)$. Hence, it is possible to hold a specific suspension condition based on an estimated frequency f_{zr} and applying an electric current. Using the measurements of the deflection acceleration and deflection velocity, it is possible to estimate the road profile frequency and to decide which damping to apply in order to achieve comfort and/or road holding conditions. By using the RMS values of the deflection velocity and deflection acceleration of the MR damper piston, the estimation of the frequency is given by,

$$\hat{f}_{zr} = \sqrt{\frac{(\dot{z}_{def1}^2 + \dot{z}_{def2}^2 + \dots + \dot{z}_{defn}^2)}{(\dot{z}_{def1}^2 + \dot{z}_{def2}^2 + \dots + \dot{z}_{defn}^2) \cdot 4\pi^2}} \quad (5)$$

where, n is the number of samples to compute the RMS value, using a sample time of 1/512, the size of window is 256 samples. The manipulation corresponding to a given frequency is obtained observing the frequency response function for the controlled variables, defining a lookup table in order to assure the desired performances for comfort and road holding.

MR DAMPER MODELING

Several mathematical models are available for modeling the nonlinear behavior of MR dampers; an exhaustive comparison among different model structures is presented in [18, 19, 20] by highlighting the advantages and modeling characteristics of each approach. Generally, the MR damper models can be grouped as parametric and non-parametric. In the non-parametric models, the coefficients do not have a physical meaning and they can reproduce the nonlinear

behavior of an MR damper with high accuracy like an artificial neural network approach; however, for its complex structure, the computing time can be higher than the parametric models. On the other hand, some parametric models have the limitation of representing the hysteretic behavior of MR dampers, e.g. the Bingham model or the viscoelastic-plastic model. In [21] a novel parametric model is proposed; this algebraic model describes the hysteretic behavior of the MR damper with an inverse tangent function by adding an inertial force term which depends on the acceleration of the piston in order to track the MR force at the highest excitation velocities. The MR damping force of this algebraic model, as function of the electric current (I), is given by:

$$F_{MR} = f_0 + c_b I \dot{z}_{def} + \frac{2}{\pi} f_y \tan^{-1}\{k[\dot{z}_{def} - \dot{z}_0 \operatorname{sgn}(\ddot{z}_{def})]\} + m \ddot{z}_{def} \quad (6)$$

where f_0 , c_b , f_y , k , $\dot{z}_0$ and m are unknown parameters to be identified. Several experiments have been proposed to explore the nonlinear behavior at different displacement frequencies with different electric current sequences [22]. Figure 1 shows the modeling performance of the algebraic model reported in [21] expressed in the characteristic force-velocity plot; the model is compared with experimental data provided from an MR damper manufactured by BWI™. In this case, the MR damper only has two levels of actuation: low and high damping force at 0 and 2.5 amperes, respectively. The force-velocity diagram shows that the MR damper is not symmetric, the span of the force is $[-6000$ to $11000]$ N; the damper stroke is approximately ± 30 mm.

For the simulations, four MR damper models are considered for representing the semi-active suspension system by using the eqn. (6).

SIMULATION TESTS

The evaluation of the different vehicle controllers using four independent ASCS, one for each corner, is focused in ride comfort, handling and stability.

The CarSim® software is the software in the loop. In this software, a commercial Pick-up vehicle is simulated according to the parameters in Figure 2.

The rear unsprung mass is 280 Kg and the front one is 163 Kg, the maximum stroke in each corner is 300 mm and the stiffness of each front corner is 43.5 N/mm and 39 N/mm for each rear corner.

A bump of 10 cm at 20 Km/h is implemented to compare the different semi-active control strategies versus the passive suspension. The force-velocity curves of the passive dampers are obtained experimentally from the baseline dampers of the pick-up. Posteriorly, the ride comfort is evaluated with a Bounce Sine Sweep Test (BSST) with a vehicle speed of 30 Km/h. The handling is evaluated using a Double Lane Change (DLC) test at 120 Km/h. The lateral stability is observed with a FiShHook (FSH) test at 80 Km/h. Other two tests are done considering normal driving situations: rough