2013 European Control Conference (ECC)
July 17-19, 2013, Zürich, Switzerland.

Suspension Control Strategy for a Fully Electrified Vehicle *

Juan C. Tudón-Martínez[1], Sébastien Varrier[2], Ruben Morales-Menendez[1] and Ricardo Ramirez-Mendoza[1]

Abstract— **The design of chassis for a truck considers a wide set of conditions in which load and unload are the most typical variety. For years, flat ride rules have been applied to achieve comfort, by providing tuning rules in the suspension as today. For an electric vehicle however, the weight distribution is different as a result of power train and batteries. A suspension control strategy is proposed in a full size pick-up truck using magneto-rheological dampers. As a first principle, the estimation of the frequency of the road profile allows the control of comfort and road holding in normal driving situations (straight roads). This strategy includes the coupling joints among the four-wheel stations of the truck and it allows the control of vertical forces in order to minimize the chassis vertical acceleration. A second principle, it uses the continuous measurement of the steering wheel to configure the four semi-active damping forces in order to reduce the wheel tramp, wheel hop, front-end dive, and other causes of vehicle instability under risk driving conditions. Simulation results in CarSim^{TM} of a full prototype show that the suspension performance of the new fully electrified truck is better in comfort (reduction up to 60% in the chassis vertical acceleration) and handling (reduction around 5% in the vehicle slip angle) than the original truck (before its modification).**

I. INTRODUCTION

Advanced technologies in automotive suspension systems have been developed to increase the comfort and safety in vehicles, mainly for sedan cars in recent years. For pick-up trucks, the load transfer into the four wheel-stations depends on the chassis design for the weight distribution; thus, semi-active or active dampers are a good solution to control the chassis motion when the vehicle is driven over different road profiles. Moreover, when the vehicle has an electrified power train system, the batteries weight modifies the weight distribution by affecting the sensitivity to road irregularities.

By comparing the actuation system between semi-active and active shock absorbers, the active ones have better isolation performance, but they are more expensive because an external supply power system is required; while, semi-active dampers are cheaper with acceptable performance. There exist different semi-active damping technologies; the main advantages of Magneto-Rheological (*MR*) devices are: 1) long bandwidth of control and 2) fast time response.

During last decades, several approaches of active and semi-active suspension control have been proposed for a full vehicle, some of them based on non-linear [1], [2], [3], [4], robust [5], [6], [7] or optimal [8] control theory. These techniques offer attractive results when there is an accurate vehicle model, or there exist enough measurements that could describe the non-linear dynamics for designing an observer/estimator. On the other hand, *On-Off* controllers, e.g. sky-hook, ground-hook, acceleration driven damper, etc., represent the easiest solution to control the damping ratio into an adaptive suspension system; but, they are based on a Quarter of Vehicle (*QoV*) model that does not analyze the inherent mechanical coupling joints between the four wheel stations and by consequence the load transfer could not be the appropriate. In [9], it is compared the comfort performance of some *On-Off* control strategies by using independent corners; the controllers are better than the passive suspension system but it is not guaranteed the best damping solution. Moreover, in [10] it is presented the compromise between comfort and road holding that some automotive semi-active suspension *On-Off* controllers have respect to advanced control methods that can manage the trade-off between both control objectives.

Some approaches are proposed to control the load transfer into the vehicle by analyzing the coupling forces between the corners [11], [12]; these heuristic methodologies are feasible to implement; however, two synchronized levels of control must be designed. Additionally, none of the aforementioned strategies include a compensation strategy for the load transfer when the vehicle is driven under risk driving situations by using the steering and/or braking signals. Moreover, most of the proposals are for medium-size vehicles.

For reducing the vertical and lateral vehicle dynamics, a novel model-free control strategy is proposed for the semi-active suspension system of an electrified pick-up truck. The proposed Combinatory quasi-Optimum Damping (*COD*) controller determines the best solution of damping force for the four semi-active dampers by using a monitoring module in the suspension and stability system. The suspension monitoring system analyzes the vertical and rotational motion (pitch and roll) in the sprung mass as well as the vertical vibrations in each wheel; and for the stability system, the lateral (sideslip angle, yaw and yaw rate) behavior of the vehicle is analyzed. The main goal is to minimize the chassis motion and ensure stability in any driving situation. The best damping combination is based on bandwidths; thus, a frequency estimation module of the road profile based on a robust $\mathscr{H}_\infty$ observer is considered.

The outline of this paper is as follows: in the next section, the semi-active suspension system is described. Section III presents the structure of the proposed Combinatory quasi-Optimum Damping (*COD*) controller. Section IV presents

* This work is partially supported by the CONACyT: PCP project 03/2010 (France - México).

[1]J.C. Tudón-Martínez, R. Morales-Menendez and R. Ramirez-Mendoza are with Tecnológico de Monterrey, Av. E. Garza Sada 2501, 64849, Monterrey N.L., México {jc.tudon.phd.mty, rmm, ricardo.ramirez}@itesm.mx

[2]S. Varrier is with the Gipsa-Lab, 11 rue des mathématiques, 38402 Grenoble, France sebastien.varrier@gipsa-lab.fr

978-3-952-41734-8/©2013 EUCA

the simulation test and discusses the results. Finally, conclusions are presented in section V.

II. SEMI-ACTIVE SUSPENSION SYSTEM

The semi-active suspension system in the truck is considered by two independent wheel stations in the front track and a solid axle in the rear one. Each corner contains an *MR* damper model to control the motion into the vehicle body. Experimental data are used to model a commercial automotive *MR* damper (BWI^{TM}) that only has two levels of actuation by mechanical design: low damping force at 0 A and high damping force at 2.5 A. Figure 1c shows that the *MR* damper has a not symmetric behavior in the jounce-rebound effects, whose range of force is [−6,000 to 11,000] N (peak to peak). The damper stroke is ± 50 mm with a time constant of 12 ms and settling time of 72 ms.

For simplicity of computing, few number of parameters and good modeling performance, the *MR* damper model used for this approach is an extension of [13], but with inclusion of the manipulation signal (electric current, *I*). Its non-linear dynamics can be modeled by the damping force F_{MR} as:

$$F_{MR} = I f_c \tanh\left(a_1 \dot{z}_{def} + a_2 z_{def}\right) + b_1 \dot{z}_{def} + b_2 z_{def} \qquad (1)$$

where, z_{def} and $\dot{z}_{def}$ are the position and velocity of the damper piston. By considering a standard test (*chirp*) of 14 *mm* of amplitude from 0.5 to 15 *Hz* at 0 A and 2.5 A, Fig. 1a-b; the coefficients in the jounce effect (positive $\dot{z}_{def}$) were $fc = 1,322.3$, $a_1 = 4.9$, $a_2 = 7.5$, $b_1 = 6,429.2$ and $b_2 = 0$, while for the rebound effect were $fc = 551.0$, $a_1 = 38.9$, $a_2 = 22.2$, $b_1 = 3,230.9$ and $b_2 = -5,897.2$.

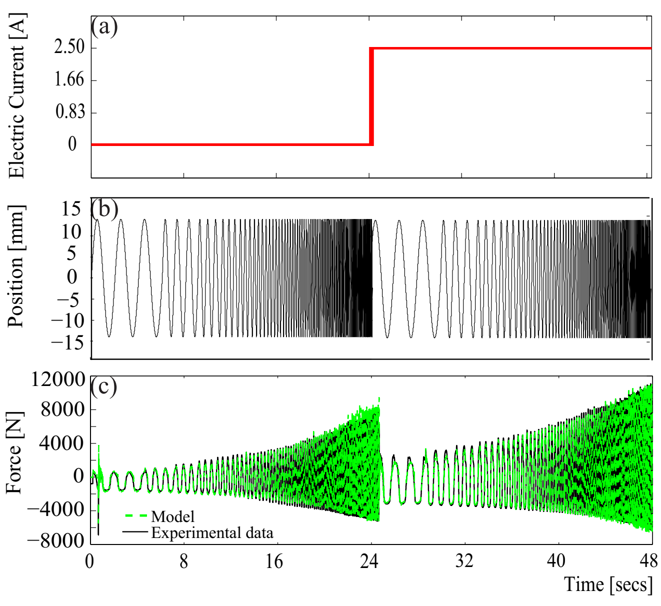

Fig. 1. Comparison between experimental data and the *MR* damper model.

III. SUSPENSION CONTROL STRATEGY

The proposed suspension control strategy is divided in two parts: 1) an off-line analysis in the frequency domain by using a suspension and stability monitoring system and 2) an on-line algorithm that computes the best damping combination (among the four *MR* dampers) according to the driving conditions. Figure 2 shows the interaction between the off and on-line task in the proposed suspension controller.

A. Suspension monitoring system

The suspension monitoring system analyzes the vertical dynamics of the chassis (sprung mass acceleration $\ddot{z}_{s_c}$, heave motion z_{s_c} and roll and pitch angles) and wheels (suspension deflection and tire compression at each corner) in the frequency domain, when the car is driven in a straight road profile whose roughness has enough frequency contents. The main idea is to analyze the suspension behavior in all damping combinations in order to detect the best damping solution at each frequency of vibration that the car is exposed.

Because the used experimental damper only has two states of actuation, it is ideal to design an heuristic *On-Off* controller. By considering the full suspension system in the pick-up truck, there are 16 combinations of actuation among the four *MR* dampers, e.g. the configuration $(0,0,0,0)$ means that the four *MR* dampers have an electric current value of 0 A, $(1,1,1,1)$ means that all dampers have 2.5 A, $(1,1,0,0)$ means that the front dampers are on (2.5 A) and the rear dampers are off (0 A), and so on.

A Bounce Sine Sweep (*BSS*) test (sinusoidal road signal with decreasing amplitude and increasing frequency) at constant vehicle velocity is used to monitor the semi-active suspension system with the 16 combinations of damping, in the whole range of frequencies of interest in an automotive application. The result of the suspension monitoring system is to determine the best damping combination in bandwidths.

1) Quasi-optimum damping combination: The suspension monitoring system defines the quasi-optimum damping combination in terms of a comfort and road holding cost function. Since comfort and road holding have a trade-off performance, based on the hybrid control strategy [14], the *COD* controller selects the best damping solution by weighting the comfort and road holding control goals as:

$$X\,|_{F_{MR_i}} = \lambda \cdot x\,|_{comf} + [1-\lambda] \cdot x\,|_{rh} \qquad (2)$$

where, $X\,|_{F_{MR_i}}$ is a cost function between the comfort and road holding in the i_{th} damping combination. According to [10], the weighting parameter λ is an optimization variable that can be obtained by a convex combination of the comfort index ($x\,|_{comf}$) and road holding index ($x\,|_{rh}$), by using the passive suspension system as reference:

$$x\,|_{a} = \frac{1}{N}\sum_{b=1}^{N} \frac{Var_b(MR_{i_{th}})}{Var_b(passive)} \qquad (3)$$

where, $a = \{comf, rh\}$ and Var is the frequency response of the variable of interest during the test; thus, $x\,|_{comf}$ is the average index among the roll, pitch, heave and vertical acceleration ($N = 4$) by including the vertical force coupling among the 4-wheel stations and $x\,|_{rh}$ is the average among the suspension deflection and tire compression of each corner ($N = 8$). $MR_{i_{th}}$ refers to the i_{th} damping combination under analysis and *passive* refers to the original suspension system by using a set of passive dampers.

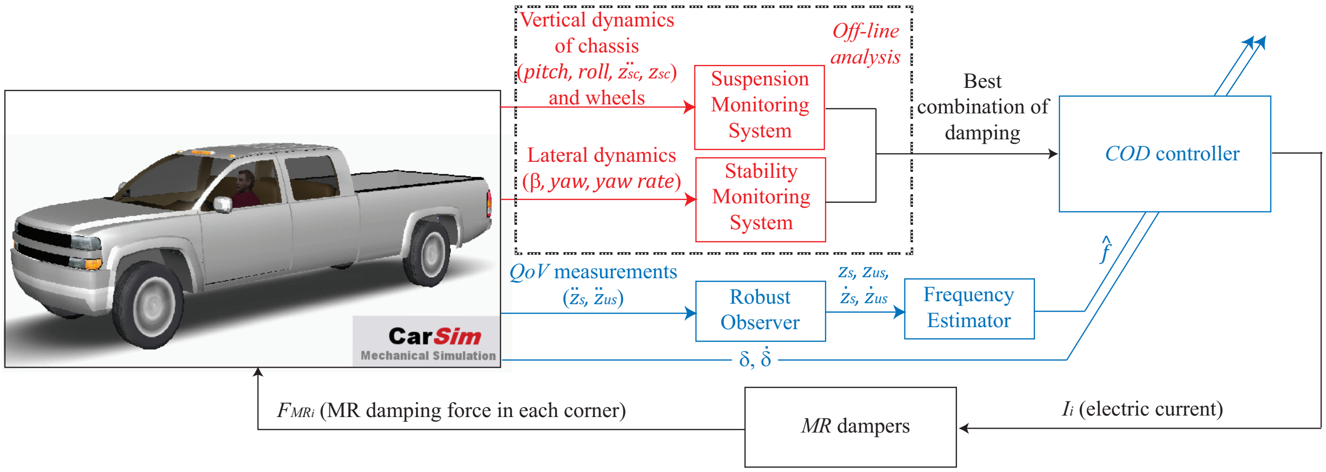

Fig. 2. Block diagram of the Combinatory quasi-Optimum Damping (*COD*) controller.

Once λ is obtained, the quasi-optimum damping combination will be the configuration that minimizes the eq. (2). Because, the performance of the semi-active suspension depends on the frequency of motion, it is desirable to build bandwidths of control with frequency dependence. The minimum of (2) represents the best possible solution for improving the vehicle comfort and ensuring the wheel-road contact at each frequency of excitation when the pick-up truck is driven in a straight way.

B. Stability monitoring system

Normally, when the road quality is enough (e.g. a highway) the driver achieves a high velocity ($>$ 100 Km/h) in the vehicle without care if the way is straight or not. At these conditions, a suspension monitoring system is not enough to determine the best damping combination because, at high velocities the car performs as a filter of vibrations. Thus, a stability monitoring system is used to analyze the lateral dynamics of the vehicle when this is driven in a road that demands changes in the vehicle steering.

A Double Lane Change (*DLC*) test is used to monitor the effect of the semi-active suspension system into the lateral dynamics of the vehicle in order to determine the best damping combination that increases the handling stability.

1) Quasi-optimum damping combination: The quasi-optimum damping combination used to increase the handling stability depends on the steering wheel angle (δ) and its rate ($\dot{\delta}$). Two rules of actuation have been defined when the car is driven over curve roads:

1) Damping solution when the car is turning on right ($\delta < 0$), and this maneuver is higher than a predefined steering threshold ($abs(\delta) \geq \delta_{TH}$) and occurs suddenly ($abs(\dot{\delta}) \geq \dot{\delta}_{TH}$), i.e. $abs(\dot{\delta})$ overshoots a predefined control limit.
2) Damping solution when $\delta \geq 0$ and $abs(\delta) \geq \delta_{TH}$ and $abs(\dot{\delta}) \geq \dot{\delta}_{TH}$.

The best damping solution is obtained by analyzing the lateral dynamics of the vehicle with the 16 combinations of damping, with:

$$Y\,|_{F_{MR_i}} = \frac{1}{N}\sum_{b=1}^{N}\frac{Var_b(MR_{i_{th}})}{Var_b(passive)} \tag{4}$$

where, $Y\,|_{F_{MR_i}}$ is the stability index in the i_{th} damping combination, which represents the average index among the vehicle slip angle (β), yaw angle and yaw rate signal ($N = 3$) by including the vertical force coupling among the four corners. Similarly to the suspension monitoring system, the quasi-optimum damping combination will be the configuration that minimizes the eq. (4), which represents the best possible damping solution to increase the handling stability when the pick-up truck is susceptible to a risk driving condition.

C. COD controller

Once the suspension and steering monitoring systems define off-line the best damping solution at different driving conditions, the *COD* controller determines on-line the electric current to apply in each *MR* damper according to the current driving situation. The control law is given by:

$$I_i = \begin{cases} I_i(\text{stab}) & if \quad f_{zr} \leq f_{TH} \wedge abs(\delta) \geq \delta_{TH} \wedge abs(\dot{\delta}) \geq \dot{\delta}_{TH} \\ I_i(\text{susp}) & if \quad otherwise \end{cases} \tag{5}$$

1) If the car is subjected to sudden changes of steering ($abs(\delta) \geq \delta_{TH} \wedge abs(\dot{\delta}) \geq \dot{\delta}_{TH}$), and these occur at high vehicle velocity (i.e. road frequency lower than a predefined threshold f_{TH}), the *COD* controller will be oriented to reduce the lateral dynamics. The electric current in each *MR* damper $I_i(\text{stab})$ follows the rules of actuation described on the section III-B, whose best damping solution was obtained off-line by the stability monitoring system. Steering wheel angle and its rate are assumed to be available from the *CAN* network.
2) Otherwise, when the car is driven in normal situations (without sudden changes in the steering), the *COD* controller will be oriented to the vertical dynamics. The control input in each *MR* damper is given by $I_i(\text{susp})$, that represents the configuration of electric

current associated to the best damping solution for comfort and road holding obtained by the suspension monitoring system. Because $I_i(\text{susp})$ is a *look-up* table of frequency - electric current, it is required an online estimation of the frequency of the road as a pointer.

1) Frequency estimation of the road profile: There are some approaches used to estimate the road profile by using visual sensors and others capable to compute directly the road roughness, e.g. profilometers; however, both are quite expensive for being implemented in commercial vehicles. A good alternative is an indirect estimation of the road irregularities by monitoring the suspension motion.

In [15] it is proposed a robust $\mathscr{H}_\infty$ observer to estimate the variables that monitor directly the suspension motion (deflection $z_{def} = z_s - z_{us}$ or velocity $\dot{z}_{def} = \dot{z}_s - \dot{z}_{us}$) of a *QoV* model by using two accelerometers, one for the sprung mass ($\ddot{z}_s$) and other for the unsprung mass ($\ddot{z}_{us}$). This estimation, robust to noise and un-modeled dynamics, is used to estimate the road profile signal.

Because a road profile satisfies a sum of sinusoidal waves according to the ISO 8608, and a sum of two or more sinusoidal waveforms can be obtained by the effective *RMS* value of the waves [16], the road profile frequency $\hat{f}_{z_r}$ can be estimated by the *RMS* values of the position ($z_{r_{RMS}}$) and velocity ($\dot{z}_{r_{RMS}}$) of the road as:

$$\hat{f}_{z_r} = \frac{\dot{z}_{r_{RMS}}}{2 \cdot \pi \cdot z_{r_{RMS}}} \quad \simeq \sqrt{\frac{\left(\dot{z}_{r_1}^2 + \dot{z}_{r_2}^2 + \cdots + \dot{z}_{r_n}^2\right)}{\left(z_{r_1}^2 + z_{r_2}^2 + \cdots + z_{r_n}^2\right) \cdot 4\pi^2}} \quad [Hz] \tag{6}$$

where, n is the number of samples in a time window that guarantees at least 2 cycles of the estimated frequency. The effectiveness of the road frequency estimator can be reviewed in [15].

IV. RESULTS AND DISCUSSION

The depicted control strategy is applied in order to improve the comfort and stability of a pick-up truck model, which was customized by a *K&C* test. The dynamics is generated via Carsim^{TM} vehicle simulator. The Kerb weight is 2,011 Kg, the rear unsprung mass is 280 Kg and the front one is 163 Kg, some physical dimensions are presented in Fig. 3. In a front *QoV* system the sprung mass is 630 Kg while for the rear is 387 Kg. The spring stiffness in each corner is non-linear; while the tire stiffness is considered constant: 230 N/mm.

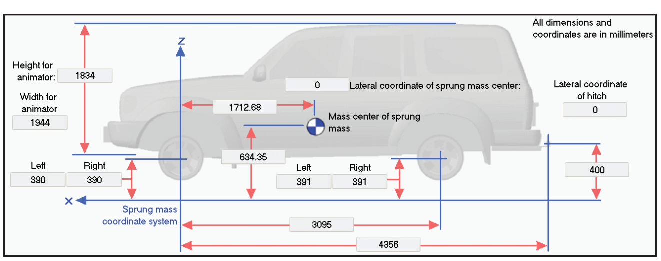

Fig. 3. Physical parameters of the vehicle in Carsim^{TM}.

For determining the best damping combination when the vehicle is driven in straight ways, a *BSS* test at 30 Km/h allows to excite the vehicle in the frequency range of interest in automotive suspensions, i.e. [0-10] Hz. Figure 4 shows the optimization result for the convex combination of comfort and road holding, eq. (2), by using the *comfort-road holding plane* [10]. Note that some frozen damping configurations improve the comfort or road holding cost function respect to the passive suspension; however, the *COD* control strategy uses the potential benefits of the semi-active suspension by reducing the comfort-road holding trade-off respect to the passive performance.

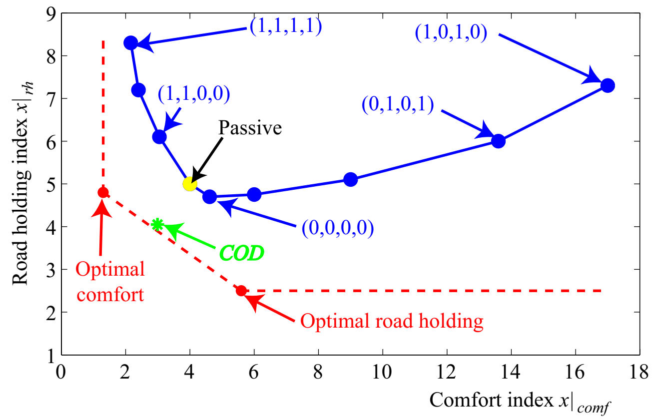

Fig. 4. Comfort-road holding trade-off for the *COD* control strategy, compared to the passive suspension system (original dampers), semi-active suspension system with frozen damping configuration (continuous blue line) and optimal comfort/road holding bounds (red dashed line).

Figure 5 shows the comfort result of the suspension monitoring system by using the comfort-road holding trade-off of the *COD* controller displayed in Fig. 4; similarly, a road holding result has been obtained. Five bandwidths of control (*BW*) have been identified in the frequency response of the vertical dynamics:

1) *BW1*.- at frequencies below than 2.4 Hz it is practically the same comfort and road holding performance in all damping combinations (similar $X|_{F_{MR_i}}$ index); thus, it is good to have the four dampers off,
2) *BW2*.- close to the frequency of resonance of the sprung mass, it is desirable to have the four dampers at high damping level for reducing the heave and pitch angle, however the road holding index is affected,
3) *BW3*.- between 3.6 and 7 Hz, the comfort-road holding trade-off is acceptable when all dampers are at 0 A,
4) *BW4*.- close to the frequency of resonance of the unsprung mass, the roll, pitch and heave motions are filtered but the sprung mass acceleration is reduced whit the configuration $(1,1,0,0)$, moreover the road holding cost function is acceptable with this damping configuration. However, the configuration $(0,0,0,0)$ is the best one for road holding; but, this increases significantly the vertical acceleration of the chassis,
5) *BW5*.- finally, at frequencies > 8.4 Hz, all chassis motions are filtered; thus, all dampers could be off.

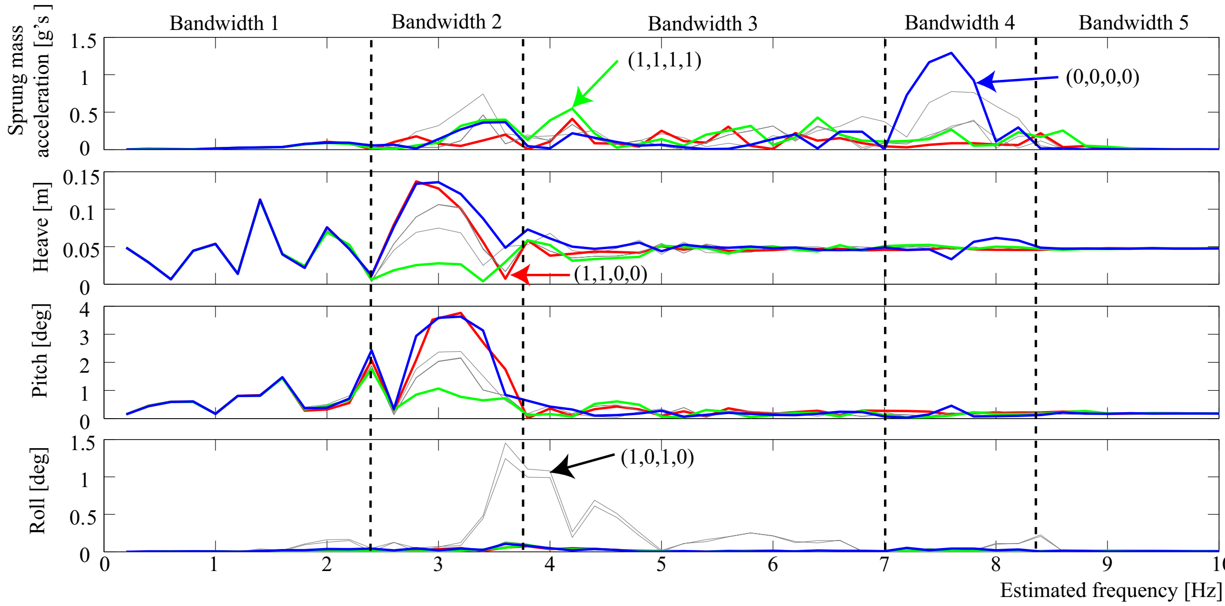

Fig. 5. Frequency response of comfort for some damping combinations. Five bandwidths of control have been selected.

By using the stability monitoring system, two damping combinations are selected to increase the handling stability when the vehicle is driven with sudden changes in the steering. Figure 6 shows the best damping solutions that minimize the Y index in the two actuation rules described in the section III-B: 1) Actuation 1.- when the steering wheel angle is negative (turn to right) and overshoots its threshold, the configuration $(0,1,0,1)$ increases the handling stability, i.e. it is desirable to have the right *MR* dampers at high level of damping to increase the vertical force in this side and help with the vehicle stability, e.g. help to an electronic stability control, 2) Actuation 2.- viceversa with the configuration $(1,0,1,0)$ when the steering angle is positive.

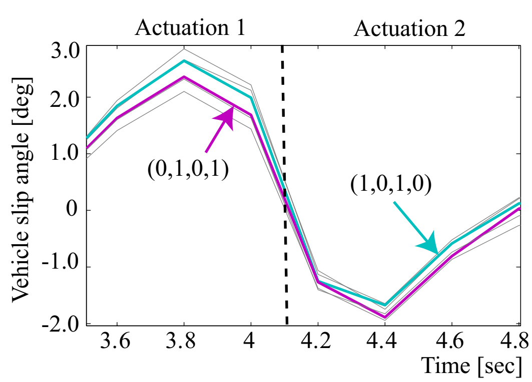

Fig. 6. Transient response of the β angle for some damping combinations. Two damping combinations are selected by analyzing the variables of interest for handling stability.

With the information obtained from the monitoring systems, the *COD* controller selects on-line the best damping combination to use according to the current driving conditions. Figures 7 and 8 show the performances of comfort and handling stability of the *COD* controller in comparison with a baseline pick-up truck model that has a passive suspension system. Clearly, both control objectives are improved with the proposed suspension control strategy, e.g. the vertical acceleration in the chassis is reduced mainly close to the frequencies of resonance of the sprung and unsprung masses, the reduction is up to 0.3 g's in the frequency of resonance of the sprung mass and up to 1 g in the frequency of resonance of the unsprung mass. On the other hand, the vehicle slip angle is reduced slightly (0.4 degrees) when the semi-active suspension control strategy is used to compensate the lateral motion of the pick-up truck, Fig. 8.

V. CONCLUSIONS

A model-free control strategy for the full semi-active suspension system of an electrified pick-up truck is proposed. The control objectives are comfort/road holding and handling stability of the vehicle. The proposed control strategy, which includes the coupling among the 4-wheel independent stations, is divided in two main tasks: 1) off-line is determined the quasi-optimum damping combination when the vehicle is driven in normal conditions or risk situations by using two monitoring systems, one for the suspension system (vertical dynamics) and other for the stability system (lateral dynamics); then, 2) the Combinatory quasi-Optimum Damping (*COD*) controller establishes on-line the damping force in each wheel station, by analyzing the current driving situation. If the vehicle is susceptible to a danger situation, the *COD* controller increases the handling stability; while, if the vehicle is driven over an irregular road, the *COD* controller is oriented to comfort and road holding. In this case, it uses a frequency estimation module of the road profile based on an $\mathscr{H}_\infty$ robust observer.

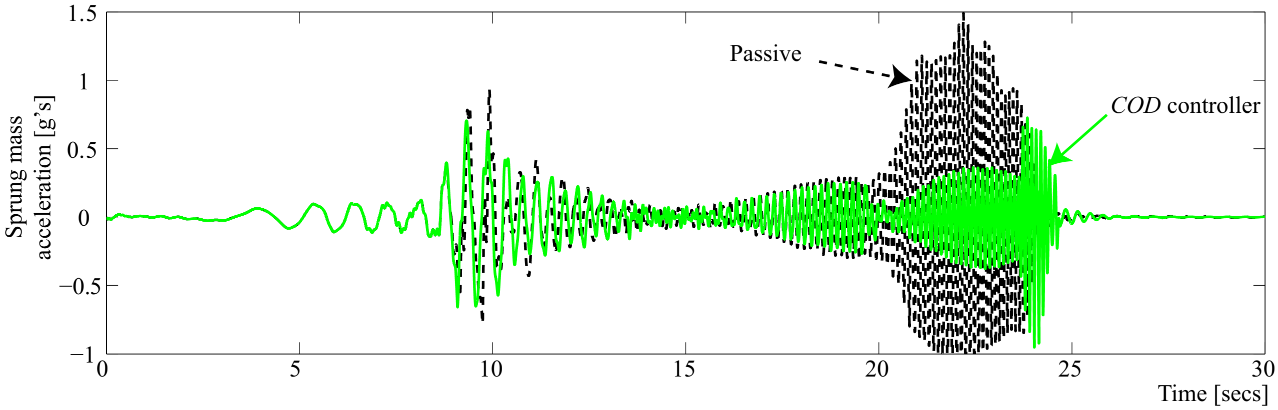

Fig. 7. Comparison of the comfort performance between the *COD* controller and the passive suspension system, by using a *BSS* test at 30 Km/h.

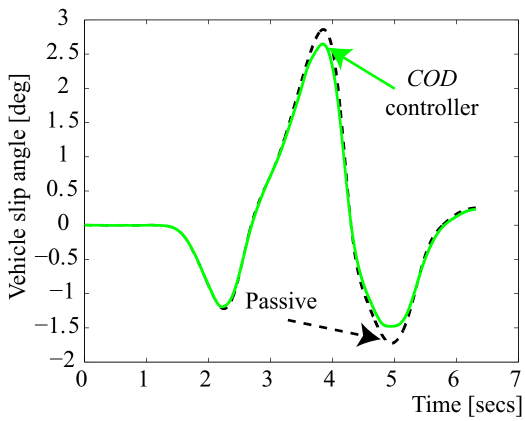

Fig. 8. Comparison of the stability performance between the *COD* controller and the passive suspension system, by using a *DLC* test at 120 Km/h.

A Magneto-Rheological (*MR*) damper model obtained from experimental data is incorporated in each corner of the vehicle; each *MR* damper has two levels of actuation. Real data of the vehicle are used in CarSim^{TM}, which is used as Software-in-the-Loop (*SiL*). Simulation results showed that the proposed controller obtained a better comfort performance in the *Bounce Sine Sweep* test, specially at frequencies close to the resonance frequency of the sprung (reduction around 20%) and unsprung (reduction around 60%) mass. In the *Double-Lane Change* test, the *COD* controller increases slightly the handling stability, the vehicle slip angle is reduced around 5%. Additionally, the simplicity and feasibility of the controller shows an attractive practical solution for being implemented into an electronic control unit of a vehicle.

REFERENCES

[1] A. Chamseddine, T. Raharijaona and H. Noura, "Sliding Mode Control Applied to Active Suspension using Nonlinear Full Vehicle and Actuator Dynamics", *Proc. of 45th IEEE Conf. on Decision and Control*, 2006, pp. 3597-3602.

[2] P. Gáspár, G. Szederkényi, Z. Szabó and J. Bokor, "The Design of a Two-level Controller for Suspension Systems", *Proc. of 17th IFAC World Congress*, 2008, pp. 3386-3391.

[3] J. Yoon, W. Cho, J. Kang, B. Koo and K. Yi, "Design and Evaluation of a Unified Chassis Control System for Rollover Prevention and Vehicle Stability Improvement on a Virtual Test Track", *Control Eng. Practice*, 18(6), 2010, pp. 585-597.

[4] C. Poussot-Vassal, O. Sename, L. Dugard, P. Gáspár, Z. Szabó and J. Bokor, "Attitude and Handling Improvements through Gain-Scheduled Suspensions and Brakes Control", *Control Eng. Practice*, 19(3), 2011, pp. 252-263.

[5] S.B. Choi, H.S. Lee and Y.P. Park, "H_∞ Control Performance of a Full-vehicle Suspension Featuring Magnetorheological Dampers", *Vehicle System Dynamics*, 38(5), 2002, pp. 341-360.

[6] J. Wang, D.A. Wilson, W. Xu and D.A. Crolla, "Active Suspension Control to Improve Vehicle Ride and Steady-state Handling", *Proc. of 44th IEEE Conf. on Decision and Control and European Control Conf.*, 2005, pp. 1982-1987.

[7] M. Chadli, A.E. Hajjaji and A. Rabhi, "H_∞ Observer-based Robust Multiple Controller Design for Vehicle Lateral Dynamics", *Proc. of American Control Conf.*, 2010, pp. 1508-1513.

[8] C. Crivellaro, "Robust Semi-Active Suspension System Control for Pickup Truck", *SAE Technical paper 2009-36-0176*, 2009.

[9] J.C. Tudón-Martínez, J. Lozoya-Santos and R. Morales-Menendez, "Efficiency of On-off Semiactive Suspensions in a Pick-up Truck", *SAE Int. J. Commer. Veh.*, 5(1), 2012, pp. 333-342.

[10] C. Poussot-Vassal, C. Spelta, O. Sename, S.M. Savaresi and L. Dugard, "Survey and Performance Evaluation on Some Automotive Semi-Active Suspension Control Methods: A Comparative Study on a Single-corner Model", *Annual Reviews in Control*, 36(1), 2012, pp. 148-160

[11] J. Swevers, C. Lauwerys, B. Vandersmissen, M. Maes, K. Reybrouck and P. Sas, "A Model-free Control Structure for the On-line Tuning of the Semi-Active Suspension of a Passenger Car", *Mechanical Systems and Signal Processing*, 21(3), 2007, pp. 1422-1436.

[12] X. Dong, M. Yu, Z. Li, C. Liao and W. Chen, "Neural Network Compensation of Semi-Active Control for a Magneto-rheological Suspension with Time Delay Uncertainty", *Smart Materials and Structures*, vol. 18, 2009, pp. 1-14.

[13] S. Guo, S. Yang and C. Pan, "Dynamical Modeling of Magneto-rheological Damper Behaviors", *J. of Intell. Mater., Syst. and Struct.*, vol. 17, 2006, pp. 3-14

[14] M. Ahmadian, "A Hybrid Semiactive Control for Secondary Suspension Applications", *ASME-ICE, Symp. on Advanced Automotive Tech.*, 1997, pp. 743-750

[15] J.C. Tudón-Martínez, S. Fergani, S. Varrier, O. Sename, L. Dugard, R. Morales-Menendez and R.A. Ramírez-Mendoza, "Road Adaptive Semi-Active Suspension in a Pick-up Truck using an LPV Controller", to appear in *Proc. of the 7th IFAC Symp. on Advances in Automotive Control*, Tokyo, Japan, 2013

[16] K.V. Cartwright, "Determining the Effective or RMS Voltage of Various Waveforms Without Calculus", *The Tech. Interface*, 8(1), 2007, pp. 1-20