

Triple-Control-Mode for Semi-Active Suspension System

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ABSTRACT

There is an increasing customer demand for adjustable chassis control features which enable adaption of the vehicle comfort and driving characteristics to the customer requirements. One of the most promising vehicle control systems which can be used to change the vehicle characteristics during the drive is the semi-active suspension system. This paper presents a Rule-Optimized Fuzzy Logic controller for semi-active suspension systems which can continuously adjust itself not only according to the road conditions but also to the driver requirements. The proposed controller offers three different control modes (Comfort, Normal and Sport) which can be switched by the driver during driving. The Comfort Mode minimizes the accelerations imposed on the driver and passengers by using a softer damping. On the other hand, the increased damping in Sport Mode provides better road holding capability, which is critical for sporty handling. The Normal Mode is adjusted to provide an overall balance between the vehicle ride comfort and road holding. The controller synthesis is performed by using an eleven degree of freedom full vehicle ride dynamics simulation model which is validated through laboratory tests performed on a hydraulic four-poster shaker. A unique optimization process is employed for obtaining the optimum Fuzzy Logic membership functions and the optimum rule-base of the proposed semi-active suspension controller. Discrete optimization is performed with Genetic Algorithm (GA) to find the global optima of the cost function which considers the ride comfort and road holding performance of the full vehicle. A comparison between the three control modes in terms of ride comfort and road holding is performed. The results show that, the proposed control modes provide three different vehicle characteristics to the driver. In addition to this, all three control modes are superior to the optimal passive suspension in terms of both ride comfort and road holding.

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INTRODUCTION

The semi-active dampers have been developed to resolve the conflict between the vehicle ride comfort and road holding [1], [2], [3]. Since, the semi-active suspension systems provide a significant improvement in vehicle ride performance over the passive systems, several control strategies have been adopted and developed to offer superior ride quality to road vehicles. For example, Crosby and Karnopp [4] have proposed the famous 'Skyhook' damper control algorithm for a vehicle suspension system. While, Ahmadian [5] modified the Skyhook strategy by combining it with Groundhook strategy to improve road holding performance parallel to the ride comfort. The optimal linear quadratic (state feedback) control was one of the first methodologies to be applied to suspension control, mainly in the case of active suspensions [1]. Some applications on the semi-active suspensions have been proposed using different control strategies based on the optimal control [6], [7], [8]. An optimal self-tuning control algorithm for active and semi-active suspension systems incorporating a weighting controller, state observer and parameter estimator, based on LQG and LQR theories is suggested by [9] and [10] respectively. On the other hand, application of intelligent control methods in the semi-active suspension control algorithms is suggested in Kaldas et al. [11]. In this study, the semi-active suspension control algorithm is derived by using the adaptive Fuzzy Logic control together with the Kalman Filter for state estimation.

However, the technology is still an emerging one, and elaboration and more research work on different theoretical and practical aspects are required. Therefore, in [12] the authors developed an automatic design procedure for Fuzzy Logic damper controller based on Binary-Discrete GA. The key advantage of combining GA and Fuzzy Logic is that almost all the tasks of Fuzzy controller design can be accomplished automatically. The parameters of Fuzzy Logic inference rules and membership functions are tuned by the optimizer to handle the conflicting requirements of ride comfort, road holding and suspension deflection. Furthermore, the authors examine the performance enhancement of the developed Rule-Optimized Fuzzy Logic controller for semi-active suspension system by adding the road preview capability in [13].

The semi-active suspension systems have shown cost effectiveness over the fully active suspension systems [1], [2]. Therefore, the damper suppliers developed semi-active dampers and controllers and made them commercially available. For example, [14] presented a semi-active suspension system with commercial name "CES-Continuously Controlled Electronic Suspension System", which continuously adjusts damping levels according to the road conditions and dynamic states of the vehicle, such as longitudinal speed and lateral acceleration. The aim of control is to deliver comfort without sacrificing the safety of sure handling. Similar to [14], [15] developed a system called "CDC-Continuous Damping Control" which adapts

suspension damping flexibly and continuously to the respective driving situation and solves the compromise between safety, comfort, and dynamics in the vehicle.

In recent years, the automobile companies have developed several commercial vehicle models with semi-active suspension systems. The control algorithms have been continuously developed and new control features have been added, which make the customers satisfied with these vehicles. One control features, which is also the concern of this paper, is the driver capability to change the comfort and driving characteristics of the vehicle through the push of a button. For example, [16] developed the DCC system (Adaptive Chassis Control) which is based on semi-active dampers. In this new DCC system, the suspension constantly adjusts itself to the road conditions, the driving situation and the driver's requirements. The semi-active shock absorbers are used to make this possible and steering assistance is also adjusted in addition to the damping. The driver can switch the vehicle driving characteristics from the Normal mode to the Comfort or Sport modes through a button in the dashboard. As well, [17] designed the CVD system (Continuously Variable Damping) which is similar to the DCC designed by [16].

In this study, the driver capability to change the comfort and driving characteristics of the vehicle during the drive is constituted by enhancing a Rule-Optimized Fuzzy Logic controller [12] through the Triple-Control Mode feature. The controller synthesis is performed through an optimization process which determines the optimum membership functions and the optimum rule-base of the Rule-Optimized Fuzzy Logic controller for each driving mode. The proposed controller has three different control modes (Comfort, Normal and Sport), which can be switched online by the driver according to his/her requirements. An experimentally verified full vehicle ride dynamics model and a measured stochastic road profile are used for evaluating the performance of the proposed Rule-Optimized Fuzzy Logic controller with Triple-Control-Mode.

VEHICLE MATHEMATICAL MODEL

In this study, a generic full vehicle suspension model with non-linear suspension component characteristics has been used. The vehicle model contains seven masses which represent the body, four wheels, engine, and driver. The model has eleven degrees of freedom which include the bounce, pitch and roll motions of the body and engine, and the bounce motion of the wheels and driver. The suspension between body mass and wheel mass is modeled using a linear spring and a viscous damper in series with a rubber top mount at each corner. The damper and top mount masses are neglected. The nonlinear characteristics of the damper and the top mount are taken into account by using detailed damper top mount model presented in [18] and non-linear Force-Velocity damper characteristics. Tyre is modeled with a non-parametric Force-Displacement-Velocity map and point contact is assumed between the road and the tyre at each corner. Tyre-road separation is taken into account. Furthermore, the anti roll bars at front and rear axles are also considered inside the model. The engine is assumed to have bounce, pitch and roll degrees of freedom with respect to the vehicle body. The engine body is supported by three equivalent vertical mounts, which are modeled using linear springs in parallel with viscous dampers. The driver is

included inside the full vehicle model as a one degree of freedom system with the vertical motion only, and it is modeled as a mass supported by a linear spring and a linear damper. The axis convention and the notations used for the vehicle body, wheels and road profile are shown in Figure 1.

The equations of motion for the bounce, pitch and roll responses of the vehicle body are written as follows:

$$M_b \ddot{Z}_b = -F_{sb1} - F_{sb2} - F_{sb3} - F_{sb4} + F_{e1} + F_{e2} + F_{e3} + F_s \quad (1)$$

$$I_b \ddot{\theta}_b = L_1 F_{sb1} - L_2 F_{sb2} + L_1 F_{sb3} - L_2 F_{sb4} + T_e^p + T_s^p \quad (2)$$

$$J_b \ddot{\phi}_b = B_{s1} F_{sb1} + B_{s2} F_{sb2} - B_{s1} F_{sb3} - B_{s2} F_{sb4} + T_{stf} + T_{str} + T_e^r + T_s^r \quad (3)$$

The equations of motion for the engine vertical, pitch and roll responses are:

$$M_e \ddot{Z}_e = -F_{e1} - F_{e2} - F_{e3} \quad (4)$$

$$I_e \ddot{\theta}_e = L_{e1} F_{e1} - L_{e2} F_{e2} - L_{e3} F_{e3} \quad (5)$$

$$J_e \ddot{\phi}_e = B_{e1} F_{e1} - B_{e2} F_{e2} - B_{e3} F_{e3} \quad (6)$$

The equations of motion for vertical response of the driver can be written as:

$$M_s \ddot{Z}_s = -F_s \quad (7)$$

The equations of motion for the bounce motion of the wheels can be written as:

$$M_{wi} \ddot{Z}_{wi} = -F_{wi} + F_{swi} \quad (8)$$

Where;

$$Z_{b1} = Z_b - L_1 \theta_b + B_{s1} \varphi_b \quad (9.a)$$

$$Z_{b2} = Z_b + L_2 \theta_b + B_{s2} \varphi_b \quad (9.b)$$

$$Z_{b3} = Z_b - L_1 \theta_b - B_{s1} \varphi_b \quad (9.c)$$

$$Z_{b4} = Z_b + L_2 \theta_b - B_{s2} \varphi_b \quad (9.d)$$

$$Z_{e1} = Z_e - L_{e1}\theta_e + B_{e1}\varphi_e \quad (10.a)$$

$$Z_{e2} = Z_e - L_{e2}\theta_e - B_{e2}\varphi_e \quad (10.b)$$

$$Z_{e3} = Z_e + L_{e3}\theta_e - B_{e3}\varphi_e \quad (10.c)$$

$$Z_{be1} = Z_b - L_{be1}\theta_b + B_{be1}\varphi_b \quad (11.a)$$

$$Z_{be2} = Z_b - L_{be2}\theta_b - B_{be2}\varphi_b \quad (11.b)$$

$$Z_{be3} = Z_b - L_{be3}\theta_b - B_{be3}\varphi_b \quad (11.c)$$

$$Z_{bs} = Z_b - L_{bs}\theta_b + B_{bs}\varphi_b \quad (12)$$

$$F_{sbi} = K_{si}(Z_{bi} - Z_{wi}) + F_{mi} \quad (13)$$

$$F_{ej} = K_{ej}(Z_{ej} - Z_{bej}) + C_{ej}(\dot{Z}_{ej} - \dot{Z}_{bej}) \quad (14)$$

$$F_s = K_s(Z_s - Z_{bs}) + C_s(\dot{Z}_s - \dot{Z}_{bs}) \quad (15)$$

$$F_{wi} = K_{ti}(Z_{wi} - Z_{oi}) \quad (16)$$

$$F_{swi} = K_{si}(Z_{bi} - Z_{wi}) + F_{di} \quad (17)$$

$$T_e^p = -L_{be1}F_{e1} - L_{be2}F_{e2} - L_{be3}F_{e3} \quad (18)$$

$$T_e^r = B_{be1}F_{e1} - B_{be2}F_{e2} - B_{be3}F_{e3} \quad (19)$$

$$T_s^p = -F_s L_{bs} \quad (20)$$

$$T_s^r = F_s B_{bs} \quad (21)$$

$$T_{stf} = 2B_{s1}K_{stf}((Z_{b1} - Z_{w1}) - (Z_{b3} - Z_{w3})) \quad (22)$$

$$T_{str} = 2B_{s2}K_{str}((Z_{b2} - Z_{w2}) - (Z_{b4} - Z_{w4})) \quad (23)$$

With $i = 1, 2, 3, 4, j = 1, 2, 3$

The damper of the semi-active suspension system used in this study can be controlled continuously between upper and lower limits of the Force-Velocity characteristic curves. Therefore, it is theoretically capable of tracking a damping force demand independently of the damper velocity. The demanded damper force F_{sci} is generated by the controller according to different control laws and strategies. Since the controller output cannot be fully implemented because of the non-linear restriction in the system, i.e. it can only dissipate energy and not input it, the system cannot work with the output control signal only [1], [6]. Therefore, an additional control law is required to keep the control force in the controllable damper limits [12]. In this study, the Rule-Optimized Fuzzy Logic controller is used to generate the demanded suspension control force F_{sci} . The additional control law defined in Equation 24 is used to keep the actual semi-active damper force F_{svi} always within the allowable force limits of the damper.

$$F_{svi} = \begin{cases} F_{mini} & \text{if } F_{sci} \leq F_{mini} \\ F_{maxi} & \text{if } F_{sci} \geq F_{maxi} \\ F_{sci} & \text{if } F_{min} < F_{sci} < F_{maxi} \end{cases} \quad (24)$$

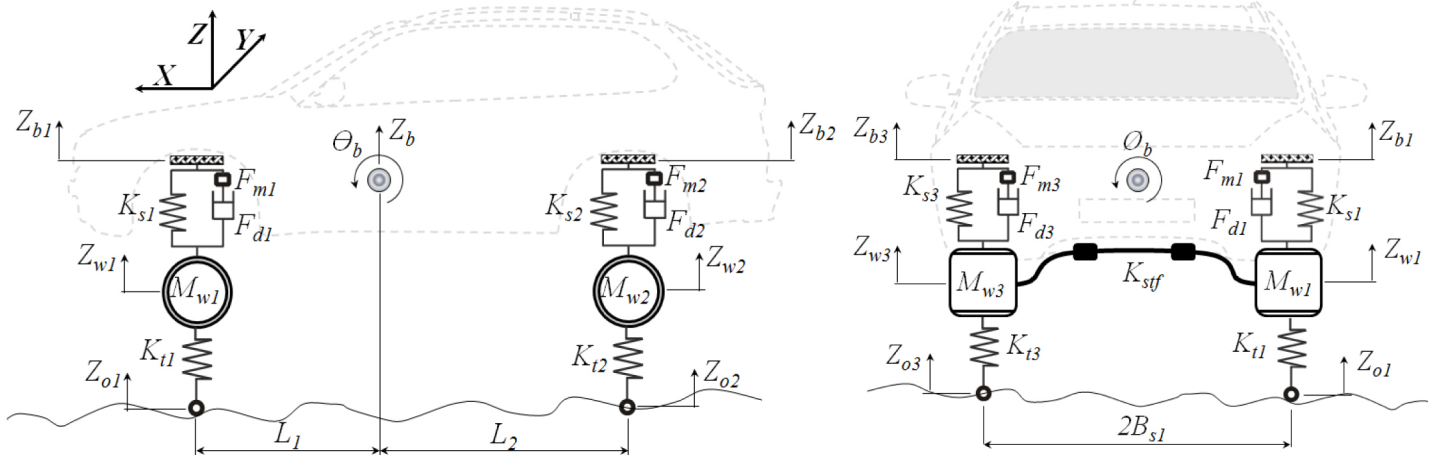


Figure 1. The 11 DoF vehicle model

The semi-active damper has its own dynamics in addition to the solenoid valve dynamics, which have significant influences upon the performance of the semi-active suspension system. Therefore, the overall dynamics of the semi-active damper and its solenoid valve is simulated as a first order transfer function which models the delay behaviour of the damper and its solenoid valve. Furthermore, the first order transfer function has employed as a filter to cutoff the controlled damping force at the allowable operating frequency of the solenoid valve. The first order transfer function shown in Equation 25 has been used for simulating semi-active damper dynamics

$$F_{svi}(s) = \frac{1}{(t_d(s)+1)} F_{svdi}(s) \quad (25)$$

In Equation 25, $F_{svdi}(s)$ is the Laplace transform of the desired damping force $F_{svdi}(t)$, while $F_{svi}(s)$ is the actual damping force whose time domain form is $F_{svi}(t)$ shown in Equation 24., and $t_d(s)$ is the Laplace transform of the time delay constant.

MODEL VERIFICATION

The four-poster hydraulic shaker test-rig, which is available at the Institute of Automotive Engineering (IAE) of the Technical University of Braunschweig, is used for the verification analysis of the mathematical vehicle model. The hydraulic cylinders have dynamic force limit of 21kN together with ± 125 mm displacement limit. The road profile and the measurement signals are filtered with 35Hz low-pass filter. The experimental vehicle is equipped with body acceleration and suspension travel sensors. A side view schematic drawing of the four-poster test-rig is shown in Figure 2.

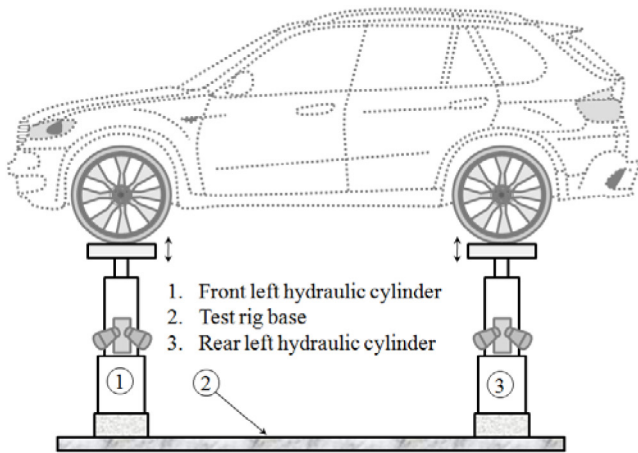


Figure 2. Hydraulic shaker test-rig

A sine sweep input signal covering a frequency range from 0.5 Hz to 15 Hz and a measured stochastic road profile are used for the model verification tests [11]. The amplitude of the sweep profile is decreasing with increasing frequency from 35 mm to 3 mm as shown in Figure 3.

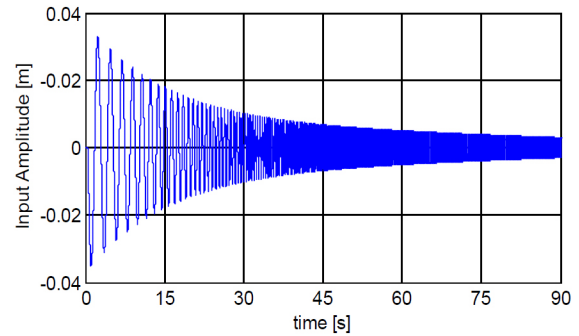


Figure 3. Sine-Sweep input profile

Since a controllable damper is used in this study, the comparison analysis is performed for the two different damper settings, which can be regarded as soft and hard settings. At the beginning of the verification analysis, an optimization process is applied for identifying the unknown vehicle parameters. The parameters that have been identified include the stiffness and damping constant of the engine mounts, and the damper force scale factors which accounts for the altering of the damper characteristics with usage. The goal of the optimization for identifying the unknown vehicle parameters is to reduce the RMSE between the measured and simulated body accelerations as shown in Equations 26.

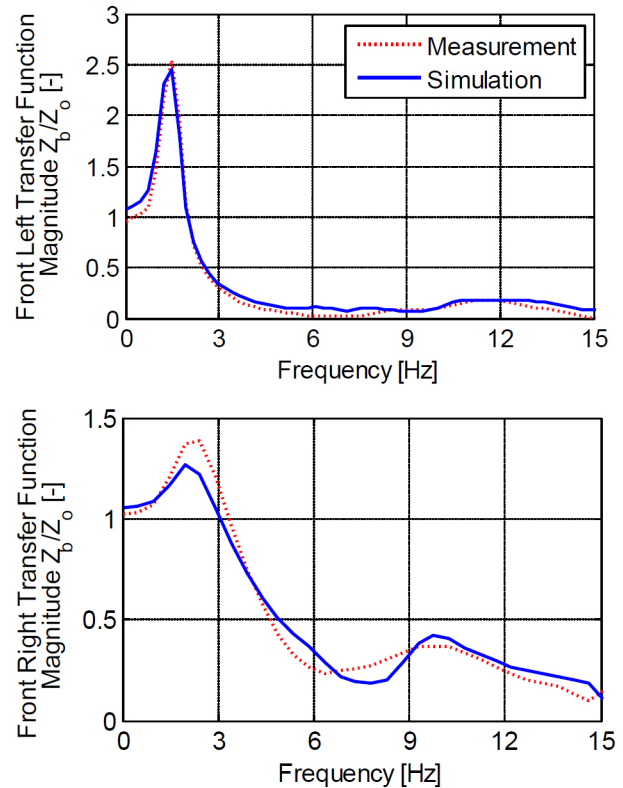


Figure 4. A comparison between the measured and simulated body displacement transfer function magnitudes for sweep road input for soft and hard damper setting

$$RMSE = \sqrt{\frac{1}{N} \sum_{i=1}^N (B.Acc_{j,i,mea} - B.Acc_{j,i,sim})^2}$$