

Performance Analysis of the Rule-Optimized Fuzzy-Logic Controller for Semi-Active Suspension

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ABSTRACT

This paper presents a performance analysis study for the Rule-Optimized controller of a semi-active suspension system. The Rule-Optimized controller is based on a Fuzzy Logic control scheme which offers new opportunities in the improvement of vehicle ride performance. An eleven degree of freedom full vehicle ride dynamics model is developed and validated through laboratory tests performed on a hydraulic four-poster shaker. An optimization process is applied to obtain the optimum Fuzzy Logic membership functions and the optimum rule-base of the semi-active suspension controller. The global optima of the cost function which considers the ride comfort and road holding performance of the full vehicle is determined through discrete optimization with Genetic Algorithm (GA). The performance analysis of the controller is performed by evaluating the Rule-Optimized controller performance under different driving conditions; including different road profiles, different vehicle speeds and different vehicle loadings. The results show that, the Fuzzy-Logic based Rule-Optimized controller for semi-active suspension provides significant improvements in both ride comfort and road holding performance of the vehicle under different operating conditions.

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INTRODUCTION

Several control methods have been studied and applied to the active/semi-active suspension systems. Karnopp et al. [1] have proposed the famous 'skyhook' damper control algorithm for a semi-active vehicle suspension system and demonstrated the improved performance of this system over the passive suspension. Ahmadian [2] had modified the skyhook strategy by combining it with groundhook strategy to improve the road holding performance parallel to the ride comfort. The linear quadratic (state feedback) control has been one of the first methodologies to be applied to suspension control, mainly in the case of active suspensions [3]. Some applications on the semi-active suspensions have been proposed using different control strategies based on the optimal control [4], [5], [6].

The control performance of traditional controllers depends mainly on the accuracy of the dynamic model used and the accuracy of the state measurements and estimations. In order to meet practical requirements in a semi-active suspension system, it is crucial to derive or to identify an appropriate model for the traditional controller design. Estimating uncertain effects is even more challenging due to the random noise caused by road inputs, [7], [8]. Some model-free and robust intelligent controllers were introduced to

solve the problems of the uncertainty effects, e.g. Fuzzy Logic Controller (FLC), [9], [10]. The FLC is an adequate methodology for designing robust controllers that are capable of delivering a satisfactory performance in the face of uncertainty and imprecision. As a result, the FLC has become a popular approach to non-linear and uncertain system control in recent years. There are different ways to construct FLCs for vehicle suspension control, with the most common method to construct the FLCs being by eliciting the fuzzy rules and its membership functions based on experts' knowledge or experience, [11], [12]. Kaldas et al. [13] developed a semi-active suspension system controller using Adaptive Fuzzy Logic control theories together with a Kalman Filter for state estimation. A quarter vehicle ride dynamics model is constructed and validated through laboratory tests performed on a hydraulic four-poster shaker. A Kalman Filter algorithm is constructed for bounce velocity estimation, and its accuracy is verified through measurements performed with external displacement sensors. The benefit of using adaptive control with Fuzzy Logic to maintain the optimum performance over a wide range of road inputs is enhanced by the accuracy of the Kalman Filter in estimating the controller inputs. A gradient-based optimization algorithm is applied for tuning FLC parameters. The quarter vehicle model is simulated with the

developed semi-active suspension controller on different road profiles. A comparison is performed between the Adaptive Fuzzy semi-active controller, the optimum LQR semi-active controller, and the optimum passive suspension system in terms of ride comfort and road holding. The results showed that the FLC based semi-active suspension system provides significant improvements in both ride comfort and road holding of the vehicle on different road profiles.

The common problem of the available Fuzzy Logic controllers for suspension control is that, using the experts' knowledge or experience to design the fuzzy controller rule base cannot fully handle or accommodate for the uncertainties associated with changing road inputs. Therefore, the authors developed an effective control concept for semi-active suspension system based on the automatic design methods and rule acquisition procedures for fuzzy systems [14]. The developed Fuzzy Logic scheme tunes the controller to treat the conflicting requirements of ride comfort and road holding parameters within a specified range of the suspension deflection. The GA has been used to find the global optima of the cost function, which considers the ride comfort and road holding performance of the full vehicle. The authors named the designed controller "Rule Optimized Fuzzy Logic Controller" because it is based on optimizing the rule base and controller structure.

In this study, a performance analysis is performed for the Rule-Optimized Fuzzy-Logic controller by evaluating the controller performance through full vehicle simulations under different driving conditions including different road profiles, different vehicle speeds and different vehicle loadings.

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 (DoF) 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 at each corner is modeled using a linear spring and a viscous damper in series with a rubber top mount. The jounce bumpers with nonlinear stiffness are also included in the model. 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 [15] together with the "current-velocity-force" map of the damper. 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:

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

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

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

The modeled vehicle is front-engine front-drive type and the engine-gearbox block is supported by three mounts. First mount is located at the left side while the second one installed at the right edge of the engine. The third engine mount is the sub-frame mount which is located at the rear right side of the engine. The equations of motion for the vertical, pitch and roll degrees of freedom of the engine are:

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

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

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

The equation 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;

$$Z_{b1} = Z_b - L_1 \phi_b + B_{s1} \chi_b \quad (9)$$

$$Z_{b2} = Z_b + L_2 \phi_b + B_{s2} \chi_b \quad (10)$$

$$Z_{b3} = Z_b - L_1 \phi_b - B_{s1} \chi_b \quad (11)$$

$$Z_{b4} = Z_b + L_2 \phi_b - B_{s2} \chi_b \quad (12)$$

$$Z_{e1} = Z_e - L_{e1} \phi_e + B_{e1} \chi_e \quad (13)$$

$$Z_{e2} = Z_e - L_{e2}\varphi_e - B_{e2}\kappa_e \quad (14)$$

$$Z_{e3} = Z_e + L_{e3}\varphi_e - B_{e3}\kappa_e \quad (15)$$

$$Z_{be1} = Z_b - L_{be1}\varphi_b + B_{be1}\kappa_b \quad (16)$$

$$Z_{be2} = Z_b - L_{be2}\varphi_b - B_{be2}\kappa_b \quad (17)$$

$$Z_{be3} = Z_b - L_{be3}\varphi_b - B_{be3}\kappa_b \quad (18)$$

$$Z_{bs} = Z_b - L_{bs}\varphi_b + B_{bs}\kappa_b \quad (19)$$

$$F_{sbi} = K_{si}(Z_{bi} - Z_{wi}) + F_{mi} \quad (20)$$

$$F_{ej} = K_{ej}(Z_{ej} - Z_{bej}) + C_{ej}(\dot{Z}_{ej} - \dot{Z}_{bej}) \quad (21)$$

$$F_s = K_s(Z_s - Z_{bs}) + C_s(\dot{Z}_s - \dot{Z}_{bs}) \quad (22)$$

$$F_{wi} = K_{ti}(Z_{wi} - Z_{oi}) \quad (23)$$

$$F_{swi} = K_{si}(Z_{bi} - Z_{wi}) + F_{di} \quad (24)$$

$$T_e^p = -L_{be1}F_{e1} - L_{be2}F_{e2} - L_{be3}F_{e3} \quad (25)$$

$$T_e^r = B_{be1}F_{e1} - B_{be2}F_{e2} - B_{be3}F_{e3} \quad (26)$$

$$T_s^p = -F_s L_{bs} \quad (27)$$

$$T_s^r = F_s B_{bs} \quad (28)$$

$$T_{stf} = 2B_{s1}K_{stf}((Z_{b1} - Z_{w1}) - (Z_{b3} - Z_{w3})) \quad (29)$$

$$T_{str} = 2B_{s2}K_{str}((Z_{b2} - Z_{w2}) - (Z_{b4} - Z_{w4})) \quad (30)$$

with $i = 1, 2, 3, 4$ and $j = 1, 2, 3$.

The damper of the semi-active suspension system used in this study can be controlled continuously between the upper and lower limits of the force-velocity characteristic curves. In order to control the force output of the semi-active damper, a controlled solenoid valve which influences the oil flow dynamics is employed inside the damper. The current-velocity-force map of the semi-active damper used in this study is shown in Figure 2. The force map is experimentally identified through testing of commercially available dampers.

The semi-active damper is theoretically capable of tracking the demanded damping force 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 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 [14]. Therefore, an additional control law is required to keep the control force within limits. 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 31 is used to keep the actual semi-active damper force F_{svi} always within the achievable force limits of the damper.

$$F_{svi} = \begin{cases} F_{mini} & \text{if } F_{sci} \leq F_{mini} \\ F_{maxi} & \text{if } F_{sci} \geq F_{maxi} \\ F_{sci} & \text{if } F_{min} < F_{sci} < F_{maxi} \end{cases} \quad (31)$$

The own dynamics of the semi-active damper as well as the dynamics of the solenoid valve have influence on the performance of the semi-active suspension system. In order to account for these influences, the overall dynamics of the semi-active damper and its solenoid valve are simulated using the first order transfer function shown in Equation 32 which models the delay behavior of the damper and its solenoid valve.

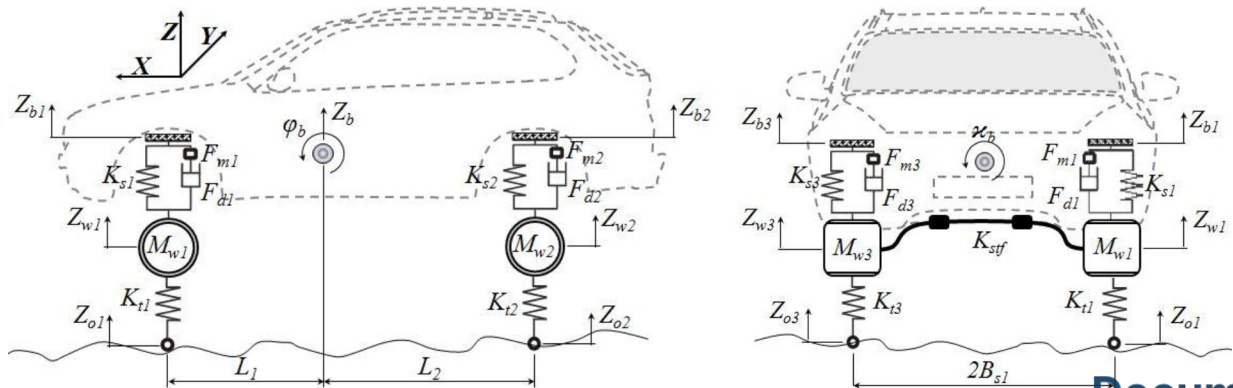


Figure 1. The 11 DoF full vehicle model

$$F_{svi}(s) = \frac{1}{(t_d s + 1)} F_{svdi}(s) \quad (32)$$

In Equation 32, $F_{svdi}(s)$ 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 $F_{svi}(t)$ given in Equation 31, and t_d is the time constant.

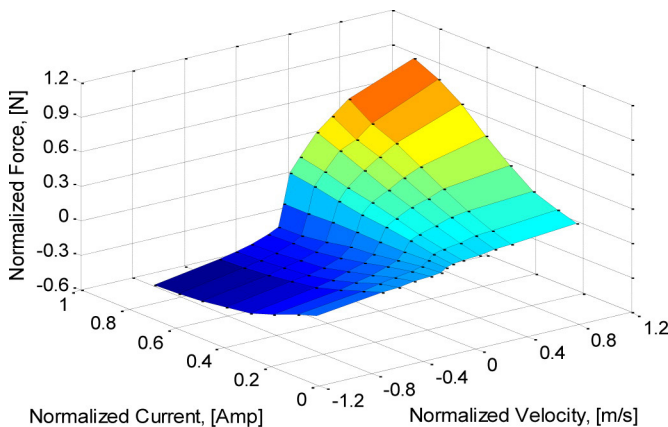


Figure 2. Normalized semi-active damper characteristics

CONTROLLER DESIGN

As known, the semi-active control algorithms are designed to obtain the optimal damping characteristics, which can be used during the control process. The semi-active control algorithm defines the relationship between the damper velocity, motion states of the vehicle body and the demanded damper force. Since the control algorithm is tuned on certain types of roads, good ride performance might not be achieved by using the same control algorithm on different roads. Therefore, some adaptation methods have been suggested to overcome this problem. One of the well-known adaptation techniques is the Adaptive Gain Scheduling [13], [5]. In this method different control algorithms (i.e. different parameter sets) are used at different road profiles by using the RMS of the body/wheel vertical acceleration as an indicator for the road roughness to switch between the algorithms. This method is very simple and cost effective; however, theoretically infinite number of control algorithms is required for optimum performance. With the advances in the optimization algorithms, self-tuning methodology has been developed to overcome the disadvantage of the Adaptive Gain Scheduling method [3], [16], [17]. The self-tuning method is designed to generate or modify the control algorithm during operation at each time step of the control process. This method is effective but difficult and expensive to apply in the vehicle, since it requires a very fast microprocessor and very accurate measurement system.

The idea behind the Rule-Optimized control algorithm is to obtain a continuous physical relationship between the body velocity, body acceleration and demanded damper force. The relationship between these constitutes a three dimensional characteristics which enables the representations of the control algorithm with a map as shown in Figure 3. The obtained map can be used directly without any adaption

algorithm, because the demand control signal depends on the road condition which is conducted from the body vertical response signal (body acceleration).

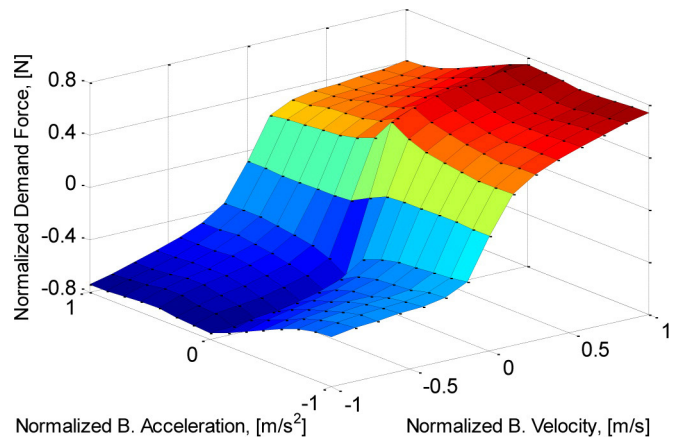


Figure 3. Force map of the Rule-Optimized controller

Controller Design Assumptions

In order to design a FLC for the semi-active suspension system, certain assumptions are required to make the design process of the controller easier. As the system is quasi symmetrical, it is assumed that the symmetrical membership functions about the y-axis will provide a valid controller. Furthermore, a symmetrical rule-base and other constraints are also introduced as follows [14]:

- The universes of discourses are normalized to lie between - 1 and 1 with scaling factors external to the FLC used to give appropriate values to the variables.
- The first and last membership functions have their apexes at -1 and 1 respectively. This can be justified by the fact that, changing the external scaling would have similar effect to changing these positions.
- The membership functions and the rule-base of the FLC for the front and rear axles are the same, but the scaling factors for the inputs and outputs are different.
- The number of fuzzy sets is constrained to be an odd integer greater than unity. In combination with the symmetry requirement, this means that the central membership function for all variables will have its apex at zero.
- The base vertices of membership functions are coincident with the apex of the adjacent membership functions. This ensures that the value of any input variable is a member of at most two fuzzy sets, which is an intuitively sensible situation. It also ensures that when a variable's membership of any set is certain, i.e. unity, it is a member of no other sets [18].
- Triangular membership functions are used. Furthermore, the rule base and interface engine are based on the Mamdani-Type of fuzzy inference, and the center of area method is used for the defuzzification process.
- The controller uses the body vertical velocity and acceleration at each corner as the input and generates the demanded damper force F_{sci} as the output.