

# A New Adaptive Controller for Performance Improvement of Automotive Suspension Systems with MR Dampers

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## ABSTRACT

A control algorithm is developed for active/semi-active suspensions which can provide more comfort and better handling simultaneously. A weighting parameter is tuned online which is derived from two components - slow and fast adaptation to assign weights to comfort and handling. After establishing through simulations that the proposed adaptive control algorithm can demonstrate a performance better than some controllers in prior-art, it is implemented on an actual vehicle (Cadillac STS) which is equipped with MR dampers and several sensors. The vehicle is tested on smooth and rough roads and over speed bumps.

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## INTRODUCTION

The job of the suspension system in an automobile is dual: to provide a comfortable ride to the passengers by isolating them from the road irregularities, bumps and potholes, and to improve the road holding capacity of the vehicle thereby providing safety. The use of suspension systems in vehicles is not new. In fact, they have been in use since the cars were actually horse drawn carriages [1]. But still, active research has been prevalent for the development of new and better suspension systems. One major reason for this can be attributed to the fact that the two requirements of ride comfort and handling which the suspension is expected to fulfill are conflicting. Figure 1 shows this conflicting nature for different suspension parameters in terms of the RMS acceleration of the chassis (comfort) and RMS dynamic tire force (handling and safety) for some particular road and driving conditions. It can be seen that for better ride comfort (as in a Limousine), a softer suspension (low  $k$  and  $d$ ) is required but it leads to higher tire forces, hence, less safety. On the other hand, for better handling (as in a sports car), a stiffer suspension (high  $k$  and  $d$ ) is required but it makes the ride less comfortable. A conventional suspension with a passive spring and damper is represented by a fixed point on this conflict diagram. Numerous efforts have been made by researchers to design a suspension system which caters to a wide range of performance requirements with as less compromise as possible by changing its properties during run-time like active and semi-active suspensions to give near optimal performance. Such suspensions with variable properties have led to the design of several control algorithms like skyhook, groundhook, clipped optimal, etc. which offer better results in different aspects of suspension performance.

An active suspension system consists of an actuator (electric or hydraulic) which can inject as well as dissipate power. Coupled with an appropriate controller, such a system can provide a performance far better than a typical semi-active system both in terms of comfort and handling. One such system developed by Bose Corporation [2] uses linear electromagnetic motors which can control each corner of the vehicle independently. Although the performance of this system is quite superior to that of a conventional suspension and Bose claims that the system can recover energy by driving the motors in generator mode, the system is yet to be integrated in a production vehicle, possibly due to high costs and power requirements. Most of the commercially available active suspensions as of now use hydraulic systems. For instance, the Dynamic Drive from BMW [3] significantly reduces roll angle during cornering by using two active rotating hydraulic actuators. Mercedes-Benz has also developed an active suspension named as Active Body Control (ABC) [4] which consists of an active hydraulic cylinder mounted in series with the spring, which can rapidly move in the vertical direction by getting energized by a high pressure hydraulic pump.

A semi-active suspension system can change the damping characteristics during run-time but cannot provide a force input. It is represented on the conflict diagram in Figure 1 by a constant stiffness line. The level of damping can either be defined by the user through an instrumental panel or can be automatically controlled for that particular state of the vehicle by an on-board CPU that takes feedback from various sensors mounted on it. There are several ways through which the damping can be varied in a semi-active system. One commercial product is MagneRide™ by E.W. Group which uses Magneto-

Rheological (MR) fluids [5]. An MR fluid has magnetically soft (easily, but temporarily magnetized) iron particles suspended in a synthetic hydrocarbon base. Application of magnetic field by the electromagnetic coil contained in the piston causes the particles to align into fibrous structures thereby increasing the viscosity. Hence, varying the magnetic flux in effect controls the viscosity of the damper. Another method of varying the damping is by changing the orifice size, as is done in the CDC®-Continuous Damping Control by ZF [6]. The CDC has a proportional valve, which offers soft damping when the opening for the oil flow is expanded and firm damping when it is restricted.

Since variable dampers are practically and economically viable, it is worth investigating the possibilities of better control algorithms for semi-active suspensions. In this research, a new controller is proposed which is benchmarked against some well-accepted algorithms in prior-art, first through simulations and then by implementation on a fully instrumented Cadillac STS equipped with MR dampers.

## CONTROL ALGORITHMS IN PRIOR-ART

A semi-active control algorithm varies the damping in order to obtain either better comfort, or handling, or both (with weights assigned to each). An adaptive control algorithm accounts for the variation in system parameters and the assignment of weights to the performance criteria. A two degree of freedom quarter car model as shown in Figure 2 will be considered throughout this research to define a suspension system with variable damping. Some of the widely popular control algorithms in literature are discussed below.

**Skyhook control:** Introduced by Karnopp et al. [7], it is one of the most popular comfort oriented control strategies. Originally developed for a single degree of freedom quarter-car model (no unsprung mass), it tries to emulate a fictitious damper  $d_{sky}$  attached between the sprung mass and the stationary sky so that its movement is minimized thereby maximizing comfort. However, since it is practically realized by a damper mounted between the sprung and the unsprung mass which can only apply force in the direction opposite to the relative velocity between them, the damping force is assumed to be zero when the passivity constraint is violated. Mathematically, it can be expressed as

$$F_d = \begin{cases} -d_{sky}\dot{z}_s & \text{for } \dot{z}_s(\dot{z}_s - \dot{z}_u) \geq 0 \\ 0 & \text{for } \dot{z}_s(\dot{z}_s - \dot{z}_u) < 0 \end{cases} \quad (1)$$

The skyhook control strategy greatly attenuates the motion of the sprung mass. However, when implemented to a little more realistic two degree of freedom model, this technique leads to extreme vibrations of the unsprung mass (wheel hop) and high dynamic tire forces, which deteriorates vehicle stability.

**Groundhook control:** Analogous to the skyhook algorithm, the groundhook control algorithm [8] tries to attenuate the dynamic tire force by emulating a fictitious damper  $d_{gnd}$  attached between the unsprung mass and a static frame on the ground. Mathematically,

$$F_d = \begin{cases} d_{gnd}\dot{z}_u & \text{for } \dot{z}_u(\dot{z}_s - \dot{z}_u) \leq 0 \\ 0 & \text{for } \dot{z}_u(\dot{z}_s - \dot{z}_u) > 0 \end{cases} \quad (2)$$

**Clipped optimal control:** In this control scheme, first an optimal controller is designed using techniques like LQR or LQG, to generate the optimal control force  $F_a$  which is assumed to take any arbitrary value [9], i.e., can act as an actuator as well as an energy dissipater. Then the passivity constraint is invoked to “clip” the force when it needs to inject power. It can be expressed mathematically as

$$F_d = \begin{cases} 0 & \text{if } F_a(\dot{z}_s - \dot{z}_u) > 0 \\ F_a & \text{if } F_a(\dot{z}_s - \dot{z}_u) \leq 0 \end{cases} \quad (3)$$

where  $F_a$  is the actuator force that would be optimally required if the system was fully active. It should be noted here that the term “optimal” is described in the performance index sense, which is defined by the designer as per requirement. The performance index might consist of cost on sprung mass movement, tire deflection, suspension deflection and input force, with weights assigned to each. The numerical value of the performance index, as such, has no physical significance.

## ADAPTIVE SEMI-ACTIVE SUSPENSION: MODELING, CONTROL AND SIMULATION RESULTS

### Estimation

Since comfort and handling are two conflicting criteria for suspension design, therefore, when designing a control algorithm for a semi-active suspension (or active suspension for that matter), it would be worthwhile to keep track of the two performance costs on-line and tune the weights assigned to the two criteria accordingly to obtain a balanced performance. Moreover, other variables like unsprung mass acceleration, suspension rattle velocity, etc. are also of interest for implementing the controller. All these quantities of interest must be attainable using commonly available and economical sensors in a vehicle. Since an accelerometer on the vehicle body (sprung mass) has become almost standard equipment on all modern cars due to its use in other systems as well (like ABS and ESC), measurement of comfort objective is relatively easy. Similarly, measurement of the suspension travel (relative displacement between sprung and unsprung mass) is also possible using sensors like a potentiometer or an LVDT. On the contrary, direct measurement of tire forces requires quite

expensive sensors, which are impractical for a passenger vehicle. Hence, in this research, a simple filter based estimator has been used as shown in Figure 3 to estimate the required quantities by directly measuring two values: sprung mass acceleration  $\ddot{z}_s$  and relative displacement between sprung and unsprung mass  $z_s - z_u$  (suspension travel), the sensors for which are relatively economical and practical to mount.

This is a very simple approach for estimation. Although more complex estimators like Extended Kalman Filter (EKF) can provide more accurate results, it gives practically useful data for the proof of concept of the controller, as shown in later sections.

### Adaptive Semi-Active Suspension Control

In this section, a control algorithm for semi-active suspensions will be developed. First, two different control schemes, one handling oriented and other comfort oriented are proposed. Then, the two are integrated by introducing a modification of the adaptive control structure presented in [10] to finally obtain a controller which is simple, economical to implement, and automatically distributes its priority between comfort and handling on-the-run as per the requirement.

From the equations of motion of Figure 2, it can be derived that the tire force would approach to zero (thereby providing better handling) for a damping force of

$$F_d = -m_u \ddot{z}_u + k_p(z_s - z_u) \quad (4)$$

However, since  $F_d$  is a passive force, this rule has to be modified to accommodate the passivity constraint.

$$F_d = \begin{cases} -m_u \ddot{z}_u + k_p(z_s - z_u) & \dots \\ \dots & \text{for } (-m_u \ddot{z}_u + k_p(z_s - z_u))(\dot{z}_s - \dot{z}_u) \leq 0 \\ 0 & \text{for } (-m_u \ddot{z}_u + k_p(z_s - z_u))(\dot{z}_s - \dot{z}_u) > 0 \end{cases} \quad (5)$$

Similarly, the acceleration of the sprung mass would approach to zero (thereby providing better comfort) for a damping force of

$$F_d = k_p(z_s - z_u) \quad (6)$$

which basically becomes the R-S controller [11] after satisfying the passivity constraint.

The two control schemes mentioned above cater to their own specific performance objective only. Hence, if there exists a weighting parameter  $\alpha \in [0, 1]$  such that it defines the weights given to the two performance indices, and its value adapts

itself online as per the requirement from driving conditions, then we have a very simple yet elegant controller for semi-active suspension systems.

$$\begin{aligned} F_{adpt} &= \alpha(-m_u \ddot{z}_u + k_p(z_s - z_u)) + (1 - \alpha)(k_p(z_s - z_u)) \\ \Rightarrow F_{adpt} &= k_p(z_s - z_u) - \alpha m_u \ddot{z}_u \end{aligned} \quad (7)$$

The damping force is

$$F_d = \begin{cases} F_{adpt} & \text{for } F_{adpt}(\dot{z}_s - \dot{z}_u) \leq 0 \\ 0 & \text{for } F_{adpt}(\dot{z}_s - \dot{z}_u) > 0 \end{cases} \quad (8)$$

One interesting point to be noted here is that without the passivity constraint, Eq. (7) can be used as a controller for a fully active suspension system as well.

Hence, for one extreme value of the weighting parameter  $\alpha = 0$ , the controller is purely comfort oriented, while for the other extreme value  $\alpha = 1$ , the controller only focuses on handling. Now the problem has been reduced to appropriately adjusting the value of  $\alpha$  for maximum performance improvement. An adaptive controller introduced in [12] had the basic idea that comfort is the objective, as long as the dynamic tire forces are within certain limits. The weighting parameter  $\alpha$  varying between zero and one defines the level of this constraint violation. On similar lines but with some modifications to suit our requirements, an adaptation scheme for  $\alpha$  will be developed in the next section.

### Calculation of Weighting Parameter $\alpha$

From the method described in Figure 3, the dynamic tire forces can be easily estimated from commonly available sensors, and thus, it can be constantly monitored for constraint violation. It is known that the RMS value is generally used to quantify a stochastic variable like tire force over time. However, during actual driving conditions, singular events like bumps and potholes suddenly change the tire forces, which might not appear in the RMS value that immediately. Hence, the absolute value also needs to be kept tracked of. The weighting parameter  $\alpha$ , therefore, is derived from two components: slow adaptation (from RMS value  $F_{tire,RMS}$ ) and fast adaptation (from instantaneous absolute value  $|F_{tire}|$ ).

Assuming stochastic road input and Gaussian probability density of the dynamic tire force, its standard deviation over a time period  $T$  will be given by

$$\sigma_{F_{tire}} = \sqrt{\frac{1}{T} \int_0^T F_{tire}^2(\tau) d\tau} \quad (9)$$



Since the dynamic tire force has zero mean, its standard deviation is the same as its RMS value, i.e.,  $F_{tire,RMS} = \sigma_{F_{tire}}$ . For a normally distributed zero mean stochastic signal, the  $6\sigma$  rule states that nearly all of its values lie within the bounds of 6 standard deviations. Applying the  $6\sigma$  rule to dynamic tire force would mean that  $F_{tire}$  should remain within the bounds of the static tire force  $F_{stat}$  for most of the time  $T$ . In other words, the wheel almost never leaves contact with the ground. Mathematically, the constraint can be formulated as

$$F_{tire,RMS} \leq \frac{F_{stat}}{6} \quad (10)$$

where the static wheel load  $F_{stat} = (m_s + m_u)g$ .

To also take into account the variation of  $F_{tire}$  due to singular events which might not reflect in its RMS value immediately, another constraint can be defined which takes into consideration its instantaneous value. It completely depends on the controller designer that what value of the ratio  $F_{tire}/F_{stat}$  is deemed permissible. Clearly,  $F_{tire}/F_{stat} \geq -1$ , with the extreme value of  $-1$  indicating that the tire has just left contact with the ground. However, it would be too late if the controller waits for the ratio to become  $-1$  before it changes the weighting parameter. Some buffer region is needed during which the controller can act and bring down the dynamic tire force. As an intelligent guess, a threshold of  $-80\%$  is selected for the ratio of dynamic to static tire force. Mathematically

$$F_{tire} + 0.8F_{stat} \geq 0 \Rightarrow F_{tire} \geq -0.8F_{stat} \quad (11)$$

It should be noted here that only negative value of  $F_{tire}$  is responsible for leaving contact with the ground. However, since tire has very low damping, a high positive value of  $F_{tire}$  would soon be followed by a high negative value. To avoid unnecessary chatter in the controller, the absolute value  $|F_{tire}|$  can be used.

$$|F_{tire}| \leq 0.8F_{stat} \quad (12)$$

With the two constraint equations defined for slow and fast adaptation, a way can be developed to quantify the level of constraint violation. The adaptation structure is shown in Figure 4, where the input is the dynamic tire force  $F_{tire}$  (which will be the estimated tire force  $\hat{F}_{tire}$  in practice) and output is the weighting parameter  $\alpha$ .

The upper branch of the structure is responsible for fast adaptation. The ratio of dynamic to static tire force is input to a heuristic function  $h(u)$  (Figure 5) which increases steeply from zero to one for input greater than 0.8. The selection of heuristic function  $h(u)$  is also based on intuitive guess. Since the absolute value of  $F_{tire}$  is considered,  $h(u)$  is symmetric about y-axis. The output from  $h(u)$  is error  $e_f$  which is then passed

through a low pass filter with time constant  $\tau_f$  to reduce the chatter and rapid variations in fast adaptation parameter  $\alpha_f$ . The lower branch of the structure outputs the slow adaptation parameter  $\alpha_s$ . The amount by which the RMS of the ratio of static to dynamic tire force deviates from the predefined threshold of  $1/6$ , designated as slow adaptation error  $e_s$ , is integrated with output limited between zero and one. Finally, the two parameters from slow and fast adaptation are added, and the minimum of the sum  $\alpha_s + \alpha_f$  and unity gives the final value of the weighting parameter  $\alpha$ .

## Simulation Results

A two degree-of-freedom quarter car model was modeled in MapleSim and simulated with different road profiles (rough, smooth, bump) as input (modeled in time domain as described in Appendix A and with different controllers. Assuming that the damping force can be varied to any arbitrary value within some bounds and all the required variables are either measured or estimated, the proposed adaptive controller was then benchmarked against passive damping, skyhook and groundhook controllers. A passive damping of 1500 Ns/m has been selected which provides a really good compromise between comfort and handling for quarter car parameters shown in Table 1, which are very close to that of the front-left suspension of a Cadillac STS (used for experimental verification in the next section).

Table 1. Quarter-car parameters used in simulation of different controllers

| Parameter | Description      | Value      |
|-----------|------------------|------------|
| $m_s$     | Sprung mass      | 493 kg     |
| $m_u$     | Unsprung mass    | 62 kg      |
| $k_p$     | Spring stiffness | 35600 N/m  |
| $k_t$     | Tire stiffness   | 277000 N/m |

Figures 6, 7, and 8 show the simulation results for a transitional road input, where initially, the road profile is smooth (type-A), but switches to rough (type-C) at  $t = 20$  s, the total simulation time being 40 s. The power spectral density of the acceleration of the sprung mass in the frequency band of maximum human sensitivity is shown in Figure 6 as an index of comfort. It is seen that over the expected result of skyhook controller being better and groundhook controller being worse than a passive system in improving comfort, the proposed adaptive algorithm shows a performance ranging from close to slightly better than the skyhook. A peak is discernible in the 1-2 Hz region which is clearly the resonant frequency of the sprung mass. The new controller seems to do a quite good job in reducing this peak as well.

It might occur to the reader that the difference in the performance of skyhook and proposed controller is not significant enough. However, the actual strength of adaptive controller can be seen in Figure 7, which compares the running RMS of the tire force with different controllers. While the skyhook controller leads to exceedingly high dynamic tire force