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AN ANALYTICAL EVALUATION OF THE TRANSIENT DYNAMICS OF SEMIACTIVE DAMPERS

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ABSTRACT

This paper provides an analytical evaluation of semiactive dampers, which have been more increasingly used for vehicle applications in the recent years, when transient dynamics are present. Although much research has been performed on various aspects of semiactive dampers, the vast majority of past studies have been performed for the steady state aspects of such dampers. This study will extend the results of those studies, by providing an evaluation of semiactive dampers when they are subjected to a transient dynamics, such as a step input. The performance of a semiactive damper with two commonly used control methods—namely, skyhook control and hybrid control—is evaluated analytically, using a single suspension model. In addition to evaluating such metrics as the peak response and settling time of the sprung body, this analysis will include an evaluation of the work performed by semiactive dampers. A comparison with passive suspensions shows that the semiactive suspensions that are considered here could improve various aspects of vehicle ride comfort and handling, within the limitation of a single suspension model. Field-testing of the semiactive dampers discussed here would, of course, provide more conclusive evidence on their benefits for controlling vehicle transient dynamics.

INTRODUCTION

Semiactive dampers change their damping force in real time by simply changing the damping coefficient according to a control policy, which is usually based on the system dynamics. The ability to vary the semiactive damping coefficient independent of damper velocity, within limits, has prompted a number of studies to explore the possibility of improving suspension performance by using semiactive damper technology. As noted by Ahmadian and Pare in [1], the vast majority of the past studies have concluded that, at least in some respect, semiactive dampers can outperform conventional passive dampers that are commonly used in vehicle suspensions [2 – 6].

Semiactive dampers can be adjusted by mechanical means or using the rheological properties of the fluid that is used in the damper. The former uses mechanical valves driven by a solenoid or stepper motor to control damper force in a hydraulic damper. The latter category uses the rheological effect of controllable fluids, such as magneto rheological or electro rheological fluids, to provide adjustable damping forces. Although mechanical and rheological control dampers have been researched and developed extensively, the rheological controllable dampers have received much more attention in the past few years, mainly due to great advances in magneto rheological fluids.

The most widely used—and the original—control policy for semiactive dampers is commonly known as the “skyhook” policy, although many other variations of the skyhook control policy have been introduced in different studies in the past two decades. Some of the variations to skyhook control are called “groundhook” control and “hybrid” control, as described in detail in several past studies and will be discussed in later sections. A comprehensive review of these studies can be found in a paper by Ivers and Miller [7].

In spite of the large body of research that exist on semiactive dampers and the control policies that can govern them, the vast majority of such studies are mainly concerned with the steady state aspects of semiactive systems. This study will provide a different look at semiactive systems, by evaluating the transient dynamics of such systems. The performance of a semiactive damper with two different control methods—namely, skyhook control and hybrid control—is evaluated analytically, using a single suspension model with two degrees of freedom. The evaluation metrics will include the peak response and settling time of the sprung body and the work performed by the dampers. The results for semiactive dampers are compared with a passive damper, in order to highlight the benefits that semiactive dampers can provide in controlling the transient dynamics of a suspension.

BACKGROUND

In studying dynamic systems, whether passive or semiactive, the terminology sprung and unsprung bodies (masses) are often used in dynamics literature to refer to the two bodies of a two-degree of freedom dynamic model, such as the single suspension model in Figure 1. The sprung body, represented by m_1, and the unsprung body, denoted by m_2, are connected together by a spring and damper. The unsprung body is connected to a movable base.

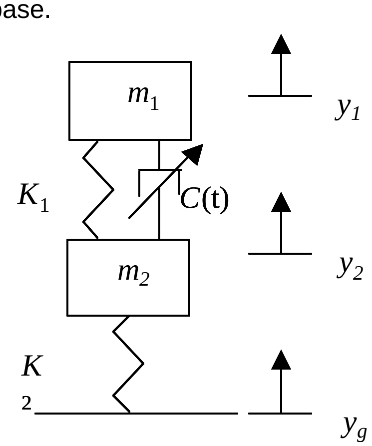

Figure 1. Two Degree of Freedom Single Suspension Model

As was mentioned earlier, the methods that have been suggested for controlling the semiactive damper in Figure 1 are several. For the purpose of this study, we will mainly focus on two of the control methods that have been most widely considered in past studies: skyhook control and hybrid control, as they will be described in detail next.

Figure 2 shows a schematic representation of a semiactive damper with skyhook control. The damper connected to the sky (a fixed y-axis coordinate) implies that if the suspension damper of Figure 1 is in rebound and the sprung body is moving upwards, then skyhook control turns the damper on and the damper pulls down on the sprung body, according to:

$$F_d = G v_1 \tag{1}$$

In other conditions, the skyhook control turns the damper off to provide no damping, therefore preventing the damper from pushing in to the suspended body. We will choose the following well-known mathematical representation for skyhook control:

$$F_{dd} = \begin{cases} G v_1 & v_1 v_{12} > 0 \\ 0 & v_1 v_{12} \le 0 \end{cases} \tag{2}$$

where,

F_{dd} = desired damper force, N

v_1 = sprung body velocity, m/s

v_2 = unsprung body velocity, m/s

v_{12} = damper velocity, m/s

G = skyhook gain, N/m/s

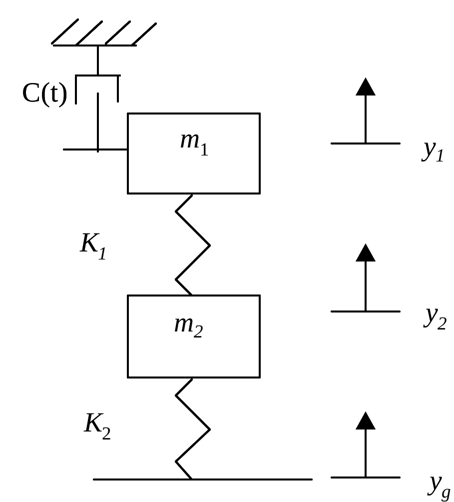

Figure 2. Schematic of a Single Suspension with Ideal Skyhook Control

One draw back to skyhook control is the dynamic jerk that can result from the rapid change in acceleration of the sprung body, as caused by the change in damping force. The discontinuous semiactive forces are also noted in [8], and will be evaluated in this study.

Hybrid control is a combination of two control methods, namely skyhook method that is intended to control the sprung body and groundhook control that is used for controlling the unsprung mass. Hybrid control can provide damping to both the sprung and unsprung bodies, independently, as is shown in an ideal realization in Figure 3. Mathematically, hybrid controller can be described by:

$$F_d = G\left(a\, s_{sky} - (1-a)s_{ground}\right) \tag{3}$$

Where

$$s_{sky} = \begin{cases} v_1 & v_1 v_{12} > 0 \\ 0 & v_1 v_{12} < 0 \end{cases}$$

$$s_{ground} = \begin{cases} v_2 & v_2 v_{12} < 0 \\ 0 & v_2 v_{12} > 0 \end{cases}$$

The terms in above equations are according to:

a_{sky} = Skyhook weighting factor, unitless

a_{ground} = Groundhook weighting factor, unitless

s_{sky} = Skyhook switch variable, m/s

s_{ground} = Groundhook switch variable, m/s

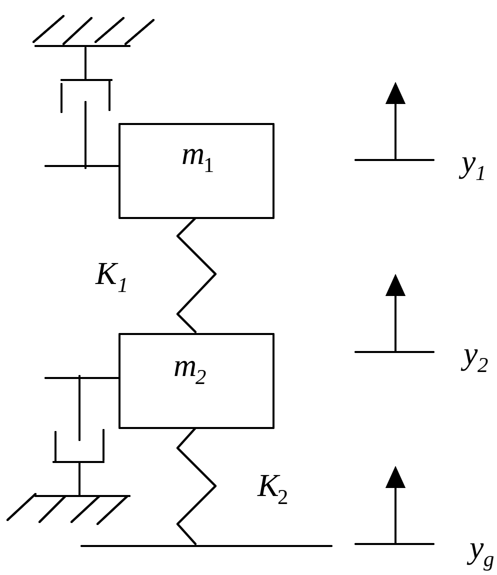

Figure 3. Idealized Schematic Representation of Hybrid Control for a Single Suspension Model

As has been described in earlier studies, hybrid control is commonly used to provide a better compromise between controlling the sprung and sprung mass in a vehicle [1 – 2].

DYNAMIC MODEL

The model that is used for presenting the results of this study, shown in Figure 1, is commonly referred to as a single suspension (quarter car) model. It represents a corner of a vehicle by including the dynamics of the sprung mass, unsprung mass, and suspension along the vertical axis of the vehicle. The sprung mass refers to the vehicle body, or the mass that is placed on the suspension. The unsprung mass refers to the bodies or components that are below the suspension, such as the vehicle axle, tire, and a portion of the suspension. The suspension includes the spring and damping elements that connect the sprung and unsprung bodies together. The terms used to describe each part of the model are listed in Table 1.

The dynamics of the single suspension model in Figure 1 can be represented by

$$m_1 \frac{d^2 y_1}{dt^2} + K_1 y_1 - K_1 y_2 + F_d = 0 \tag{4a}$$

$$m_2 \frac{d^2 y_2}{dt^2} + (K_1 + K_2) y_2 - K_1 y_1 - F_d = K_2 y_g \tag{4b}$$

Table 1. Single Suspension Model Parameters

Symbol	Description	Units
F_d	Damper force	N
F_1	Spring force	N
F_2	Tire force	N
m_1	Sprung mass	kg
m_2	Unsprung mass	kg
$C(t)$	Damping coefficient	N/m/s
K_1	Suspension spring stiffness	N/m
K_2	Tire stiffness	N/m
y_1	Sprung body displacement	m
y_2	Unsprung body displacement	m
y_g	Ground position	m

The determination of the damper force, F_d, will depend upon the damper model used. For a passive damper, we simply use a linear representation of the damper described by:

$$F_d = C v_{12} \tag{5}$$

where

$C =$ passive damping coefficient, N/m/s

For semiactive dampers, the damper characteristics are according to Figure 4, in which regions A and B indicate the force that can be provided by the damper in rebound and jounce, respectively. The maximum damper force is bounded by the force line with slope C_{on} and the minimum damping force is bounded by the damper force line with slope C_{off}. This means that the damping force indicated in Eqs. (2) and (3) cannot exceed the limits set by C_{on} and C_{off} in extension and compression. In practice, these limits are dictated by several elements, including the damper size and design. Nonetheless, in modeling semiactive dampers it is important to properly represent the damper force limitation, as will be the case for this study.

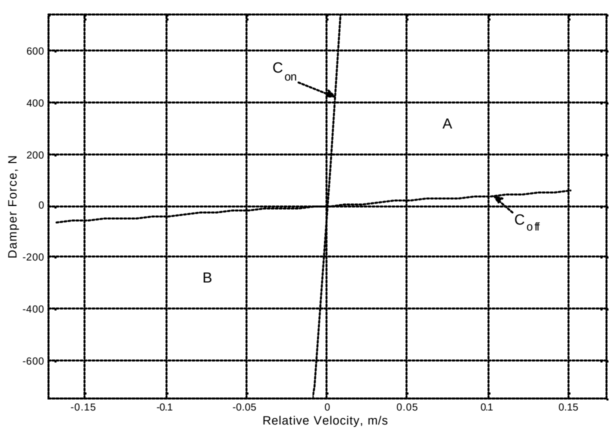

Figure 4. Damper Force-velocity Characteristic for a MR Damper Model. $C_{on} = 85000$ N/m/s $C_{off} = 381.9$ N/m/s

Another important aspect of semiactive dampers that must be represented in the model is the damper force delay, which always exists in a closed-loop system. Here, damper force delay—or simply, damper delay—is referred to the short interval of time that elapses between the time the system controller commands a damping force and when the damper actually provide that damping force. This short interval can vary from a couple to several milliseconds, depending on the damper and electronics design [9].

We will use the first-order filter:

$$\mathrm{TF}_{\mathrm{filter}} = \frac{1}{ts+1}, \tag{6}$$

since it has been successfully used by others in the past for this purpose [10]. The time constant τ is chosen to be 10 milliseconds, based on the capabilities of the semiactive suspensions that we have worked with in the past.

In order to better understand the effect of the damper delay on the dynamic system in Eq. (4), one can refer to the bode plot of the filter in Eq. (6), shown in Figure 5. At very low frequencies, the filter does not distort the damper force magnitude or phase. As the frequency increases, however, the filter scales down the damper force and introduces a phase delay in damping force, which is representative of the time delay that exists in practice. In our simulations, we made sure that the damper force magnitude is modeled such that the attenuated magnitude is representative of the force that can be achieved by semiactive dampers in practice.

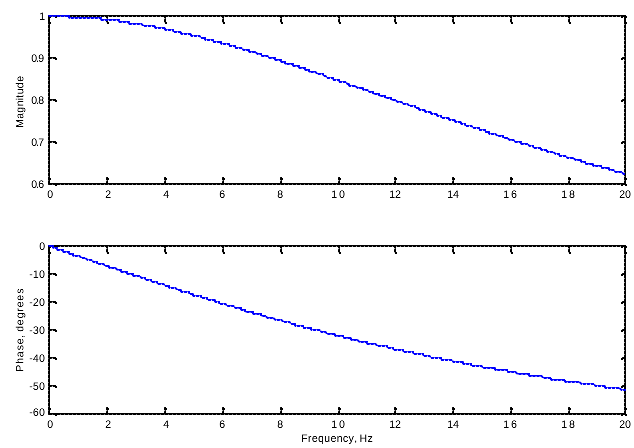

Figure 5. Frequency Response of the 1st Order Filter used for Semiactive Damper Delay

The input that will be used is a step input of the magnitude 1.27 cm, which is selected based on the amplitude of transient inputs that a vehicle suspension can be subjected to. For each suspension, we will evaluate the system output for different damping conditions, in terms of

- Sprung body settling time (simply called "settling time" on the plots): A measure of vehicle handling
- Sprung body displacement, acceleration, and jerk: Ride comfort measures
- Unsprung body displacement: A measure of vehicle handling

Additionally, for each of the three systems that we will present in the next section, we will choose a "most suitable" damping condition for the system, and evaluate the sprung body velocity, acceleration, and jerk; relative velocity across the suspension; tire acceleration and force; and damper force versus time for 2 seconds. It is, of course, recognized that the "most suitable" damping conditions that we have selected may not necessarily be the "optimal" damping, as in practice that will be depend on several different vehicle factors that the model may not adequately represent. These cases are merely intended to represent the damping conditions that look most favorable in our damping sensitivity analysis.

RESULTS

Passive Damper: The results for passive damper are presented in Figures 6 – 8, which show the commonly known compromise between the ride and handling measures [2]. As damping increases from 5% to 80 %, the sprung mass peak-to-peak acceleration and jerk increase from 2.7 m/s^2 up to 4.2 m/s^2 and from 120 m/s^3 up to 639.8 m/s^3, respectively. These increases detract from the ride quality. As damping is increased from 5% to 80%, however, settling time decreases from 14.6 seconds down to 0.75 seconds. In addition, the sprung mass peak-to-peak displacement decreases from 0.0224m to 0.006m and the unsprung mass displacement decreases from 0.014m to 0.003m. The decrease in the settling time and sprung mass displacement should benefit ride quality, while the decrease in unsprung mass displacement should improve handling. The trade off evident in Figure 6 occurs between the sprung mass acceleration and jerk verses the sprung mass settling time and displacement as well as the unsprung mass displacement. Damping cannot simply be increased without bound in an attempt to improve ride comfort and handling. Excessive amounts of damping locks out the suspension and negate its benefits.

Figure 7 shows the superposition of the step response damper force, damper velocity (relative velocity), and the sprung mass velocity. The results are as one would expect from a linear viscous damper. Figure 8 shows a large spike in sprung body acceleration due to the large damping forces that results from the large relative velocity across the suspension, caused by the step input. As we will show next, this spike will be decreased greatly by semiactive suspensions.

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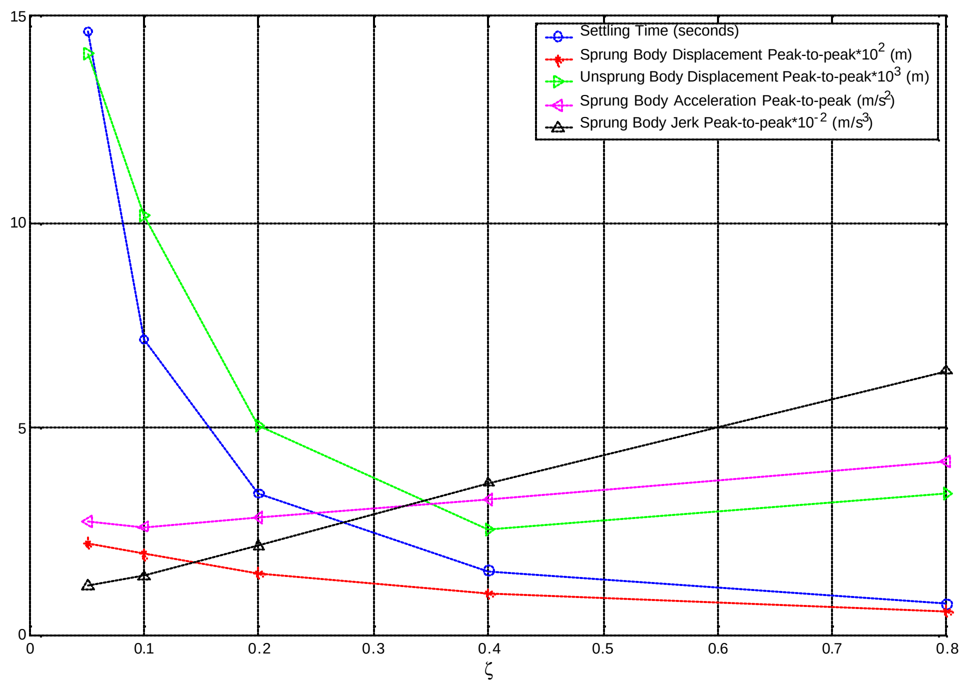

Figure 6. Effect of Varying Damping Ratio, z , on the Peak-to-peak Step Response Performance of a Single Suspension with Passive Control

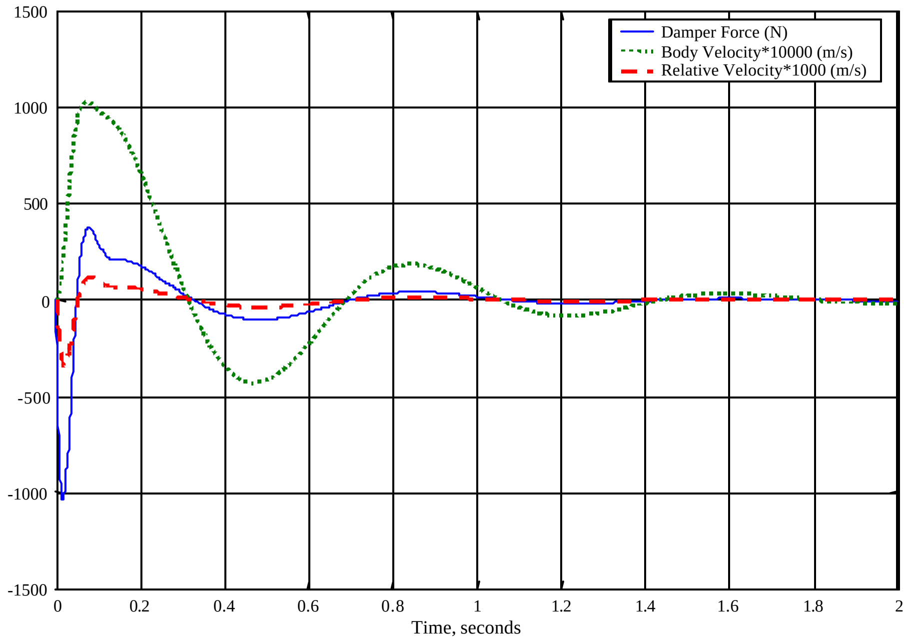

Figure 7. Step Response of a Passive Suspension for $z = 0.4$

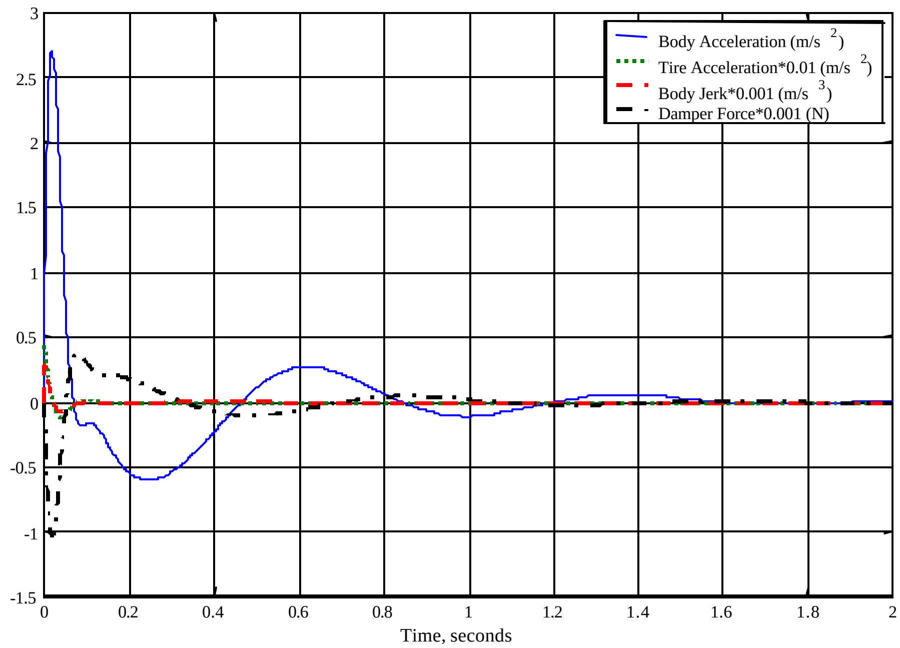

Figure 8. Step Response of a Passive Suspension for $z = 0.4$

Skyhook Damper: The tradeoffs involved in maximizing the skyhook single suspension response performance to a step input are shown in Figure 9. The transient response trade off that occurs as *G* changes from 500 N/m/s to 8000 N/m/s is between increasing the peak-to-peak jerk and decreasing the settling time, sprung and unsprung mass peak-to-peak displacements, and the sprung mass peak-to-peak acceleration. For this gain range, the settling time reduces from 9 seconds to 0.623 seconds. The sprung and unsprung mass displacements decrease from 0.022m to 0.001m and from 0.015m down to 0.012m, respectively. In addition, the peak-to-peak acceleration is reduced from 3.13 m/s^2 to 2.73 m/s^2. The peak-to-peak jerk, however, increases from 146 m/s^3 to 230 m/s^3. Since we know that jerk negatively impacts ride comfort, then an upper limit for jerk provides a practical limit on how much we should increase *G* in order to improve the other performance measures.

One advantage to using skyhook control over passive dampers is the relative insensitivity of the sprung mass acceleration step response to the control gain, *G*, as shown in Figure 9. The skyhook gain to acceleration relationship is more favorable for the ride performance than the results for passive damper, shown in Figure 6. The explanation for this relative insensitivity to changes in gain is that the skyhook control in Eq. (2) turns the damper off for the first 0.05 seconds of the solution and prevents the damper from transmitting the impact of the step to the body, as shown in Figure 10. The desired damper force is determined by the controller alone. As was discussed earlier, the actual damping force is determined by the physical constraints of the damper. As is shown in Figure 10, the actual damper force is approximately equal to the desired (or ideal) damper force when the damper is on. However, there is a minimum damping coefficient and, therefore, we cannot get a zero damper force when the damper is moving. This situation occurs during the first 0.05 seconds of the solution where the desired damper force is zero. Of course, in practice, zero damping cannot be realized, although the semiactive damper can be designed to have very low damping in its off state.

One effect of semiactive control in Eq. (2) that should not be ignored is the dynamic jerk caused by switching the damping force among different levels. Jerk for semiactive control in Figure 11 can be loosely compared with the results for passive damper in Figure 8.

The damper force-velocity characteristic gives a clear picture of defining the input/output relationship for the damper. Figure 12 contains four such plots; one each for the gains of 1000, 2000, 4000, and 8000. The damper time delay introduces a small amount of hysteresis in force-velocity plots, in the sense that at times we have positive damper forces when the damper velocity is negative. In practice, however, the hysteresis does not greatly affect the system performance, as has been shown in several past studies (e.g., [1 -2]) and is corroborated in Figure 9.

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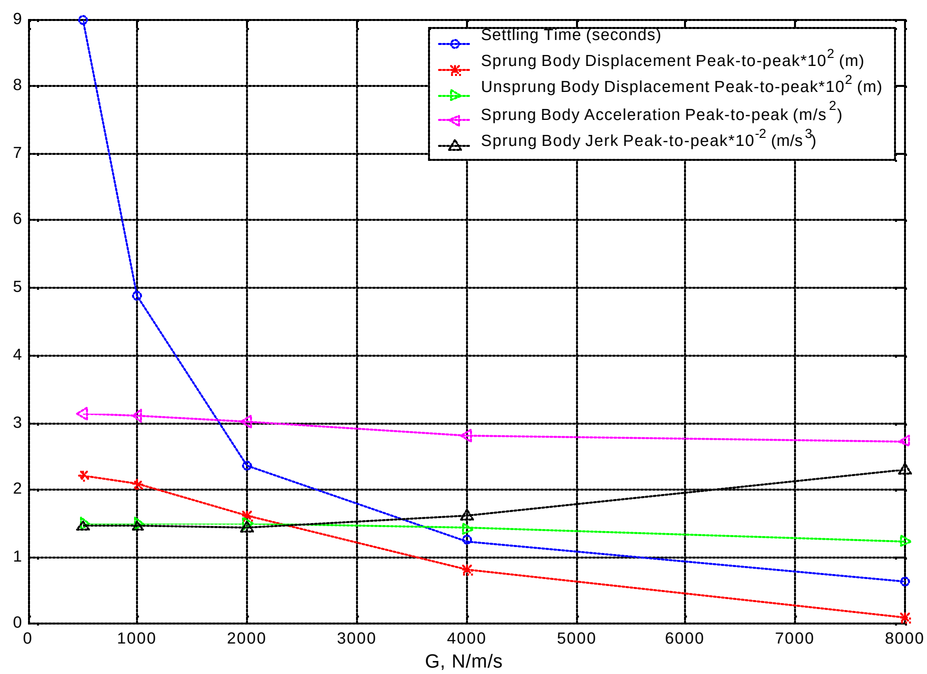

Figure 9. Effect of Varying *G* on the Peak-to-peak Step Response Performance Measures for a Single Suspension with Skyhook Damper

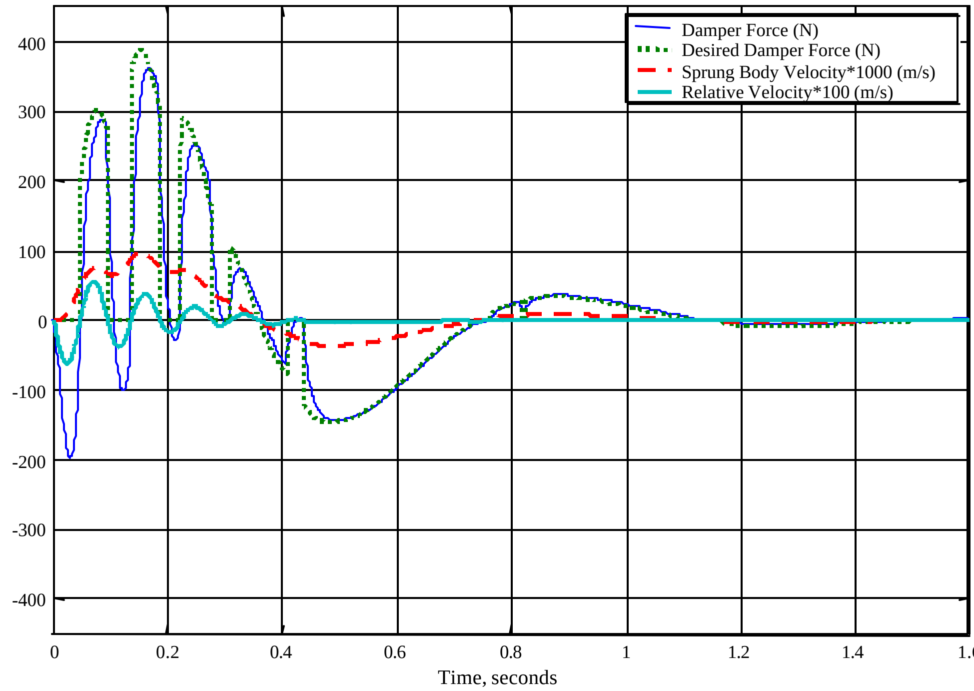

Figure 10. Step Response of a Single Suspension with Skyhook Damper for G = 4000

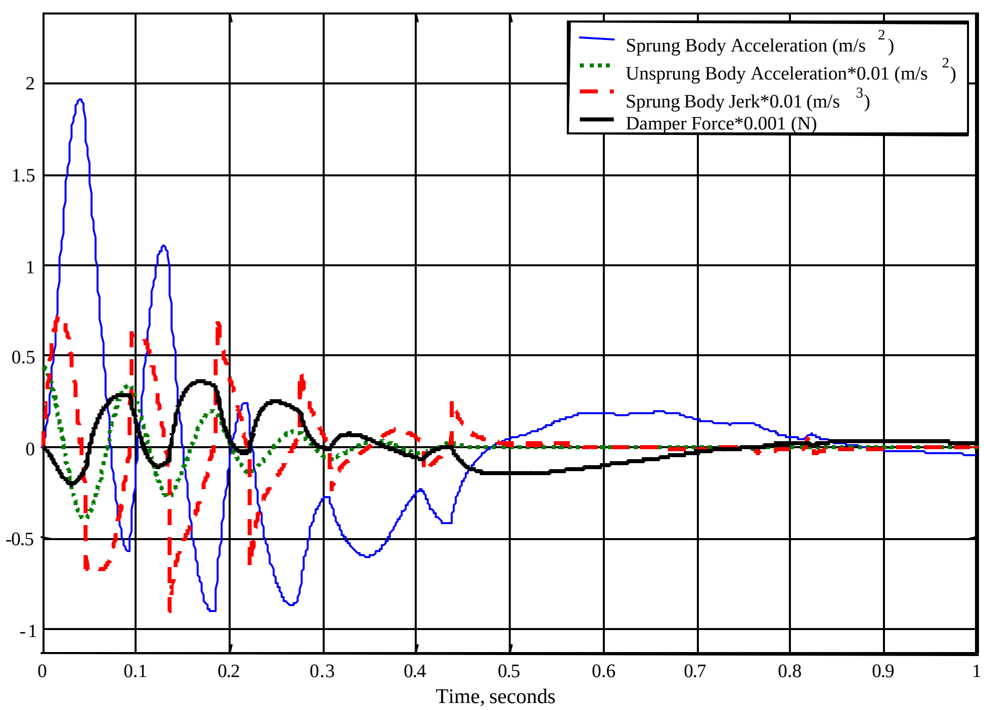

Figure 11. Step Response of a Single Suspension with Skyhook Damper for G = 4000

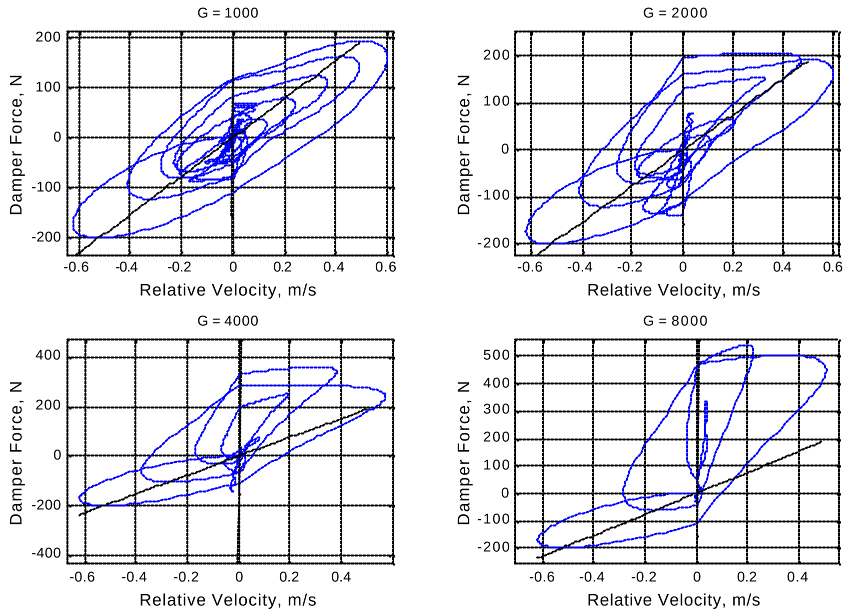

Figure 12. Comparison of Force-velocity Trajectories for a Single Suspension with Skyhook damper Subjected to a Step Input

Hybrid Control: Referring to Eq. (3), when a increases from 0.5 to 1, a trade off occurs between the unsprung mass peak-to-peak displacement, and the settling time and the sprung mass peak-to-peak displacement, acceleration, and jerk, as shown in Figure 13. Over the lower gain range of a = 0.25 to 0.5, the unsprung mass peak-to-peak displacement decreases from 0.0045m to 0.0032m, while over the upper gain range of a = 0.5 to 1 the displacement increases to 0.0143m. The increase in unsprung mass displacement would decrease the transient handling ability of the vehicle. Settling time decreases from 4.5 seconds to 1.2 seconds over the entire range of a = 0.25 to 1, indicating an improvement to ride comfort. Over the same range, we are able to observe the following:

- sprung mass peak-to-peak displacement decreases from 0.0197m to 0.0081m
- peak-to-peak acceleration decreases from 4.21 m/s^2 to 2.81 m/s^2
- peak-to-peak jerk decreases from 364 m/s^3 to 162 m/s^3

The decrease in these performance measures also indicates improvement to ride comfort. In summary, the results in Figure 13 indicate that increasing a beyond 0.5 would improve the ride comfort, but could cause degradation in vehicle handing.

As shown in Figure 14, increasing *G* for hybrid control in Eq. (3) has a similar effect as increasing passive damping in Figure 6. As *G* increases from 1000 to 8000, the settling time and the sprung and unsprung body peak-to-peak displacements decrease while the peak-to-peak acceleration and jerk increase. Over this gain range, the settling time decreases from 9 seconds to 0.86 seconds, therefore potentially improving ride comfort. The sprung and unsprung peak-to-peak displacement decreases from 0.0225m down to 0.0065m and from 0.0139m down to 0.0028m, respectively. This decrease would improve ride comfort as well as vehicle handling. Increasing G from 1000 to 8000, however, increases acceleration from 3.11 m/s^2 up to 4.50 m/s^2, and jerk from 155 m/s^3 up to 491 m/s^3. These increases should detract from ride comfort. Overall, Figure 14 indicates that increasing the gain G would diminish ride comfort, but would improve vehicle handling.

The transient behavior of hybrid control can be studied further by examining Figure 15, which show that the actual damper force is approximately equal to the desired (or ideal) damper force. One reason for the similarity in the actual and desired damper force is that the desired damper force usually has the same sign as the damper velocity (relative velocity). Since the damper velocity direction is consistent with the desired damper force direction, the switches do not intercede in a way that turns the damper force to zero. The reason for the damper being always on can be best explained by studying the hybrid control in Eq. (3). For this example, we will concentrate on the first ½ cycle of the transient response. During the ½ cycle, the relative velocity, v_{12}, is of opposite sign to the sprung body velocity, v_1, and their product is therefore negative. According to Eq. (2), the skyhook component of the desired damper force is zero. Additionally, the product of the relative velocity, v_{12}, and the unsprung velocity, v_2, is negative, indicating that the groundhook component of Eq. (3) is nonzero. Therefore, the total desired damper force is nonzero, meaning that the damper would be on.

Comparing Figures 16 and 11 reveals that hybrid control causes a larger jerk than skyhook control. Additionally, Figures 17 and 18 show the force-velocity trajectories for G= 4000 and α = 0.5, respectively. The trajectory in Figures 17 and 18 include the damper delay, although the sense of time is lost in these trajectories, due to their convoluted nature. It suffices to mention that our experience has shown that the damper delay does not have a significant effect on the benefits of semiactive dampers for vehicle suspensions.

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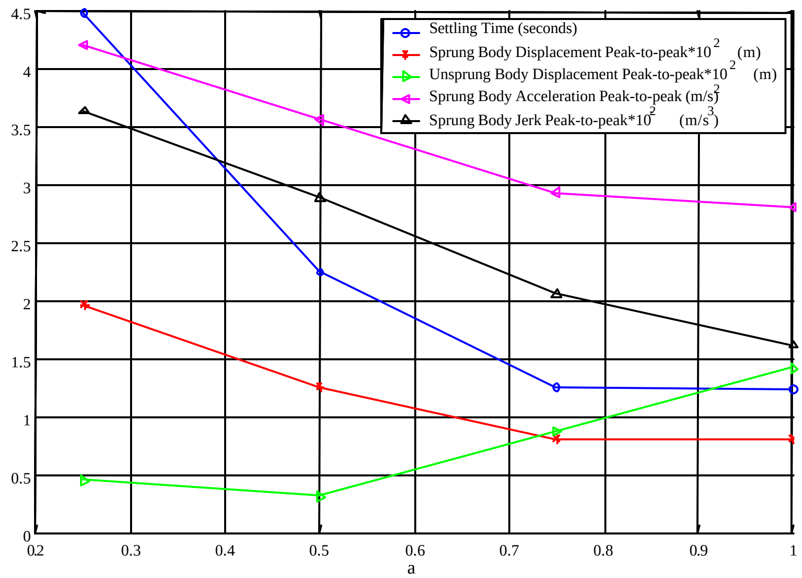

Figure 13. Effect of Varying *a* on the Peak-to-peak Response of a Single Suspension with Hybrid Control for *G* = 4000, Subjected to a Step Input

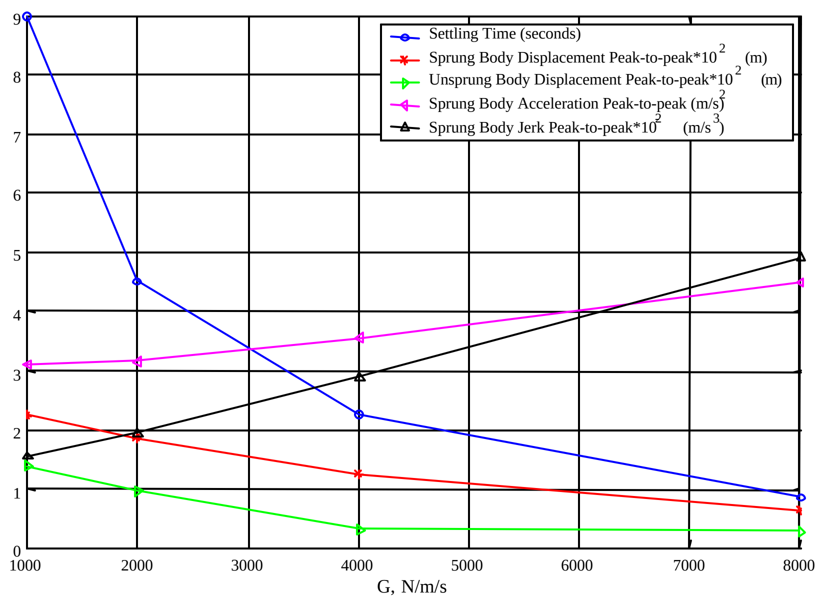

Figure 14. Effect of Varying *G* on the Peak-to-peak Response of a Single Suspension with Hybrid Control for *a* = 0.5, Subjected to a Step Input

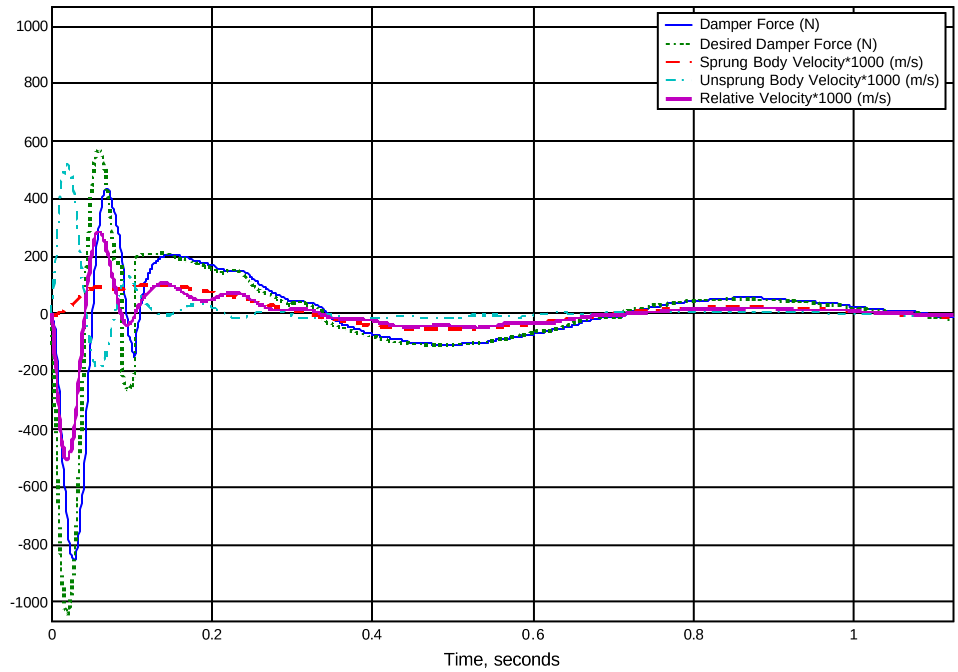

Figure 15. Step Response of a Single Suspension with Hybrid Control for G = 4000 and α = 0.5

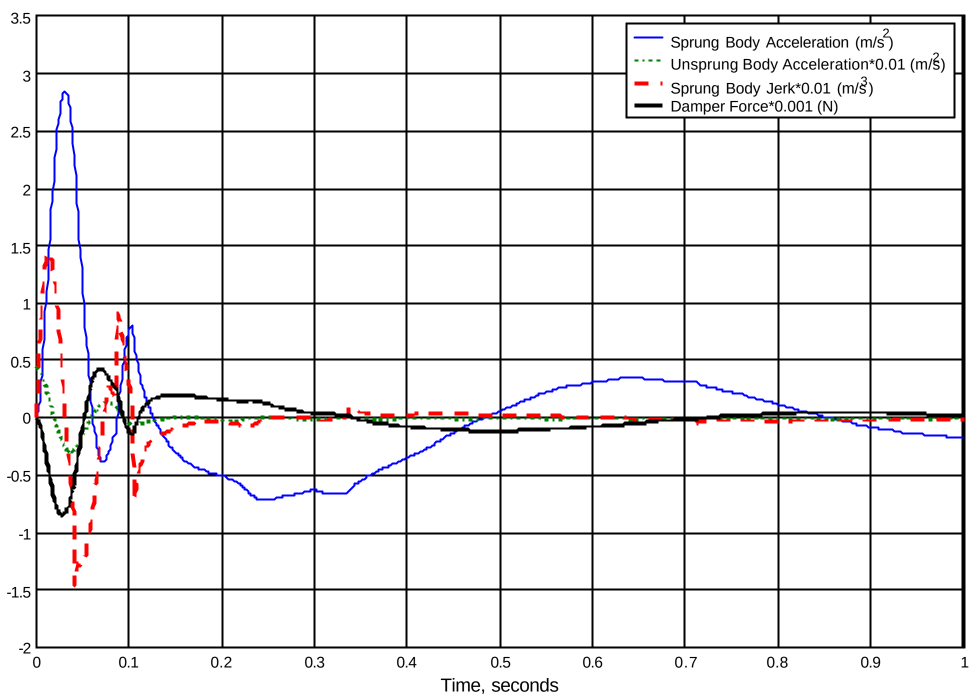

Figure 16. Step Response of a Single Suspension with Hybrid Control for
G = 4000 and α = 0.5

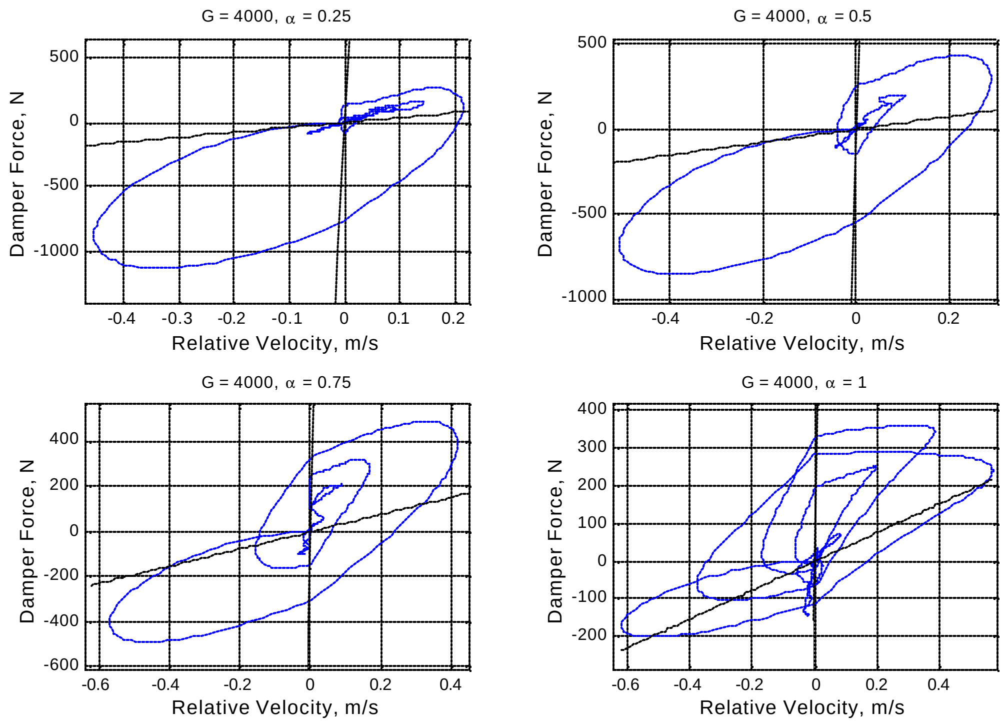

Figure 17. Comparison of Force-velocity Trajectories for a Single Suspension with Hybrid Control for *G* = 4000

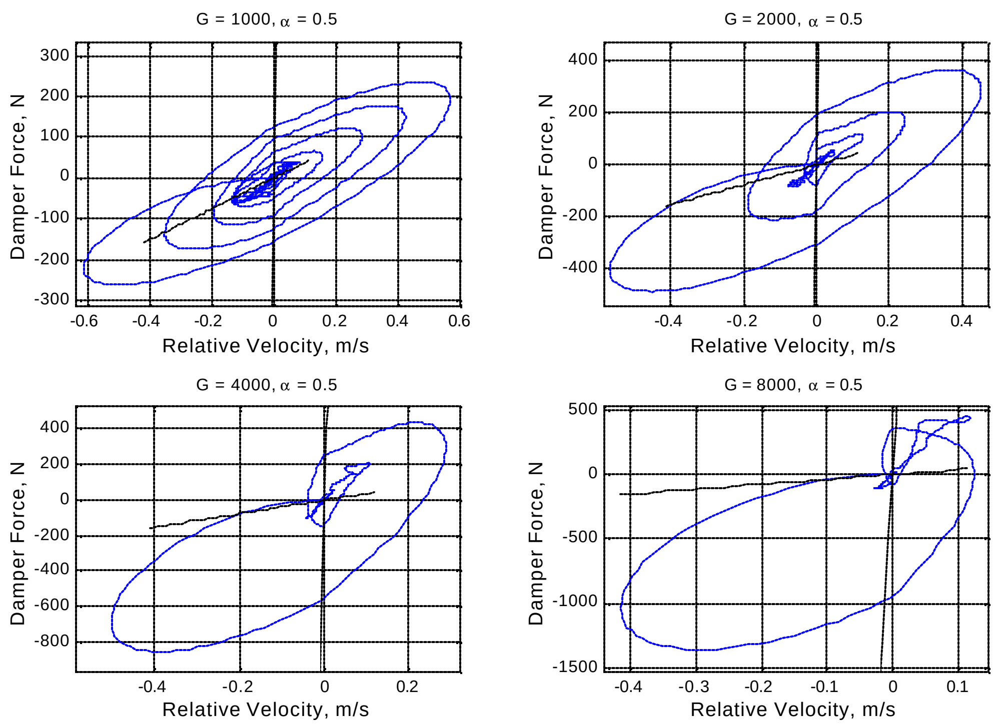

Figure 18. Comparison of Force-velocity Trajectories for a Single Suspension with Hybrid Control for a = 0.5

CONCLUSIONS

An introduction was provided to semiactive dampers that have been widely considered for vehicle suspension applications. The studies for analyzing the steady-state dynamics of semiactive systems were extended by evaluating the transient dynamics of such systems, for two of the control methods that are most often considered for semiactive systems. Using a single suspension model, we evaluated the effect of skyhook and hybrid control on improving the transient dynamics of a vehicle suspension, in terms of providing better ride comfort and vehicle handling. Using measures such as sprung body (vehicle body) displacement, acceleration, and jerk, as well as unsprung body (axle-tire assembly) displacement, velocity, and acceleration, we were able to show that the semiactive control techniques that were considered in this study could improve various aspects of vehicle ride comfort and handling, as compared with conventional passive dampers. Of course, the preliminary results that are offered in this study through the analysis of a single suspension model need to be evaluated in the field with vehicle testing, in order to provide further validation and verification of the true impact of semiactive suspensions on improving vehicle transient dynamics.

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