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INFLUENCE OF SUSPENSION ON THE VEHICLE DYNAMIC PERFORMANCE

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Abstract. A typical vehicle suspension is made up of two components: a spring and a damper. The spring is chosen based solely on the weight of the vehicle, while the damper is the component that defines the suspension's placement on the compromise curve.

Depending on the type of vehicle, a damper is chosen to make the vehicle perform best in its application. Ideally, the damper should isolate passengers from low-frequency road disturbances and absorb high-frequency road disturbances.

1. INTRODUCTION

A typical vehicle suspension is made up of two components: a spring and a damper. The spring is chosen based solely on the weight of the vehicle, while the damper is the component that defines the suspension's placement on the compromise curve.

Depending on the type of vehicle, a damper is chosen to make the vehicle perform best in its application. Ideally, the damper should isolate passengers from low-frequency road disturbances and absorb high-frequency road disturbances. Passengers are best isolated from low-frequency disturbances when the damping is high. However, high damping provides poor high frequency absorption. Conversely, when the damping is low, the damper offers sufficient high-frequency absorption, at the expense of low-frequency isolation.

Three types of suspensions that will be analyzed here are passive, fully active, and semi-active suspensions. A conventional passive suspension is composed of a spring and a damper. The suspension stores energy in the spring and dissipates energy through the damper. Both components are fixed at the design stage. For this reason, this type of suspension falls victim to the classic suspension compromise.

If the damper is replaced with a force actuator, the suspension becomes a fully active suspension. Hindered by its complexity and its power consumption, fully active suspensions have yet to be accepted for conventional use. The idea behind fully active suspensions is that the force actuator is able to apply a force to the suspension in either jounce or rebound. This force is actively governed by the control scheme employed in the suspension. The third and final type of suspension that will be mentioned here is a semi-active suspension. In a semi-active suspension, the passive damper is replaced with a semi-active damper. A semi-active damper is capable of changing its damping characteristics.

Whether through mechanically changing orifices or fluid with adjustable viscosity, a semi-active damper offers greater variation in damping.

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2. PASSIVE SUSPENSION SYSTEM

A typical vehicle primary suspension can be modeled as shown in Figure 1. The model represents a single suspension from one of the four corners of the vehicle.

The input to this model is a displacement input which is representative of a typical road profile. The input excites the first degree of freedom (the unsprung mass of a quarter of the vehicle, representing the wheel, tire, and some suspension components) through a

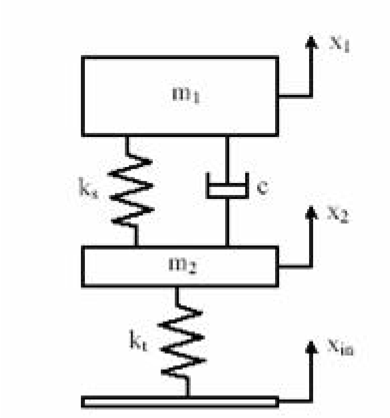

Figure 1. Passive suspension

spring element which represents the tire stiffness. The unsprung mass is connected to the second degree of freedom (the sprung mass, representing the body of the vehicle) through the primary suspension spring and damper.

As the damping is increased, the resonant peaks are attenuated, but isolation is lost both at high frequency and at frequencies between the two natural frequencies of the system. The lack of isolation between the two natural frequencies is caused by the increased coupling of the two degrees of freedom with a stiffer damper. The lack of isolation at higher frequencies will result in a harsher vehicle ride.

The equations of motion for the system can be written in matrix form as

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_1 \end{bmatrix}\begin{Bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{Bmatrix} + \begin{bmatrix} c & -c \\ -c & c \end{bmatrix}\begin{Bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{Bmatrix} + \begin{bmatrix} k_1 & -k_1 \\ -k_1 & k_1+k_2 \end{bmatrix}\begin{Bmatrix} x_1 \\ x_2 \end{Bmatrix} = \begin{bmatrix} 0 \\ k_2 \end{bmatrix} x_{in} \tag{1}$$

Knowing the physical parameters of the system, we can calculate the damping ratio. To facilitate this calculation, we will treat the system as two systems. In order to present the transmissibility plots as a function of damping ratio rather than damping coefficient, we can decouple the equations of motion by neglecting the off-diagonal terms, and then estimate the damping ratio for each mass as

$$V_1 = \frac{c}{2\sqrt{k_1 m_1}} \tag{2}$$

$$V_2 = \frac{c}{2\sqrt{(k_1+k_2)m_2}} \tag{3}$$

While this method of calculating the damping ratio is only valid at low damping, the intent is not to precisely define the damping ratio, but rather to show the effects of increased damping on transmissibility.

3. CONTROL SCHEMES FOR SYSTEM

Skyhook, groundhook, and hybrid semi-active control will be presented and compared with a typical passive suspension.

3.1 SKYHOOK CONTROL

As the name implies, the skyhook configuration shown in Figure 2 has a damper connected to some inertial reference in the sky. With the skyhook configuration, the tradeoff between resonance control and high-frequency isolation, common in passive suspensions, is eliminated [3].

Notice that skyhook control focuses on the sprung mass: as c_{sky} increases, the sprung mass motion decreases. This, of course, comes at a cost. The skyhook configuration excels at isolating the sprung mass from base excitations, at the expense of increased unsprung mass motion.

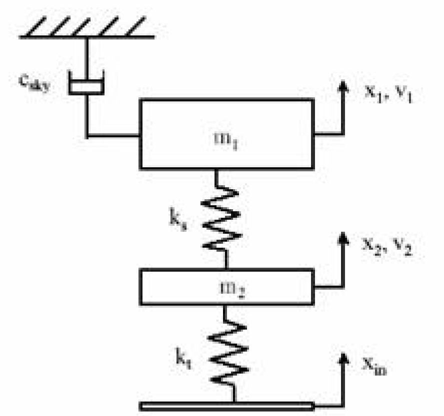

Figure 2. Skyhook Damper Configuration

In essence, this skyhook configuration is adding more damping to the sprung mass and taking away damping from the unsprung mass. The skyhook configuration is ideal if the primary goal is isolating the sprung mass from base excitations [4], even at the expense of excessive unsprung mass motion. An additional benefit is apparent in the frequency range between the two natural frequencies. With the skyhook configuration, isolation in this region actually increases with increasing c_{sky}. Because this damper configuration is not possible in realistic automotive applications, a controllable damper is often used to achieve a similar response to the system modeled in Figure 2. The semi-active damper is commanded such that it acts like a damper connected to an inertial reference in the sky. Figure 3 shows the semi-active equivalent model with the use of a semi-active damper.

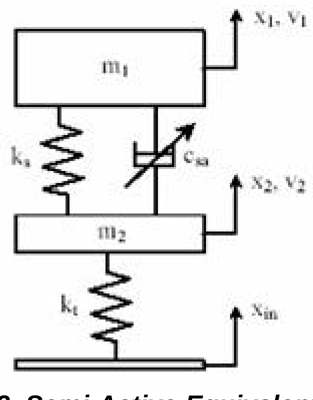

Figure 3. Semi-Active Equivalent Model

Several methods exist for representing the equivalent skyhook damping force with the configuration shown in Figure 3. Perhaps the most comprehensive way to arrive at the equivalent skyhook damping force is to examine the forces on the sprung mass under several conditions. First, let us define certain parameters and conventions that will be used throughout controller development. Referring to Figure 3, the relative velocity, v_{12}, is defined as the velocity of the sprung mass (m_1) relative to the unsprung mass (m_2). When the two masses are separating, v_{12} is positive. For all other cases, up is positive and down is negative. Now, with these definitions, let us consider the case when the sprung mass is moving upwards and the two masses are separating.

Under the ideal skyhook configuration we find that the force due to the skyhook damper is

$$F_{sky} = -c_{sky} v_1 \tag{4}$$

where F_{sky} is the skyhook damping force.

Next we examine the semi-active equivalent model and find that the damper is in tension and the damping force due to the semi-active damper is

$$F_{sa} = -c_{sa} v_{12} \tag{5}$$

where F_{sa} is the semi-active damping force.

Now, in order for the semi-active equivalent model to perform like the skyhook model, the damping forces must be equal, or

$$F_{sky} = -c_{sky} v_1 = -c_{sa} v_{12} = F_{sa} \tag{6}$$

We can solve for the semi-active damping in terms of the skyhook damping (7) and use this to find the semi-active damping force needed to represent skyhook damping when both v_1 and v_{12} are positive (8).

$$c_{sa} = \frac{c_{sky} v_1}{v_{12}} \tag{7}$$

$$F_{sa} = c_{sky} v_1 \tag{8}$$

Next, let us consider the case when both v_1 and v_{12} are negative. Now the sprung mass is moving down and the two masses are coming together. In this scenario, the skyhook damping force would be in the positive direction, or

$$F_{sky} = c_{sky} v_1 \tag{9}$$

Likewise, because the semi-active damper is in compression, the force due to the semi-active damper is also positive, or

$$F_{sa} = c_{sa} v_{12} \tag{10}$$

Following the same procedure as the first case, equating the damping forces reveals the same semi-active damping force as the first case. Thus, we can conclude that when the product of the two velocities is positive, the semi-active force is defined by equation (8).

Now consider the case when the sprung mass is moving upwards and the two masses are coming together. The skyhook damper would again apply a force on the sprung mass in the negative direction. In this case, the semi-active damper is in compression and cannot apply a force in the same direction as the skyhook damper. For this reason, we would want to minimize the damping, thus minimizing the force on the sprung mass.

The final case to consider is the case when the sprung mass is moving downwards and the two masses are separating. Again, under this condition the skyhook damping force and the semi-active damping force are not in the same direction. The skyhook damping force would be in the positive direction, while the semi-active damping force would be in the negative direction. The best that can be achieved is to minimize the damping in the semi-active damper.

Summarizing these four conditions, we arrive at the well-known semi-active skyhook control policy:

$$\begin{Bmatrix} v_1 v_{12} \geq 0 & F_{sa} = c_{sky} v_1 \\ v_1 v_{12} < 0 & F_{sa} = 0 \end{Bmatrix} \tag{11}$$

It is worth emphasizing that when the product of the two velocities is positive that the semi-active damping force is proportional to the velocity of the sprung mass. Otherwise, the semi-active damping force is at a minimum.

3.2 GROUNDHOOK CONTROL

The groundhook model differs from the skyhook model in that the damper is now connected to the unsprung mass rather than the sprung mass. This modified configuration is shown in Figure 4.

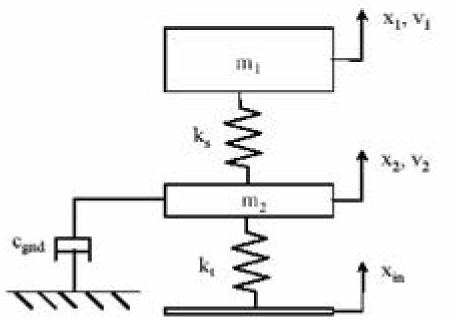

Figure 4. Groundhook Damper Configuration

Under the groundhook configuration, the focus shifts from the sprung mass to the unsprung mass. As skyhook control excelled at isolating the sprung mass from base excitations, groundhook control performs just as well at isolating the unsprung mass from base excitations. Again, this performance comes at the cost of excessive sprung mass motion. The groundhook configuration effectively adds damping to the unsprung mass and removes it from the sprung mass.

Through the same reasoning used for skyhook control, it can easily be shown that the groundhook semi-active control policy reduces to:

$$\begin{Bmatrix} -v_1 v_{12} \geq 0 & F_{sa} = c_{gnd} v_2 \\ -v_1 v_{12} < 0 & F_{sa} = 0 \end{Bmatrix} \qquad (12)$$

3.3 HYBRID CONTROL

An alternative semi-active control policy known as hybrid control has been shown to take advantage of the benefits of both skyhook and groundhook control [1]. With hybrid control, the user has the ability to specify how closely the controller emulates skyhook or groundhook. In other words, hybrid control can divert the damping energy to the bodies in a manner that eliminates the compromise that is inherent in passive dampers. The hybrid configuration is shown in Figure 5.

Using hybrid control, the user can specify how closely the controller resembles skyhook or groundhook. Combining the equations (11) and (12) we arrive at the semi-active hybrid control policy:

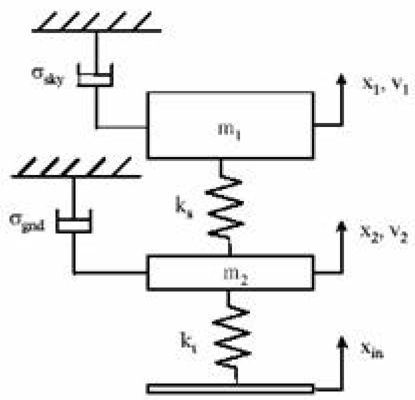

Figure 5. Hybrid Configuration

$$\begin{Bmatrix} v_1 v_{12} \geq 0 & d_{gnd} = v_1 \\ v_1 v_{12} < 0 & d_{gnd} = 0 \end{Bmatrix} \quad \begin{Bmatrix} -v_1 v_{12} \geq 0 & d_{gnd} = v_2 \\ -v_1 v_{12} < 0 & d_{gnd} = 0 \end{Bmatrix} \qquad \{F_{sa} = G\lfloor a\, d_{sky} + (1-a) d_{gnd} \rfloor\} \tag{13}$$

where s_{sky} and s_{gnd} are the skyhook and groundhook components of the damping force. The variable a is the relative ratio between the skyhook and groundhook control, and G is a constant gain.

3.4 ALTERNATIVE SKYHOOK CONTROL POLICIES

This research also explored the benefits of two alternative skyhook based control strategies. The physical representation of these two policies remains unchanged from conventional skyhook. Distinguishing these two controllers from skyhook is the condition by which the damper force is applied. The first alternative skyhook policy is a displacement based skyhook algorithm and can be described by:

$$\begin{Bmatrix} x_1 v_{12} \geq 0 & F_{sa} = c_{sky} v_1 \\ x_1 v_{12} < 0 & F_{sa} = 0 \end{Bmatrix} \tag{14}$$

In the case of displacement skyhook, the semi-active force, F_{sa}, remains the same.

However, now the governing condition is the product of x_1 and v_{12} rather than v_1 and v_{12}.

The second alternative skyhook control policy is relative displacement skyhook.

Relative displacement skyhook can be expressed as:

$$\begin{Bmatrix} v_1 x_{12} \geq 0 & F_{sa} = c_{sky} v_1 \\ v_1 x_{12} < 0 & F_{sa} = 0 \end{Bmatrix} \tag{15}$$

where now the governing condition is the product of v_1 and x_{12}. The semi-active force remains the same. The two alternative skyhook policies were evaluated for their benefits relative to traditional skyhook control.

4. CONCLUSIONS

A semi-active damper is capable of changing its damping characteristics. Whether through mechanically changing orifices or fluid with adjustable viscosity, a semi-active damper offers greater variation in damping.

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