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# Adaptive hybrid control strategy for semi-active suspension system

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**Abstract**: A hybrid control strategy for semi-active suspension provides a good compromise between comfort and handling. Based on the hybrid coefficient the suspension can be biased towards a skyhook or a groundhook system. To provide an optimum value of the hybrid coefficient an adaptive hybrid strategy is presented in this paper. The proposed adaptive hybrid system is implemented on a two-state damper and is compared with a conventional hybrid system with fixed value of the hybrid coefficient. A quarter-car model is used to validate the results. The simulation results indicate that the proposed adaptive hybrid control system can perform better than a conventional hybrid control system.

## 1. Introduction

There is no need of suspension for smooth roads. But, the road surfaces are not uniform and it has bumps, potholes, etc. Therefore, it is necessary for the Suspension system to isolate this unevenness to the vehicle structure. Suspension is one of the most important components in the vehicles because it helps to improve better drivability, ride comfort, passenger safety, manoeuvrability of vehicle [1]. The major component in suspension system of a vehicle consists of spring, damper, wishbone. The spring is used to isolate the vehicle body from the road disturbances to provide better ride comfort. The damper should always provide a better contact between tire and road surface to make better drivability [2]. Based on the vehicle type the damper is chosen. A vehicle with 'hard' suspension refers to stiffer spring and firm damper. This offers good control over the motion of vehicle body and wheels vibrations, and it provides best handling. However, this system is not capable of providing effective body isolation i.e. better rider comfort. In contrast, a suspension system with lower spring stiffness and soft damping, is called 'soft' suspension provides nominal body isolation from road disturbances and provides better ride comfort. However, this system is not capable of providing effective road handling [3].

        To overcome these limitations and the effects of this compromise gave the development of three types of suspension system in automotive industries namely, Passive suspension system, active suspension system, semi active suspension system. The passive system consists of a spring and damper. In this, the energy is stored in spring during bump and dissipates this stored energy through the damper. In this, both the components characteristics are fixed at the design stage.
If the damper is replaced by the force actuator, then the suspension becomes fully active suspension system. The main drawback of this system is increased complexity and its more power consumption to actuate the force actuator during rebound or jounce. In case of power failure, the entire active suspension stalls [4]. The disadvantages of active suspension system lead to development of semi

active suspension in which, only damper from conventional sprung mass system is replaced by variable damper. This variation in damping can be achieved by two ways. One is the method of varying the orifice diameter in the damper and second method is by using special fluids like magneto rheological (MR) and electro rheological (ER) fluids. These fluids have varying viscosity as the function of magnetic or electrical excitation. Due to their low power consumption, simplicity and performance, semi active suspensions are preferred over active suspension [5].

The main purpose of this study is to present an adaptive hybrid control system which provides the value of the hybrid coefficient dynamically in order to optimize the performance of a conventional hybrid control system.

## 2. Suspension Modelling

### 2.1. 2DOF Quarter-Car Model

To evaluate the response of a semi-active suspension control strategy, a 2 degree of suspension model is used. As it represents one single suspension unit it is also known as the quarter car model. This model includes the sprung mass, unsprung mass, suspension spring and damper. It models the tire as a spring element with a pre-defined stiffness value.
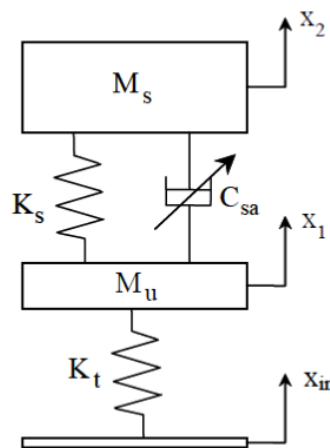


Figure 1. 2DOF Quarter-Car Model

The damping coefficient can be constant for a passive suspension system whereas for a semi-active suspension system it is a variable and depends upon the control strategy. The input to the quarter car model is the road disturbance based on the road profile.
The equation of motion for the 2DOF system can be given as [6],

$$M_s\ddot{x}_2 + K_s(x_2 - x_1) + C_{sa}(\dot{x}_2 - \dot{x}_1) = 0$$

$$M_u\ddot{x}_1 - K_s(x_2 - x_1) + K_t(x_1 - x_{in}) - C_{sa}(\dot{x}_2 - \dot{x}_1) = 0$$

The model parameters used in this study are represented as follows,

| Parameter | Value |
|---|---|
| Sprung Mass ($M_s$) | 240 kg |
| Unsprung Mass ($M_u$) | 36 kg |
| Suspension Spring Stiffness ($K_s$) | 16000 N/m |
| Tyre Stiffness ($K_s$) | 160000 N/m |

Table 1. Quarter-car parameters

The value of the damping coefficient is chosen to model the system. Based on the value of damping coefficient the damping ratio can be given as,

$$\zeta_s = \frac{C_s}{2\sqrt{K_s M_s}}$$

Here, $C_s$ is the damping coefficient of a passive damper. The value of $C_s$ is considered as,

$$C_s = 196 \; Ns/m \qquad (\zeta_s = 0.050)$$

For a semi-active suspension system the values of $C_{on}$ and $C_{off}$ for an ON-OFF state can be chosen as follows [7],

$$C_{on} = 2.2 C_s \text{ and } C_{off} = 0.2 C_s$$

For a conventional hybrid systems the following pair of damping coefficients are chosen,

$$C_{on} = 431.2 \; Ns/m, C_{off} = 39.2 \; Ns/m$$

*2.2. Hybrid Control Algorithm*

An ideal skyhook configuration mainly prioritizes the sprung mass damping and reduces the same by attaching the sprung mass to a static point in the sky via a damper. This configuration highly dampens the high frequency at of the sprung mass. But the downside of this is that as the sprung mass damping is increased, the effective damping at the unsprung mass will decrease.
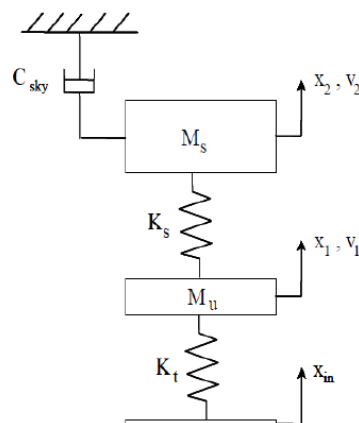


Figure 2. Skyhook Configuration

But this configuration is not realistic as we cannot attach a damper to a fiction point to the sky. Hence, we need to resolve this configuration to a realistic setup where it behaves in the same way as like the system in Figure 1. The value of the damping for system in Figure 2 is given by

$$C_{sa} = \frac{C_{sky} \times \dot{x}_2}{\dot{x}_{12}}$$

where,

$C_{sky}$      = skyhook damping coefficient
$\dot{x}_2$   = Sprung mass displacement
$\dot{x}_{12}$  = Relative displacement between Sprung and unsprung mass

Another way to achieve the skyhook configuration is by implementing a two-state control to a varying damper. This control is governed by two conditions [8]:

$$if \quad \dot{x}_2 \times \dot{x}_{12} \geq 0 \; ; then \; C_{sa} = C_{max}$$
$$else \; \dot{x}_2 \times \dot{x}_{12} < 0; then \; C_{sa} = C_{min}$$

Where, $C_{min}$ and $C_{max}$ are the minimum and maximum values of the damping coefficients respectively. The damping factor is being governed based on the sprung mass displacement and the relative displacement of the sprung and unsprung masses.

Unlike the Skyhook configuration, the Groundhook configuration has the Unsprung mass attached to a fiction point on the ground via a damper. This configuration prioritizes the unsprung mass and dampens it from road excitations. Likewise, this has a downside of reduction of the damping on the sprung mass with an increase in damping of the unsprung mass.
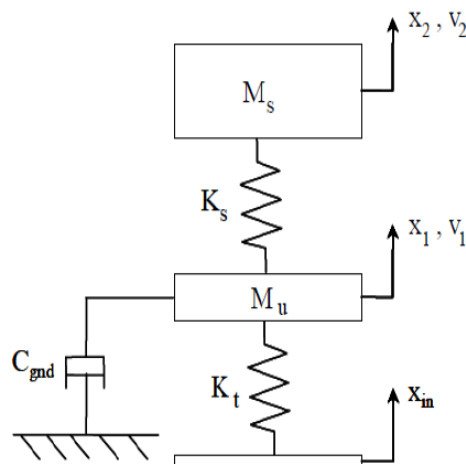


Figure 3. Groundhook Configuration

A similar control method can be implemented to this configuration where a variable damper (between the unsprung mass and the sprung mass) is controlled by varying the damping coefficient between two states based on the following conditions [9]:

$$if \quad -\dot{x}_1 \times \dot{x}_{12} \geq 0 \; ; then \; C_{sa} = C_{max}$$
$$else \; -\dot{x}_1 \times \dot{x}_{12} < 0; then \; C_{sa} = C_{min}$$

The hybrid control strategy combines both the skyhook and groundhook controls. This strategy allows the user to define a variable α which can specify the ratio of the skyhook or groundhook damping coefficient.
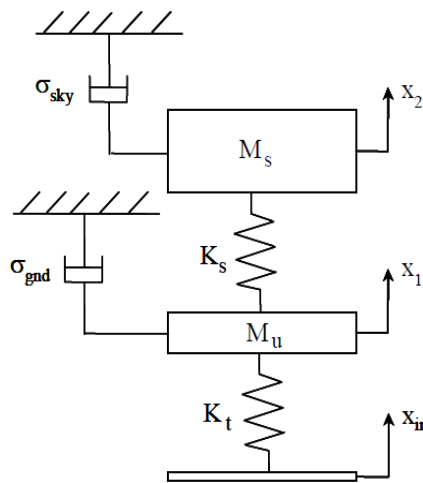
Figure 4. Hybrid Configuration

The hybrid control is given as [10],

$$\dot{x}_2(\dot{x}_2 - \dot{x}_1) \geq 0 \; \sigma_{sky} = \dot{x}_2$$
$$\dot{x}_2(\dot{x}_2 - \dot{x}_1) < 0 \; \sigma_{sky} = 0$$
$$-\dot{x}_1(\dot{x}_2 - \dot{x}_1) \geq 0 \; \sigma_{gnd} = \dot{x}_1$$
$$-\dot{x}_1(\dot{x}_2 - \dot{x}_1) < 0 \; \sigma_{gnd} = 0$$
$$F_{sa} = G\big[\alpha\sigma_{sky} + (1-\alpha)\sigma_{gnd}\big]$$

Where $\sigma_{sky}$ and $\sigma_{gnd}$ are skyhook and groundhook components of the damping force. When $\alpha$ is 1 the control is purely skyhook, whereas when $\alpha$ is 0 the control is purely groundhook.

The passive representation of semi-active dampers controlled by hybrid policy is shown in figure 5.
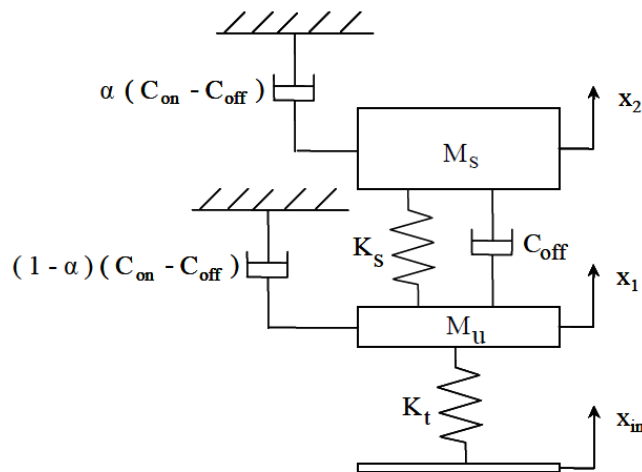


Figure 5. Passive Representation of Hybrid Control

The 2DOF quarter car model based on the shown fig is modelled in Simulink.

### 2.3. Adaptive Hybrid Control

The response of the hybrid control system is based on the α value according to which the system will be biased to some extent towards skyhook or groundhook control policy. The system can be improved further by defining a method to control the value of α dynamically for a system. This ensures that an optimised value of α is chosen based on the state of system. The benefit of implementing such a strategy ensures reduction in complexity of the control system as well as the onboard processing time. In this paper, a strategy which gives three different values of α based on the following conditions are implemented.

$$if, \qquad \dot{x}_2(\dot{x}_2 - \dot{x}_1) \geq 0 \qquad \alpha = 0.9$$
$$else\ if, \qquad -\dot{x}_1(\dot{x}_2 - \dot{x}_1) \geq 0 \qquad \alpha = 0.1$$
$$else, \qquad \alpha = 0.5$$

This strategy ensures that the system is biased towards skyhook when condition for skyhook is suitable i.e. the sprung mass requires damping, and towards groundhook when condition for groundhook is suitable i.e. the unsprung mass requires damping. The value of α is chosen such that the system never becomes a full skyhook or groundhook configuration. This also ensures that there is no sudden jerk in the system due to shifting from skyhook to groundhook or vice-versa. If the skyhook or groundhook conditions are not satisfied this strategy gives a value of α as 0.5 which means that the damping force is distributed equally between the sprung and unsprung mass.

## 3. Simulation and discussion

### 3.1. Responses of Interest
To compare the proposed strategy's performance with a conventional hybrid control system (α = 0.5) the following key responses of the system are studied. The responses of the system are shown in Appendix.

*3.1.1. Sprung and Unsprung Mass Transmissibility:* The transmissibility curve gives us the amplitude values at resonance frequencies of the system. From this analysis the resonance frequencies of the system are found to be at 1.24 Hz and 11 Hz. The sprung mass transmissibility curve shows a decrease in amplitude at 1.24 Hz from 8.27 to 6.32 (Appendix Figure 4). For the unsprung mass there is a decrease in amplitude from 6.57 to 5.83 at 11 Hz (Appendix Figure 5).

*3.1.2. Sprung Mass Acceleration:* This criterion is a representation of vehicle ride comfort. Higher the sprung mass acceleration for a given road input lesser the comfort. The proposed system could reduce the amplitude for this criterion from 65.8 to 49.1 (Appendix Figure 6).

*3.1.3. Tire deflection:* This criterion represents the road holding ability of the vehicle and is concerned with the ride handling. Higher the tire deflection lower will be the road holding capability. The proposed system was able to reduce the amplitude for this criterion from 0.1 to 0.08 (Appendix Figure 7).

*3.1.4. Suspension deflection:* This criterion is concerned with the amount of space required for the suspension. It is always better to have a suspension system which can provide better performance while taking lesser area as the vehicle's centre of gravity can be lowered by using a suspension system which takes smaller space. The proposed system was able to reduce the amplitude of this criterion from 0.97 to 0.723 (Appendix Figure 8).

| Parameter | Hybrid | Adaptive Hybrid |
|---|---|---|
| Sprung Mass Transmissibility | 8.27 @ 1.23 Hz | 6.32 @ 1.23 Hz |
| Unsprung Mass Transmissibility | 6.57 @ 11 Hz | 5.83 @ 11 Hz |
| Sprung Mass Acceleration | 65.8 @ 1.24 Hz | 49.1 @ 1.24 Hz |
| Road Holding | 0.1 @ 1.24 Hz<br>0.0966 @ 11.1 Hz | 0.0749 @ 1.23Hz<br>0.08 @ 11.1 Hz |
| Suspension Deflection | 0.97 @1.24 Hz | 0.723 @ 1.24 Hz |

Table 2. Comparison of Hybrid with Adaptive Hybrid

## 4.  Conclusion and Future work

An adaptive hybrid control strategy was presented which provides the value of the hybrid coefficient dynamically. The proposed strategy was implemented on a two-state damper. The performance of the system was compared with that of a conventional hybrid control with hybrid coefficient as 0.5. The response of the system was found to be better in all the performance parameters. In future, a linear control can be designed which can be implemented on a linear hybrid control system rather than a two-state system. The value of the hybrid coefficient can also be optimized for the different performance parameters.

## References

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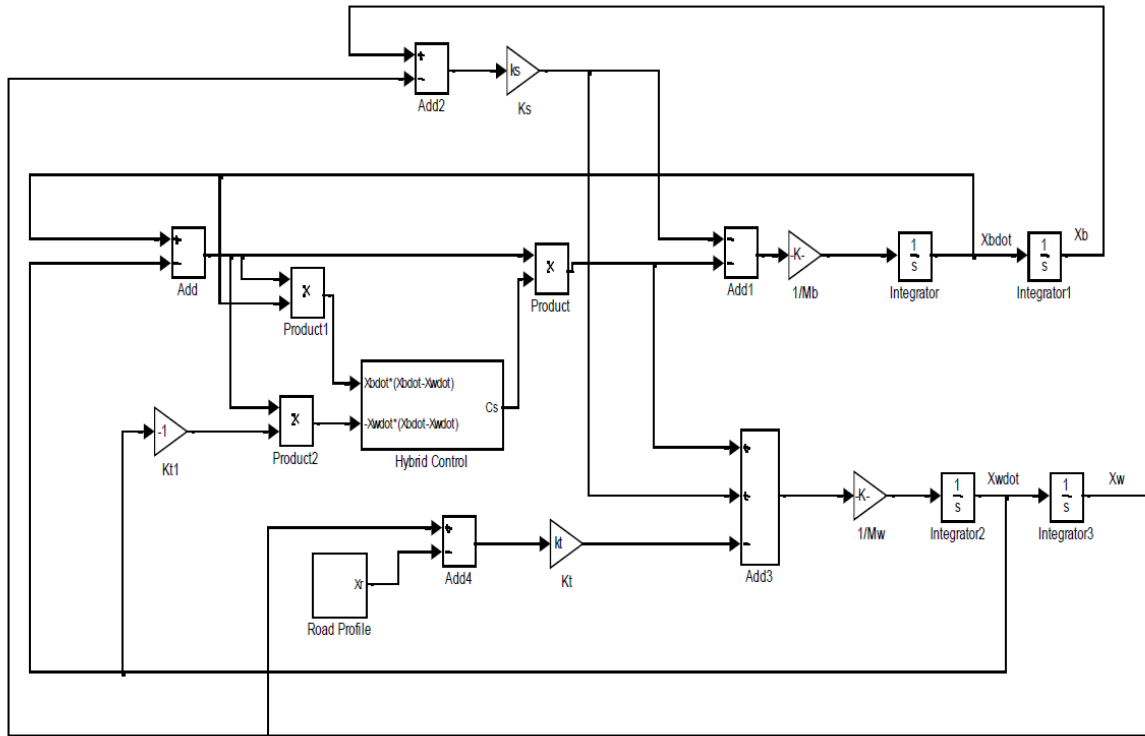
## APPENDIX



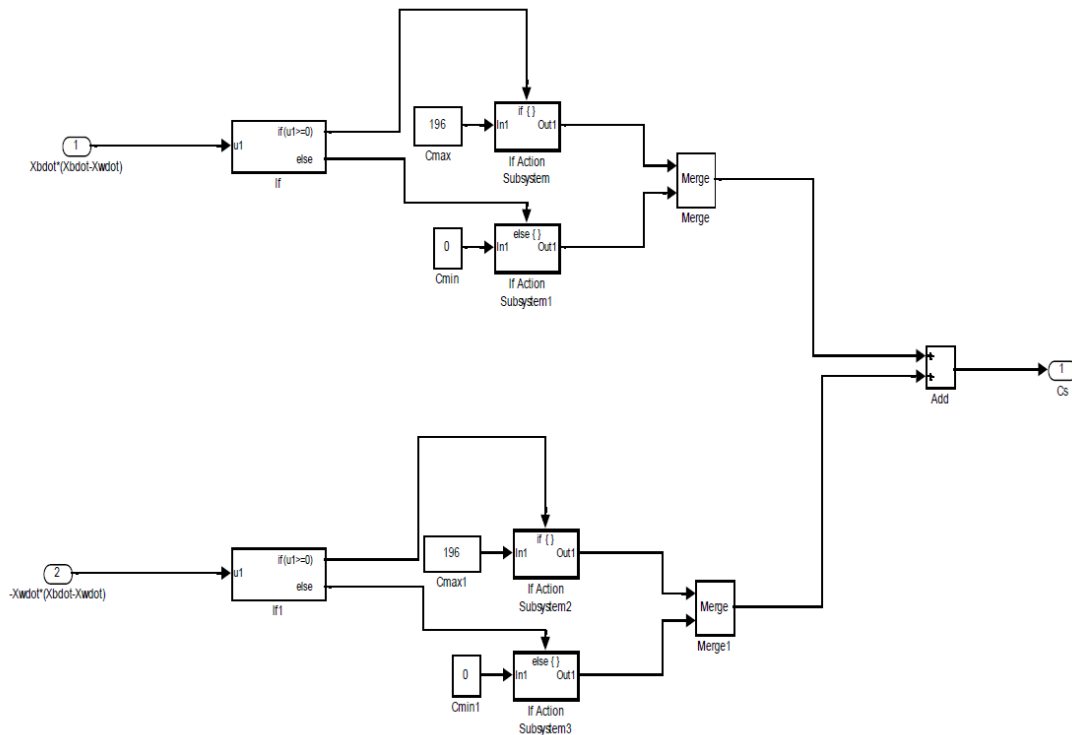Figure 1. Simulink model of Hybrid Semi-Active Suspension system
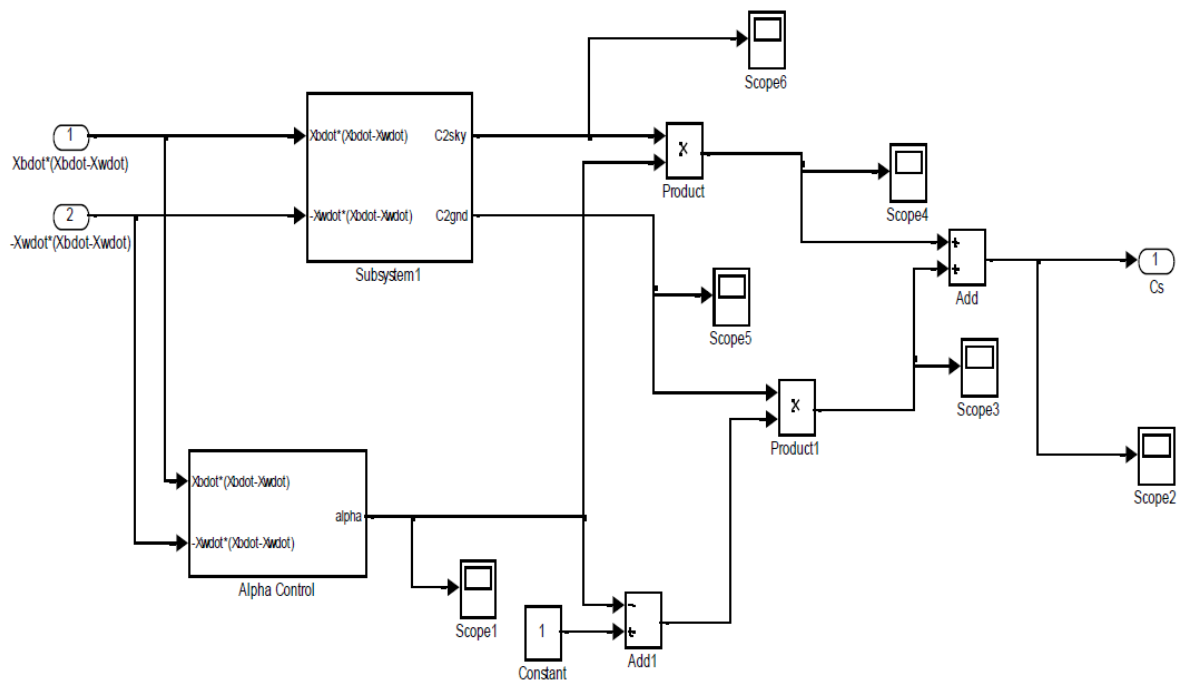


Figure 2. Simulink model of Hybrid Control Block
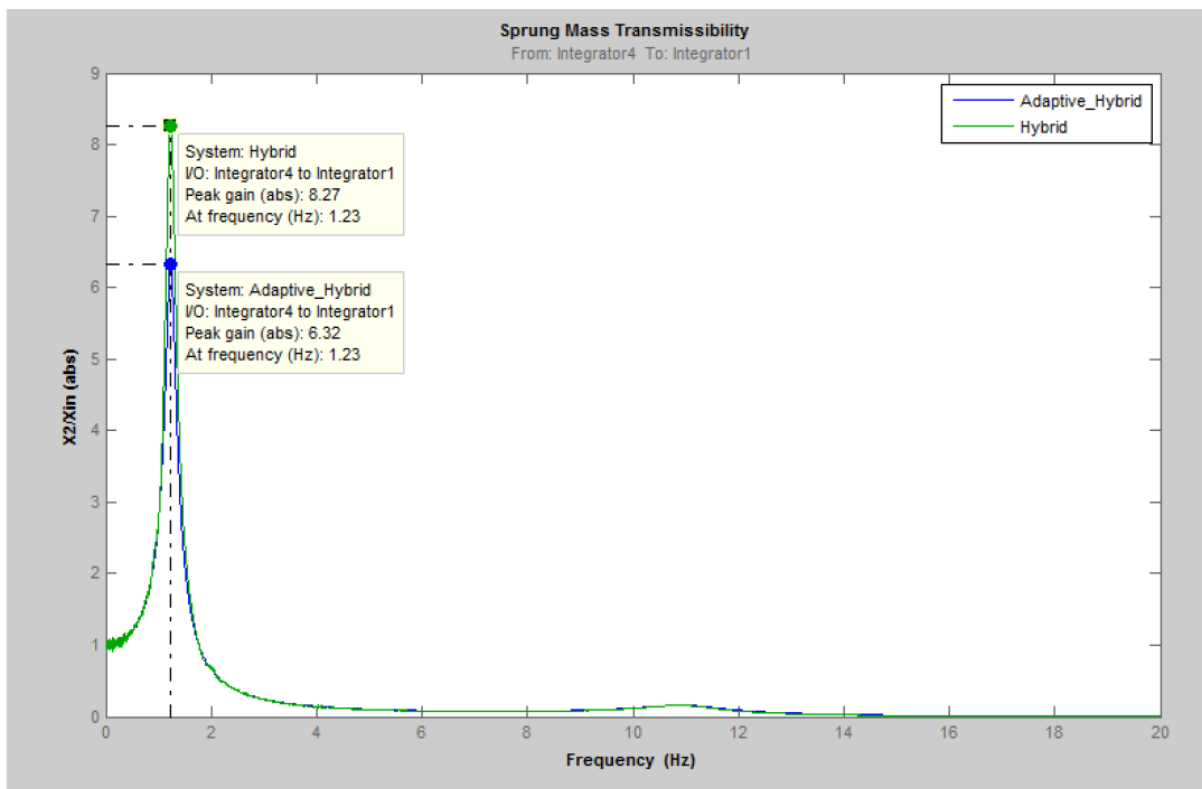
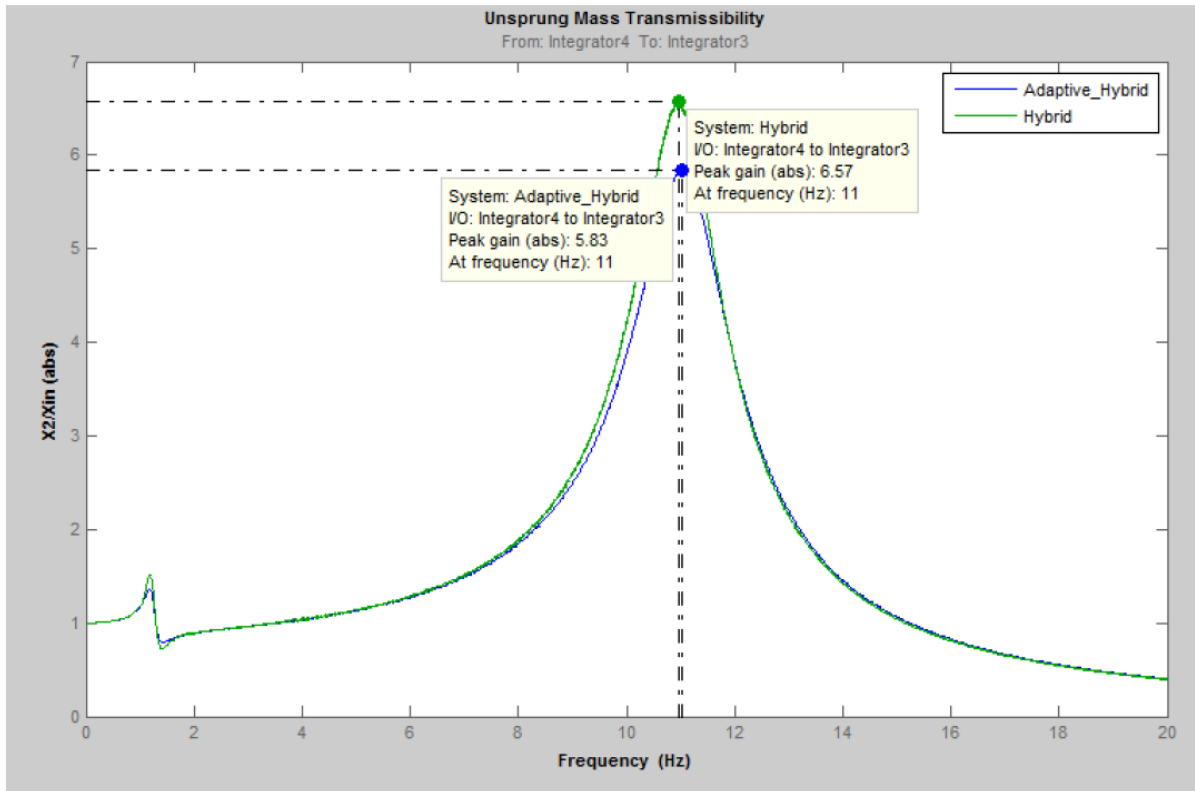Figure 3. Simulink model of Adaptive Hybrid Control Block



Figure 4. Sprung Mass Transmissibility
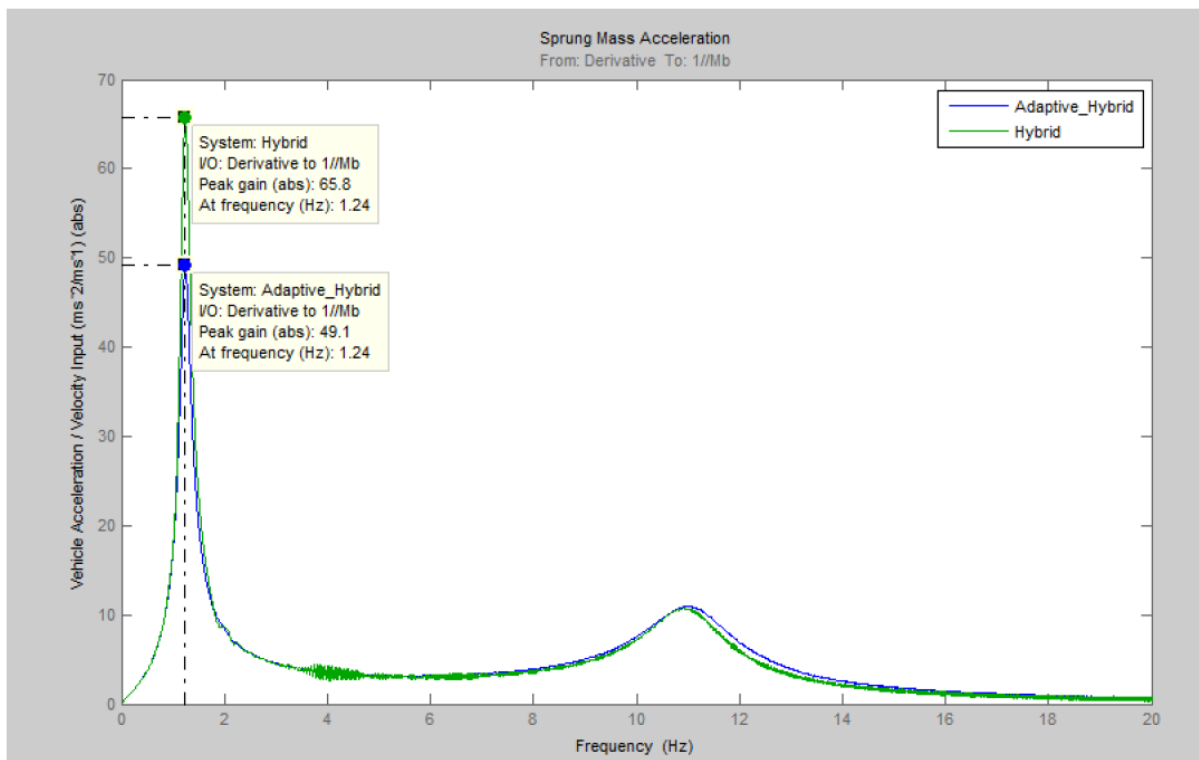
Figure 5. Unsprung Mass Transmissibility
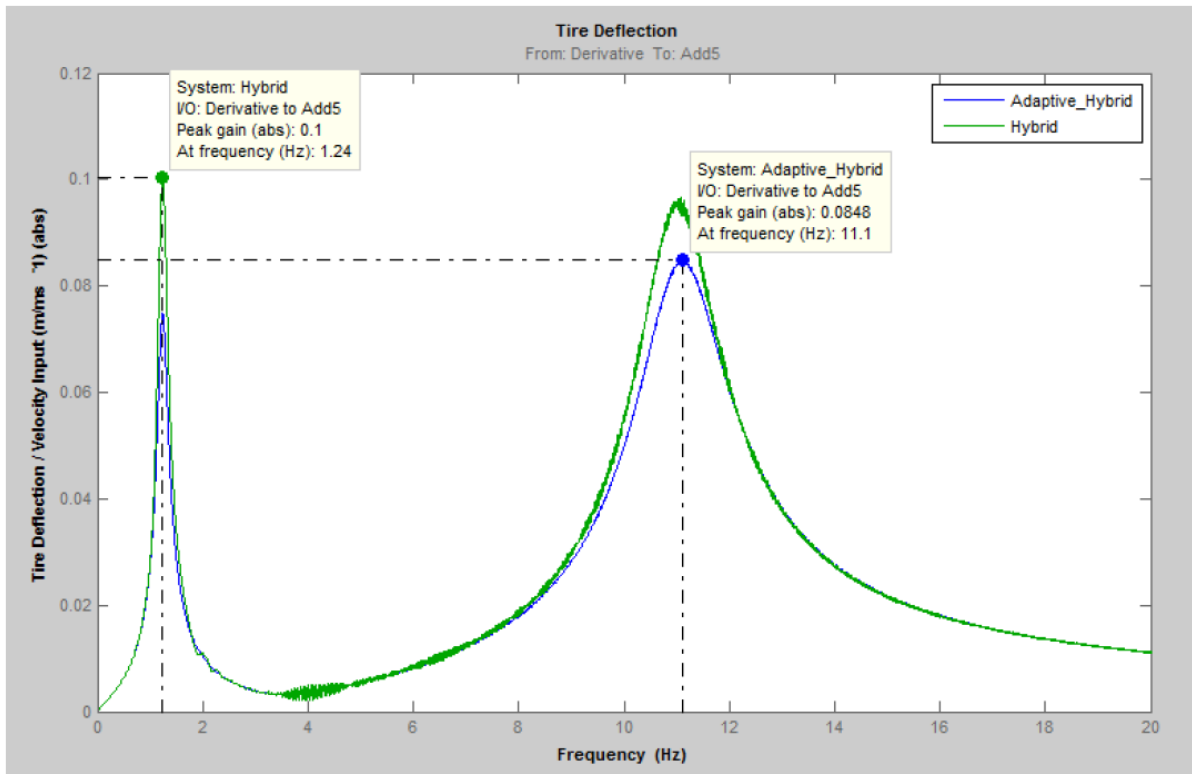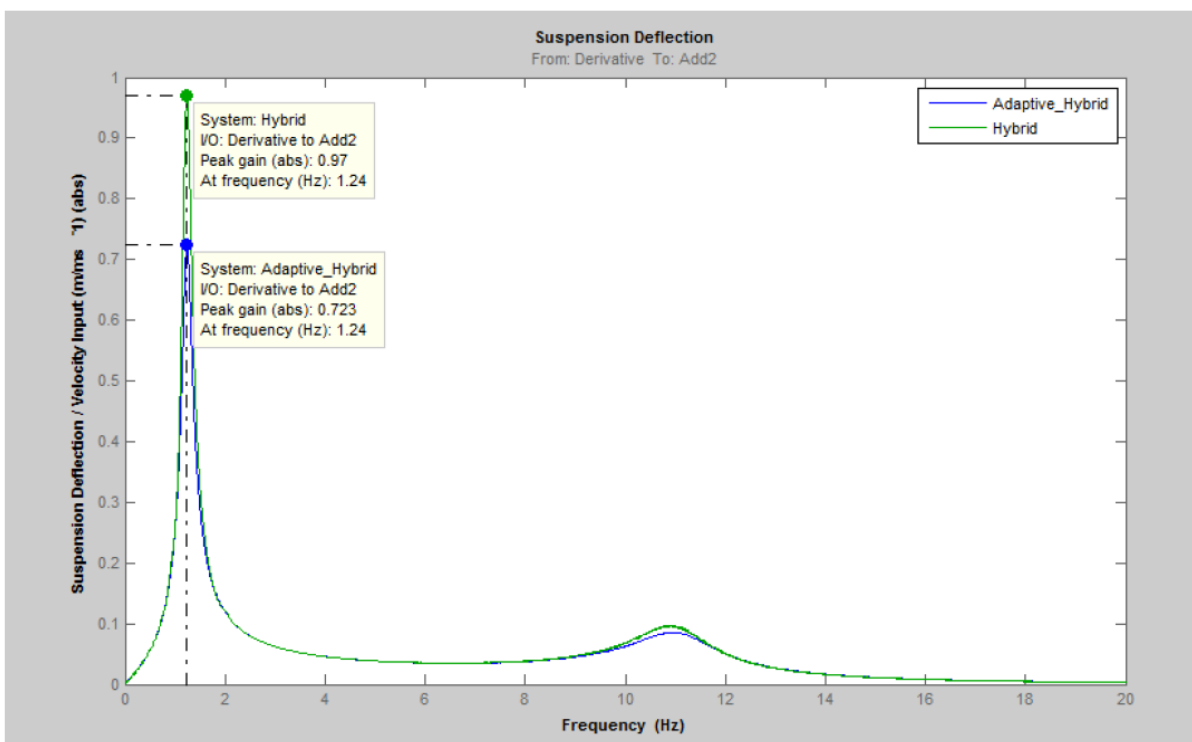


Figure 6. Sprung Mass Acceleration

Figure 7. Tire Deflection



Figure 8. Suspension Deflection