

Analysis of control policies and dynamic response of a Q-car 2-DOF semi active system

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Abstract. Several control policies of Q-car 2-DOF semiactive system, namely skyhook, groundhook and hybrid controls are presented. Their ride comfort, suspension displacement and road-holding performances are analyzed and compared with passive system. The analysis covers both transient and steady state responses in time domain and transmissibility response in frequency domain. The results show that the hybrid control policy yields better comfort than a passive suspension, without reducing the road-holding quality or increasing the suspension displacement for typical passenger cars. The hybrid control policy is also shown to be a better compromise between comfort, road-holding and suspension displacement than the skyhook and groundhook control policies.

Keywords: Ride comfort, quarter-car model, semiactive system, groundhook, skyhook, hybrid, passive suspension system

1. Introduction

Suspension systems are often used to control response of various rigid and flexible multi-body systems [1,2] and the most commonly used suspension systems in vehicular applications, where they are used particularly to control the tire deflection or wheelhop for handling performance and vehicle body deflection and acceleration for passenger ride comfort [3]. Other vehicular applications of such systems include engine, cab and seat suspensions. These so called secondary suspensions increase ride comfort by reducing vibration transmission to the operator compartment [4]. Many works has been conducted to find factors that affect the ride comfort. Demic et al. [5] for example investigated human behavior under random vibrations and shows that human are generally very sensitive to frequency less than 1 Hz.

Since first introduced by Crosby and Karnopp [6,7], semiactive suspension systems continue to gain considerable attention in vehicle applications. This is due to its advantageous characteristics over passive system in overcoming the traditional conflict between vehicle safety and handling, and ride comfort, as well as its significantly less complexity and power requirement than active suspension system [8–10]. Semiactive dampers draw small amounts of energy to operate a valve to adjust the damping level and thus reduce the amount of energy transmitted from the source to the suspended body.

Semiactive dampers can generally be classified based on their control scheme as either on-off skyhook, continuous skyhook, on-off groundhook or continuous groundhook. Skyhook control is known to be able to improve the vehicle body resonance, but at the expense of the tire resonance (wheelhop), while groundhook control provides a better

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Table 1
States and input description

Symbol	Description	Units
x_1	Suspension deflection	Meters
x_2	Sprung mass vertical velocity	Meters/Sec
x_3	Tire deflection	Meters
x_4	Unsprung mass vertical velocity	Meters/Sec
v_{in}	Velocity input	Meters/Sec

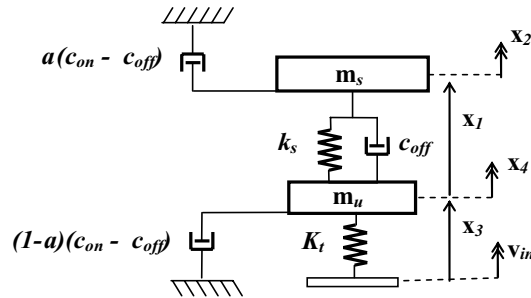


Fig. 1. Quarter-car 2-DOF model.

control of tire resonance, but at the expense of increasing vehicle body resonance. Hybrid control, which combine the effect of both skyhook and groundhook has been introduced and has been shown to provide a compromise between the two while still performing better than the passive system [4,11].

This work intended to look at the performance of each semiactive control policy and compare among themselves and the conventional passive system. The specific performance criteria of interest are the ride comfort, suspension displacement and road-holding responses. A quarter-car model that commonly used in many past suspension studies is considered. This model includes a single suspension system that connects two bodies representing one-quarter of the vehicle body (sprung mass) and one-half of an axle (unsprung mass). The tire stiffness is included as part of the axle.

Three different analyses are presented. The first one is the time-domain transient response analysis, in which the Peak-to-Peak value and settling time are compared. The second is the time-domain steady state response analysis, where the Peak-to-Peak values of pure tone input at resonance frequencies are compared. Finally, frequency responses analyses were conducted and the response curves are compared.

2. Modeling, derivations and analysis

A general quarter-car model is used in this analysis and is shown in Fig. 1. From this single model, all semiactive control scheme as well as passive system can be derived. This way of defining the state variables was first introduced by Chalasani [1] and is advantageous in getting the variables of interest in a more direct way. The state variables and input descriptions are provided in Table 1.

The equations of motion derived from the model are shown below:

$$k_s x_1 + m_s \dot{x}_2 + (c_{off} + \alpha (c_{on} - c_{off})) x_2 - c_{off} x_4 = 0 \quad (1)$$

$$-k_s x_1 - c_{off} x_2 + k_t x_3 + (c_{off} + (1 - \alpha) (c_{on} - c_{off})) x_4 + m_u \dot{x}_4 = 0 \quad (2)$$

where the state relation can further be related as:

$$\dot{x}_1 = x_2 - x_4 \quad (3)$$

$$\dot{x}_3 = x_4 - v_{in} \quad (4)$$

Table 2
Model parameters

Symbol	Description	Units
m_s	Sprung mass	240 Kg
m_u	Unsprung mass	36 Kg
c_s	Suspension damping coefficient	980 N s/m
k_s	Suspension stiffness coefficient	16000 N/m
k_t	Tire stiffness coefficient	160000 N/m

Table 3
Natural frequencies of each system and control policy

System	ω_{n1} (m_s natural frequency), rad/s	ω_{n2} (m_{us} natural frequency), rad/s
Passive	7.8581	69.2702
Skyhook	7.7801	69.9642
Groundhook	7.7813	69.9539
Hybrid	7.7840	69.9296

Notice that from this configuration, skyhook and groundhook controls can be obtained by varying the value of α . The system becomes a skyhook when $\alpha = 1$ and groundhook when $\alpha = 0$. The system is said to be in hybrid control when the value of α is between these values. For the purpose of this analysis, hybrid control is defined as when $\alpha = 0.5$.

The equations of motion for passive system can be easily obtained from the same figure by letting $c_{off1} = c_{s1}$, $c_{on1} = c_{s1}$, $c_{off2} = c_{s2}$, and $c_{on2} = c_{s2}$. With these assumptions, the following equations of motion are obtained:

$$k_s x_1 + m_s \ddot{x}_2 + c_s x_2 - c_s x_4 = 0 \quad (5)$$

$$-k_s x_1 - c_s x_2 + k_t x_3 + c_s x_4 + m_u \ddot{x}_4 = 0 \quad (6)$$

where the state relation remain the same as Eqs (3) and (4).

The motion variables of interest are suspension deflection (x_1), sprung mass acceleration (\ddot{x}_2), and tire deflection (x_3). Unsprung mass acceleration (\ddot{x}_4) is also presented for the sake of comparison, although no significant ride quality can be implied from its result.

The same model parameters are used for every configuration. Their numerical values which are typical for passenger vehicle are summarized in Table 2.

Typical semiactive damping coefficients are chosen using the relationships of $c_{on} = 2.2c_s$ and $c_{off} = 0.2c_s$ [12]. These relationships yield to $(c_{on} - c_{off}) = 2c_s$. Using a built-in MATLAB function, the natural frequencies of each system are obtained and shown in Table 3.

These different analyses were conducted to compare the passive to the semiactive control techniques (skyhook, groundhook and hybrid), that are transient and steady state responses in time domain and frequency response. In the transient response analysis, the system is excited with a step input signal. Peak-to-peak value and settling time is noted for each control technique and then compared. In the steady state response analysis, the system is excited with a sinusoidal input with input frequency is set to be equal to the natural frequencies of the system, ω_{n1} and ω_{n2} . This is to simulate the worst case scenario in which resonances occur. Finally, transmissibility ratios over frequency span for all variables of interest are presented and the control techniques are compared.

3. Results and discussion

3.1. Transient state response

The responses for each control technique and passive system are presented in Fig. 2 for sprung mass acceleration, Fig. 3 for unsprung mass acceleration, Fig. 4 for suspension deflection and Fig. 5 for tire deflection. Their Peak-to-peak value and settling time are summarized in Figs 6 and 7.

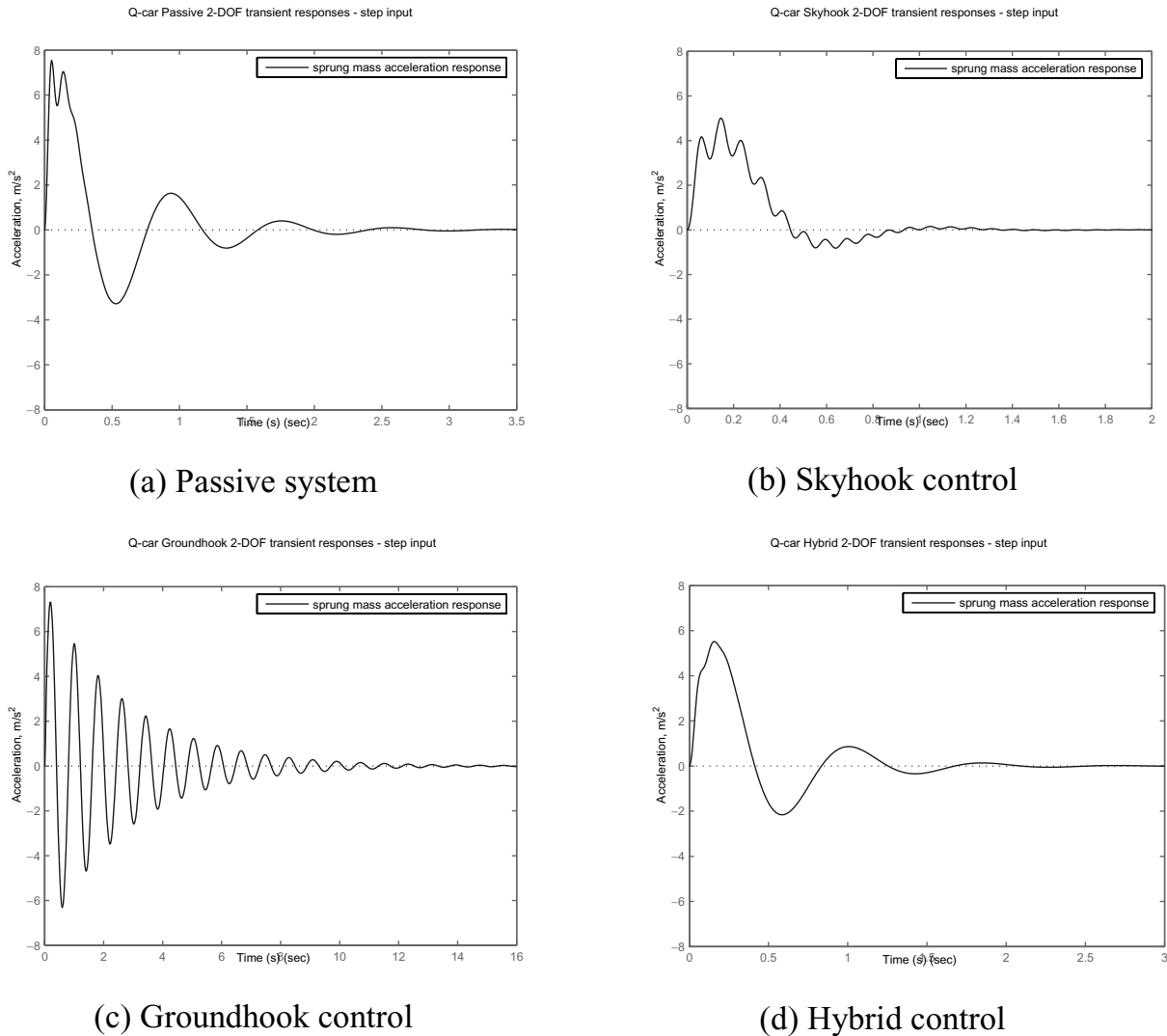


Fig. 2. Sprung mass acceleration response.

Figure 6 shows that relative to the passive system, skyhook control improves all the performance criteria in the sprung mass response at the expense of the unsprung mass response, where it performs significantly worse than the passive system. Exactly the opposite case is true for the groundhook control. Hybrid control, however, improves all performance criteria in both sprung mass and unsprung mass responses.

Figure 7 shows that relative to the passive system, skyhook control improves all the performance criteria in the suspension deflection response as well as the settling time in the tire deflection response. Its Peak-to-Peak value for the tire deflection however is larger than the passive system. Groundhook control performances are worse than the passive system in all cases. Again, hybrid control improves all performance criteria in both suspension and tire deflection responses.

3.2. Steady state response

A summary of the Peak-to-Peak value for each control technique and passive system for both input frequencies, are presented in Fig. 8 for sprung mass and unsprung mass acceleration responses, and Fig. 9 for suspension and tire deflection responses.

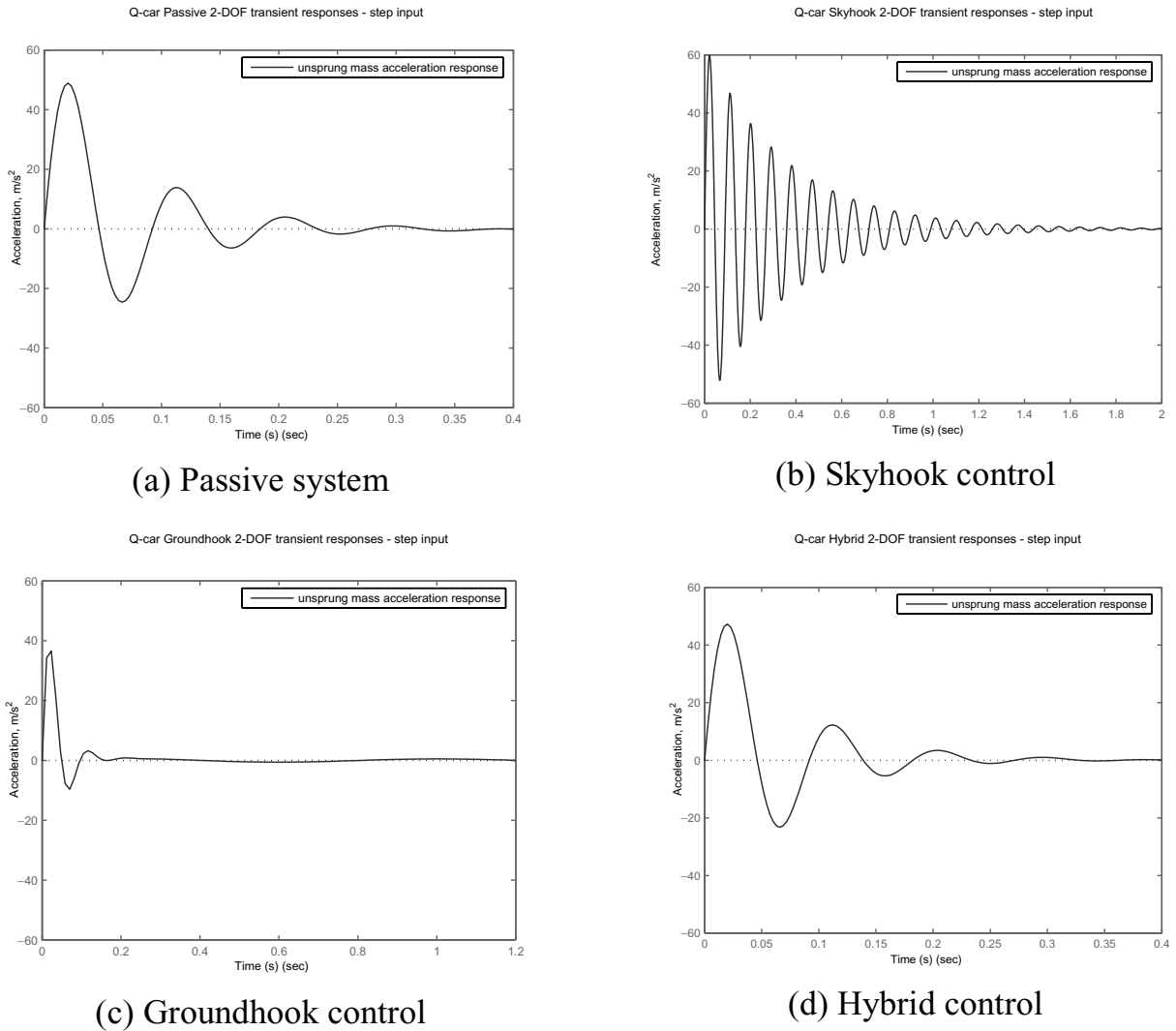


Fig. 3. Unsprung mass acceleration response.

Figure 8 shows that relative to the passive system, skyhook control performs worse in all cases except for the $\omega_{\text{input}} = \omega_{n1}$ case in the sprung mass acceleration response. Groundhook control shows improvement in the $\omega_{\text{input}} = \omega_{n2}$ case in both the sprung mass and unsprung mass acceleration response. It performs worse in the $\omega_{\text{input}} = \omega_{n1}$ case in both responses. Hybrid control, however performs better in all cases.

Figure 9 shows that relative to the passive system, skyhook control performs better in the $\omega_{\text{input}} = \omega_{n1}$ case in both responses but worse in the case of $\omega_{\text{input}} = \omega_{n2}$. On the other hand, groundhook control shows improvement in the $\omega_{\text{input}} = \omega_{n2}$ case in both the sprung mass and unsprung mass acceleration response. It performs worse in the $\omega_{\text{input}} = \omega_{n1}$ case in both responses. Again, hybrid control performs better in all cases.

3.3. Frequency response

The transmissibility plot are shown in Fig. 10 for sprung mass and unsprung mass accelerations and Fig. 11 for suspension and tire deflection.

Generally it can be concluded that skyhook control improves the response at the sprung mass natural frequency while increasing the response at the unsprung mass natural frequency. On the other hand, groundhook control

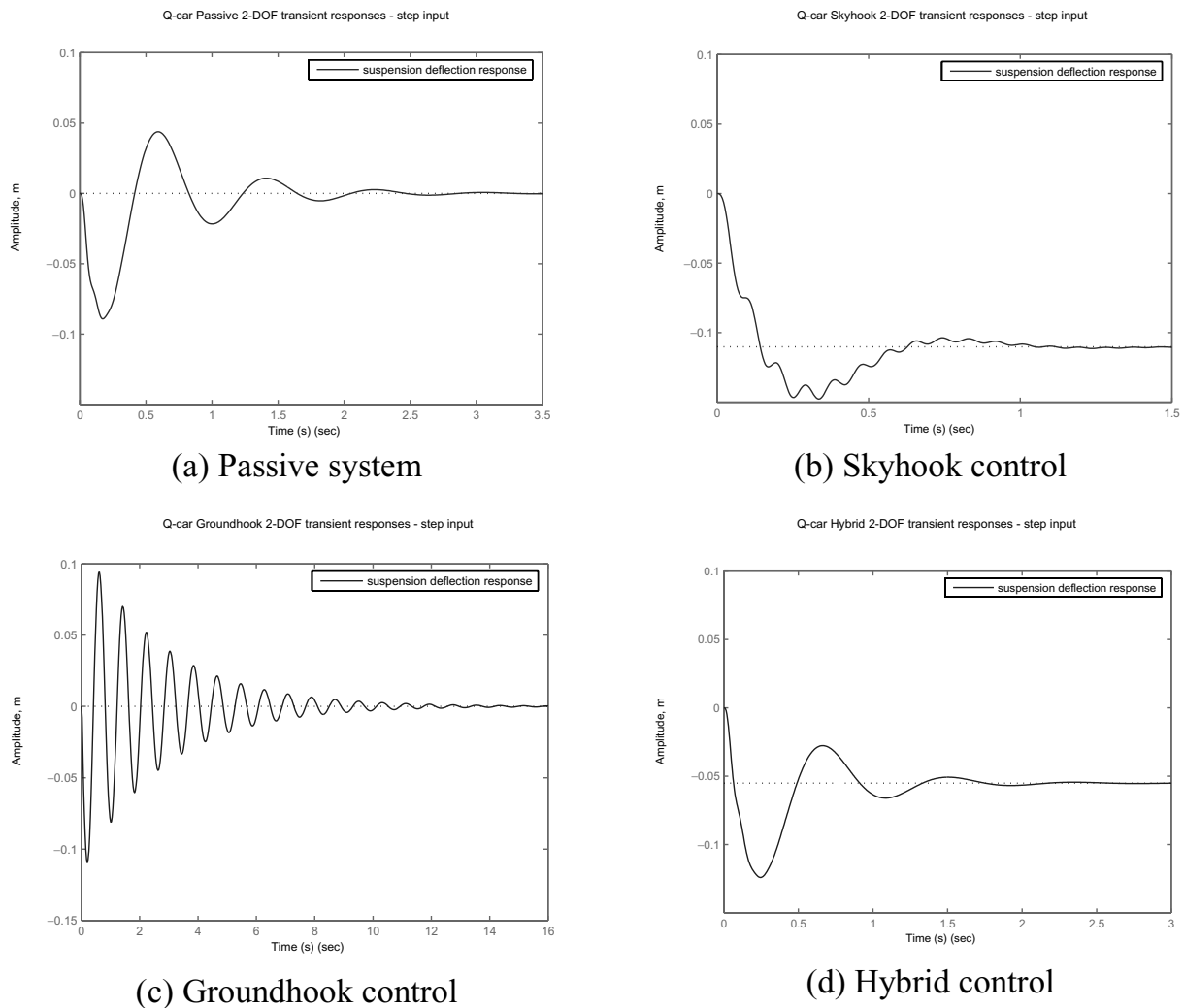


Fig. 4. Suspension deflection response.

improves the response at the unsprung mass natural frequency while increasing the response at the sprung mass natural frequency. Consistent with the earlier analysis, hybrid control provides sort of compromise response of the semiactive control techniques, while still performs better than the passive system. Significant increase for the skyhook and hybrid controls in the suspension deflection response as well as groundhook control in the tire deflection response are noted.

4. Conclusion

Several control policies of Q-car 2-DOF semiactive system, namely skyhook, groundhook and hybrid controls were derived. Their ride comfort, suspension displacement and road-holding performances in time and frequency domains were analyzed and compared with passive system. The results show that the hybrid control policy yields better comfort than a passive suspension, without reducing the road-holding quality or increasing the suspension displacement for typical passenger cars. The hybrid control policy is also shown to be a better compromise between comfort, road-holding and suspension displacement than the skyhook and groundhook control policies. This work is a step in a series of investigative research on different car models and strategies.

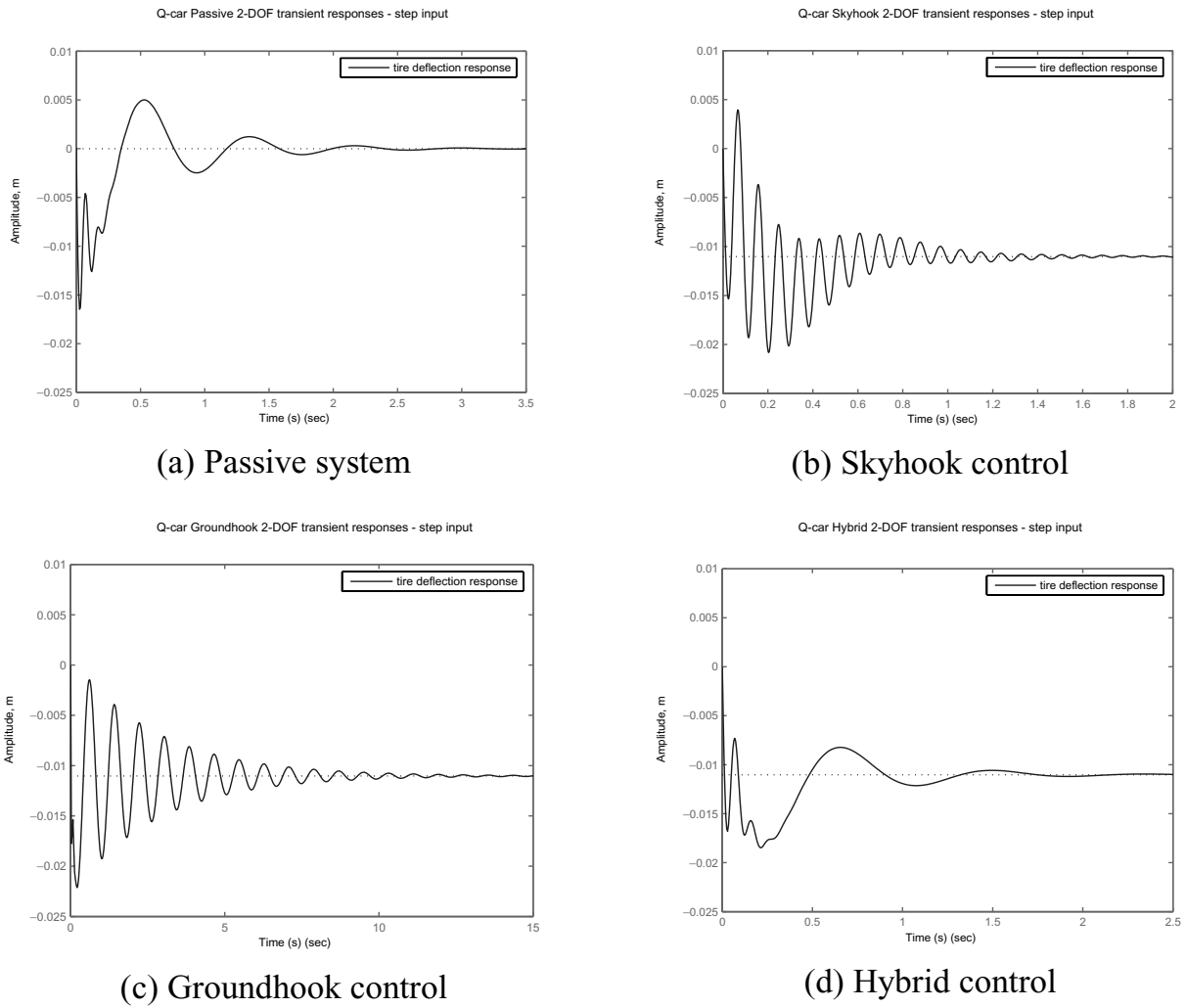


Fig. 5. Tire deflection response.

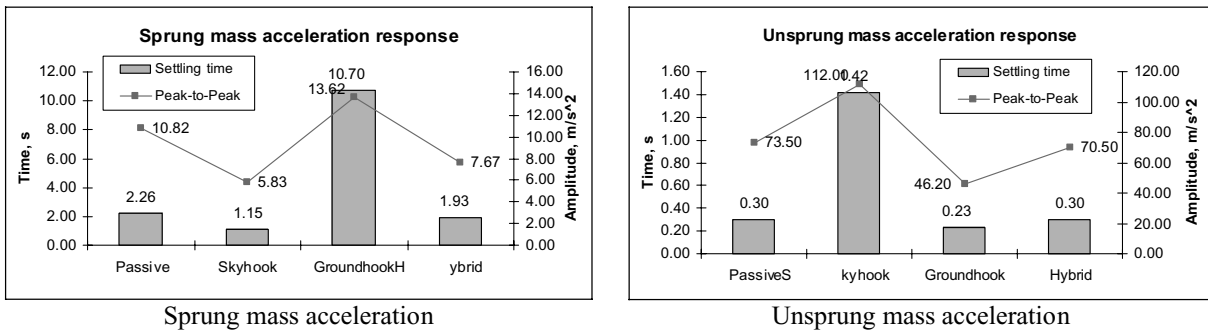


Fig. 6. Summary of results for sprung mass and unsprung mass acceleration response.

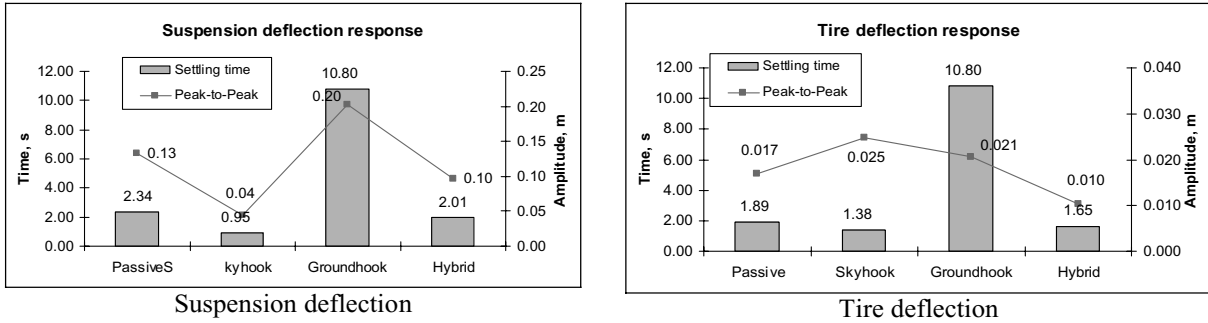


Fig. 7. Summary of results for suspension and tire deflections response.

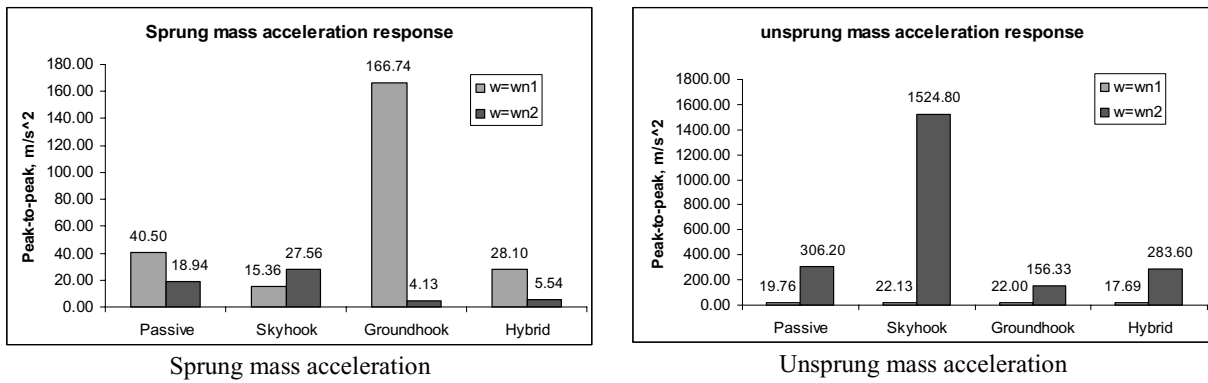


Fig. 8. Peak-to-Peak value for sprung mass and unsprung mass acceleration response.

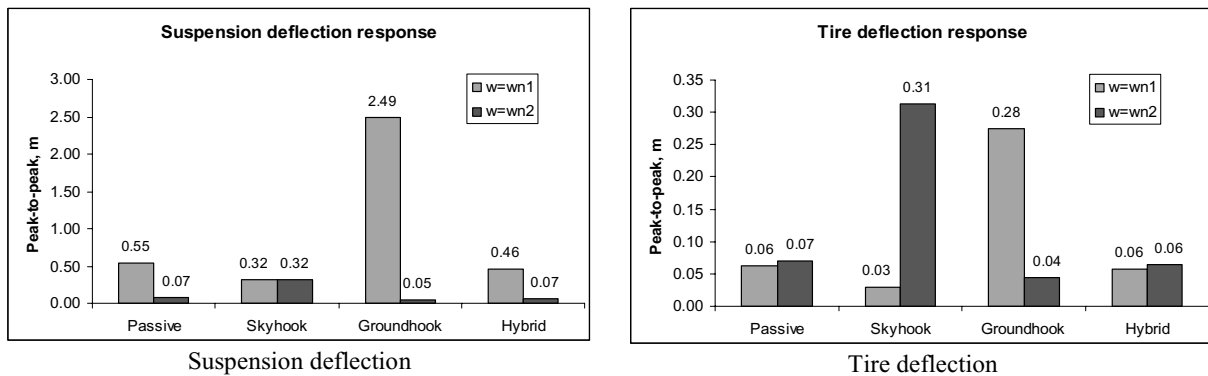


Fig. 9. Peak-to-Peak value for suspension and tire deflection response.

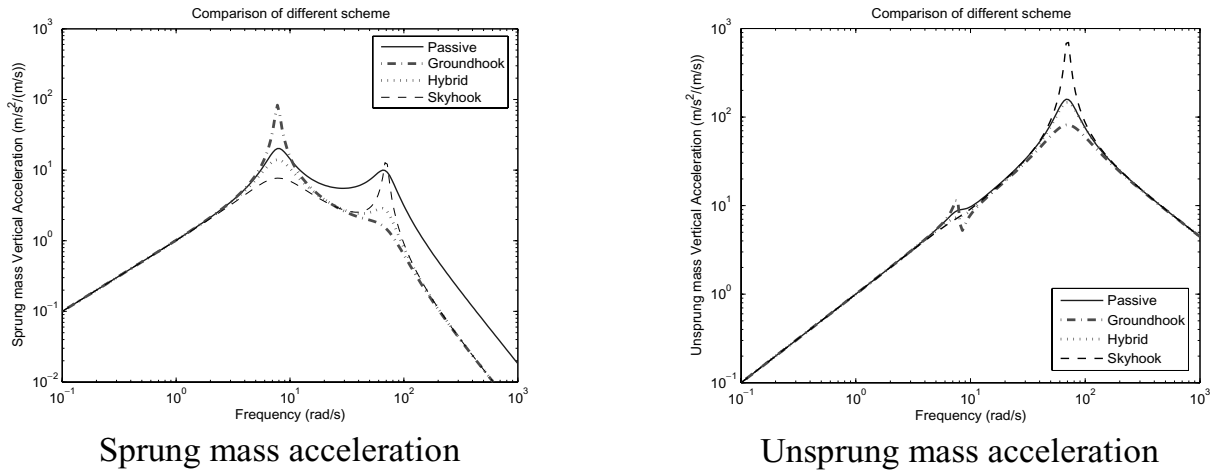


Fig. 10. Sprung mass and unsprung mass acceleration frequency response.

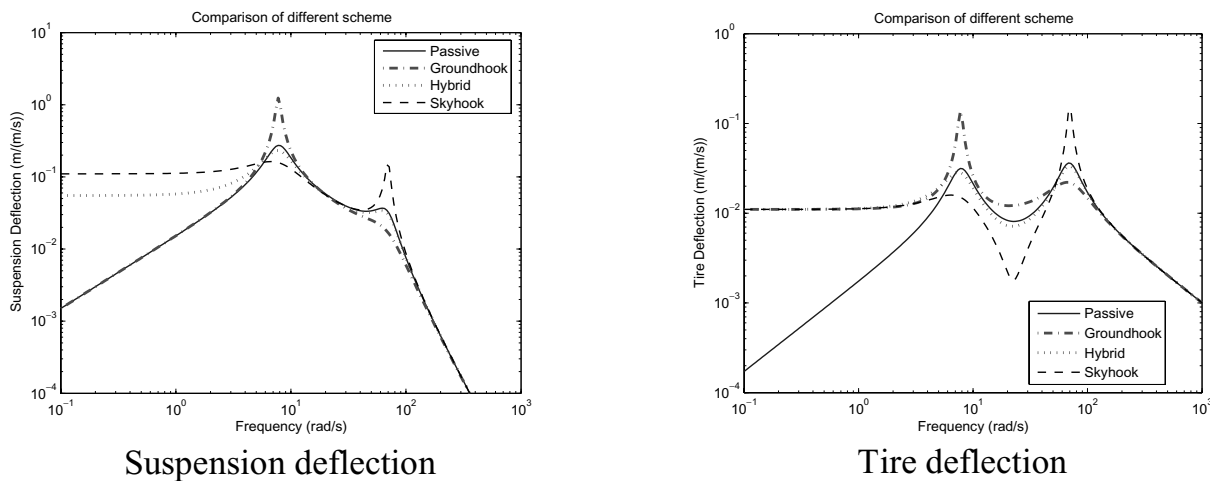


Fig. 11. Suspension and tire deflection frequency response.

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