Performance Comparison Between Skyhook and Semi Active Damping Force Estimator (SADE) Algorithms for Semi Active Suspension System

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ABSTRACT

An investigation on the performance of algorithms in semi active suspension system using a quarter car model has been performed. A Magnetorheological damper model has been developed and used as an actuator in the semi active suspension system. The Skyhook and Semi Active Damping Force Estimator (SADE) algorithms are used to control the operation of the MR damper in quarter car model's suspension system. The damping coefficient value for both of the algorithms are intended to use the same value, for a fair untuned comparison in investigating the algorithm's advantages in improving ride comfort performance of a quarter car model. The simulation model is subjected to a random road profiles with its amplitudes are ranging from -7.5 cm to 7.5 cm. The model's vertical motions (vertical jerk, acceleration and, displacement, suspension working space and tyre dynamic loads) are observed in this studies. The mentioned responses are studied in both time domain and frequency domain results. It was found that the overall performance of untuned SADE-controlled semi active suspension system are at 52.1 percent, compared to the performance of the Skyhook semi active suspension system, which is at 58.1 percent. The use of SADE algorithm however ensures a small trade-off of suspension working space and tyre dynamic load when compared to the passive and Skyhook semi active suspension system.

Keywords: *Skyhook Control, Semi Active Damping Force Estimator (SADE), Ride Comfort, Quarter Car*

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Introduction

Modern passenger vehicles are normally being equipped with a suspension system which act by supporting the weight of the chassis as well as preventing the road disturbance from reaching the passenger in the passenger cabin. A typical or conversional suspension system applied by the vehicle manufacturers is typically consists two main components: damper and spring. However, due to the technological advancement in the development of the vehicle's suspension system, advanced of suspension systems has been introduced. The semi and active suspension system types started to be applied in the modern passenger vehicles.

The operation of conventional suspension system has its own limitation. It works best during the isolation of low-frequency disturbances, when the damping is high. However, it comes with a trade-off where high damping would cause poor high frequency shock absorption. Conversely, at low damping, the damper performs its best in high-frequency absorption, sacrificing low-frequency isolation.

In order to ensure that a semi active suspension system could perform at its best during low or high frequency shock absorption, an advanced suspension system had been introduced. The advanced suspension system is divided into two types: the semi active and active suspension systems.

The semi active suspension system mainly consists of a spring and variable damping damper. The variable damping damper used in the semi active suspension system can either be a variable orifice damper or a Magnetorheological (MR) damper. For a variable orifice damper, the variable damper characteristics are given through the controlled openings of the solenoid valve in the damper. For the MR damper, the variable damping characteristics are obtained by exposing the magnetorheological fluid to the magnetic field. The viscosity of the MR fluid are high when exposed to a strong magnetic field and vice versa. The changes of the MR fluid viscosity at the orifice opening in the MR damper disrupt the MR fluid flow thus giving variable damping characteristics.

As for the active suspension system, it can further be divided into two types which is the fully active and slow-active suspension systems [1]. In a fully active suspension system, the shock absorption task is mainly done by an actuator, with no presence of a spring. The active suspension system has the ability to operate over a wide range of isolation frequencies as well as having the ability to introduce external forces in the system [2]. However, the operation of the active suspension system requires a severe power consumption as well as high operating costs when compared to the conventional and semi active suspension systems.

The application of the slow active suspension system is more suitable to be applied in a passenger vehicle compared to the active suspension system.

The main components in the slow-active suspension system are low bandwidth actuator, spring and damper which can be arranged in a series or parallel configuration. The purpose of the low bandwidth actuator is just to maintain the chassis levelling during vehicle’s maneuvers, as the shock observations are still being done in the spring and damper. The power consumption of this system is moderate and costs lower than the active suspension system [2].

Based on the literature review done, designing a controller for an advance suspension system is always a challenging task. The controller need to be design in order to ensure that the actuator in the advance suspension systems will always operates at its optimum level, giving the best ride comfort performance that can be experienced by the passengers. Typical controller or algorithms that were used in advance suspension system are the skyhook control theory [3,4], linear optimal control [5,6] and state feddback controller [7,8].

This paper will be emphasizing on the performance of a quarter car semi active suspension when controlled by a notable skyhook algorithm and a potentially commercialized Semi Active Damping Force Estimator (SADE) [9]. The performance of the studied suspension systems, are being assessed quantitatively, in terms of three type of responses; discomforted parameters (sprung mass’s jerk, acceleration and displacement), suspension working space (SWS) and dynamic tire load (DTL).

Simulation Model

The performance of the studied semi active suspension systems were done using a simulation model which consists of 4 main elements which are the quarter car model, control algorithm, current generator model, and MR damper model. Figure 1 shows in general, the signal layout used in modelling the semi active suspension system model.

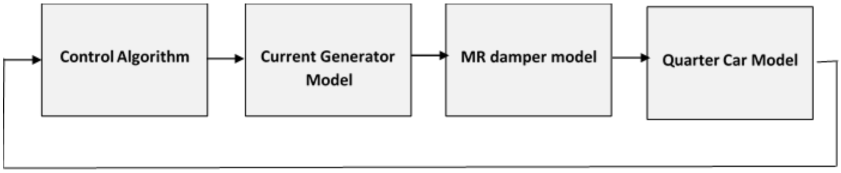

Figure 1: Signal layout for the semi active suspension system model

Quarter Car Model

A quarter car modelling is chosen to be used in this study in order to investigate the ride comfort performance of the studied semi active suspension systems. The quarter car model consists of two degree of freedom which involves vertical motions of the sprung and unsprung masses. Figure 2 shows the vehicle's quarter car model.

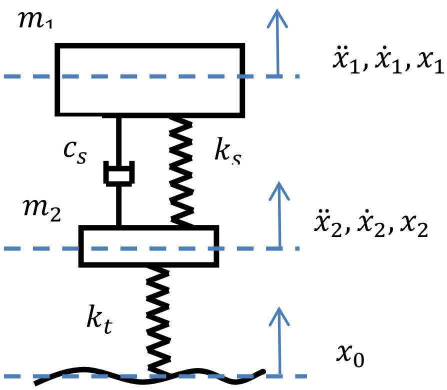

Figure 2: Two degree of freedom of vehicle model

There are some assumptions made in this study. First assumption is the model aerodynamic effect is neglected and the road is assumed to be level except for road disturbance. Second assumption is the parameters of the model are assumed to be constant throughout the simulation process such as tire stiffness, spring stiffness, and damper coefficient. Based on the single degree of freedom model in Figure 1, the acceleration of sprung mass is defined by:

$$m_1\ddot{x}_1 = F_D + F_s \quad (1)$$

where m_1 is the sprung mass, $\ddot{x}_1$ is the sprung mass acceleration, F_D is the suspension damper force and F_s is the suspension spring force. The suspension damper force, F_D is given by:

$$F_D = C_s(\dot{x}_2 - \dot{x}_1) \quad (2)$$

where C_s is the suspension's damping coefficient, $\dot{x}_2$ is the unsprung mass velocity and $\dot{x}_1$ is the sprung mass velocity. The suspension spring force, F_s is given by:

$$F_s = k_s(x_2 - x_1) \quad (3)$$

where k_s is the suspension spring coefficient, x_2 is the unsprung mass displacement and x_1 is the sprung mass displacement. The acceleration of unsprung mass is defined by:

$$m_2\ddot{x}_2 = F_t - F_D - F_s \tag{4}$$

where m_2 is the unsprung mass, $\ddot{x}_2$ is the unsprung mass acceleration and F_t is the tire force. The tire force, F_t is given by:

$$F_t = k_t(x_2 - x_0) \tag{5}$$

where k_t is the tyre spring stiffness and x_0 is random road input ranging from -7.5 cm to 7.5 cm. Table 1 shows the parameters value used for the quarter car model [10].

Table 1: Quarter car model's parameters

Definitions	**Value**
Sprung mass, m_1	250 kg
Unsprung mass, m_2	50 kg
Spring stiffness, k_s	17900 N/m
Tire stiffness, k_t	190 000 N/m
Damping coefficient, c_s	1500 N.s/m

Semi Active Suspension System Algorithms

The Skyhook control system was introduced by Karnopp [3], emphasizing in isolating the sprung mass from the excessive base excitations by assuming that there is an imaginary damper between the sprung and the stationary sky. The configurations works as if there is a high damping force between the sprung mass and the stationary mass. This prevents the sprung mass from oscillating excessively. The unsprung mass is free to oscillate excessively. The equation governing skyhook control is given by:

$$\begin{aligned} &If\ \dot{x}_1(\dot{x}_1 - \dot{x}_2) \geq 0\ then\ F_d = C_{sky}\dot{x}_1 \\ &If\ \dot{x}_1(\dot{x}_1 - \dot{x}_2) < 0\ then\ F_d = 0 \end{aligned} \tag{6}$$

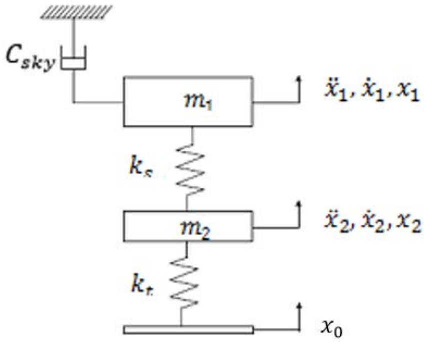

Figure 3: Skyhook control system

with $\dot{x}$ is sprung mass velocity, $\dot{x}_1$-$\dot{x}_2$is relative velocity between sprung and unsprung mass, F_d is the estimated damping force and C_{sky} is the skyhook damping coefficient.

As for the second algorithm, the Semi Active Damping Force Estimator (SADE), the algorithm estimates damping forces based on the assumption the semi active actuator could produce any amount of required damping force [1]. Figure 4 shows the idea of the SADE algorithm. The SADE algorithm is given by:

$$F_{SADE} = [C_s(\dot{x}_2 - \dot{x}_1) + m_1\ddot{x}_1]A \tag{7}$$

where C_s is a damping coefficent, $\dot{x}_2$ is velocity for the unsprung mass, $\dot{x}_1$ is velocity for the sprung mass, m_1 is weight of the sprung mass and A is a tunable gain.

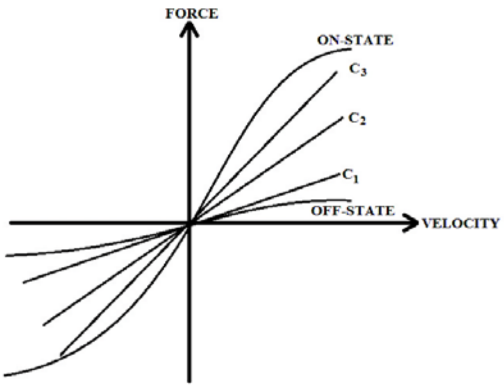

Figure 4: The concept of SADE

Performance Evaluations

A constant value of 1500 Ns/m for the damping coefficient, C_s has been used in this study. The value was used in all equations that estimates damping forces (Eq. 2, Eq. 6 and Eq. 7). The purpose of this approach is to study the performance of the semi active suspension system over the passive suspension system without going through proper tuning procedures. Three responses were evaluated and they are the sprung mass' vertical motions (jerk, acceleration, and displacement), suspension working space (relative velocity between sprung and unsprung masses) and tyre dynamic load. The best performance of ride comfort is achieved when at least two out of these three responses are having the lowest root mean square value (RMS). The RMS value is given by:

$$RMS = \sqrt{\frac{1}{T}\int \|u(t)\|^2 dt} \qquad (8)$$

The performance of the semi active suspension systems are evaluated based on the comparison with the passive suspension system in terms of overall percentage improvement performance [10].

$$P_{\max} = \frac{1}{5}\sum\left[\left(\frac{RMS_{passive} - RMS_{semi\ active}}{RMS_{passive}}\right)_k \times 100\right] \qquad (9)$$

with k = 1, 2, … 5 and 1 = jerk, 2 = acceleration, 3 = displacement, 4 = suspension working space, 5 = tyre dynamic load.

Magnetorheological (Mr) Model

A non-parametric Delphi's Magnetorheological (MR) damper model was used in this study. Figure 3 shows the characteristic of the damper. The MR damper model was modelled based on the experimental data of the damper using non-parametric modelling approach [1]. Based on Figure 5, the data mapping involves the damping force data ranging from -1 m/s to 1 m/s characterized at 0 to 5 Ampere.

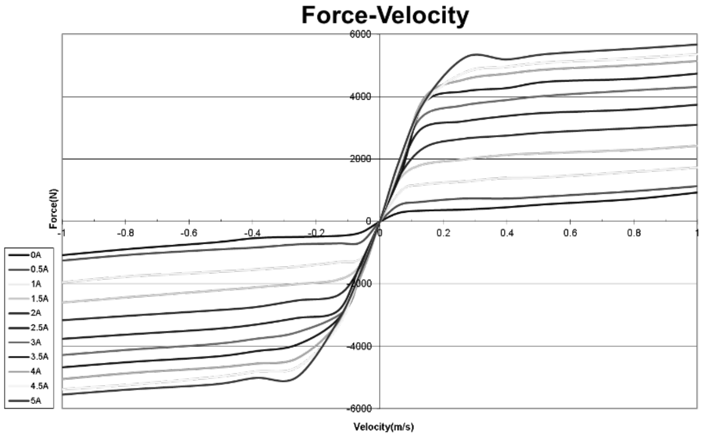

Figure 5: Characteristics of Delphi's MR damper

If the input values (velocity data) are other than the provided hardpoint velocity data, a linear interpolation-extrapolation data will be done [1].

$$If\ V > V_i\ \&\ V < V_{i+1}\ then\ F = \left(\frac{V-V_i}{V_{i+1}-V_i}\right)(F_{i+1} - F_i) \qquad (10)$$

where V_i is velocity hard point data; V is for the velocity value; F_i is damping force hard-point data; F is desired damping force value; i is the data number (e.g. 0,1,2, 3, 4, 5, 6,…).

Linear interpolation-extrapolation between the force-velocity curves is made when the input current to the MR damper is between two current points.

The interpolation-extrapolation between the force-velocity curves is made when the input current is between two current hardpoint data [1].

$$If\ i > i_n\ \&\ i < i_{i+1}\ then\ F = \left(\frac{i-i_i}{i_{i+1}-i_i}\right)(F_{n+1} - F_n) + F_n \qquad (11)$$

where *i* is the data value; n is the number of current increment (1, 2, 3, ….10) ; i_0 is the current value of 0 Ampere; i_1 is the current value of 0.5 Ampere; i_2 is the current value of 1 Ampere; i_3 is the current value of 1.5 Ampere; i_4 is the current value of 2 Ampere; i_5 is the current value of 2.5 Ampere; i_6 is the current value of 3 Ampere; i_7 is the current value of 3.5 Ampere; i_8 is the current value of 4 Ampere; i_9 is the current value of 4.5 Ampere; i_{10} is the current value of 5 Ampere; F is the desired damping force value; F_n is the damping force value at current hard-point data.

Current Generator Model

The current generator model is basically an inverse characteristic model of the applied MR damper model in this paper. Its function is to estimate the amount of current applied to the MR damper model, enabling the MR damper model to execute shock absorption as calculated by the controller [1]. The current signals are produced based on the relative velocity information of the quarter car model and the estimated damping force from the control algorithm. The equation for the current generator model is given by [1]:

$$If\ F_d \geq F_I\ \&\ F_d \leq F_{I+1}\ \ then\ \ I = \left(\frac{F_d - F_I}{F_{(I+1)} - F_I}\right)\left(I_{(i+1)} - I_i\right) + I_i \quad (12)$$

where F_d is the desired force from the algorithm; F_I is the hardpoint data of the damping force; *I* is the desired current; I_i is for I_1 , I_2 , I_3 … I_9 where I_0 , I_1 , I_2…. I_{11} is for 0 Ampere, 0.5 Ampere, 1 Ampere, 1.5 Ampere, 2 Ampere, 2 Ampere, 2.5 Ampere, 3 Ampere, 3.5 Ampere, 4 Ampere, 4.5 Ampere and 5 Ampere respectively.

Results and Discussion

The comparison between the passive and semi active suspension systems were done by subjecting the quarter car model with random road profile inputs which its amplitudes are ranging from ±7.5cm.The responses observed during the attenuations are the sprung mass's vertical jerk, acceleration and displacement. Other observed responses are the suspension working space (relative velocity between the unsprung mass and the sprung mass) as well as the tire dynamic load (the forces between the unsprung mass and the contact surface). Table 2 shows the root-mean-square (RMS) value results for these responses for both passive and semi active suspension systems.

Based on Table 2, the overall average improvements made by the the Skyhook and SADE systems are more than 50 percent when compared to the passive suspension system. However, further observation indicates that only the sprung mass's vertical jerk and acceleration were improves significantly, by both of the algorithms. This is expected as the semi active suspension system are mainly to improve the sprung mass motions with an allowable trade-off; excessive motion of unsprung mass and higher dynamics tyre load hitting the contact surface.

Referring to the same results (Table 2), it can be seen that the overall average improvement made by the Skyhook suspension system is slightly better than the SADE suspension system. This was mostly contributed by a higher RMS value of vertical jerk, which is at 49.7 percent when compared to the passive system performance. As for SADE suspension system's performance, the improvement made for the same response is at 46.9 percent.

It is also observed based on Table 2, the vertical displacement for both of the skyhook-controlled and SADE-controlled systems response are higher than the passive suspension system response. This is acceptable as the motions of the sprung mass is actually occurring at lower magnitudes of vertical jerk and acceleration. As for the suspension working space and the tire dynamic load, the responses provided by the SADE-controlled system can be seen to decline for only 3.7 percent. This shows that the SADE-controlled system only allows a small trade-offs of the suspension working space and tyre dynamic load responses. Figures 6 to 10 shows the results of the studied responses in time domain graphs.

Table 2: RMS value of the studied responses

Responses\Suspension	**Passive**	**Skyhook**	**(%)**	**SADE**	**(%)**
Jerk (m/s^3)	1232	620.1	49.7	654.5	46.9
Acceleration (m/s^2)	10.27	5.74	41.7	5.99	41.7
Displacement (m)	0.01334	0.0158	-18.4	0.0602	-20.1
Suspension space(m/s)	1.604	1.819	-13.4	1.819	-3.7
Tyre dynamic loads (N)	10590	1100	-3.9	10980	-3.7
Average Performance			58.1		52.1

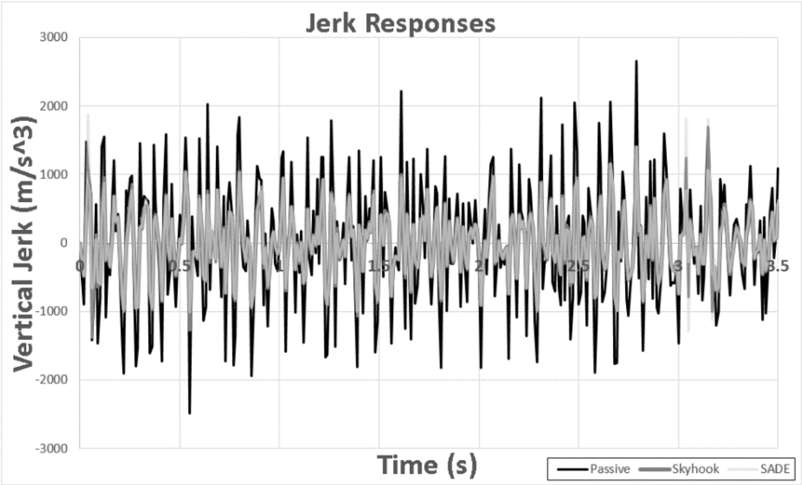

Figure 6: Jerk responses

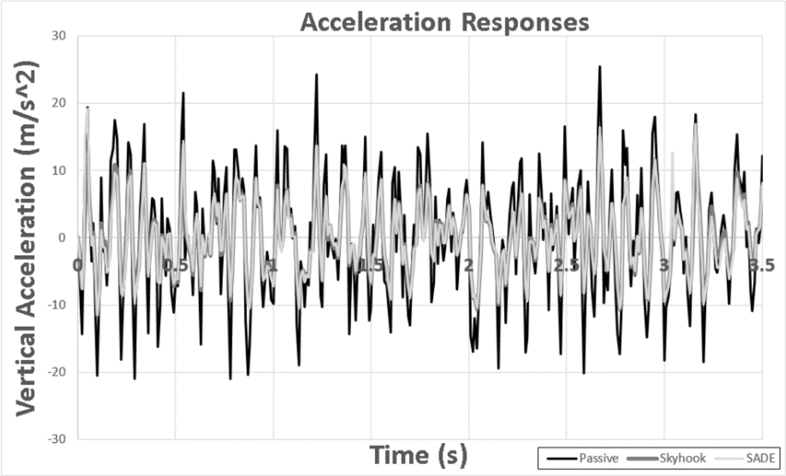

Figure 7: Acceleration responses

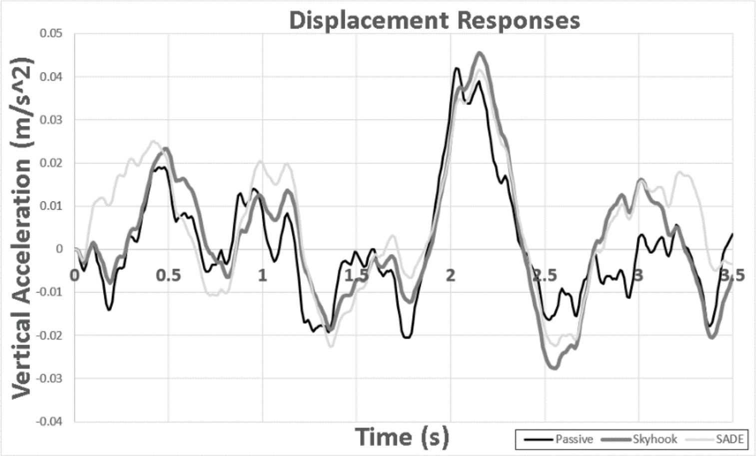

Figure 8: Displacement responses

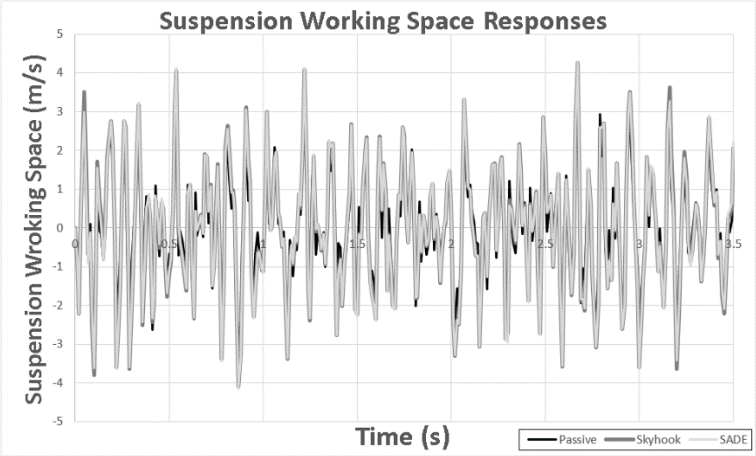

Figure 9: Suspension working space responses

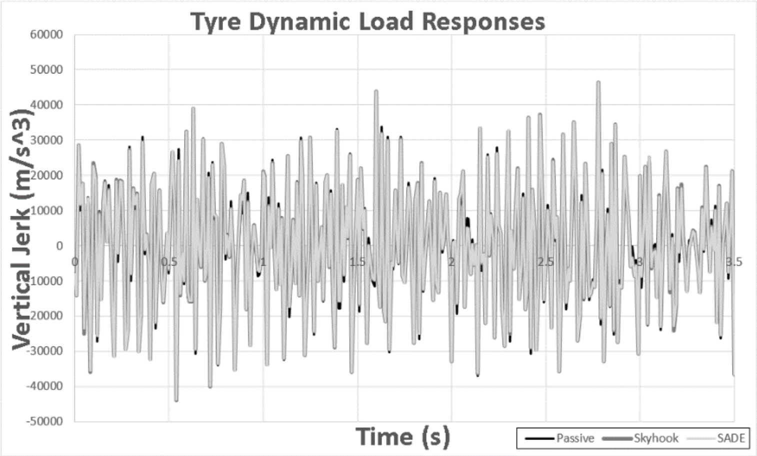

Figure 10: Tyre dynamic load responses

Table 3 shows the dominant frequency of the studied responses, obtained from the time domain data, converted into frequency domain results, using Fourier Fast Transform (FFT) method. The purpose of investigating the studied responses in frequency domain results (Figures 11 to 14) is to investigate and determine the occurrences/frequencies of the signals of the studied responses.

Only the most dominant frequency belongs to the signals, which has the highest magnitude, will be highlighted in this paper. It can be seen in Table 3, in general that most of the signals of studied responses oscillates at the same dominant frequency, even though the types of suspension system changes i.e. dominant jerk frequency for passive and semi active suspension systems are the same which is at 14.53 Hz. Even though that it seems that there's no improvement which were made in terms of the most dominant oscillation frequency for the studied responses, the results of frequency domain need to be interpreted in parallel with the time domain results e.g. for the jerk response, in semi active suspension systems, the sprung mass is having a jerking motion at a lower magnitudes with its dominant occurrences are remained the same as the passive suspension system.

Table 3: Responses dominant frequencies

Responses\Suspension Types	**Passive (Hz)**	**Skyhook (Hz)**	**SADE (Hz)**
Jerk	14.53	14.53	14.53
Acceleration	6.838	6.838	6.838
Displacement	0.5698	1.14	1.14
Suspension working space	6.838	6.838	6.838
Tyre dynamic loads	45.58	45.58	45.58

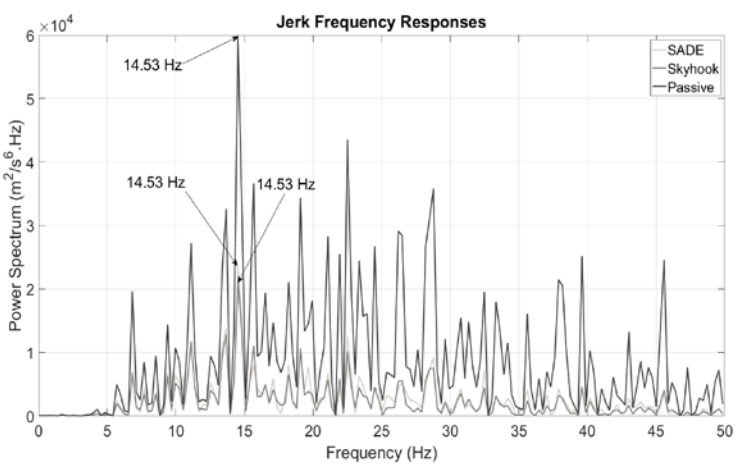

Figure 11: Jerk's frequency responses

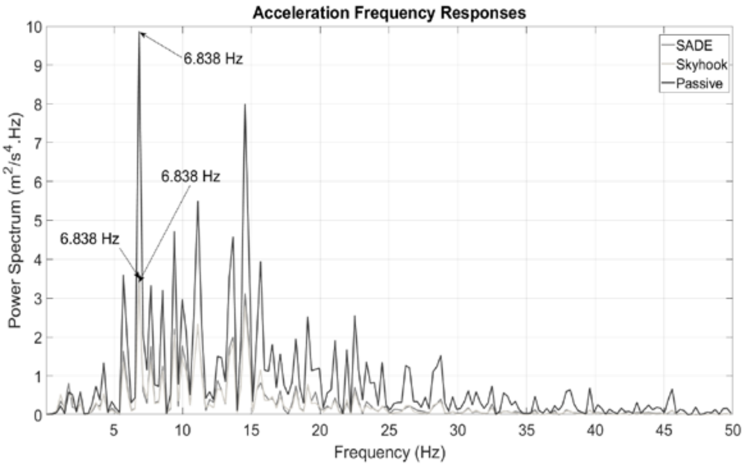

Figure 12: Acceleration's frequency responses

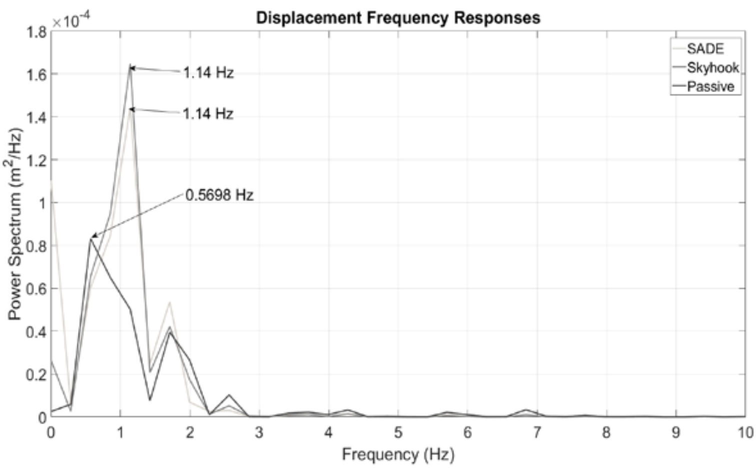

Figure 10: Displacement's frequency responses

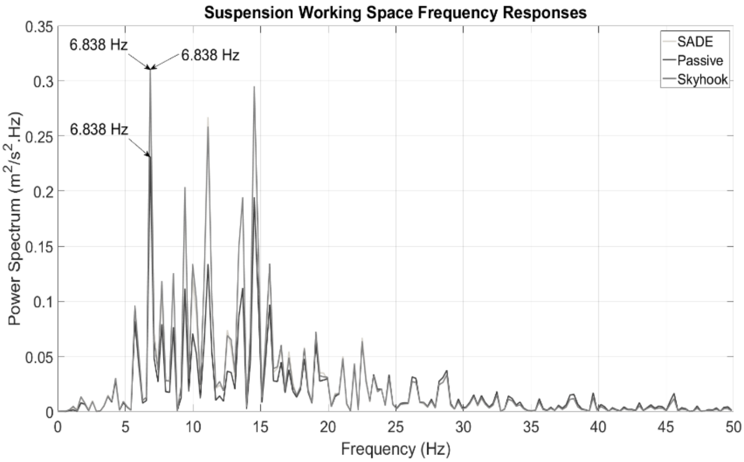

Figure 13: Suspension working space frequency responses

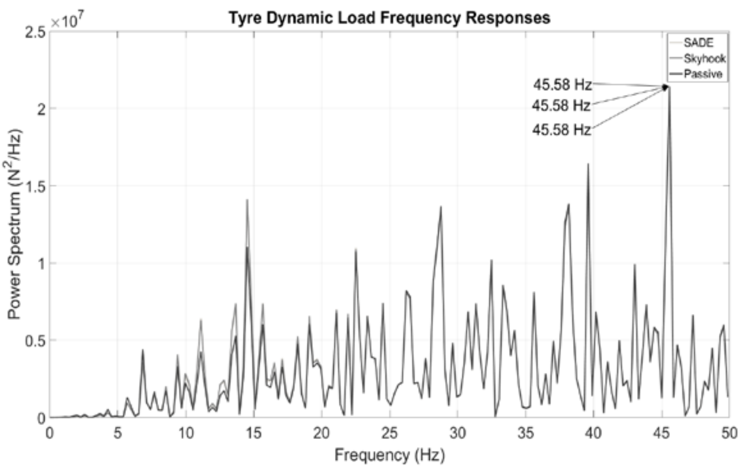

Figure 14: Tyre dynamic load's frequency responses

Conclusions

The semi active suspension systems performed better than the normal passive suspension system. However in term of the comparison made between the algorithms used to control the operation of MR damper model in study, it was found that the performance of skyhook-controlled and SADE-controlled systems in attenuating random road profiles are slightly different, with the difference of overall percentage performance is only 6 percent. Both of the algorithms improves sprung mass's vertical jerk and acceleration by at least 50 percent with some trade-off of suspension working space and tyre dynamics load. However, the trade-off which exists for the suspension working space and tyre dynamic loads responses in the SADE-controlled semi active suspension system, were observed to be lesser than the trade-off made by the skyhook-controlled system. The dominant occurrences of the magnitudes in the studied responses were also observed to remain the same as the passive suspension system, except for the displacement response where the studied semi active suspension systems is causing the sprung mass to oscillate slightly more.

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