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Research paper

Evaluation of an Electric Vehicle Ride Dynamics under ISO-2631 Criteria

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Abstract

The work is aimed to study the vertical response of ride performance of an electric vehicle. The issues related to the design of vehicle model with passive suspension system are discussed. A full-car seven-degree-of-freedom model is used to investigate the dynamics response by applying road disturbances in sinusoidal road input excitation. The time response of the heave, roll, and pitch of the sprung mass is obtained for the need of studying the effect of given variation of both suspension stiffness coefficient and suspension damping coefficient. Finally, the resulted responses in the time domain are then evaluated using ISO-2631 criteria to assess the passenger comfortability.

Keywords: *Electric vehicle, suspension system, comfortability, electric vehicle, and ISO-2631..*

1. Introduction

Ride and handling characteristics of an automobile depend on the characteristics of the tires. Reaction point between the vehicle and roadway occurs on the tires. Tire manages the input of forces and disturbances from the road. Tire characteristics are therefore a key factor in the effect of the road on a vehicle. It also affects the output forces which controls vehicle stability and cornering characteristics. The tire's basic characteristics are managed by the system of springs, dampers, and linkages that control the way in which tires move and react to disturbances and control inputs [1].

Road condition can show real performance of vehicle ride just like ride over bump. The center of gravity height, relative to the track, determines load transfer, also called weight transfer, from side to side and caused body rolling. Centrifugal force acts at the center of gravity to lean the car toward the outside of the curve, increasing downward force on the outside tires. The center of gravity height, relative to the wheelbase, determines load transfer between front and rear. The momentum of car acts at its center of gravity to twist the car forward or backward, respectively during braking and acceleration. Since it is only downward force that changes and not the location of the center of gravity, the effect on over/understeer is opposite to that of an actual change in the center of gravity. When a car is braking, the downroad load on the front tires in- creases and that on the rear decreases, with a corresponding change in their ability to take sideways load, causing oversteer [2].

The quality referred to as ride comfort is affected by a variety of factors including high-frequency vibrations, body booming, body roll, and pitch, as well as the vertical spring action normally associated with a smooth ride [2]. If the vehicle is noisy, or it rolls excessively in turns, or pitchs during accelerations and braking, or the body produces a booming resonance, occupants will experience the uncomfortable ride.

The ride quality normally associated with the vehicle's response to bumps is a factor of the relatively low-frequency bounce and rebound movements of the suspension system [1]. Following a bump, the un-damped suspension of a vehicle will experience a series of oscillations that will cycle according to the natural frequency of the system.

According to Newton's first law, a moving body will continue moving a straight line until it is acted upon by a disturbing force. Newton's second law refers to the balance that exists between the disturbing force and the reaction of the moving body. In the case of the automobile, weather the disturbing force in the form of a wind-gust, an incline in the roadway, or the cornering forces produced by tires, the force causing the turn and the force resisting the turn will always be in balance [3].

Vehicle handling characteristics have to do with the way in which the vehicle's inertial forces and the cornering forces of the tires act against each other. The magnitude and vector of the inertial forces are established by the vehicle's weight and balance. In a turn, angular acceleration results in a force that is centered at the vehicle center of gravity and acts in a direction away from the turn center. The ability to overcome these forces and produce a controlled, stable turn depends upon the combined characteristics of the suspension and tires. The job of the suspension system is to support, turn, tilt and otherwise manage the tires and their relationship to the vehicle and the ground in a way that will maximize their capabilities.

From the description above, then studying about the ride dynamics model of Semar-T is very important to obtain the appropriate parameters in order to develop the comfortability of Semar-T including the improvement on vertical response. So that passengers will experience a comfortable ride.

2. Vehicle Model

Numerous papers about the theoretical and experimental investigation on the dynamic behavior of passively and actively suspended road vehicles have been published to improve ride quality and handling performance [9-11]. The quarter-vehicle model [3,8], half-vehicle model [4,6,7] and complete-vehicle model [12] have been developed with researches related to the dynamic behavior of vehicle and its vibration control. Suppression of vibration in passive suspensions depends on the spring stiffness, damping coefficient and car mass [5]. Some assumptions in order to develop a complete-vehicle model such as the tires are modeled as a linear spring without damping; there is no rotational motion in wheels; the behavior of springs and dampers are linear; the tires are always in contact with the road surface and effect of friction is neglected so that the residual structural damping is not considered into vehicle modelling; the center of gravity is located in the center of vehicle.

In any vehicle dynamics simulation, there are some calculations for a particular vehicle axis system as illustrated in Figure 1 The vehicle fixed coordinate system is right-hand orthogonal, originates at the body centre of gravity (CG) and travels with the vehicle. This standard coordinate system will be used to describe the forces on the vehicle.

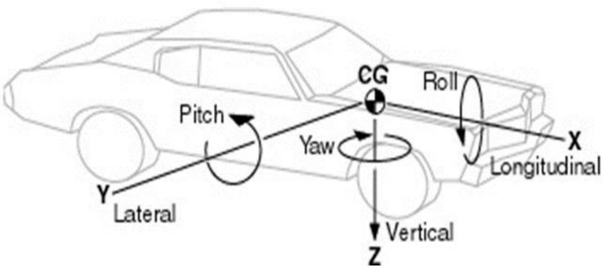


Fig. 1: Vehicle coordinate system

The complete-vehicle model is represented as a linear seven degree-of-freedom (DOF) system as shown in Figure 2. It consists of a single sprung mass m_s (car body) connected to four unsprung masses m_{u1}, m_{u2}, m_{u3}, and m_{u4} (front-left, front-right, rear-left, rear-right wheels) at each corner. The sprung mass is free to bounce, pitch and roll while the unsprung masses are free only to bounce vertically with respect to the sprung mass. All other mo- tions are neglected for this model. Hence this system has seven degrees-of-freedom and allows simulation of tire load forces in all four tires, body acceleration and vertical body displacement as well as roll and pitch motion of the car body. The suspensions between sprung mass and unsprung masses are modeled as linear viscous dampers and linear spring elements, while the tires are modeled as simple linear springs without damping. For simplicity, all pitch and roll angles are assumed to be small [2].

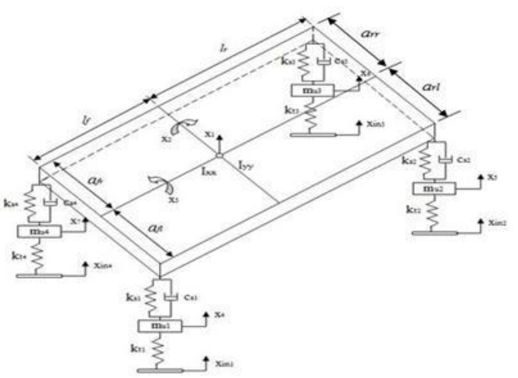


Fig. 2: Seven DOF full-car model

3. Arrangement of Variations

From the press testing of damper and spring has been conducted known that the values of suspension damping coefficient (cs) and suspension stiffness coefficient (ks) to a passenger car 1000 kg are worth 982 Ns/m and 18218 N/m as shown in Figures 3 and 4. Hence, the value variations of dampers and springs for Semar-T are taken in around those values. The values of suspension damping coefficient varied between 700-1500 Ns/m, while the values of suspension stiffness coefficient varied between 15000-27000 N/m.

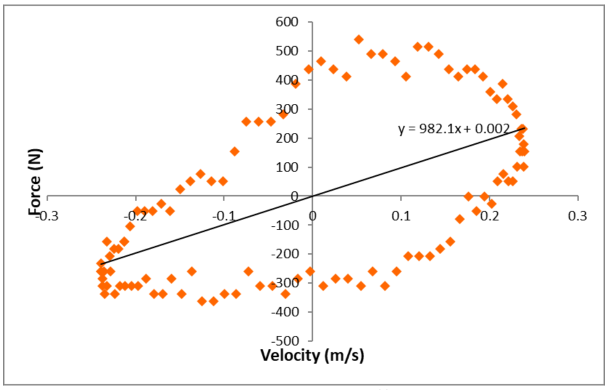


Fig. 3: Suspension damping coefficient testing result

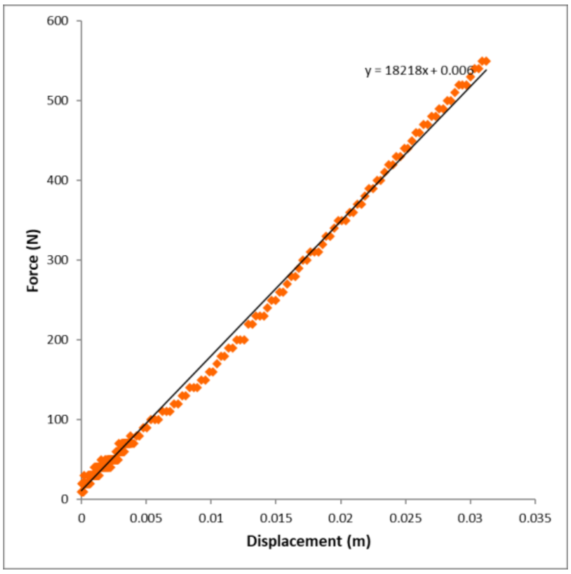


Fig. 4: Suspension stiffness coefficient testing result

4. Simulation and Validation

Simulation was performed for a period of five seconds and a fix step size of 0.025 seconds. The type of road disturbance considered in this study is sinusoidal function with the amplitude of 0.075 m and the excitation frequency of 1 Hz. The numerical values of the full car model parameters and parameters variations are set in the Tables 1 and 2.

Table 1: Simulation setup

Symbol	Value m_s 1400 kg
I_{xx}	450.1 kg.m^2
I_{yy}	1263.5 kg.m^2
m_{u1} and m_{u4}	40 kg
m_{u2} and m_{u3}	35 kg
k_{s1}, k_{s2}, k_{s3} and k_{s4}	are variated
c_{s1}, c_{s2}, c_{s3} and c_{s4}	are variated

k_{t1}, k_{t2}, k_{t3} and k_{t4}	200,000 N/m
a_{fl}, a_{fr}, a_{rl} and a_{rr}	0.65 m
l_f and l_r	0.95 m

Table 2: List of Parameter Variations

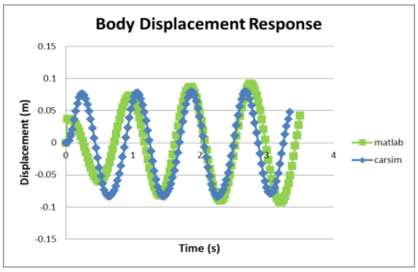
Variation		k_s $[N/m]$	c_s $[N/m]$
1	A	18,000	700
	B	18,000	900
	C	18,000	1100
	D	18,000	1300
	E	18,000	1500
2	A	15,000	900
	B	18,000	900
	C	21,000	900
	D	24,000	900
	E	27,000	900

In simulation study, it is essential to justify that the model is valid. Therefore, the vehicle ride model must be compared with other validated model. The validated model used in this case is CARSIM software which is known as vehicle dynamic software developed by University of Michigan. The other common software used to simulate ride dynamics is MATLAB-SIMULINK. However, due to the computational cost and validity of the simulation results, CARSIM is preferable than the counterpart. This software can be easily downloaded from internet and freely installed into a personal computer.

5. Results and Discussion

5.1. Validation Results

A complete-vehicle seven-degree-of-freedom model is verified with CARSIM software with sinusoid road profile mode. Model verification is performed for a period of 3.5 seconds. The numerical values of the vehicle model parameters are obtained from the design of Semar-T 2nd generation as shown in Table 1. Figures 3 show the responses of model and vehicle behaviors obtained from CARSIM in terms of body displacement, body acceleration, pitch angle, roll angle, front-left suspension travel, front-left wheel acceleration, rear-left suspension travel, rear-left wheel acceleration, rear-right suspension travel, rear-right wheel acceleration, front-right suspension travel and front-right wheel acceleration responses.



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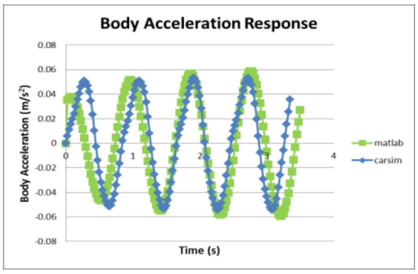

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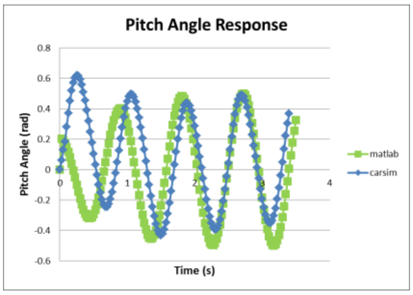

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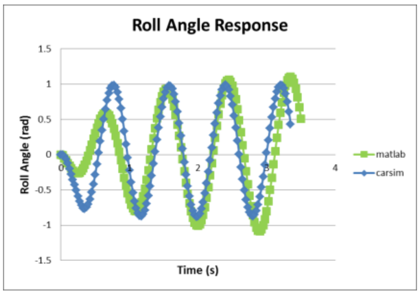

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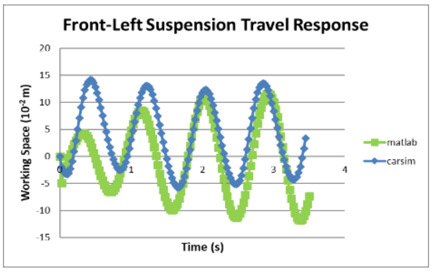

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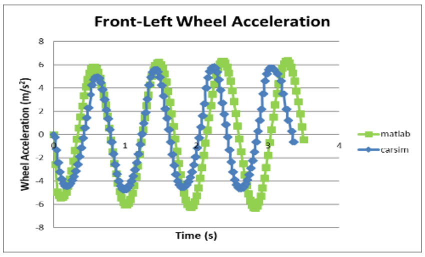

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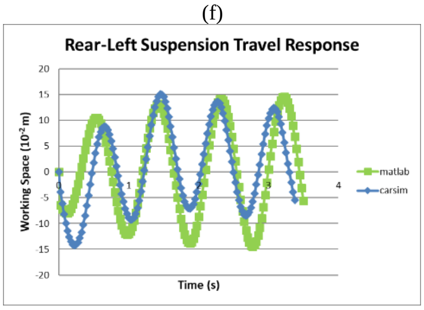

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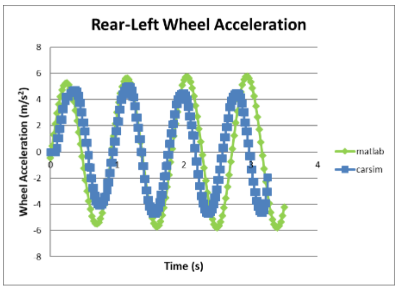

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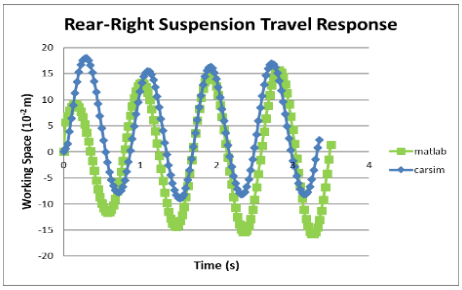

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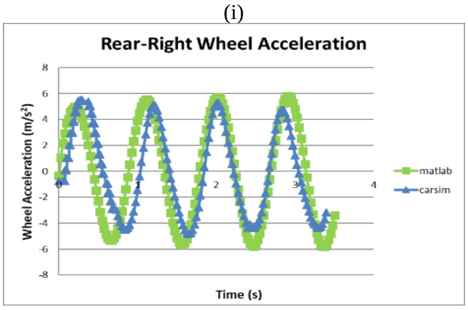

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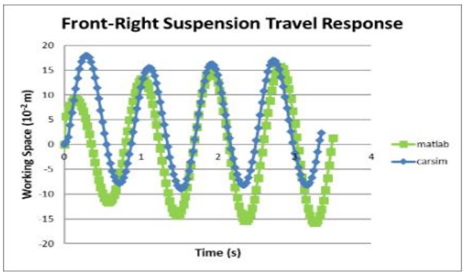

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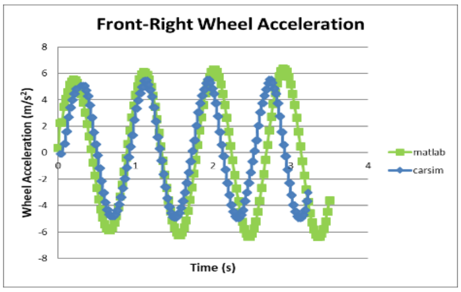


(l)

Fig 5: Validation graphic of: (a) body displacement response, (b) body acceleration response, (c) pitch angle response, (d) roll angle response, (e) front-left suspension travel response, (f) front-left wheel acceleration response, (g) rear-left suspension travel response, (h) rear-left wheel acceleration response, (i) rear-right suspension travel response, (j) rear-right wheel acceleration response, (k) front-right suspension travel response, and (l) front-right wheel acceleration response.

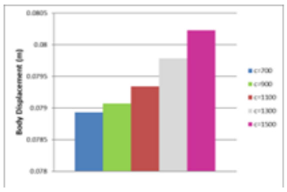
It can be seen that the trend between model developed and CARSIM results are almost similar, but slightly different in magnitude. The slight difference in the magnitude may be due to the fact that the suspension in CARSIM which used double wishbone, whereas in the developed model, the suspension is assumed to be linear suspension. In overall, similar responses and trend can be obtained from the developed model compared to the behaviors obtained with CARSIM. As long as the trend of the model response is closely similar with the CARSIM results, it can be said that the model is valid.

5.2. Simulation Results

The effectiveness of five suspension system variations is also investigated in time domain simulation. The simulation results are shown in Figure 4 where ks constant. In which the blue block indicates 1.a (c_s = 700 Ns/m) variation, green block indicates 1.b (c_s = 900 Ns/m) variation, red block indicates 1.c (c_s = 1100 Ns/m) variation, gray block indicates 1.d (c_s = 1300 Ns/m) variation and violet block indicates 1.e (cs = 1500 Ns/m) variation. From Figures 4 (a) and (b), it can be seen that the 1.a variation shows significant improvement on two performance criteria namely body displacement and body acceleration. Unwanted vibratory motions of vehicle body can be suppressed by the 1.a variation resulting in improved ride performance.

Figures (c) and (d) shows the roll angle and pitch angle responses, it can be seen that the improvement of 1.a variation system in term of pitch angle better than four variations other, but there is no improvement in term of roll angle response. Figures (e)-(l) show the suspension travel and wheel acceleration responses of five suspension system variations are compared for each corner. It can be seen that the 1.a variation shows significant improvement on wheel acceleration response, but there is no improvement on suspension travel response.

Time domain of 1st variation: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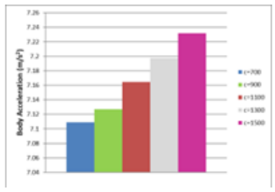
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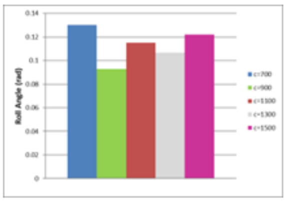
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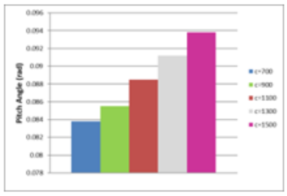
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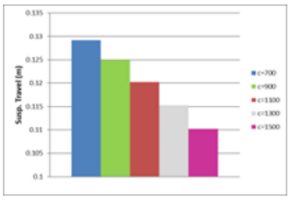
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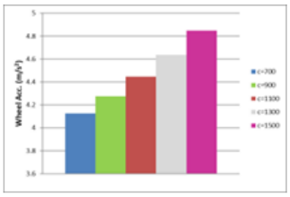
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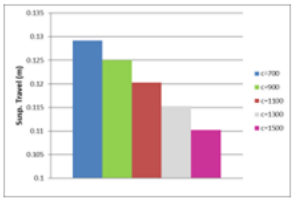
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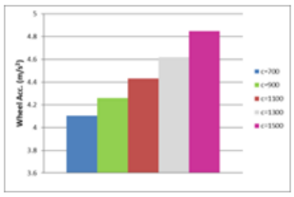
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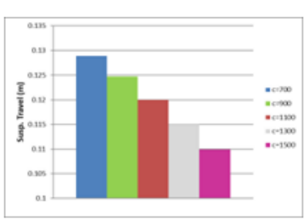
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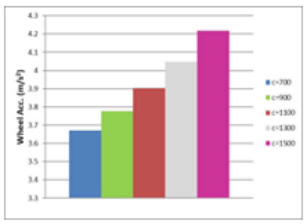
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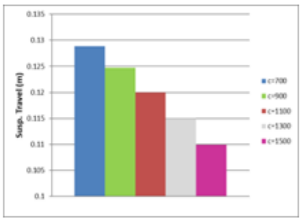
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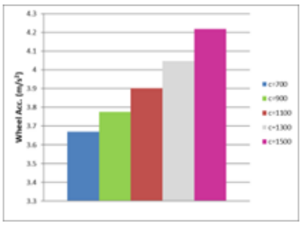
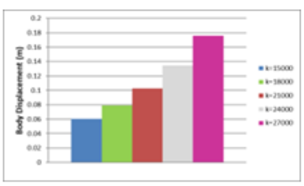

(l)

Fig 4: Peak-to-Peak values of: (a) body displacement response, (b) body acceleration response, (c) roll angle response, (d) pitch angle response, (e) front-left suspension travel response, (f) front-left wheel acceleration response, (g) rear-left suspension travel response, (h) rear-left wheel acceleration response, (i) rear-right suspension travel response, (j) rear-right wheel acceleration response, (k) front-right suspension travel response, and (l) front-right wheel acceleration response, for different values of the suspension damping coefficients (c_s) and constant value of the suspension stiffness coefficients (k_s = 18000 N/m)

The effectiveness of five suspension system variations is also investigated in time domain simulation. The simulation results are shown in Figure 5 where c_s constant. In which the blue block indicates 2.a (k_s = 15000 N/m) variation, green block indicates 2.b (k_s = 18000 N/m) variation, red block indicates 2.c (k_s = 21000 N/m) variation, grey block indicates 2.d (k_s = 24000 N/m) variation and violet block indicates 2.e (k_s = 27000 N/m) variation. From Figures 5 (a) and (b), it can be seen that the 2.a variation shows significant improvement on two performance criteria namely body displacement and body acceleration. Unwanted vibratory motions of vehicle body can be suppressed by the 2.a variation resulting in improved ride performance.

Figures (c) and (d) shows the roll angle and pitch angle responses, it can be seen that the improvement of 2.a variation system in term of roll and pitch angle better than four variations other. Figures (e)-(l) show the suspension travel and wheel acceleration responses of five suspension system variations are compared for each corner. It can be seen that the 2.a variation shows significant improvement on suspension travel response, but there is no improvement on wheel acceleration response.

Time domain of 2nd variation: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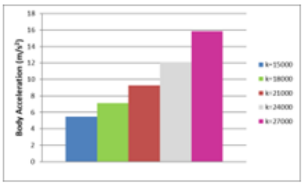
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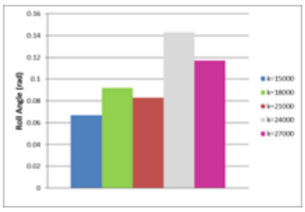
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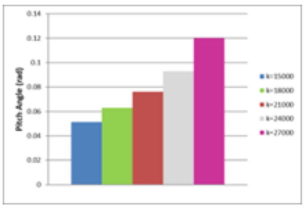
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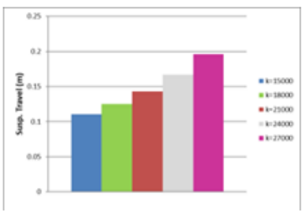
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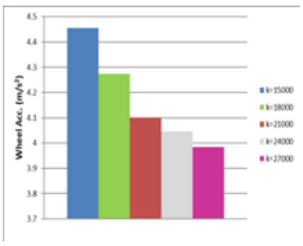
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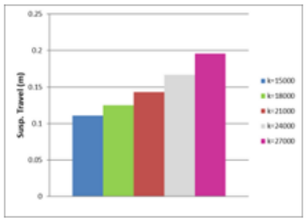
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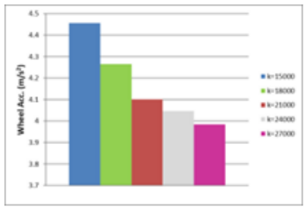
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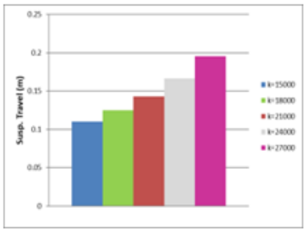
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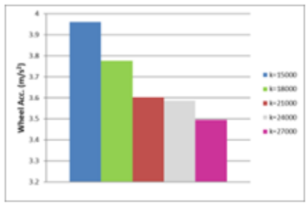
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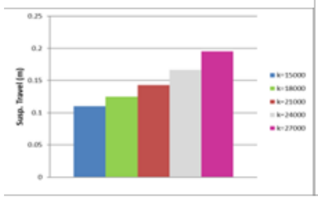
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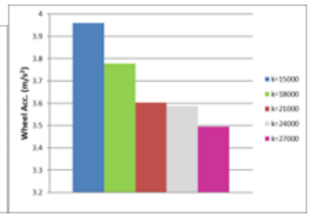
(l)

Fig 5: Peak-to-Peak values of: (a) body displacement response, (b) body acceleration response, (c) roll angle response, (d) pitch angle response, (e) front-left suspension travel response, (f) front-left wheel acceleration response, (g) rear-left suspension travel response, (h) rear-left wheel acceleration response, (i) rear-right suspension travel response, (j) rear-right wheel acceleration response, (k) front-right suspension travel response, and (l) front-right wheel acceleration response, for different values of the suspension stiffness coefficients (k_s) and constant value of the suspension stiffness coefficients (c_s = 900 Ns/m)

5.3. ISO 2631 Analysis

Referring to the first variation, it can be seen that the performance of the 1.b (ks = 18,000 N/m, cs = 900 Ns/m) suspension system variation shows better performance compared to its counterpart. Hence, it can be noted that the performance of the 1.b suspension system variation is reducing unwanted body acceleration significantly better than other suspension system variations, especially at frequency range of body natural frequency to wheel natural frequency. However, for the frequency of excitation exactly at the wheel natural frequency is slightly worse, because passengers are only able to survive for 2.5 hours. However, the passengers will be able to last longer to feel the vibration on the 1.b suspension system variation.

From the explanation above, so can be concluded that 1.b (cs = 900 Ns/m) suspension system variation is the most optimal variation better than other variations, because it is reducing unwanted body motions. So that, it can be improve ride performance of Semar-T 2nd generation and provide the experience comfortable ride for passengers.

From the second variation, it can be seen that the performance of the 2.a (ks = 15,000 N/m, cs = 900 Ns/m) suspension system variation shows better performance compared to its counterpart. Hence, it can be noted that the performance of the 2.a suspension system variation is reducing unwanted body acceleration significantly better than other suspension system variations, especially at frequency range of body natural frequency to wheel natural frequency. However, for the frequency of excitation exactly at the wheel natural frequency is slightly worse, because passengers are only able to survive for 2.5 hours. However, the passengers will be

able to last longer to feel the vibration on the 2.a suspension system variation.
From the explanation above, so can be concluded that 2.a (ks= 15,000 N/m) suspension system variation is the most optimal variation better than other variations, because it is reducing unwanted body motions. So that, it can be improve ride performance of Semar-T 2nd generation and provide the experience comfortable ride for passengers.

6. Conclusions

A computer simulation of ride performance of the electric car has been conducted to study the effect of both stiffness and damping constants on the comfortability. The variation of stiffness and damping in the suspension showed a positive sign for further spring and damper selection. Increasing the value of suspension stiffness coefficient will cause increasing the value of vertical response at around body natural frequency (0.5-2 Hz approximately). While, increasing the value of suspension damping coefficient will cause increasing the value of vertical response at around wheel natural frequency (5-20 Hz approximately). So that, low stiffness coefficient is needed to improve ride performance of vehicle. The variation having value of k_s = 15,000 N/m and c_s = 900 Ns/m shows superior performance than other variations and able to improve all selected performance criteria, although the body roll response is slightly worse. Overall, the ride dynamic response of the electric vehicle has been in the range of ISO-2631 criteria about passenger ride quality.

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