Lightweight Vehicle and Driver’s Whole-Body Models for Vibration Analysis

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Abstract. Vehicle vibration is a main factor for driving fatigue, discomfort and health problems. The ability to simulate the vibration characteristics in the vehicle and its effects on driver’s whole-body vibration will give significant advantages to designers especially on the vehicle development time and cost. However, it is difficult to achieve optimal condition of ride comfort and handling when using passive suspension system. This paper presents mathematical equations that can be used to describe the vibration characteristics of a lightweight electric vehicle that had been developed. The vehicle’s model was combined with the lumped-parameter model of driver to determine the whole-body vibration level when the vehicle is passing over a road hump using Matlab Simulink. The models were simulated at a constant speed and the results were compared with the experimental data. The simulated vibration level at the vehicle floor and seat were almost similar to the experimental vibration results. The suspension systems that are being used for the solar vehicle are able to reduce the vibration level due to the road hump. The models can be used to simulate and choose the optimal parameters for the suspensions.

1. Introduction

Vehicle vibration is one of the major factors that results to driving discomfort, fatigue and health problems to drivers. Improvement on wheel and seat suspension systems, and in-vehicle ergonomics consideration for drivers to minimise the effect of the vibration had long been carried out [1-3]. Whole body vibration (WBV) exposure is a common phenomenon in vehicles where vibration from road surface and structures are transmitted through the legs, buttocks and back of seated individuals. Prolonged WBV exposure may lead to adverse affects on health, activities and comfort, and may also cause motion sickness. Over-exposure may result in lumbar disc disease and tissue failure or metabolic interference. The lumbar disc could also be damaged if sitting in incorrect posture in over-exposure vibration situation. The most common injury reported due to WBV is low back pain (LBP) even though there is non-exact dose-response relationship between them [4-5].

Most of the numerical analysis studies related to the vehicle vibration were based on mass-spring-damper systems since both the spring and the damper are the two main components that form the basic system of the vehicle suspension. From the mechanical models of the systems, mathematical
models were developed to analyse the characteristics of the vibration and results were compared with the allowable international standard [6] of vibration exposure to the driver. Generally, the studies were done by adopting parameters of ready made systems and necessary modification to existing systems to overcome those problems would be very limited because of the complexity of the system. However, the used of simulation software to analyse vibration characteristics for structures has enabled designers to analyse and determine the components and suitable parameters that are able to overcome the vibration problems.

2. Method
The integration of ergonomics at the early stage of design process will reduce product development time and cost [7-8]. However, the process requires tools which are able to compute the numerical analysis with respect to various human requirements and expectations. Therefore, the need for cost-effective computing tools which can incorporate ergonomics consideration during the product design stage is crucial [9].

In this study the dynamic behaviour of the vehicle and the driver is investigated using Matlab Simulink software. The combination of both vehicle and driver models will give early indication on how the vehicle suspension system will affect the driver especially on the driving comfort levels. The vehicle on suspensions can be modelled as a lumped parameters system with a rigid body connected to the wheels by the springs and dampers. The suspensions must be a linear or nonlinear elastic system and incorporate the damping function to avoid the onset of resonant oscillations [10]. A six degrees of freedom (DOFs) full vehicle model was developed by combining three quarter-vehicle suspension models [11] presenting the vertical motion and angular pitch and roll motions. The seated driver can be represented by three DOFs system consists of three body components and a seat which were coupled by spring and damping elements. The input signal to the seat is the vertical displacement of the vehicle floor assuming that the seat was placed on the centre of gravity (CG) of the vehicle. The seat is also using a passive suspension system consists of the spring and damping components. In this study, the simulation parameters used were almost similar to the components being used for the electric vehicle. The input signal or the road profile for the simulation is a road hump.

2.1. Lightweight vehicle
An electric car which uses solar energy to charge its battery has been designed and fabricated. The design has considered of using lightweight material and component because weight is an important factor for the vehicle in terms of electrical energy saving. The vehicle main body structure was fabricated using 38 mm aluminium hollow pipe of 3.14 mm thickness. Other components used such as the wheels, spring suspensions, steering and seat are common parts available off-the-shelf.

The car uses a 48V direct current (DC) permanent magnet motor of 3000 maximum r.p.m. The swing arm of the rear wheel is designed so that it can move freely in vertical direction. Four units of motorcycle spring damper suspensions are being used at the front wheels and the rear swing arm. The rear wheel consist of two 70 inches tyres which was coupled together to support the weight of the DC motor on the swing arm. The front left wheel and right front wheel consists of 21 inches tyre with aluminium sport rim. Figure 1 shows the location of the suspension systems and seat.

![Figure 1. Lightweight electric car.](image-url)
2.2. Simulation
Simulation was done using Matlab/Simulink. The mathematical model of the electric vehicle suspension system and the lumped-parameter model of a driver were developed and transferred into the Simulink block command. This analysis will help in analysing the vibration level that could be transmitted to the driver’s body resulted from the road hump, through the vehicle suspension, the vehicle floor and the seat.

The simulation uses an input signal of the road hump’s profile with a maximum height and width of 0.09 metres and 3 metres respectively. The velocity of the vehicle is assumed to be constant at 15 km/h, a preferred velocity when driving over the hump [12-14]. The equations of motion for the 6 DOFs full-vehicle and 3 DOFs seated driver models were programmed accordingly using blocks in Simulink. The parameters for the models are given in table 1 and table 2. There were three inputs for the simulation at both front wheels and rear wheel which represented by $Z_{rfl}$, $Z_{rfr}$ and $Z_{rr}$. As mentioned previously, the input was a road hump signal where $Z_{rfl}$ and $Z_{rfr}$ were having the same signal assuming that both front wheels reached the road hump simultaneously. The signal for $Z_{rr}$ was delayed which depends on the velocity of the vehicle, $v$. For the simulation, the velocity, $v$ was set to 5.5 m/s. The time taken for the vehicle (or the rear wheel) to pass the hump is approximately 1 sec.

**Table 1.** The parameters for full-vehicle model.

| Description                        | Symbol | Value   |
|------------------------------------|--------|---------|
| Sprung mass                        | $M_s$  | 310 kg  |
| Left front - unsprung mass         | $M_{ufl}$ | 15 kg  |
| Right front - unsprung mass        | $M_{ufr}$ | 15 kg  |
| Rear - unsprung mass               | $M_{ur}$ | 30 kg  |
| Roll - moment of inertia           | $I_{xx}$ | 130 kg/m$^2$ |
| Pitch - moment of inertia          | $I_{yy}$ | 340 kg/m$^2$ |
| Left front - susp. stiffness       | $K_{sfl}$ | 92 kN/m |
| Right front - susp. stiffness      | $K_{sfr}$ | 92 kN/m |
| Rear - suspension stiffness        | $K_{sr}$  | 184 kN/m |
| Left front - susp. damping         | $C_{sfl}$ | 500 Ns/m |
| Right front - susp. damping        | $C_{sfr}$ | 500 Ns/m |
| Rear - suspension damping          | $C_{sr}$  | 1000 Ns/m |
| Left front tyre stiffness          | $K_{tfl}$ | 38 kN/m |
| Right front tyre stiffness         | $K_{tfr}$ | 38 kN/m |
| Rear tyre stiffness                | $K_{tr}$  | 35 kN/m |
| Left front tyre damping            | $C_{tfl}$ | 150 Ns/m |
| Right front tyre damping           | $C_{tfr}$ | 150 Ns/m |
| Rear tyre damping                  | $C_{tr}$  | 100 Ns/m |
| Length - front wheel to C.G        | $a$     | 1.4 m   |
| length - rear wheel to C.G         | $b$     | 1.5 m   |
| Width of sprung mass               | $w$     | 1.4 m   |
| Seat mass                          | $M_{se}$ | 5 kg    |
Table 2. The parameters for full-vehicle model.

| Description                        | Symbol | Value   |
|------------------------------------|--------|---------|
| Buttock and legs mass              | $M_{bl}$ | 20 kg   |
| Body mass                          | $M_{bo}$ | 40 kg   |
| Head mass                          | $M_{he}$ | 5 kg    |
| Seat suspension stiffness          | $K_{se}$ | 7 kN/m  |
| Seat suspension damping            | $C_{se}$ | 900 Ns/m|
| Buttock and legs stiffness         | $K_{bl}$ | 40 kN/m |
| Buttock and legs damping           | $C_{bl}$ | 2500 Ns/m|
| Body stiffness                      | $K_{bo}$ | 35 kN/m |
| Body damping                       | $C_{bo}$ | 750 Ns/m|
| Head stiffness                      | $K_{he}$ | 130 kN/m|
| Head damping                       | $C_{he}$ | 250 Ns/m|

2.3. Experimental set-up

An experiment was done to measure the vibration at the vehicle floor and seat when the vehicle passing over the speed hump. Two Kistler (Type: 8690C) piezobeam accelerometers were used and placed on the vehicle floor (under the seat) and also on the seat. Both accelerometers were connected to National Instrument NI9233 four-channel, ± 5V, 50 kS/s per channel, 24-Bit IEPE dynamic signal acquisition module and NI CompactDAQ. The analogue input was analysed using NI Sound and Vibration software using LabVIEW SignalExpress interface. The measurement rate and sample size were set to 50k Hz and 500k respectively using continuous samples acquisition mode. The data was taken for a period of 10 second for a distance of approximately 40 metres between the start and stop points. The signal was processed using Butterworth lowpass filter with 20 Hz cut-off frequency. During the experiment, the speed of the vehicle was kept constant when passing the hump.

3. Full-vehicle model

A full-vehicle suspension model can be represented as a linear six DOFs system consisting of the front and rear wheels, unsprung masses, sprung mass and suspension components as shown in figure 2.

![Figure 2. Six DOFs full-vehicle model.](image-url)
In the model, wheels are represented by tyres having spring characteristics. The unsprung mass is made up of the weights of the wheels, axle and the components located geometrically below the suspension system while the sprung mass represents the body or chassis of the car. This six DOFs model was adapted from Ikinega [15] and the pitch and roll angles were assumed to be small. Using the Newton’s second law, the equations of motion for the six DOFs model of the solar vehicle are given by (1) to (6).

(1), (2) and (3) show the equations of motion in the vertical direction for the unsprung masses \( M_{ugf} \), \( M_{ugr} \) and \( M_{ur} \) in terms of stiffness, \( K \), damping coefficient, \( C \), pitch, \( \theta \) and roll, \( \phi \). The vertical displacement, velocity and acceleration are given by \( Z \), \( \dot{Z} \) and \( \ddot{Z} \) respectively. Similarly for pitch and roll motions, the ‘dot’ and ‘double dot’ above the symbols represent the velocity and acceleration respectively. Details of the parameters used for the equations are described in table 1.

\[
\begin{align*}
M_{ugf} \ddot{Z}_{ugf} &= K_{ugf} Z_{ugf} + C_{ugf} \dot{Z}_{ugf} - a K_{ugf} \theta - a C_{ugf} \dot{\theta} + 0.5 w K_{ugf} \phi + 0.5 w C_{ugf} \dot{\phi} - K_{ugf} Z_{ugf} - C_{ugf} \dot{Z}_{ugf} \\
- K_{ugf} Z_{ugf} + K_{ugf} Z_{ugf} + C_{ugf} \dot{Z}_{ugf} + C_{ugf} \dot{Z}_{ugf}
\end{align*}
\]

\[
\begin{align*}
M_{ugr} \ddot{Z}_{ugr} &= K_{ugr} Z_{ugr} + C_{ugr} \dot{Z}_{ugr} - a K_{ugr} \theta - a C_{ugr} \dot{\theta} - 0.5 w K_{ugr} \phi - 0.5 w C_{ugr} \dot{\phi} - K_{ugr} Z_{ugr} - C_{ugr} \dot{Z}_{ugr} \\
- K_{ugr} Z_{ugr} + K_{ugr} Z_{ugr} + C_{ugr} \dot{Z}_{ugr} + C_{ugr} \dot{Z}_{ugr}
\end{align*}
\]

\[
\begin{align*}
M_{ur} \ddot{Z}_{ur} &= K_{ur} Z_{ur} + C_{ur} \dot{Z}_{ur} + b K_{ur} \theta + b C_{ur} \dot{\theta} - K_{ur} Z_{ur} - C_{ur} \dot{Z}_{ur} + K_{ur} Z_{ur} - C_{ur} \dot{Z}_{ur} + C_{ur} \dot{Z}_{ur} + C_{ur} \dot{Z}_{ur}
\end{align*}
\]

(4), (5) and (6) show the equations of motion for the sprung mass \( M_s \), in vertical, pitch and roll motions respectively.

\[
\begin{align*}
M_s \ddot{Z}_s &= -K_s Z_s - K_s Z_s - K_s Z_s - C_s \dot{Z}_s - C_s \dot{Z}_s - C_s \dot{Z}_s + a K_s \theta + a K_s \dot{\theta} - b K_s \theta + a C_s \dot{\theta} \\
+ a C_s \dot{\theta} - b C_s \dot{\theta} + K_s \dot{Z}_{ugf} + C_s \dot{Z}_{ugf} + K_s Z_{ugf} + C_s Z_{ugf} + K_s Z_{ur} + C_s Z_{ur} + C_s Z_{ur}
\end{align*}
\]

\[
\begin{align*}
I_{yy} \dddot{\theta} &= a K_{yy} Z_{yy} + a K_{yy} Z_{yy} - b K_{yy} Z_{yy} - a C_{yy} \dot{Z}_{yy} + a C_{yy} \dot{Z}_{yy} - b C_{yy} \dot{Z}_{yy} - a^2 K_{yy} \theta + a^2 K_{yy} \dot{\theta} - b^2 K_{yy} \theta \\
- a^2 C_{yy} \dot{\theta} - a^2 C_{yy} \dot{\theta} - b^2 C_{yy} \dot{\theta} - a C_{yy} Z_{ugf} + a C_{yy} Z_{ugf} - a C_{yy} Z_{ugf} - a C_{yy} Z_{ugf} + b K_{yy} Z_{ur} + b C_{yy} Z_{ur}
\end{align*}
\]

\[
\begin{align*}
I_{zz} \dddot{\phi} &= -0.25 w^2 K_{zz} \phi - 0.25 w^2 K_{zz} \phi - 0.25 w^2 C_{zz} \phi - 0.25 w^2 C_{zz} \phi + 0.5 w K_{zz} Z_{zz} + 0.5 w C_{zz} Z_{zz} \\
- 0.5 w K_{zz} Z_{zz} - 0.5 w C_{zz} Z_{zz}
\end{align*}
\]

From the above equations, the Simulink model was constructed. Figure 3 shows the model for the unsprung mass \( M_{ugf} \) which was given an input signal of the speed hump as shown in figure 4.
4. Driver’s model

Behaviour of human body seated in a car can be modelled as a 3 DOFs system as shown in figure 5. The model was adopted from Liang and Chiang [16] and comprised of three masses which are coupled with spring and damping elements. It was assumed that the model is linear in which the driving point mechanical impedance did not vary significantly over the range of excitation type and levels in the frequency range. The three masses are comprised of the head and neck, $M_{he}$, the body, $M_{bo}$ and the buttock and legs, $M_{bl}$ in contact with the vehicle seat, $M_{se}$. The masses of the lower legs and feet, and the hand and arm are not included based on the assumption of their negligible contributions to the biodynamic response of the seated body.

The addition of the driver and seat models to the vehicle has changed the equation of motion for the sprung mass $M_s$ (4) to that given by (7). The parameters used for the equations are described in table 1.

$$M_s \ddot{Z}_s = K_{sfl} (-Z_s + a\theta + Z_{ufl}) + K_{sfr} (-Z_s + a\theta + Z_{ufr}) + K_{se} (-Z_s - b\theta + Z_{se}) + K_{bo} (Z_{bo} - Z_s)$$

$$+ C_{sfl} (-\dot{Z}_s + a\dot{\theta} + \dot{Z}_{ufl}) + C_{sfr} (-\dot{Z}_s + a\dot{\theta} + \dot{Z}_{ufr}) + C_{se} (-\dot{Z}_s - b\dot{\theta} + \dot{Z}_{se}) + C_{bo} (\dot{Z}_{bo} - \dot{Z}_s)$$

(7)
The vertical equation of motion for the vehicle seat and the lumped-parameter model of seated driver are given by (8) to (11). The parameters used for the equations are described in table 2.

\[
M_{sw}\ddot{z}_w = K_w(z_s - z_{sw}) + K_{bl}(Z_{bl} - Z_{sw}) + C_{sw}(\dot{z}_s - \dot{Z}_{sw}) + C_{bl}(\ddot{z}_{bl} - \ddot{Z}_{sw})
\]  

(8)

\[
M_{bl}\ddot{z}_b = K_{bl}(Z_{sw} - Z_{bl}) + K_{bo}(Z_{bo} - Z_{bl}) + C_{bl}(\dot{z}_w - \dot{Z}_{bl}) + C_{bo}(\ddot{z}_{bo} - \ddot{Z}_{bl})
\]  

(9)

\[
M_{ho}\ddot{z}_{ho} = K_{ho}(Z_{bl} - Z_{ho}) + C_{ho}(\dot{z}_{bl} - \dot{Z}_{ho})
\]  

(10)

\[
M_{he}\ddot{z}_{he} = K_{he}(Z_{ho} - Z_{he}) + C_{he}(\dot{z}_{bl} - \dot{Z}_{he})
\]  

(11)

Figure 6 shows the Simulink model for the seat and the buttock and leg masses of the driver. It can be seen that the inputs to the seat were given by the \(Z_s\) and \(\dot{Z}_s\).

5. Results and discussion
The input signals for left front wheel \(Z_{rf}\) and rear wheel \(Z_{rr}\) were represented by the solid line and dashed line respectively as shown in figure 7. The input signal for right front wheel \(Z_{rfr}\) was similar to \(Z_{rf}\).

Figure 7. Input signals for the simulation.

Figure 8. \(M_{ufl}\), \(M_{ur}\) and \(M_s\) vertical accelerations.

The simulation results for vertical acceleration of the front wheel unsprung mass \(M_{ufl}\), rear wheel unsprung mass \(M_{ur}\) and sprung mass \(M_s\) are shown in figure 8, while the frequency power spectrum of \(M_s\) is shown in figure 9. When the vehicle was driven over the hump, the accelerations of \(M_{ufl}\) and \(M_{ur}\) were between 7 ms\(^{-2}\) and -10 ms\(^{-2}\) and decaying after passing the hump. However, the acceleration of \(M_s\) was between 4 ms\(^{-2}\) and -4 ms\(^{-2}\). Based on the simulation, the passive suspension system
implemented has attenuated the vibration acceleration from the wheels to the vehicle floor quite reasonably and the significant frequency of $M_s$ falls below 5 Hz.

![Figure 9. Frequency spectrum of $M_s$.](image)

The $M_{se}$ acceleration was lower compared to $M_s$ acceleration as in figure 10. However, the vibration acceleration of driver’s masses $M_{bl}$, $M_{bo}$ and $M_{he}$ were between 3.5 ms$^{-2}$ and -4.8 ms$^{-2}$. These accelerations were slightly higher compared to the $M_{se}$ acceleration because of the stiffness characteristics given to the masses. As can be seen in figure 11, the significant frequency of the seat was also below 5 Hz.

![Figure 10. $M_{se}$, $M_{bs}$, $M_{bo}$, $M_{he}$ vertical accelerations.](image)

![Figure 11. Power spectrum of $M_{se}$.](image)

Displacements of $M_{ul}$, $M_{ur}$ and $M_s$ are shown in figure 12. The maximum displacement of $M_{ul}$, $M_{ur}$ and $M_s$ was 0.09 m, 0.12 m and 0.06 m respectively.

An experiment was done to measure the vertical vibration at the vehicle floor and seat. The experimental acceleration results are shown in figure 13 and figure 14. Comparison with the vertical acceleration of $M_s$ and $M_{se}$ shows close correlation between the simulation and experimental results.

![Figure 12. Vertical displacement of $M_{ul}$, $M_{ur}$ and $M_s$.](image)
One of the possibilities to improve the comfort level for driving the vehicle is to change the damping coefficient of the suspension. Reducing the mass of the vehicle could also improve the comfort level but it is normally not possible as it has been considered at the design stage. Reducing the suspension stiffness may result to poor vehicle handling capability and also to cater for the mass of the vehicle.

6. Conclusion
Full-vehicle model of a three-wheeled lightweight electric vehicle and driver’s 3 DOFs model were developed and combined to investigate the performance of the passive suspension system to attenuate the vibration when passing a road hump. The results from the simulation show that the suspensions were able to reduce the vibration level reasonably well. The experimental results were also in close agreement with the simulation. Therefore, the models can be used further for choosing optimum parameters for the suspension systems when the vehicle is driven on different type of road conditions. A simulation analysis gives great advantages to designer for selecting and determining the most effective system and parameters of a vehicle before the implementation is done.

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