Vibration transmission simulation model of a seated vehicle passenger

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Abstract. The paper presents development of vertical vibration simulation for a seated passenger in a moving vehicle is resulting from the bounce effect of the vehicle under various conditions. Although extensive research has been conducted in this field of study, the existing analysis were conducted on either the suspension of vehicle or the human body and not both. In this paper, the simulation model consists of three sub-systems, namely, vehicle suspension, seat suspension and human body model in which the vertical vibration is transmitted. By incorporating these sub-systems into the simulation, a correlation between mechanical and biological aspects can be formed between the three sub-systems. The transmission of vertical vibration in the validated simulation model provides a more realistic approach which can result to a better comparison to the real-life scenario. Parametric analysis of passive suspension system shows that lower mass ratio, higher stiffness ratio and lower damping coefficient results in better ride comfort. The incorporation of variable damper into the suspension system shows significant improvement in settling time, peak displacement and velocity, lesser discomfort rating and higher safety in passenger body.

1. Introduction
In recent years, statistics show that there has been an increase in vehicle usage all around the world. This is due to the convenience and the advantages which vehicles has to offer for society. Based on the Road Transport Department of Federal Territory in Malaysia, there has been an increment of 1156% of registered private vehicles from the year 1983 to 2004 [1]. The increase in registered vehicles is a testament to how many people are reliant on their vehicles in their daily life. The usage of these vehicles subject the users to various vibratory waves. These waves are transmitted to the human body due to direct or indirect vibratory sources from external environment which usually occur in low frequencies [2]. This natural occurrence is called Whole-Body-Vibration (WBV). Over-exposure of the human body to WBV is known to cause harmful effects towards users due to the sensitivity of the human body towards dynamic stress environments just like every other mechanical structure [3]. The harmful effects which are present consists of physical injury, impairment towards human work efficiency, safety and also health difficulties [3-4]. This is because vibrations are known to affect humans through information processing by impairment of senses and also human performance through excitement or fatigue [5]. Therefore, it is of vital importance to address these problems and work towards improving the existing technology to mitigate the risks and effects of users when operating a vehicle in their daily lives. The objectives that the research aims to achieve is to develop a suitable simulation model for vertical-WBV that a passenger is subjected to when operating a vehicle under various conditions and assess its validity using comparison of numerical simulation results and existing experimental data. Furthermore, the
research aims to assess the severity of vertical-WBV that is transmitted to the passenger and propose modifications which can be made to the existing suspension system of vehicles that contribute to the improvement of ride comfortability and safety.

There are various parameters of a vehicle which could possibly affect the vibrations experienced by a seated passenger. For an example the aerodynamics and suspension of a vehicle. The drag of a vehicle is known to result in the amplification of vibrations oscillations [6]. However, modification of the aerodynamics would often require a change in design of the vehicle body itself which could be tedious and undesirable for vehicle manufacturers. On the other hand, automotive suspension systems are designed in order to achieve three main criteria which are passenger comfort, vehicle road handling, and also capacity of the suspension system to carry load. These three criteria often conflicts each other whereby to improve one criteria, would be at the expense of another [7]. Therefore, extensive research is being conducted to obtain the optimal parameters of a suspension system. However, in this research the primary focus is to improve the suspension system in terms of passenger comfort as it is identified that there is a direct causable relationship whereby the low frequency vertical vibration resulting from bounce effect of the vehicle is the dominant vibration affecting passenger comfort [8].

Furthermore, there are three existing types of suspension systems that are in use today. Namely, passive, semi-active and active suspension systems which vary in terms of their ability to attenuate vibrations. The passive suspension system is a rigid set of spring and shock absorbers as the damper with fixed settings. Semi-active suspension systems on the other hand has a variable damper or spring control system which conducts adaptation to the road irregularities and mitigates the vibration effects and lastly, active suspension systems which provides an additional force input to counteract the vibration signals that is received by the control system [9]. However, many existing vehicles that are in use and produced today still commonly uses the passive suspension system due to the advantages that it has in terms of cost. Although studies which are carried out to determine the optimal parameter values would eliminate the dilemma of which criteria should be more prioritized, it is still unable to ultimately achieve the desired goal of the suspension system as the vehicle undergoes frequent changes in external environmental conditions which would require different sets of parameters. This led to the emergence of active and semi-active suspension systems. Due to the active suspension systems that demands high amount of external power in order to function optimally which results in higher complexity, weight, consistency and also cost [10-11], a semi-active suspension system with variable dampening is further studied in this paper.

There are several types of variable dampers which are available in the market. Namely, valve dampers using servo or solenoid, electromagnetic dampers, and also magnetorheological (MR) or electrorheological (ER) dampers. Out of all the available options, it is found that the MR damper standout due to its relatively simple mechanical design, lower cost of production and most importantly, its ability to generate high levels of damping with low power [7]. The body of the MR damper is a hydraulic cylinder and inside it consists of polarizable particles which are micron sized which exist in the MR fluid. These materials in the cylinder has the ability to convert from viscous fluids that flow freely to semi-solid state in an extremely short period of time when they are exposed to magnetic fields. This occurs due to the suspended iron particles in the fluid that forms a chain structure along the flux lines of the magnetic field [12]. The viscosity of the fluid in the MR damper which is manipulated enables the damping properties of the variable shock absorber in a vehicle to be controlled accordingly [11].

The research focuses on assessing modifications which are made to passive suspension system and also the incorporation of semi-active suspension system and how these modifications directly effects the acceleration responses of the human body. Hence, the discomfort rating and health safety can be further assessed according to the international standards set by ISO 2631 which will further be discussed in the coming sections.

In previous years, much research has been done on the study of vibration in the automotive industry. Analysis has been conducted on many vehicle suspension models of simulation or experiment utilizing various software to assess the vibrations acting on the suspension which are transmitted to vehicles provide extensive knowledge on evaluation of comfortability [7,8,9,10,15,17,18,19]. However, the mechanical spring and damping within the human body is not taken into consideration which would
result in less accurate data. The study on the vibration of human body models in [2, 3, 4, 14] may provide a better assessment of discomfort as the vibration transmitted in the body is taken into account but the input source of vibration may not be characterizable to real life scenarios as the input does not undergo the mechanical system of the vehicle and seat. Thus, this paper aids in tackling this issue by transmitting a vibration source from the road roughness to the vehicle body, from vehicle body to the seat, and then finally transmitted to mass segments in the human body to assess the discomfort levels experienced, linking both mechanical and biological aspects of the vibration transmission.

2. Methodology
The followings are the considerations made for the simulation:
- The simulation model is solely for the assessment of vertical vibration as a result from the bounce motion of the vehicle for assessment of discomfort.
- The Hand-Transmitted-Vibration (HTV) and Foot-Transmitted-Vibration (FTV) are deemed negligible as it has minimal effects on the ride comfort.
- The seating position of the vehicle passenger is assumed to be erect in an upright position without backrest.

In this section, the processes which were carried out in order to achieve the objectives of this research are explained thoroughly. Firstly, the initial step was the structure and modelling of the simulation model in Simulink to represent the different sub-systems which exist that are previously mentioned. Then, the section provides a better understanding of how the responses from the simulation was extracted to MATLAB and was further processed by utilizing existing HVlab function. Lastly, the method that was used to analyze the vibration transmission using the relevant international standard of ISO 2631 that is available which allows the author to quantify the levels of discomfort and the safety level of vibration exposure experienced by the vehicle passenger.

2.1 Structure and modelling
The vibration model of the 4 subsystems, namely, the passive suspension subsystem, semi-active suspension subsystem, the seat subsystem and the human body subsystem can be used to obtain the free-body-diagrams of the respective systems. Then, Newtonian methods were used to derive the equation of motions which enables the modelling of the simulation model.

2.1.1 Modelling of suspension subsystem. In this subsection, the vibration model for the suspension subsystem and free-body-diagram are shown in figure 1 and figure 2 respectively.

Equations (1-4) were used for the vibration model for sprung mass.
\[ \sum F = ma \]  
\[ \sum F = -F_d - F_{s2} = M_s \ddot{Z}_s \]  
\[ = -C_s(\ddot{Z}_s - \dot{Z}_u) - K_s(\dot{Z}_s - Z_u) = M_s \ddot{Z}_s \]  
\[ \dot{Z}_s = \frac{1}{M_s} [C_s(\dot{Z}_u - \ddot{Z}_s) + K_s(Z_u - Z_s)] \]  

Equations (5-7) were used for the vibration model for unsprung mass.

\[ \sum F = F_d + F_{s2} - F_{s1} = M_u \ddot{Z}_u \]  
\[ = C_s(\ddot{Z}_s - \dot{Z}_u) + K_s(\dot{Z}_s - Z_u) - K_t(Z_u - Z_r) = M_u \ddot{Z}_u \]  
\[ \dot{Z}_u = \frac{1}{M_u} [C_s(\dot{Z}_s - \dot{Z}_u) + K_s(Z_s - Z_u) + K_t(Z_r - Z_u)] \]  

Where,

- $M_s$ = sprung mass
- $M_u$ = unsprung mass
- $K_s$ = spring stiffness
- $K_t$ = tire stiffness
- $C_s$ = damping coefficient
- $Z_i$ = displacement of mass
- $\dot{Z}_i$ = velocity of mass
- $\ddot{Z}_i$ = acceleration of mass

### 2.1.2 Modelling of semi-active suspension system.

In this subsection, the vibration model for the semi-active suspension system and its respective free-body-diagram are shown in figure 3 and figure 4 respectively which aid in determining the equation of motion of the system. Furthermore, the mathematical formulation used to model the variable damper is also explained and implemented. These steps which are taken ultimately result in successfully incorporating the semi-active suspension system into the simulation model.

![Figure 3. Vibration model of semi active suspension subsystem.](image1)

![Figure 4. Free-body-diagram for semi-active suspension subsystem.](image2)

Equations (8-9) were used for the vibration model for sprung mass.

\[ \sum F = -F_d - F_{s2} - B_c = M_s \ddot{Z}_s \]  

(8)
\[5 = -C_s(Z_s - Z_u) - K_s(Z_s - Z_u) - B_c = M_s \ddot{Z}_s \quad (9)\]
\[\dot{Z}_s = \frac{1}{M_s} [C_s(\dot{Z}_u - \dot{Z}_s) + K_s(Z_u - Z_s) - B_c] \quad (10)\]

Equations (11-13) were used for the vibration model for unsprung mass.

\[\sum F = F_d + F_{s2} - F_{s1} + B_c = M_u \ddot{Z}_u \quad (11)\]
\[\dot{Z}_u = \frac{1}{M_u} [C_s(\dot{Z}_s - \dot{Z}_u) + K_s(Z_s - Z_u) + K_t(Z_r - Z_u)] + B_c \quad (13)\]

Where,
\[B_c = \text{Damping force exerted}\]

The MR variable damper in the semi-active suspension system was modelled using dynamic hysteresis behaviors. Dynamic hysteresis was used to define a system whereby its response was dependent on both its past history and also current state. Therefore, it is suitable in describing the application of a variable damper due to its constantly changing environment according to the road input. There are many ways in which dynamic hysteresis could be incorporated by using existing mathematical models. In this research, the mathematical model used was the Bingham model as it is known to be one of the least complex models. The first order differential equation in (14) depicts the general equation used in describing hysteresis using Bingham’s model. It is important to note that the displacement and velocity values of the mathematical model relates to the sprung mass of the vehicle as the variable dampering was changed according to the vertical motion that is experienced by the body of the vehicle, in order to improve the ride comfort. However, the discontinuities which exist within the signum function (sgn) increases complexity when incorporating into numerical simulations. Therefore, equation (15) provides an alternative equation by using an inverse tangent function to express the signum function [13]. Therefore, the variable damper can now be incorporated into the simulation model in Simulink as shown in figure 5.

\[F_{mr0} = F_c \text{sgn } \dot{z} + c_o \dot{z} + K_o \dot{z} + F_o \quad (14)\]
\[F_{mr0} = \frac{2F_c (d.\dot{z})}{\pi} + c_o \dot{z} + K_o \dot{z} + F_o \quad (15)\]

Where,
\[F_{mr0} = B_c\]
\[F_c = \text{frictional force}\]
\[c_o = \text{damping constant}\]
\[K_o = \text{spring stiffness}\]
\[F_o = \text{force constant}\]
\[d = \text{form factor}\]
2.1.3 Modelling of seat subsystem.

In this subsection, the vibration model for the seat subsystem and free-body-diagram are shown in figure 6 and figure 7 respectively.

\[ \sum F = -F_d - F_s = M_o \ddot{Z}_o \]
\[ = -C_o (\ddot{Z}_o - \ddot{Z}_s) - K_o (Z_o - Z_s) = M_o \ddot{Z}_o \]
\[ \ddot{Z}_o = \frac{1}{M_o} [C_o (\ddot{Z}_s - \ddot{Z}_o) + K_o (Z_s - Z_o)] \]

2.1.4 Modelling of human body subsystem.

In this subsection, the vibration model for the human body subsystem and free-body-diagram are shown in figure 8 and figure 9 respectively. The 4-degree-of-freedom (DoF) model used is obtained from Boileau & Rakheja [14] as it was found to have a good accuracy in representing human body [2].
Equations (19-21) were used for the human body vibration model for $M_1$.

$$
\sum F = F_{d2} - F_{s2} - F_{d1} - F_{s1} = M_1 \ddot{Z}_1
$$

$$
= C_2(\ddot{Z}_2 - \ddot{Z}_1) + K_2(Z_2 - Z_1) - C_1(\dot{Z}_1 - \dot{Z}_0) - K_1(Z_1 - Z_0) = M_1 \ddot{Z}_1
$$

$$
\ddot{Z}_1 = \frac{1}{M_1} [C_2(\ddot{Z}_2 - \ddot{Z}_1) + K_2(Z_2 - Z_1) - C_1(\dot{Z}_1 - \dot{Z}_0) - K_1(Z_1 - Z_0)]
$$

Equations (22-24) were used for the human body vibration model for $M_2$.

$$
F_{d3} + F_{s3} - F_{d2} - F_{s2} = M_2 \ddot{Z}_2
$$

$$
C_3(\ddot{Z}_3 - \ddot{Z}_2) + K_3(Z_3 - Z_2) - C_2(\ddot{Z}_2 - \ddot{Z}_1) - K_2(Z_2 - Z_1) = M_2 \ddot{Z}_2
$$

$$
\ddot{Z}_2 = \frac{1}{M_2} [C_3(\ddot{Z}_3 - \ddot{Z}_2) + K_3(Z_3 - Z_2) + C_2(\ddot{Z}_2 - \ddot{Z}_1) + K_2(Z_2 - Z_1)]
$$

Equations (25-27) were used for the human body vibration model for $M_3$.

$$
F_{d4} + F_{s4} - F_{d3} - F_{s3} = M_3 \ddot{Z}_3
$$

$$
C_4(\ddot{Z}_4 - \ddot{Z}_3) + K_4(Z_4 - Z_3) - C_3(\ddot{Z}_3 - \ddot{Z}_2) - K_3(Z_3 - Z_2) = M_3 \ddot{Z}_3
$$

$$
\ddot{Z}_3 = \frac{1}{M_3} [C_4(\ddot{Z}_4 - \ddot{Z}_3) + K_4(Z_4 - Z_3) + C_3(\ddot{Z}_3 - \ddot{Z}_2) + K_3(Z_2 - Z_3)]
$$

Equations (28-30) were used for the human body vibration model for $M_4$.

$$
-F_{d4} - F_{s4} = M_4 \ddot{Z}_4
$$

$$
-C_4(\ddot{Z}_4 - \ddot{Z}_3) - K_4(Z_4 - Z_3) = M_4 \ddot{Z}_4
$$

$$
\ddot{Z}_4 = \frac{1}{M_4} [C_4(\ddot{Z}_3 - \ddot{Z}_4) + K_4(Z_3 - Z_4)]
$$

Where,

$M_1$ = mass of thigh and pelvis

$M_2$ = mass of lower torso

$M_3$ = mass of chest and upper torso

$M_4$ = mass of head and neck

Figure 10, 11 and 12 depicts the passive suspension subsystem, seat subsystem and human body subsystem respectively whereas figure 13 illustrates the final simulation model with passive suspension system. The displacement of sprung mass was used for the input of the seat subsystem and the displacement and velocity of the seat subsystem was used as the input for the human body subsystem.
Figure 10. Passive suspension subsystem in Simulink.

Figure 11. Seat subsystem in Simulink.

Figure 12. Human body subsystem in Simulink.
2.2 Vibration transmission analysis method

ISO 2631 standard provides a guideline to the effect whole-body-vibrations in terms of comfort, activities and health. Very often in real life cases, the WBV which passengers are subjected to occur with complex and random vibration signals at various axes. Hence, ISO 2631 standard specifies the procedures which could be done in order for these varying signals to be specified into a distinct value which can further be used for qualitative indication to assess the safety level and expected discomfort experienced. In order to obtain a single value from the large number of random data retrieved, the root-mean-square (rms) method is applied according to equation 31 below [15]. The multi-frequency response is subjected to a frequency weighting depending on its respective axis according to table 1. The frequency weightings exist to ensure that the effects of passenger discomfort would be highly substantial and the vibration at given axis has a realistic comparative influence on the overall vibration as these weightings can manipulate the contents of frequency from the raw data obtained [16].

$$a_{wor} = \frac{1}{T} \int_0^T a_w(t) \, dt$$  \hspace{1cm} (31)

| Location   | Health Weighting | Comfort Weighting |
|------------|-----------------|------------------|
| x seat     | 1.4Wd           | Wd               |
| y seat     | 1.4Wd           | Wd               |
| z seat     | Wk              | Wk               |
| x back     | 0.8Wc           | 0.8Wc            |
| y back     | Nil             | 0.5Wd            |
| z back     | Nil             | 0.4Wd            |
| x feet     | Nil             | 0.25Wk           |
| y feet     | Nil             | 0.25Wk           |
| z feet     | Nil             | 0.4Wk            |

Table 1. ISO 2631 Multiplication Factors and Frequency Weightings [15].
The frequency weighted \( \textit{rms} \) acceleration values were compared to the Health Guidance Caution Zone as shown in figure 14. This is to ensure that the well-being of users will not be affected when operating the vehicle.

![Figure 14. Graph of health guidance caution zone [15].](image)

In terms of comfort, the quantifiable discomfort scale as shown in table 2 was used as a reference to quantify the discomfort levels that are experienced by the user when experiencing vertical-WBV. The frequency weighted \( \textit{rms} \) acceleration response transmitted from the driving point which is the seat to the specific mass segments, provides a qualitative response to the degree of discomfort experienced at different mass segments by the user according to the discomfort scale.

**Table 2.** Discomfort scale according to frequency weighted acceleration \( \textit{rms} \) [15].

| \( a_{w\textit{ rms}} \) (m/s\(^2\)) | Discomfort rating                 |
|-------------------------------------|-----------------------------------|
| Less than 0.315 m/s\(^2\)          | not uncomfortable                 |
| 0.315 m/s\(^2\) to 0.63            | a little uncomfortable            |
| 0.5 to 1                            | fairly uncomfortable              |
| 0.8 to 1.6                          | Uncomfortable                     |
| 1.25 to 2.5                         | very uncomfortable                |
| Greater than 2                      | extremely uncomfortable           |

2.3 Utilization of simulation responses

The raw numerical simulation data was extracted into MATLAB to tabulate in Microsoft Excel with the first column representing the time domain \( x \)-axis and the second column representing the acceleration responses in the \( y \)-axis to prepare the data for further signal processing. The Excel sheet was utilized in
the HVlab human response to vibration toolbox MATLAB function to obtain the \textit{rms} values of acceleration data obtained from the ten seconds simulation time. Once the \textit{rms} of acceleration was obtained, the function was then utilized once again to apply the frequency weightings as stated by ISO 2631 shown in table 1 above. The corresponding commands used to execute the operation of the HVlab toolbox is shown in figure 15.

```matlab
1 - T = table(out.head.Time, out.head.Data)
2 - writetable(T, 'FYP_1_head.xls')
3 - data = xlsread(“FYP_1_head.xls”);
4 - x = data(:,1);
5 - y = data(:,2);
6 - hvlab
7 - [ds] = hvcreate(y, x, ’accl’, ’m/s2’, ’s’)
8 - [rms] = hvstats(ds)
9 - [weight] = hvweight(ds, ’Wk’)
10 - [rms] = hvstats(weight)
```

Figure 15. Commands to execute HVlab function in MATLAB.

### 3. Results & Discussion
Before carrying out the simulation, firstly the simulation model underwent validation process in order to gauge its accuracy as well as the ability to represent real life situations. The results which were obtained from the validation process is then used as the benchmark for the improvement of the passive suspension system. Modification process to the suspension system was conducted and the simulation was carried out again to analyze the changes in responses to provide validated recommendations for future production.

#### 3.1 Validation process of simulation model
The simulation model which was validated with Kim et.al [17], Hegazy and Sharaf [18] and Thite [19] for step input, sinusoidal input and random input respectively. The process is vital to ensure that the simulation model developed is able to predict relatively accurate responses and represent an actual passenger seated in a vehicle under various conditions.

##### 3.1.1 Step input.
The base excitation of the simulation model was set up as a step input. In this scenario, the input of the system represents the elevation of the passenger vehicle as it travels from lower to higher ground or even passing through an obstacle or bump while travelling. In this subsection, the parameters of the simulation model were set according to the parameters identified in [17] for validation purposes and are listed in the table 3.

Table 3. Parameter values of simulation model.

| Parameters                  | Value  |
|-----------------------------|--------|
| Sprung mass, \( M_s \)     | 660.65kg |
| Unsprung mass, \( M_u \)   | 133.14kg |
| Spring stiffness, \( k_s \) | 36542 N/m |
| Damping coefficient, \( c_s \) | 2167.8 Ns/m |
| Tire stiffness, \( k_t \)   | 332250 N/m |
Figure 16-23 illustrates the signal of the step input with an amplitude of 0.025m and the corresponding displacement values of both sprung and unsprung masses as well as the validation response for step input from [17].

Figure 16. Response of sprung mass displacement.

Figure 17. Displacement response of sprung mass in validation reference [17].

Figure 18. Response of sprung mass velocity.

Figure 19. Velocity response of sprung mass in validation reference [17].
Figure 20. Response of unsprung mass displacement.

Figure 21. Displacement response of unsprung mass in validation reference [17].

Figure 22. Response of unsprung mass velocity

Figure 23. Velocity response of sprung mass in validation reference [17].

The displacement and velocity peak responses for the sprung mass that were obtained are 0.0402 m and 0.1741 m/s respectively whereas unsprung mass having peak response of 0.0373 m and 0.957 m/s respectively. By comparing the displacement and velocity responses of both the sprung mass and unsprung mass, a good correlation can be seen between both the simulation responses and the reference responses in terms of peak response and response patterns. It is important to note that the settling time of these responses do not seem to match due to the variation in simulation time of the responses. However, the settling time of the simulation is 5.6 seconds. This value is utilized in the coming section. Thus, the simulation model is considered validated for the step input disturbance.

3.1.2 Sine input.

In this condition, the excitation of base input of the simulation model was set as a sinusoidal input. This condition was modelled to represent the vehicle travelling on a road at various speeds. Since the simulation model was designed to measure vibration at a single axis, in order to take into consideration the velocity of the car, the different input velocities were subbed into equation (32) which manipulates the frequency of the base excitation. The road roughness was assumed to be similar to a harmonic sinusoidal waveform. In this subsection, the parameters of the simulation model were set according to the parameters identified in [18] for validation purposes and are listed in the table 4.

$$\omega = \frac{2\pi v}{L}$$  

Where,

$\omega$ = base excitation frequency (rad/s)
$v$ = velocity of vehicle (m/s)
$L$ = length of sine wave (m)

| Parameters         | Value  |
|--------------------|--------|
| Sprung mass, $M_s$| 240kg  |
Unsprung mass, $M_u$ 43kg  
Spring stiffness, $k_s$ 12000 N/m  
Damping coefficient, $c_s$ 1500 Ns/m  
Tire stiffness, $k_t$ 200000 N/m

The simulation was conducted at two different velocities of moving vehicle having base excitation of 8.67 rad/s and also 11.624 rad/s with an amplitude of 0.025 m. The responses of the simulation results were then used to compare with the theoretical and experimental results as provided in [18] as shown in table 5.

| Excitation Frequency (rad/s) | Parameter  | Simulation ($a_{rms}$) | Theoretical ($a_{rms}$) | Experimental ($a_{rms}$) |
|-----------------------------|------------|------------------------|------------------------|------------------------|
| 8.671(1.38hz)               | Sprung mass| 1.667                  | 1.49                   | 1.37                   |
|                             | Unsprung mass | 2.213              | 1.5                    | 2.15                   |
| 11.624(1.85hz)              | Sprung mass | 1.922                  | 1.29                   | 1.19                   |
|                             | Unsprung mass | 2.972              | 2.31                   | 1.71                   |

Based on the results obtained, the comparison illustrates that there is a good correlation between the response of the sprung and unsprung masses in terms of $rms$ acceleration with the maximum deviation of 42.5% between the simulation response and the experimental response. This may be due to the simulation being modelled as an ideal system where the friction forces between the springs and dampers are deemed negligible whereas in the experimental test, these forces are naturally taken into consideration. However, in general, most of the responses seems to provide a relatively good correlation. Hence, it can be said that the simulation model with sinusoidal wave input source was validated. The figure A1-A8 in the appendix provides a useful graphical illustration of the correlation between the different acceleration responses in time domain.

### 3.1.3 Resonant frequency of simulation model.

In this last validation subsection, the frequency response of the passive suspension system was used for validation purposes. As seen in figure 13, two linear analysis points were placed on the simulation model. There was an input perturbation point that was located at the second order derivation of the input signal which represented the acceleration of base excitation and an output measurement point located at the acceleration response of the sprung mass. These linear analysis points function to set the boundaries and desired section when linearizing the model before conducting the frequency response estimation of the model. The input perturbation point represents the location at which an input signal was injected to the simulation model whereas the output measurement was the location at which the response was measured.

The frequency response function of the body acceleration due to the random input is illustrated in figure 24. The importance of the random signal used enables the author to validify the system when the response is in frequency domain. When comparing with the validation reference, it can be seen that the frequency response estimation of the simulation is aligned with the vehicle body response of the validation reference [19] whereby 2 peak responses can be seen in the range of 1-2 Hz and the other at 13-15 Hz as shown in figure 25. The first peak response due to the sprung mass of vehicle whereas the
second peak response occurs due to the unsprung mass of the vehicle. These peak responses take place when the frequency of vibration and natural frequency of the specific mass segment are close which causes the resonance of the system. Therefore, this is a successful validation process of the simulation model in frequency domain.

Figure 24. Frequency response estimation of simulation model for vehicle body.

Figure 25. Frequency response of reference paper for vehicle body [19].

3.2 Modification suggestion to improve comfortability of ride.

In this subsection, modifications are to be conducted within the parameters of the suspension system to observe its effects on the acceleration responses which the user experiences at different mass segments at a constant velocity of 60 km/h with 0.025 m amplitude at 10 m wavelength. The parameters of the seat suspension system and the 4-DoF human body model were extracted from references [20] and [14] respectively. The values of these parameters are included in table 6. The suspension parameters which are used in this subsection will be according to subsection 3.1.2. Then, the responses of the vehicle passenger from initial suspension system was as a benchmark when comparing to the responses of the vehicle passenger seated in the modified suspension system to observe the individual changes in different parameters. Modifications was made to the mass ratio of the vehicle \((Mu/Ms)\), stiffness ratio of the vehicle \((Kt/Ks)\) and also the damping coefficient of the shock absorber. Each modification was conducted at +25%, +50%, -25% and -50% of initial ratio in order to analyse the effects of these modifications towards human discomfort. Although the research aims to focus on the ride comfortability of passenger, it is important to note that these modifications should also take into consideration the road
holding ability of vehicle as it is also a key aspect in the optimal functioning of a suspension system. According to Shirah [21], the tire displacement is an indication of the road holding ability of a suspension system. Therefore, apart from the acceleration responses, the tire displacement from each modification was also tabulated and assessed.

**Table 6.** Parameter values of simulation model.

| Parameters                                | Value     |
|-------------------------------------------|-----------|
| Mass of thigh and pelvis, $M_1$           | 12.733 kg |
| Mass of lower torso, $M_2$                | 8.588 kg  |
| Mass of chest and upper torso, $M_3$      | 28.386 kg |
| Mass of head and neck, $M_4$              | 5.29 kg   |
| Thigh and pelvis spring stiffness, $k_1$  | 90000 N/m |
| Thigh and pelvis damping coefficient, $c_1$ | 2064 Ns/m |
| Lower torso spring stiffness, $k_2$       | 162800 N/m|
| Lower torso damping coefficient, $c_2$    | 4548 Ns/m |
| Chest and upper torso spring stiffness, $k_3$ | 183000 N/m |
| Chest and upper torso damping coefficient, $c_3$ | 4750 Ns/m |
| Head and neck spring stiffness, $k_4$     | 310000 N/m|
| Head and neck damping coefficient, $c_4$  | 400 Ns/m  |
| Mass of seat, $M_0$                       | 13.545 kg |
| Seat spring stiffness, $k_0$              | 72.3 N/m  |
| Seat damping coefficient, $c_o$           | 357 Ns/m  |

3.2.1 Modification of ratio between sprung and unsprung masses.

In this section, the mass ratio between both unsprung and sprung mass were manipulated while the other parameters were kept constant in order to visualize the effects of the modification on the human body model. Results of the modifications are shown in table 7 and plotted in figure 26 and 27 for better visualization.

**Table 7.** Acceleration response due to the modification of mass ratio.

| Modification | Sprung Mass (kg) | Unsprung Mass (kg) | Ratio (M_u/M_s) | $M_1$ acceleration (m/s^2) | $M_2$ acceleration (m/s^2) | $M_3$ acceleration (m/s^2) | $M_4$ acceleration (m/s^2) | Tire displacement (m) |
|--------------|------------------|--------------------|----------------|----------------------------|----------------------------|----------------------------|----------------------------|-----------------------|
| -50%         | 240              | 21.48              | 0.09           | 0.9206                     | 0.9189                     | 0.9023                     | 0.8775                     | 0.01802               |
| -25%         | 240              | 32.22              | 0.134          | 0.9263                     | 0.9246                     | 0.9079                     | 0.8829                     | 0.01821               |
| 1 (initial)  | 240              | 43                 | 0.179          | 0.9322                     | 0.9304                     | 0.9137                     | 0.8883                     | 0.01832               |
| +25%         | 240              | 53.7               | 0.224          | 0.9381                     | 0.9362                     | 0.9195                     | 0.8939                     | 0.01843               |
3.2.2 Modification of ratio between suspension stiffness and tyre stiffness.

In this section, the stiffness ratio between both tire and spring suspension were manipulated while the other parameters were kept constant in order to visualize the effects on the human body model. Results of the modifications are shown in table 8 and plotted in figure 28 and 29 for better visualization.

### Table 8. Acceleration response due to modification of stiffness ratio.

| Modification | Spring stiffness (N/m) | Tire stiffness (N/m) | Ratio (kt/ks) | $M_1$ acceleration $rms$ (m/s$^2$) | $M_2$ acceleration $rms$ (m/s$^2$) | $M_3$ acceleration $rms$ (m/s$^2$) | $M_4$ acceleration $rms$ (m/s$^2$) | Tire displacement (m) |
|--------------|------------------------|----------------------|--------------|-----------------------------------|-----------------------------------|-----------------------------------|-----------------------------------|----------------------|
| -50%         | 23995.2                | 200000               | 8.335        | 1.8957                            | 1.8921                            | 1.8586                            | 1.8071                            | 0.01828              |
| -25%         | 15996.8                | 200000               | 12.503       | 1.1952                            | 1.1929                            | 1.1719                            | 1.1392                            | 0.0183               |
| 1 (initial)  | 12000                  | 200000               | 16.67        | 0.9322                            | 0.9304                            | 0.9137                            | 0.8883                            | 0.01832              |
| +25%         | 9598.1                 | 200000               | 20.838       | 0.8071                            | 0.8055                            | 0.7910                            | 0.7690                            | 0.01835              |

3.2.3 Modification of ratio between suspension stiffness and tyre stiffness.

![Figure 26. Graph of acceleration response against mass ratio.](image1.png)

![Figure 27. Graph of tire displacement against mass ratio.](image2.png)

![Figure 28. Graph of acceleration response against stiffness ratio.](image3.png)

![Figure 29. Graph of tire displacement against stiffness ratio.](image4.png)
In this section, the damping coefficient was modified and the other parameters were kept constant in order to analyse its effect on the human body model. Results of the modifications are shown in Table 9 and plotted in figure 30 and 31 for better visualization.

**Table 9.** Acceleration response due to change in damping coefficient.

| Modification | Damping coefficient (Ns/m) | $M_4$ acceleration rms (m/s²) | $M_3$ acceleration rms (m/s²) | $M_2$ acceleration rms (m/s²) | $M_1$ acceleration rms (m/s²) | Tire displacement (m) |
|--------------|-----------------------------|-------------------------------|-------------------------------|-------------------------------|-------------------------------|-----------------------|
| -50%         | 750                         | 0.8232                        | 0.8217                        | 0.8075                        | 0.7851                        | 0.01705               |
| -25%         | 1125                        | 0.8801                        | 0.8784                        | 0.8630                        | 0.8390                        | 0.01771               |
| 1 (initial)  | 1500                        | 0.9322                        | 0.9304                        | 0.9137                        | 0.8883                        | 0.01832               |
| +25%         | 1875                        | 0.9751                        | 0.9731                        | 0.9559                        | 0.9288                        | 0.01882               |

**Figure 30.** Graph of acceleration response against damping coefficient.

**Figure 31.** Graph of tire displacement against damping coefficient.

### 3.2.4 Evaluation of modifications.

The tabulation of data according to the modifications which were made to the parameters of the suspension system enables the assessment of the level of discomfort experienced by the vehicle passenger at respective mass segments based on Table 7-9. The results show that there is an increase in $rms$ acceleration responses of the human body when the mass ratio is higher. Not only that, it can be seen that a higher mass ratio results in larger displacement of the vehicle tires. Therefore, with increasing mass ratio, road holding of suspension system decreases. The main cause of this phenomenon is due to the increase of inertial forces within the unsprung mass as it is increased. The higher forces require more workload from the suspension system in order to keep the vehicle tires in contact with the road. Since the other parameters are kept constant and workload from suspension system is not changed, therefore road holding ability of vehicle is negatively affected with the increment of mass ratio.

The results obtain shows that the higher the stiffness ratio, the lower the acceleration responses in the human body resulting in better ride comfortability. The tire stiffness is not modified as this parameter is not able to be realistically manipulated unless there is a change in type of tire or pressure in tire. With the tire stiffness being kept constant, a high stiffness ratio would mean that a softer spring is used in the suspension system, hence, representing a ‘soft’ suspension system, prioritizing the ride comfortability. However, the tire displacement can be seen to increase with a softer suspension system. Hence, the results prove that the improvement of vibration attenuation in the suspension system would be at the expense of the road holding ability of the vehicle and vice versa. This highlights the main
disadvantage of a rigid suspension system whereby parameters cannot be changed according to the ride conditions.

The results show that a better $rms$ acceleration response of human body can be seen with lower damping coefficient. Similarly, lower damping coefficient also shows lower tire displacement, indicating good road holding ability. However, it is important to note that these results would differ in pattern depending on the frequency of excitation experienced. Once again, highlighting the disadvantage of a passive suspension system as it is unable function optimally to cater to the varying speed when a vehicle is in operation. The damping coefficient in a suspension system exists within the shock absorber. The importance of the shock absorber is that it determines the rate at which the spring is moving. Therefore, higher damping coefficients would prevent the proper functioning of spring, therefore causing an overdamped scenario. When the spring is not able to return to its original state in time, jerks will take place. These jerks cause the acceleration within the passenger to be higher, hence, causing more discomfort. The overdamped scenario can cause the wheels of the vehicle to take a longer time to return to its initial state after compression, the possibility of the tires losing contact with the road is highly likely, hence, negatively affecting the road holding ability. However, it is also important to note that if the damper coefficient is too low, more oscillations of the sprung mass may occur due to inability of shock absorber to effectively dissipate the energy. This scenario is called under-damped. Therefore, further study can be conducted to obtain the optimal values of damping coefficient to achieve critical damping of the vehicle.

3.3 Incorporation of semi active suspension system
Apart from the modifications which can be implemented into existing suspension system to improve ride comfortability, a semi-active suspension system is incorporated into the simulation model using Bingham’s model to represent the variable MR damper. The simulation responses obtained are then compared to the responses of the existing passive suspension system in order to gauge the significance of the semi-active suspension system in improving ride comfortability of the passenger. The parameters of the simulation model used are similar as in section 3.1.1 and 3.1.2 and is simulated using both step input and sinusoidal input respectively to conduct the comparison. The parameters which are used for the semi-active suspension are provided in table 10 below.

| Parameters               | Value  |
|--------------------------|--------|
| Frictional force, $F_c$  | 210 N  |
| Damping constant, $c_o$ | 650 Ns/m |
| Spring stiffness, $K_o$ | 300 N/m |
| Force constant, $F_o$   | -7.5 N |
| Form factor, $d$        | 20     |

3.3.1 Response and comparison of semi-active suspension system to step input.
The simulation model subjected to a step input similar to section 3.1.1 and the results are shown figure 32 and 33. Table 11 is tabulated to make a comparison between response when passive suspension system is used compared to semi-active suspension system.
Figure 32. Response of sprung mass displacement.

Figure 33. Response of sprung mass velocity.

Table 11. Comparison of response using step input.

| Suspension system | Sprung mass displacement, $Z_u$ (m) | Sprung mass velocity, $Z_u$̇ (m/s) | Settling time (s) |
|-------------------|------------------------------------|----------------------------------|------------------|
| Passive           | 0.0402                             | 0.1741                           | 5.6              |
| Semi-active       | 0.025 (step)                       | 0.1448                           | 1.1              |

Evidently, it can be seen that there is significant improvement for all comparisons. The peak displacement for semi-active system indicates the optimal functioning of the system as its value is the same as the input with no oscillations or overshoot and undershoot. A 37.8% improvement can be seen for displacement of sprung mass. Noticeable differences can be seen when comparing the velocity and settling time of both the systems with an improvement of 16.83% and 80.36% respectively. The changes in responses can ultimately improve the ride comfortability of vehicle passenger when moving across an elevation in real life conditions.

3.3.2 Response and comparison of semi-active suspension system to sinusoidal input.

In this section, the simulation model is subjected to parameters and a sinusoidal input similar to section 3.1.2 and the results are tabulated in table 12 below.

Table 12. Comparison for sinusoidal input.

| Suspension system | $M_1$ acceleration $rms$ (m/s²) | $M_2$ acceleration $rms$ (m/s²) | $M_3$ acceleration $rms$ (m/s²) | $M_4$ acceleration $rms$ (m/s²) | Tire displacement (m) |
|-------------------|---------------------------------|---------------------------------|---------------------------------|---------------------------------|-----------------------|
| Passive           | 0.9322                          | 0.9304                          | 0.9137                          | 0.8883                          | 0.01832               |
| Semi-active       | 0.0032                          | 0.0033                          | 0.0034                          | 0.0035                          | 0.01802               |

Using the results in table 12, an evaluation can be done by utilizing ISO 2631 in determining the discomfort levels of the passenger when seated in vehicle with different suspension systems. Based on table 2, the frequency weighted $rms$ acceleration values at each mass segment are used to obtain quantifiable discomfort levels experienced by the vehicle passenger. The ratings of discomfort is shown in table 13. The frequency weighted $rms$ acceleration values of the passengers in the vehicle with passive suspension system shows that the passengers experience a fairly uncomfortable and uncomfortable
discomfort rating at all mass regions whereas the passenger seated in the vehicle with semi-active suspension system has no discomfort for all mass regions.

Table 13. Discomfort rating of mass segments.

| Mass Segment | Rating of discomfort |
|--------------|----------------------|
|              | Passive              | Semi-active          |
|              | No                   | No                   |
|              | A little             | A little             |
|              | Fairly               | Fairly               |
|              | Uncomfortable        | Uncomfortable        |
|              | Very                 | Very                 |
|              | Extreme              | Extreme              |
| M1           | ✓                    | ✓                    |
| M2           | ✓                    | ✓                    |
| M3           | ✓                    | ✓                    |
| M4           | ✓                    | ✓                    |

Similarly, the frequency weighted $rms$ acceleration values of each mass segment were also used to intersect with the lower boundary of the bolded dashed line in figure 14 to obtain the permissible exposure time in the $x$-axis of the graph according to ISO 2631. This is because the area below the lower boundary is the area of no risks, the area between the upper and lower boundary is the caution zone, and the area above the upper boundary is the area which is likely for risks which could damage health of passenger. Therefore, by intersecting with lower boundary of the graph, the maximum permissible time before passenger’s health is in the caution zone can be obtained. The results show that the allowable exposure limit can be increased approximately from 2.2-3 hours for passive suspension to more than 24 hours for the semi-active suspension system which is well above any reasonable time to be reached in a continuous journey. The results prove the significance of the variable damper in the semi-active suspension system in reducing discomfort and increasing safety of vehicle passenger.

4. Conclusion
In conclusion, the paper provides a clear methodology on how the simulation model is developed using Newtonian methods to derive equation of motions. The simulation model is then validated using 3 different input sources to the simulation model, namely, step, sine wave and random. All the validation process shows a clear correlation between both simulation data as well as the validation data. The parametric analysis of the passive suspension system has shown that increase in mass ratio, a decrease in stiffness ratio and an increase in damping coefficient would increase $rms$ vibration response, hence, reducing ride comfortability. The modifications of the suspension system has aided the achievement of the second objective of this paper. However, it is important to consider other aspects to which suspension system has significant importance to. Furthermore, the incorporation of the semi-active suspension system to the simulation model has shown an improvement when the model is excited with both step and sinusoidal inputs. The decrease peak displacement and velocity of the sprung mass, and also settling time can be seen when comparing both suspension systems. Furthermore, the semi-active suspension system was proven to be successful in reducing the frequency weighted acceleration $rms$ of the human
body at all mass segments and is deemed not uncomfortable and much safer when assessing with ISO 2631 standard. A simulation model with higher DoF can be proposed for future work to consider other axes of vibration which will help the assessment of the suspension design to consider other aspects such as vehicle handling.

5. Appendices

![Simulation response of sprung mass acceleration at 8.67 rad/s.](image1)

**Figure A1.** Simulation response of sprung mass acceleration at 8.67 rad/s.

![Theoretical and experimental response of sprung mass acceleration at 8.67 rad/s.](image2)

**Figure A2.** Theoretical and experimental response of sprung mass acceleration at 8.67 rad/s [18].
Figure A3. Simulation response of sprung mass acceleration at 11.624 rad/s.

Figure A4. Theoretical and experimental response of sprung mass acceleration at 11.624 rad/s [18].
Figure A5. Simulation response of unsprung mass acceleration at 8.67 rad/s.

Figure A6. Theoretical and experimental response of unsprung mass acceleration at 8.67 rad/s [18].
Figure A7. Simulation response of unsprung mass acceleration at 11.624 rad/s.

Figure A8. Theoretical and experimental response of unsprung mass acceleration at 11.624 rad/s [18].

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