Parametric Studies on Tramcar Suspension System

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Abstract: Suspension system plays an important role in the performance of a vehicle, especially the handling and ride comfort. The role of suspension parameter, particularly spring stiffness, in relation to ride quality is being analysed in this paper. This study focused on the suspension system of a non-commercial transport for recreational purposes designed by the Mechanical Engineering Faculty of Universiti Teknologi Malaysia, which is commonly known as 'tramcar'. For the purpose of the analysis, a full car model for the tramcar suspension system was developed. The simulation on the model was performed using MATLAB Simulink software. The spring stiffness value was varied in the simulation, and the suspension response was observed. From the suspension parameter analysis, it was concluded that the ride comfort of the tramcar can be improved to an optimum level by having the lowest practical spring stiffness value. Lower suspension spring stiffness was shown to provide better ride comfort in term of lower acceleration, pitch rate and roll rate responses. However, the spring stiffness should not produce response frequencies lower than 1Hz in avoiding sensations assimilated to motion sickness.

Introduction

A driver judges his vehicle based on subjective aspects. Vehicle dynamic characteristics including ride and handling have a major impact on this evaluation. For this reason, vehicle manufacturers have grown investments in order to improve this vehicle dynamic behaviour [1]. The perceived comfort level and ride stability of a vehicle are the two of the most important factors in a vehicle's subjective evaluation. There are many aspects of a vehicle that influence these two properties, most importantly the primary suspension components, which isolate the frame of the vehicle from the axle and the wheel assemblies.

In the design of a conventional suspension system there is a trade off between the two quantities of ride comfort and vehicle handling. A conventional suspension may be optimised for handling performance, or it may be optimised to isolate the occupants from road disturbances, but it cannot excel at both. In practice, the performance of conventional vehicle suspensions is a compromise between ride and handling [2, 3]. If a suspension is designed to optimise the handling and stability of the vehicle, the passenger often perceives the ride to be rough and uncomfortable. On the other hand, if the suspension is designed to optimise the comfort level, the vehicle will be comfortable, but may not be too stable during vehicle manoeuvres.

The quality referred to as "ride comfort" is affected by 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. If the vehicle is noisy, if it rolls excessively in turns, or lurches and pitches during acceleration and braking, or if the body produces a booming resonance, the passengers will experience an "uncomfortable ride." The intention of this study is to perform parametric analysis on a suspension system, analysing the relationship between the parameters of the suspension system with the factors affecting ride comfort that was mentioned above.

This study is focused on the suspension system of a non-commercial transport for recreational purposes designed by the Mechanical Engineering Faculty of Universiti Teknologi Malaysia (UTM), which is commonly known as 'tramcar'. The vibration level the passengers of the tramcar are exposed to would be the focal point. Other factors affecting ride comfort including body pitch and roll were also looked into.

Suspension Parameters

The two suspension parameters, which are the suspension spring stiffness and suspension damping, have a direct impact on the suspension response due to road disturbances. A suspension spring allows the wheel to move up and down with the road surface without causing similar movement of the body. A different spring rate would give different response for a given road profile. Thus, the spring rate of the suspension has a direct effect on the ride comfort. A damper is the main energy dissipater in a vehicle suspension. It dampens the vibration to provide a good compromise between low sprung mass acceleration for a good ride and adequate control of the unsprung mass for good road holding.

As mentioned in the introduction, the suspension performance is a compromise between ride and handling. While soft suspension springs enhance vehicle ride quality, hard suspension springs are suitable for improving vehicle handling and control performance [2]. In relation with the role of vehicle suspensions, different types of vehicles would have different primary requirements. Table 1 shows the suspension parameter values used for various types of vehicles, as found out from various literatures.

Table 1: Suspension parameters for various types of vehicles

Vehicle Types	Spring Rate (N/m)	Damping Rate (N/(m/s))
IMSA GTP [4]	460,630 (front) 538,620 (rear)	4170
Indy Light Lola [4]	397,968 (front) 582,116 (rear)	4170
Race car [5]	302,000 (front) 245,000 (rear)	1600 (rebound) 50 (bump)
Mini Baja [6]	65,000	300
Passenger car [7]	17,658	1950
Passenger car [2]	20,225	1950
Proton Waja 1.6 [8]	16,000	2133 (extension) 579 (compression)

From a general observation on Table 1, it can be noted that the suspension spring rate for race cars are significantly higher than that of passenger cars. For race cars, where vehicle handling and control performance are of paramount importance, hard suspension springs are used. Ride comfort is of least importance for race cars. On the other hand, for passenger vehicles, where ride quality is obviously an important factor, suspension springs with lower spring stiffness are used.

The effects of spring stiffness value on the ride quality can be understood from the understanding on the connections between the spring characteristics related to spring stiffness and the suspension system output parameters related to ride quality. The spring characteristics that are related to spring stiffness are the force-displacement characteristics of the spring. The suspension system output parameters related to ride quality are the sprung mass displacement and acceleration responses.

The force-displacement characteristics of the spring have a direct effect on the level of disturbances transmitted from the road profile to the sprung mass, and thus the passengers. When a tyre hit a bump, the unsprung mass will follow the road profile and in doing so, exerts a certain force and displacement to the suspension spring. The spring response depends on its force-displacement characteristic, or in other words, its stiffness.

For a certain force being transmitted when a tyre hit a bump, spring with lower stiffness will transmit lower displacement to the sprung mass, while spring with higher stiffness will transmit higher displacement to the sprung mass. For a certain spring compression displacement when a tyre hit a bump, spring with lower stiffness will cause lower acceleration of the sprung mass, while spring with higher stiffness will cause higher acceleration of the sprung mass.

From the above discussion on the effects of suspension spring stiffness value on the ride quality, it can be said that lower suspension spring stiffness will give a better ride quality in term of lower sprung mass displacement and acceleration responses. On the other hand, a higher suspension spring stiffness will give a higher displacement and acceleration responses, which is associated with poor ride quality. In other words, to attain a better ride quality, a lower suspension spring stiffness value should be used.

Vehicle Model

In the process of analysing a system, two tasks must be performed: modelling the system, and solving for the model's response. The combination of these steps is referred to as system analysis [9]. In this study, the ride comfort of the existing tramcar is being studied in term of the tramcar suspension system response to disturbances. In addition to vertical acceleration response, other parameters that are being analysed are the pitch and roll angular velocity of the sprung mass.

Fig. 1 illustrates the model: a full car model with seven degree of freedom; the vehicle body is represented by a three degree of freedom mass (vertical position, pitch rotation and roll rotation are considered); $M_{u,fr}$, $M_{u,fl,}$ $M_{u,rr}$, and $M_{u,rl}$ are defined as front right, front left, rear right, and rear left unsprung masses.

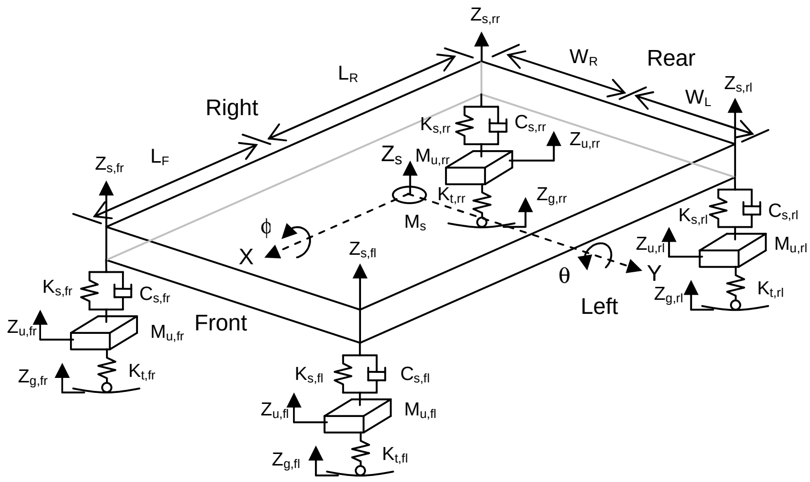

Figure 1: Full vehicle model

The force balance on the sprung mass is obtained as;

$$F_{s,fr} + F_{s,fl} + F_{s,rr} + F_{s,rl} = M_s \ddot{Z}_s \tag{1}$$

where

$F_{s,fr}$, $F_{s,fl}$, $F_{s,rr}$, and $F_{s,rl}$ are the front right, front left, rear right and rear left suspension force

M_s is the sprung mass

$\ddot{Z}_s$ is the sprung mass vertical acceleration at the centre of gravity

The moment balance on the sprung mass, summing the moments counterclockwise about the CG of the vehicle, is obtained as;

$$I_{yy}\ddot{\theta} = L_r\left(F_{s,rr} + F_{s,rl}\right) - L_f\left(F_{s,fr} + F_{s,fl}\right) \tag{2}$$

$$I_{xx}\ddot{\phi} = W_l\left(F_{s,fl} + F_{s,rl}\right) - W_r\left(F_{s,fr} + F_{s,rr}\right) \tag{3}$$

where

θ and ϕ are the pitch and roll angles at body centre of gravity

I_{xx} and I_{yy} are the roll and pitch axis inertias

L_f and L_r are the distances from the CG of sprung mass to the front and rear suspension respectively L_r is the distance between the rear and the C.G of sprung mass

W_r and W_l are the distances from the C.G of sprung mass to the right and left suspension respectively

The suspension forces acting on each corner of the sprung mass are defined as the sum of the forces produced by the suspension components, namely the spring and the damper.

$$F_{s,fr} = K_{s,fr}\left(Z_{u,fr} - Z_{s,fr}\right) + C_{s,fr}\left(\dot{Z}_{u,fr} - \dot{Z}_{s,fr}\right) \tag{4}$$

$$F_{s,fl} = K_{s,fl}\left(Z_{u,fl} - Z_{s,fl}\right) + C_{s,fl}\left(\dot{Z}_{u,fl} - \dot{Z}_{s,fl}\right) \tag{5}$$

$$F_{s,rr} = K_{s,rr}\left(Z_{u,rr} - Z_{s,rr}\right) + C_{s,rr}\left(\dot{Z}_{u,rr} - \dot{Z}_{s,rr}\right) \tag{6}$$

$$F_{s,rl} = K_{s,rl}\left(Z_{u,rl} - Z_{s,rl}\right) + C_{s,rl}\left(\dot{Z}_{u,rl} - \dot{Z}_{s,rl}\right) \tag{7}$$

where

$K_{s,fr}$, $K_{s,fl}$, $K_{s,rr}$ and $K_{s,rl}$ are the front right, front left, rear right and rear left suspension spring stiffnesses

$C_{s,fr}$, $C_{s,fl}$, $C_{s,rr}$ and $C_{s,rl}$ are the front right, front left, rear right and rear left suspension dampings

$Z_{u,fr}$, $Z_{u,fl}$, $Z_{u,rr}$ and $Z_{u,rl}$ are the front right, front left, rear right and rear left unsprung mass displacement

$Z_{s,fr}$, $Z_{s,fl}$, $Z_{s,rr}$ and $Z_{s,rl}$ are the front right, front left, rear right and rear left sprung mass displacement

$\dot{Z}_{u,fr}$, $\dot{Z}_{u,fl}$, $\dot{Z}_{u,rr}$ and $\dot{Z}_{u,rl}$ are the front right, front left, rear right and rear left unsprung mass velocity

$\dot{Z}_{s,fr}$, $\dot{Z}_{s,fl}$, $\dot{Z}_{s,rr}$ and $\dot{Z}_{s,rl}$ are the front right, front left, rear right and rear left sprung mass velocity

The displacement of each corner of the sprung mass can be expressed in term of bounce pitch and roll.

$$Z_{s,fr} = Z_s - L_f \sin\theta - W_r \sin\phi \quad (8)$$
$$Z_{s,fl} = Z_s - L_f \sin\theta + W_l \sin\phi \quad (9)$$
$$Z_{s,rr} = Z_s + L_r \sin\theta - W_r \sin\phi \quad (10)$$
$$Z_{s,rl} = Z_s + L_r \sin\theta + W_l \sin\phi \quad (11)$$

Assuming that all angles are small,

$$Z_{s,fr} = Z_s - L_f \theta - W_r \phi \quad (12)$$
$$Z_{s,fl} = Z_s - L_f \theta + W_l \phi \quad (13)$$
$$Z_{s,rr} = Z_s + L_r \theta - W_r \phi \quad (14)$$
$$Z_{s,rl} = Z_s + L_r \theta + W_l \phi \quad (15)$$

The force balances on the unsprung masses for the four wheels are obtained as;

$$F_{t,fr} - F_{s,fr} = M_{u,fr} \ddot{Z}_{u,fr} \quad (13)$$
$$F_{t,fl} - F_{s,fl} = M_{u,fl} \ddot{Z}_{u,fl} \quad (14)$$
$$F_{t,rr} - F_{s,rr} = M_{u,rr} \ddot{Z}_{u,rr} \quad (15)$$
$$F_{t,rl} - F_{s,rl} = M_{u,rl} \ddot{Z}_{u,rl} \quad (16)$$

where,

$F_{t,fr}$, $F_{t,fl}$, $F_{t,rr}$ and $F_{t,rl}$ are the tyre force at front right, front left, rear right and rear left corners

$M_{u,fr}$, $M_{u,fl}$, $M_{u,rr}$ and $M_{u,rl}$ are the front right, front left, rear right and rear left unsprung masses

$\ddot{Z}_{u,fr}$, $\ddot{Z}_{u,fl}$, $\ddot{Z}_{u,rr}$ and $\ddot{Z}_{u,rl}$ are the front right, front left, rear right and rear left unsprung mass vertical acceleration

Simulation Analysis Condition

For the analysis condition, the same analysis condition as used by Hudha was employed [8]. The test performed in this simulation is ride over bump. Two modes of bump were used, and they are pitch mode bump and roll mode bump. In pitch mode bump, the vehicle moves forward at a constant speed and hit the bump that is positioned laterally perpendicular to the vehicle's direction of travel. The two front tyres would hit the bump simultaneously and the two rear tyres will do the same shortly afterwards. The vertical motions of tyres will be transferred to the vehicle body resulting in vertical and pitch motions of the body. Fig. 2 shows the bump geometry and vehicle's forward speed of pitch mode bump test.

In roll mode bump test, the vehicle moves forward at a constant speed and cross the bump that is positioned longitudinally in the same direction with the vehicle's direction of travel. In this test, either left or the right tyres will hit the bump. Fig. 3 shows the bump geometry and vehicle's forward speed of the roll mode bump test. The figure also shows that the two left tyres of vehicle will cross the bump, whereas the two right tyres will follow a smooth road. The vertical motions of tyres will be transferred to the vehicle body resulting in vertical and roll motions of the body.

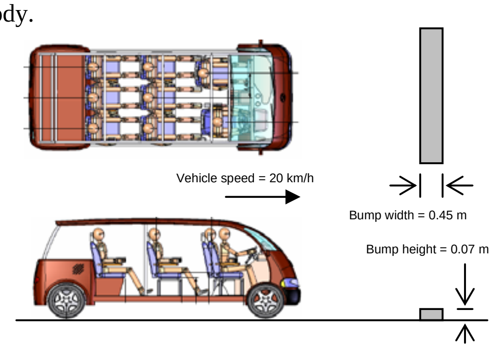

Figure 2: Pitch mode bump test

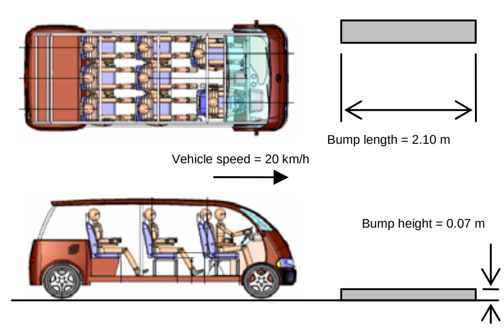

Figure 3: Roll mode bump test

Vehicle Parameters

For the simulation using the Simulink models, the input parameters for both pitch and roll mode simulation are shown in Table 2.

Table 2: Tramcar specifications

Parameter	Tramcar
Velocity (m/s)	5.56
Sprung Mass (kg)	1354
Unsprung Masses (kg)	59
Tyre Stiffness (N/m)	190000
Suspension Stiffness (N/m)	Polynomial function shown in Fig. 4
Suspension Damping (N/(m/s))	Polynomial function shown in Fig. 5
Roll Axis Moment of Inertia (kg-m^2)	729.67
Pitch Axis Moment of Inertia (kg-m^2)	2813.79
Front Tyre – CG Distance (m)	1.784
Rear Tyre – CG Distance (m)	1.633
Right Tyre – CG Distance (m)	0.805
Left Tyre – CG Distance (m)	0.795

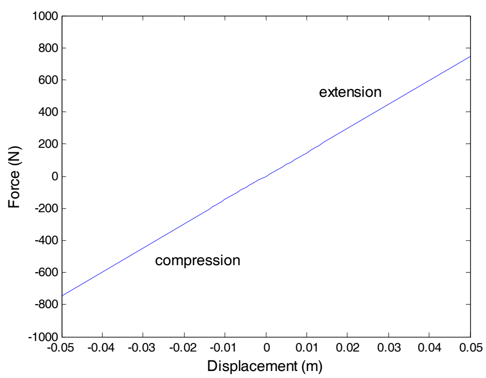

Figure 4: The force-displacement characteristic of the tramcar suspension spring

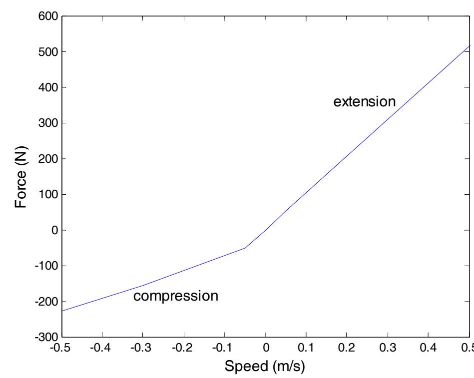

Figure 5: The force-velocity characteristic of the tramcar suspension damping

Parametric Study on Spring Stiffness

For the analysis, the suspension spring stiffness value would be varied, and the suspension system responses would be observed. Referring to Fig. 4, the tramcar suspension spring stiffness can be observed to be linear. The suspension spring stiffness was approximated to be 15000N/m. In the analysis, simulation would be performed using suspension parameters with a higher value and a lower value, in addition to the approximated value mentioned above.

For the analysis, the simulation for tramcar suspension system was performed using a number of suspension spring stiffness values. For pitch mode test, four sets of suspension spring stiffness values were used, while three sets of values were used for roll mode test. For pitch mode, the four spring stiffness values used are 15000N/m (approximated value for the actual parameter), 12000N/m (a value less than the actual value), 20000N/m and 25000N/m (two values larger than the actual value). For roll mode, the three spring stiffness values used are 15000N/m (approximated value for the actual parameter), 12000N/m (a value less than the actual value), and 25000N/m (a value larger than the actual value).

Pitch Mode Test Simulation Results

The sprung mass displacement and acceleration responses are shown in Fig. 6 and Fig. 7 respectively. It can be seen from the figures that for both the displacement and acceleration responses, the plot for 12000N/m have the lowest maximum amplitude, while the plot for 25000N/m have the largest maximum amplitude. For comparison purpose, the displacement and acceleration responses first peak magnitudes are presented in Table 3.

The RMS values for the displacement and acceleration responses have also been obtained and listed in Table 4. From the table, similar trend can be observed where the RMS values for both displacement and acceleration responses are greater when the suspension spring stiffness gets higher.

The sprung mass pitch angle and pitch rate responses are shown in Fig. 8 and Fig. 9 respectively. For both the pitch angle and pitch rate responses, the plot for 12000N/m generally have the lowest maximum amplitude, while the plot for 25000N/m generally have the largest maximum amplitude. For comparison purpose, the pitch angle and pitch rate responses first (negative) peak magnitudes are presented in Table 5.

For the purpose of observing the effect of varying spring stiffness values on the frequency of the response, the power spectrum plot of the displacement response is presented in Fig. 10. The plot shows that the frequency of the response increases with spring stiffness, where the frequency is higher when the stiffness is higher. The natural frequency magnitudes of displacement response are tabulated in Table 6.

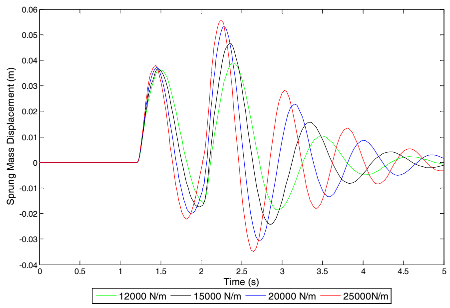

Figure 6: Sprung mass displacement at body centre of gravity for different suspension spring stiffness from pitch mode test simulation

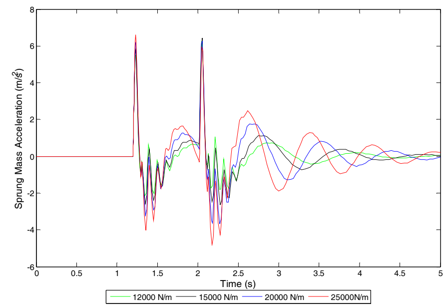

Figure 7: Sprung mass acceleration at body centre of gravity for different suspension spring stiffness from pitch mode test simulation

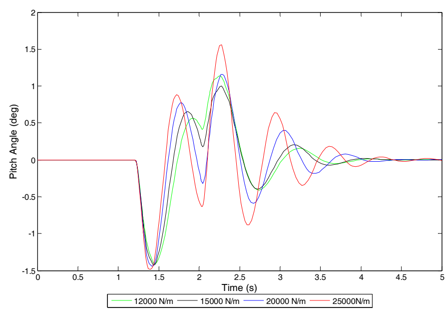

Figure 8: Sprung mass pitch angle for different suspension spring stiffness from pitch mode test simulation

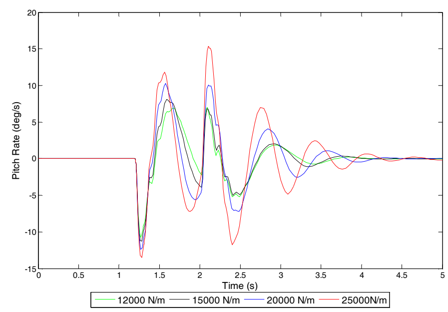

Figure 9: Sprung mass pitch rate for different suspension spring stiffness from pitch mode test simulation

Table 3: Peak amplitude comparison for displacement and acceleration responses from pitch mode test simulation

Suspension spring stiffness (N/m)	Displacement response first peak amplitude (mm)	Acceleration response first peak amplitude (m/s^2)
12000	36.1	5.5
15000	36.6	5.8
20000	37.5	6.2
25000	38.0	6.6

Table 4: RMS displacement and acceleration values for different suspension spring stiffness from pitch mode test simulation

Suspension spring stiffness (N/m)	RMS Displacement (mm)	RMS Acceleration (m/s^2)
12000	0.0135	0.8125
15000	0.0145	0.9202
20000	0.0161	1.1174
25000	0.0168	1.2908

Table 5: Peak amplitude comparison for pitch angle and pitch rate responses from pitch mode test simulation

Suspension spring stiffness (N/m)	Pitch angle response first (negative) peak amplitude (degree)	Pitch rate response first (negative) peak amplitude (m/s^2)
12000	1.41	10.71
15000	1.43	11.25
20000	1.44	12.42
25000	1.48	13.51

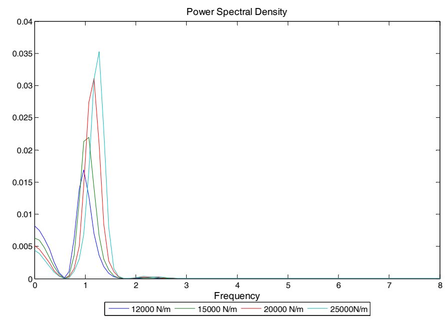

Figure 10: Power spectrum plot of displacement response for pitch mode test simulation

Table 6: Natural frequency of displacement response for pitch mode test simulation

Suspension spring stiffness (N/m)	Frequency of displacement response (Hz)
12000	0.977
15000	1.074
20000	1.172
25000	1.270

Roll Mode Test Simulation Results

The sprung mass displacement and acceleration responses are shown in Fig. 11 and Fig. 12 respectively. It can be seen from the figures that for both the displacement and acceleration responses, the plot for 12000N/m generally have the lowest maximum amplitude, while the plot for 20000N/m generally have the largest maximum amplitude. For comparison purpose, the displacement and acceleration responses highest peak magnitudes are presented in Table 7.

The RMS values for the displacement and acceleration responses have also been obtained and listed in Table 8. From the table, similar trend can be observed where the RMS values for both displacement and acceleration responses are greater when the suspension spring stiffness is higher.

The sprung mass roll angle and roll rate responses are shown in Fig. 13 and Fig. 14 respectively. It can be seen from the figures that for both the roll angle and roll rate responses, the plot for 12000N/m generally have the lowest maximum amplitude, while the plot for 20000N/m generally have the largest maximum amplitude. For comparison purpose, the pitch angle response highest peak magnitudes and the pitch rate responses first (negative) peak magnitudes are presented in Table 9.

The power spectrum plot of the displacement response is presented in Fig. 15. From the plot, it can be observed that the frequency of the response increases with spring stiffness, where higher stiffness gives higher response frequency. The natural frequency magnitudes of displacement response are tabulated in Table 10. Note that the frequencies obtained from the roll mode test simulation are the same as those obtained from the pitch mode test simulation, which are shown earlier in Table 6.

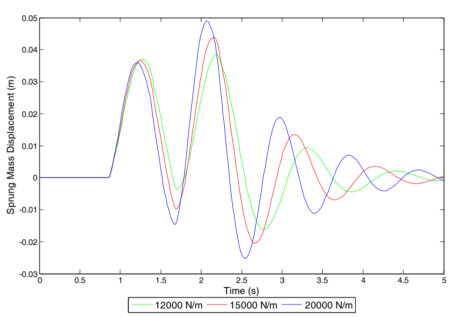

Figure 11: Sprung mass displacement at body centre of gravity for different suspension spring stiffness from roll mode test simulation

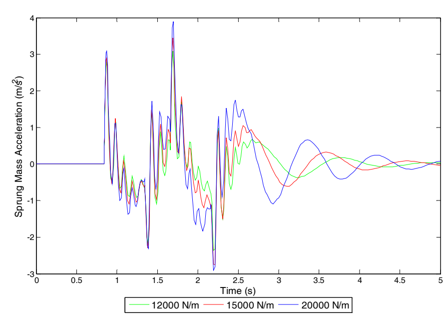

Figure 12: Sprung mass acceleration at body centre of gravity for different suspension spring stiffness from roll mode test simulation

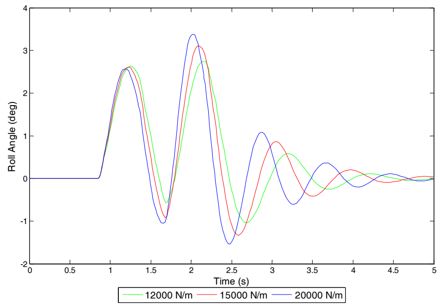

Figure 13: Sprung mass roll angle at body centre of gravity for different suspension spring stiffness from roll mode test simulation

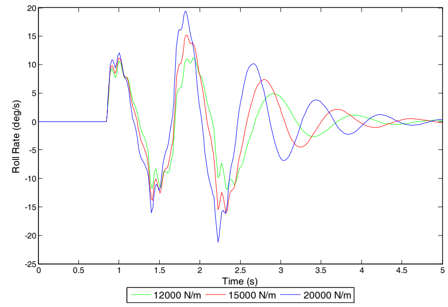

Figure 14: Sprung mass roll rate at body centre of gravity comparison for different suspension spring stiffness from roll mode test simulation

Table 7: Peak amplitude comparison for displacement and acceleration responses from roll mode test simulation

Suspension spring stiffness (N/m)	Displacement response highest peak amplitude (mm)	Acceleration response highest peak amplitude (m/s^2)
12000	38.67	3.08
15000	43.67	3.45
20000	48.96	3.89

Table 8: RMS displacement and acceleration values for different suspension spring stiffness from roll mode test simulation

Suspension spring stiffness (N/m)	RMS Displacement (mm)	RMS Acceleration (m/s^2)
12000	0.0144	0.5630
15000	0.0152	0.6557
20000	0.0161	0.8087

Table 9: Peak amplitude comparison for roll angle and roll rate responses from roll mode test simulation

Suspension spring stiffness (N/m)	Roll angle response highest peak amplitude (degree)	Roll rate response first (negative) peak amplitude (m/s^2)
12000	2.8	11.76
15000	3.1	13.83
20000	3.4	15.97

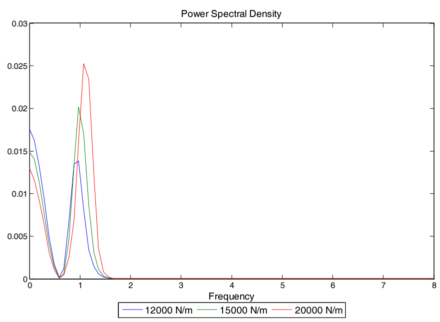

Figure 15: Power spectrum plot of displacement response for roll mode test simulation

Table 10: Natural frequency of displacement response for pitch mode test simulation

Suspension spring stiffness (N/m)	Frequency of displacement response (Hz)
12000	0.977
15000	1.074
20000	1.172

Discussion on Results from Parametric Study on Spring Stiffness

From the simulation analysis, it can be said that a low suspension spring stiffness value would provide better ride comfort in term of acceleration rate, pitch rate and roll rate responses. The results from both pitch mode and roll mode bump test simulation indicated that the acceleration, pitch rate, and roll rate responses have the lowest corresponding amplitude for the lowest spring stiffness, and have the highest corresponding amplitude for the highest spring stiffness. Thus, base on these observations, it can be deduced that to improve the ride quality of the tramcar, the suspension spring stiffness should be lowered to the lowest practical limit. This deduction agrees with what was pointed out in the discussion on suspension parameters earlier.

However, there is a limit to which the suspension spring stiffness can be reduced. One of the factors that have to be considered is the frequency of the response. From the simulation results it is observed that the response frequency increases with spring stiffness. Thus, if the spring stiffness is reduced to improve the ride quality, the frequency of the response will be lowered. In this aspect Genta has noted that frequencies lower than 1Hz produce sensations which can be assimilated to motion sickness [10]. Thus the effect of the stiffness of the springs on comfort is in a way contradictory: on one hand, the need of reducing the acceleration suggests to reduce the stiffness as much as possible, but this would lead to very low natural frequencies which can in turn cause motion sickness and similar effects [10].

In this aspect, the displacement response frequency of 1.074Hz obtained from the simulation results for the actual suspension spring stiffness of 15000N/m is already very close to the 1Hz threshold. Thus it can be said that the suspension spring stiffness is in the region of the lowest possible magnitude in the sense of avoiding a very low natural frequencies assimilated to motion sickness. It can be seen from the results that if the spring stiffness is lowered to 12000N/m, the frequency will be 0.977Hz, which is lower than the 1Hz threshold.

There are other interrelated factors that should also be looked into first before the optimum suspension parameters can be suggested. One of the factors is the vehicle handling. This is an important consideration in designing a suspension system. Even though a better ride can be obtained by reducing the spring stiffness, it is doing so in the expense of vehicle handling. Obviously there are certain levels of vehicle handling that need to be satisfied for safe driving. As the scope for this study, only ride analysis is performed, and the vehicle handling analysis is neglected. Thus, the minimum suspension spring stiffness for an acceptable vehicle handling would not be able to be determined. Hence, for a suggestion to have the best possible ride comfort for the tramcar, it can only be generally said that the suspension spring stiffness should be at the lowest practical value, such that the vehicle handling will still be at an acceptable level.

Parametric Study on Sprung Mass

This analysis was performed to observe how the suspension system response would differ for different number of passengers riding on the tramcar. For the simulation, the sprung mass for unladen, half-laden and full-laden tramcar was considered as having no passenger, four passengers, and eight passengers respectively, with every passenger was considered to weight 70kg.

Pitch Mode Test Simulation Results

The sprung mass displacement and acceleration responses are shown in Fig. 16 and Fig. 17 respectively. It can be seen from the figures that for both the displacement and acceleration responses, the plot for unladen tramcar have the largest maximum amplitude, while the plot for full-laden tramcar have the lowest maximum amplitude. For comparison purpose, the displacement and acceleration responses first peak magnitude is presented in Table 11.

The RMS values for the displacement and acceleration responses are listed in Table 12. From the table, similar trend can be observed, where the RMS values for both displacement and acceleration responses are lower when the passenger loading is higher.

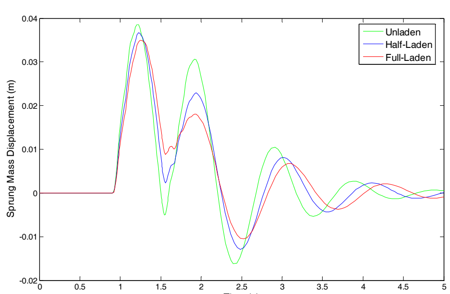

Figure 16: Sprung mass displacement at body centre of gravity comparison for different passenger loadings

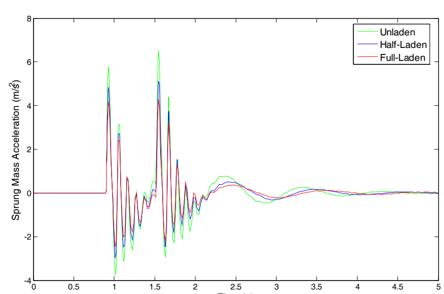

Figure 17: Sprung mass acceleration at body centre of gravity comparison for different passenger loadings

Table 11: Peak amplitude comparison for displacement and acceleration responses from pitch mode test simulation

Passenger Loading	Displacement response first peak amplitude (mm)	Acceleration response first peak amplitude (m/s^2)
Unladen	38.5	5.8
Half-laden	36.7	4.8
Full-laden	34.9	4.5

Table 12: RMS displacement and acceleration values for different passenger loadings from pitch mode test simulation

Passenger Loading	RMS Displacement (mm)	RMS Acceleration (m/s^2)
Unladen	0.0141	1.5889
Half-laden	0.0130	1.2940
Full-laden	0.0126	1.0722

Roll mode Test Simulation Results

The sprung mass displacement and acceleration responses are shown in Fig. 18 and Fig. 19 respectively. It can be seen from the figures that for both the displacement and acceleration responses, the plot for unladen tramcar have the largest maximum amplitude, while the plot for full-laden tramcar have the lowest maximum amplitude. For comparison purpose, the displacement and acceleration responses first peak magnitude is presented in Table 13.

The RMS values for the displacement and acceleration responses are listed in Table 14. From the table, similar trend can be observed the RMS values for both displacement and acceleration responses are lower when the passenger loading is higher.

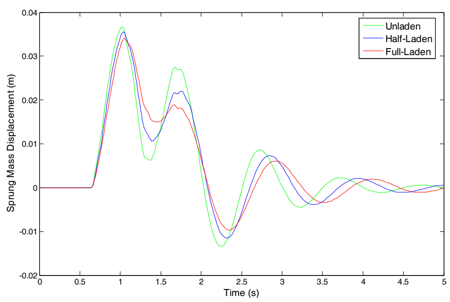

Figure 18: Sprung mass displacement at body centre of gravity comparison for different passenger loadings

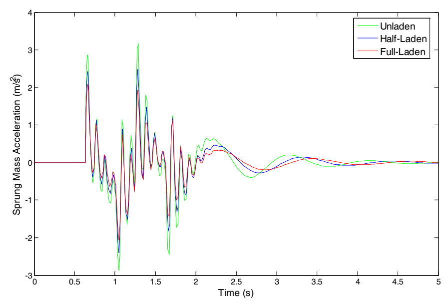

Figure 19: Sprung mass acceleration at body centre of gravity comparison for different passenger loadings

Table 13: Peak amplitude comparison for displacement and acceleration responses from roll mode test simulation

Passenger Loading	Displacement response first peak amplitude (mm)	Acceleration response first peak amplitude (m/s^2)
Unladen	36.7	2.9
Half-laden	35.6	2.4
Full-laden	34.1	2.1

Table 14: RMS displacement and acceleration values for different passenger loadings from roll mode test simulation

Passenger Loading	RMS Displacement (mm)	RMS Acceleration (m/s^2)
Unladen	0.0142	0.8337
Half-laden	0.0135	0.6717
Full-laden	0.0134	0.5632

Discussion on Results from Parametric study on Sprung Mass

From the comparison between simulation results for unladen, half-laden and full-laden tramcar, it was observed that for both the displacement and acceleration responses, the plot for unladen tramcar have the largest maximum amplitude, while the plot for full-laden tramcar have the lowest maximum amplitude. It was also observed that the RMS values for both displacement and acceleration responses are lower when the passenger loading is higher. Thus it can be concluded that the suspension system response will be better in term of ride quality when the sprung mass is larger. As the simulation analysis that was performed on the tramcar suspension system considered the sprung mass as the unladen tramcar body mass, it can be predicted that the obtained responses and results would be relatively better in term of ride quality if half-laden or full-laden was considered.

Conclusion

From the suspension parameter analysis, it was concluded that the ride comfort of the tramcar can be improved to an optimum level by having the lowest practical spring stiffness value. Lower suspension spring stiffness was shown to provide better ride comfort in term of lower acceleration, pitch rate and roll rate responses. However, the spring stiffness should not produce response frequencies lower than 1Hz in avoiding sensations assimilated to motion sickness.

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