CHAPTER VI

SIMULATION OF SUSPENSION MODEL AND PARAMETRIC STUDY INTO THE EFFECTS ON RIDE COMFORT

In field test, operators are subjected to many influences in their environment. Therefore, with the help of modern computer tools/simulation to determine the influence of some suspension parameters on ride comfort of a light commercial vehicle, a simulation model was designed and used to investigate these parameters. In this thesis, the quarter-car model, with sub-system which represents the characteristics of suspension system was used to study the response of the vehicle to the road profile excitation, based on road surface roughness classification approved by ISO 8608 [14]. Power spectral density of the response to different road profiles with corresponding values of leaf spring parameters is analyzed and the average ride comfort/discomfort level (ride indices) as the evaluation of ride comfort in quantitative numbers are calculated through methods suggested by ISO 2631-1. The results obtained from preliminary study were developed with the simulating tools where interesting parameters were examined and properly adjusted.

6.1. Quarter-Car Model

The quarter car model is a well-known model that has been widely used in vehicle simulation to study the vehicle response to excitation inputs which can reflect characteristics of vehicle structures and performance. In this thesis, the quarter-car model with sub-system, representing the characteristics of suspension of a light commercial vehicle is used to investigate the effect of parameters settings for leaf spring and damper on ride comfort. The concept of model construction is to build up a simple model that represents suspension system with components. The parameters of leaf spring and damper are put into the investigation of this study. Fig.6-1 illustrates the conceptual design of the quarter-car model that was used in the simulation.

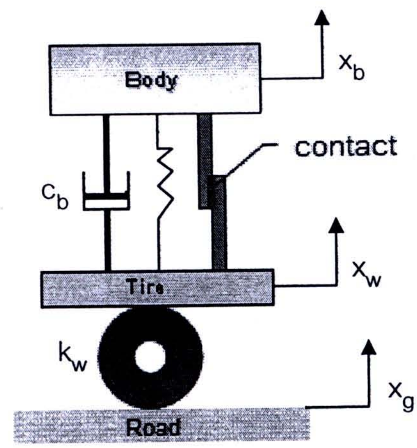

Fig.6-1 Conceptual design of the quarter-car model for suspension system

As shown in the picture, the suspension system between sprung mass (vehicle body) and unsprung mass (tire) is composed of a linear damper component, a spring, and two thin metal strips having a contact with each other which represents the hysteresis characteristics of a leaf spring. At this contact point, the relative motion occurs when the sprung and unsprung mass is moving in vertical direction, due to the road elevation that the vehicle is experienced. The relative movement at the contact point between the two thin strips also contributes to the dry friction force that lead to the appearance of nonlinear phenomena in suspension component.

6.1.1. <u>Model Characteristics</u>

The model was constructed based on following assumptions,

1. The vehicle bodies were considered to be rigid bodies.
2. The roll and pitch motions of the vehicle were small and neglected.
3. The tires were assumed to behave as a linear spring without damping.
4. The tires were not allowed to leave the road surface.
5. A nonlinear leaf spring model was developed used.
6. A suspension damper is linear function related to velocity.

6.1.2. Mathematical Equations of the Model

For the previous mentioned quarter-car model, the equations of motion can be obtained by summing the vertical forces acting on each of the two bodies in the system, and applying Newton's second law of motion, *F=ma*. The resulting two equations are second-order nonlinear, ordinary differential equations. The mathematical derivation, regarding the effect of nonlinearity property in leaf spring can be written as:

$$f_s + f_d = m_b \ddot{x}_b \qquad 6\text{-}1$$

$$k_w(x_g - x_w) + c_w(\dot{x}_g - \dot{x}_w) - f_s - f_d = m_w \ddot{x}_w \qquad 6\text{-}2$$

where f_s = *nonlinear spring force (N)*

f_d = *damping force (N)*

m_b = *mass of the car body (kg)*

m_w= *mass* x_w *of the wheel/tire (kg)*

k_w= *wheel/tire stiffness (N/m)*

c_w= *tire damping coefficient (N-s/m)*

x_b = *vertical displacement of the car body (m)*

x_w = *vertical displacement of the wheel/tire (m)*

x_g = *vertical road elevation (m)*

$x_g - x_w$ *represents suspension displacement (m)*

6.2. Road Input Profile

In real situation, vehicle system may be excited from multiple excitation sources such as the engine, rotating elements, etc. However, the road roughness is the primary concerned excitation source for simulation test performed in this thesis. In simulation test, the synthetic road profile was generated by the computational program and fed into a vehicle model as an input. The random road profile was constructed in MATLAB program according to the road surface roughness classification proposed by ISO 8608. The quality of road is classified based on the amplitudes of power spectral density as shown in Fig.6-2. The road type A is the classification of the "very good" quality road, type B for the "good" road, type C for the "average" road, type D for the "poor" road, and type E for the "very poor" road. The result of simulated road inputs compared with the classification standard is shown in Fig.6-3 and the simple road model used for this research is shown in Fig.6-4.

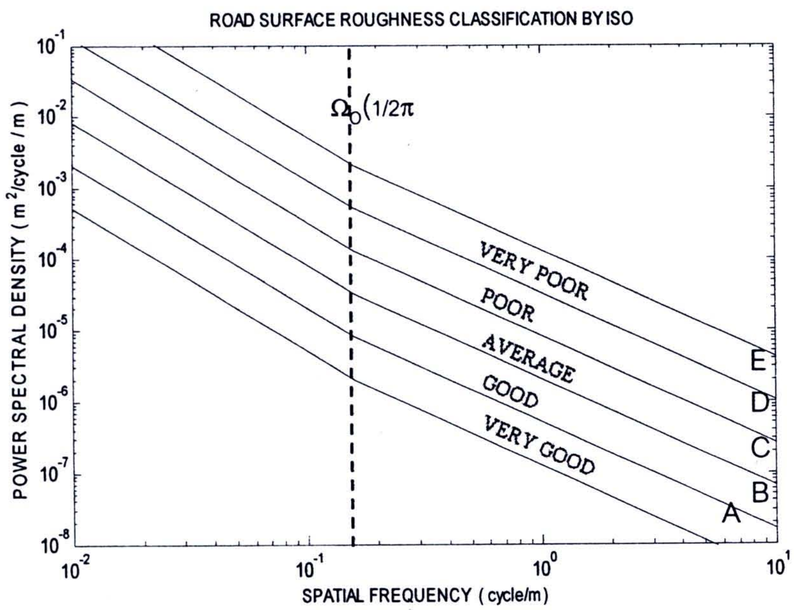

Fig.6-2 Road surface roughness classification by ISO

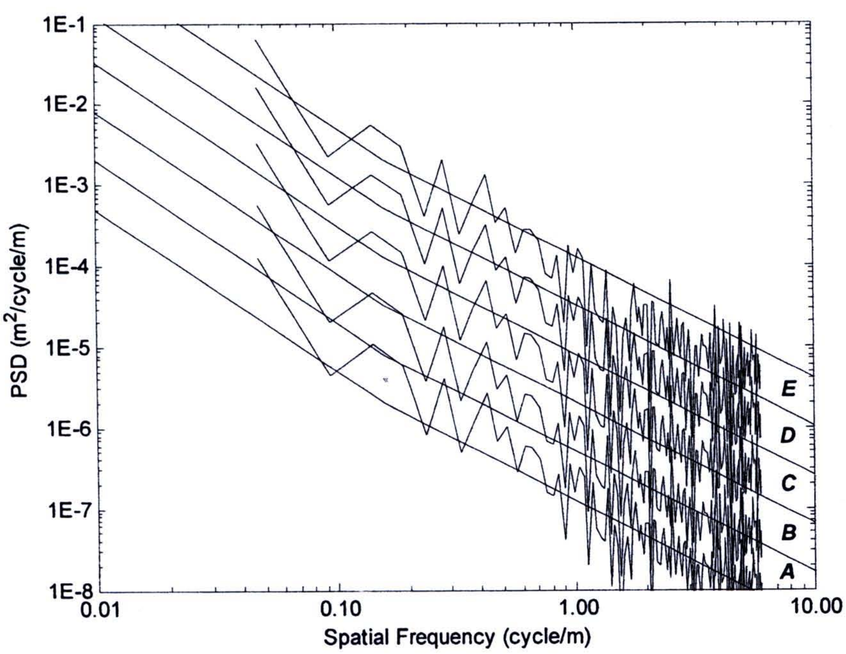

Fig.6-3 The result of simulated road inputs compared with the classification standard

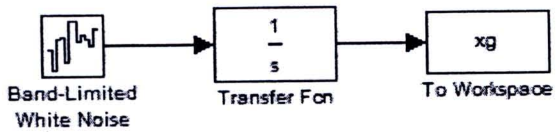

Fig.6-4 A simple road model used to simulate road inputs

6.3. Completed Vehicle Suspension Model

Full vehicle suspension model was completed by including the leaf spring model described in the previous chapter into a quarter-car model. This model represents a 1/4 vehicle system with a leaf spring type suspension that contains nonlinearities which were related to the parameters of hysteresis component.

6.4. Parametric Study by Simulation

The parametric study of the proposed leaf spring suspension model was performed by simulation. The synthetic road profile, representing road irregularities for five road classes were generated by the simple road model introduced in the previous section at the constant velocity 60 km/hr. The frequency weighting technique was employed in ride comfort evaluating process so that the value of weighted RMS acceleration for each type of road was calculated upon different values of shackle angle at initial installation of leaf spring.

Parameters	Symbols	Values	Units
Mass of vehicle body	m_b	467.5	kg
Mass of wheel/tire	m_w	30	kg
Damping coefficient of vehicle body	c_b	2369.3	N-s/m
Wheel/tire stiffness	k_w	118440	N/m

Table 6-1 Parameters of the simulated quarter-car model

Parameters used for simulating the quarter-car model is tabulated in Table 6-1. The damping coefficient of vehicle body was calculated based on value of damping ratio at 0.3 by equation 6-3.

$$\eta = \frac{c_b}{2k_l m_b} \qquad \text{6-3(a)}$$

$$c_b = 2\eta k_l m_b \qquad \text{6-3(b)}$$

where η = *damping raio* = *0.3*

k_l = *linear elastic stiffness which was set to be equal to the value of nominal stiffness obtained from leaf spring experimental testing.*

m_b = *mass of vehicle body*

The wheel/tire stiffness was calculated based on the assumption that it's natural frequency is approximately at 10 Hz and was determined by the following equation,

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k_w}{m_w}} \qquad \text{6-4(a)}$$

hence

$$k_w = m_w(2\pi f_n)^2 \qquad \text{6-4(b)}$$

where f_n = *natural frequency of wheel/tire* = *10 Hz*

m_w = *mass of wheel/tire*

k_w = *wheel/tire stiffness*

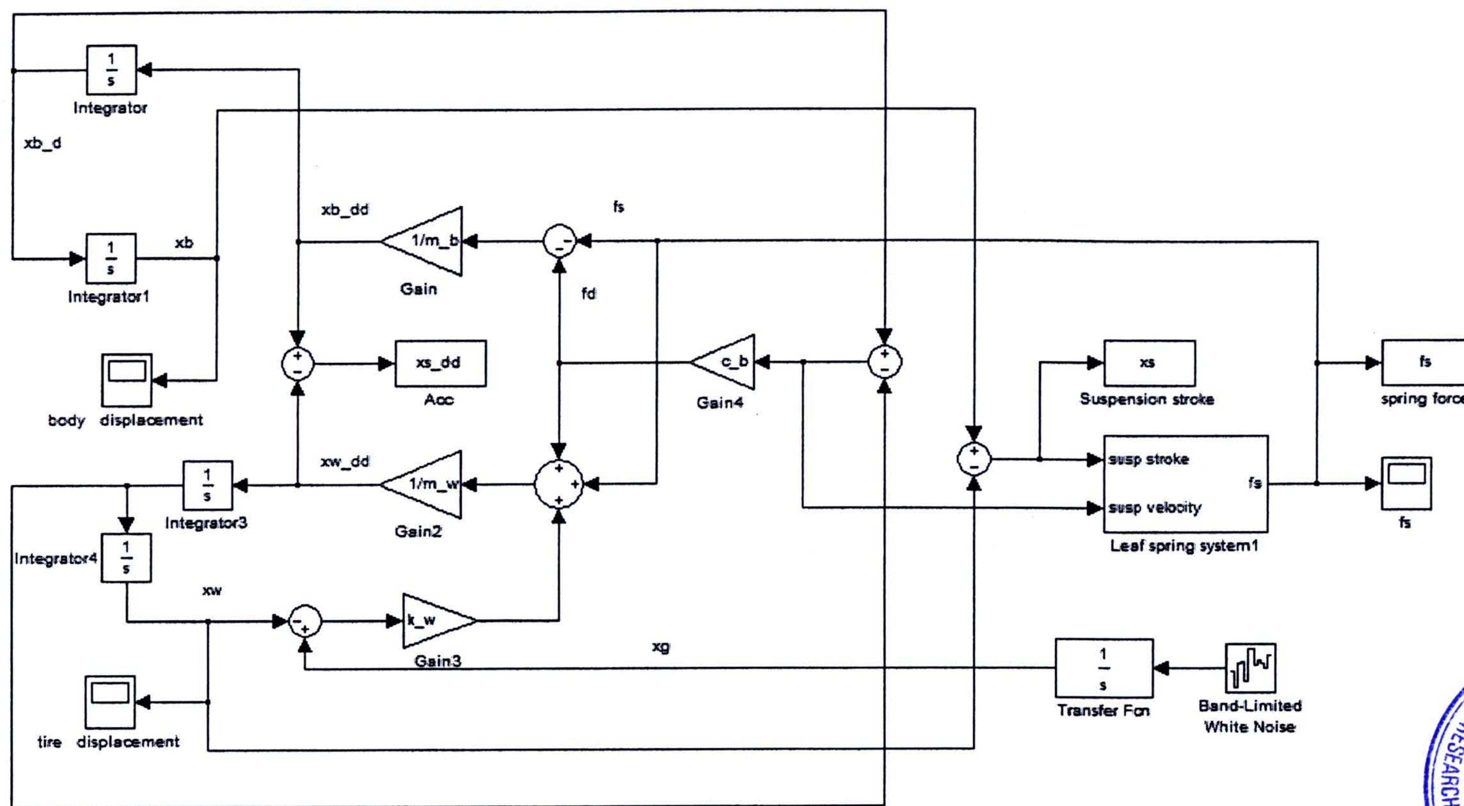

Fig.6-5 Completed vehicle suspension model

6.5. Ride Comfort Results from Simulating Model Subjected to Random White-Noise Input

The completed vehicle suspension model was used for parametric study. For simulation subjected to different types of roads, the shackle angles were varied from 60 degree to 120 degree measured counterclockwise from the horizontal plane. The relationship between torque and angular displacement was also monitored and displayed as shown below,

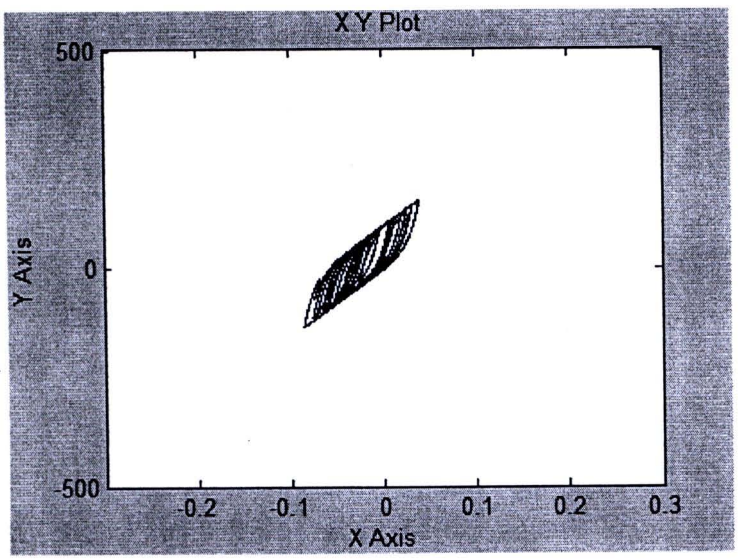

Fig.6-6 The relationship between torque and angular displacement at the rotational joints

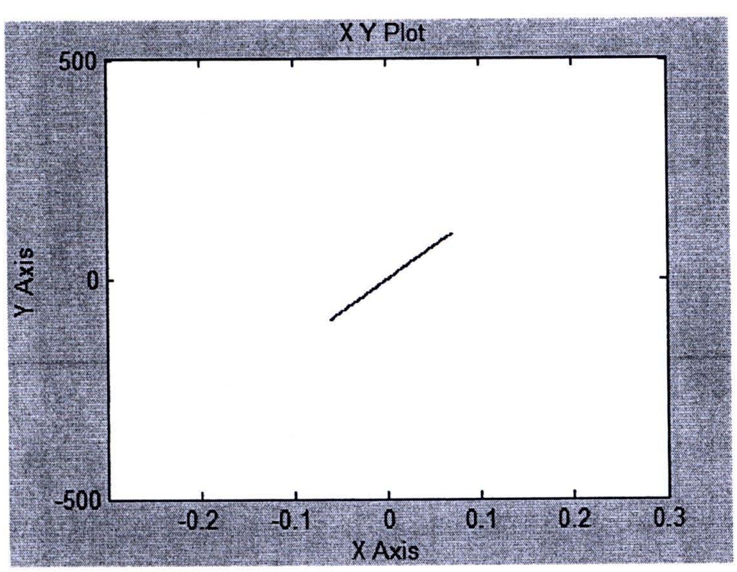

Fig.6-7 The relationship between torque and angular displacement obtained from linear-leaf spring suspension model

For simulation by linear leaf spring suspension model, the hysteresis component was excluded so that the relation between load and displacement was simple as linear function to each other as shown in Fig.6-7. The weighted RMS accelerations, representing the level of ride comfort/discomfort subjected to different classes of roads were tabulated in Table 6-1.

	Very good	Good	Average	Poor	Very Poor
60 degree	0.2211	0.6399	1.1315	2.6125	5.6427
70 degree	0.3870	0.6112	0.8351	2.1261	4.9186
80 degree	0.3067	1.2364	1.8286	5.0628	7.7244
90 degree	0.3109	0.9155	1.7317	3.7572	12.0981
110 degree	0.4877	0.8339	0.9617	3.2023	7.0052
120 degree	0.4950	1.0797	1.9207	2.8572	8.4904

Table 6-2 The weighted RMS accelerations, representing the level of ride comfort/discomfort subjected to different classes of roads (non-linear suspension)

From simulated results and calculation from the frequency-weighting method, the highest value was found at 90° of shackle angle by simulation of the "very poor" road while the lowest value was found at 60° of shackle angle by simulation of the "very good" road. The vertical acceleration of the suspension, vehicle body, and wheel/tire are also represented by the RMS of the 130 data/interval plots as shown in Figs 6-8(a), (b) and 6-9(a), (b). As seen from the graph, the magnitude of the RMS acceleration is highest at the tire, lower at the suspension and lowest at the vehicle's body for both cases. This is reasonable as the suspension perceived the vibration transmitted from vehicle tire and tried to absorb and to dissipate the energy so that the amount of vibration transferred to the vehicle's body was deteriorated.

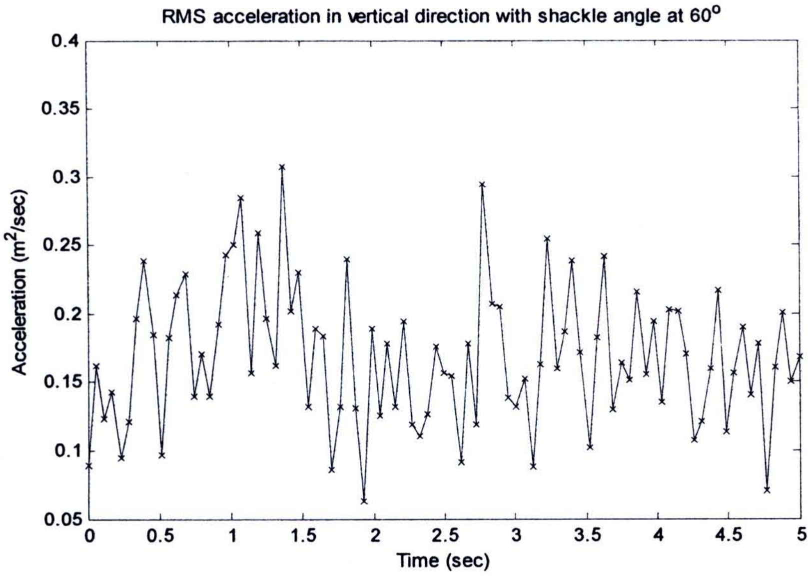

Fig.6-8(a) The plots of RMS vertical acceleration of vehicle suspension at 60° of shackle angle

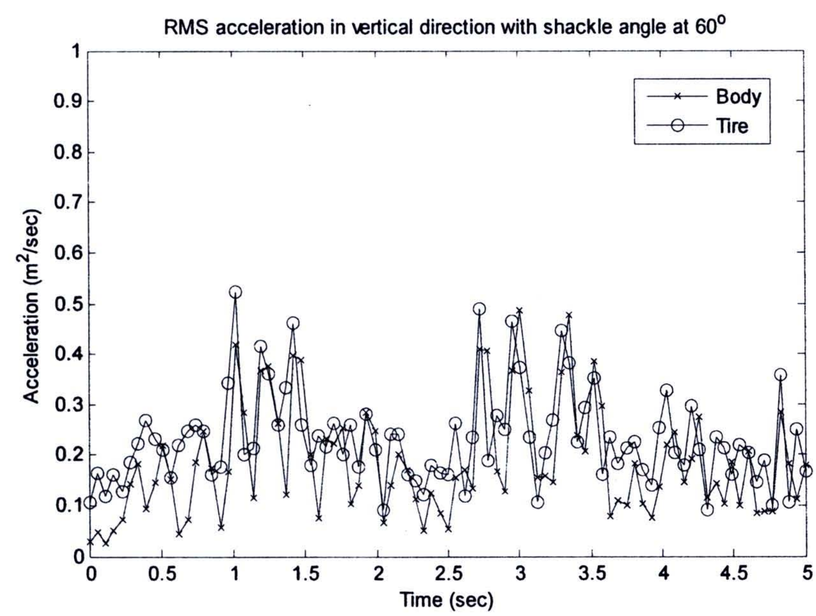

Fig.6-8(b) The plots of RMS vertical acceleration of vehicle body (cross line) and tire (circle line) at 60° of shackle angle

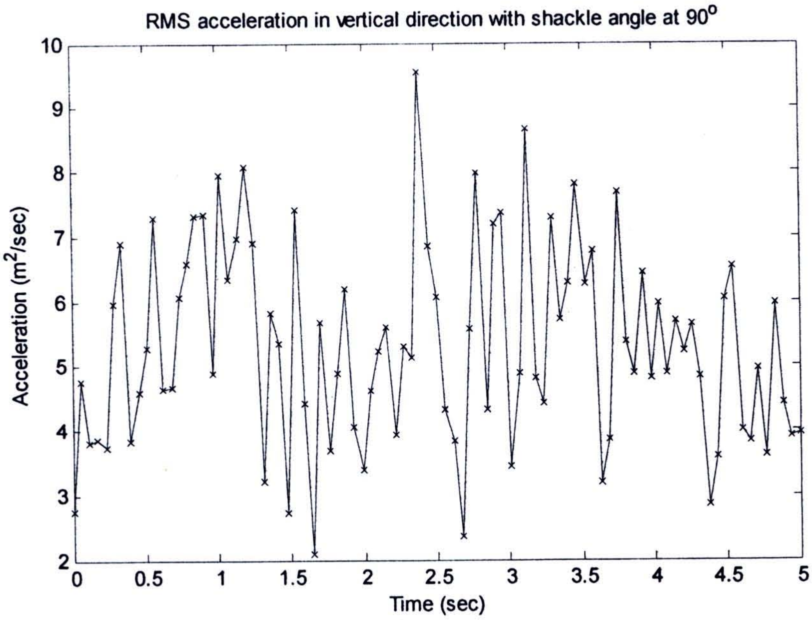

Fig.6-9(a) The plots of RMS vertical acceleration of vehicle suspension at 90° of shackle angle

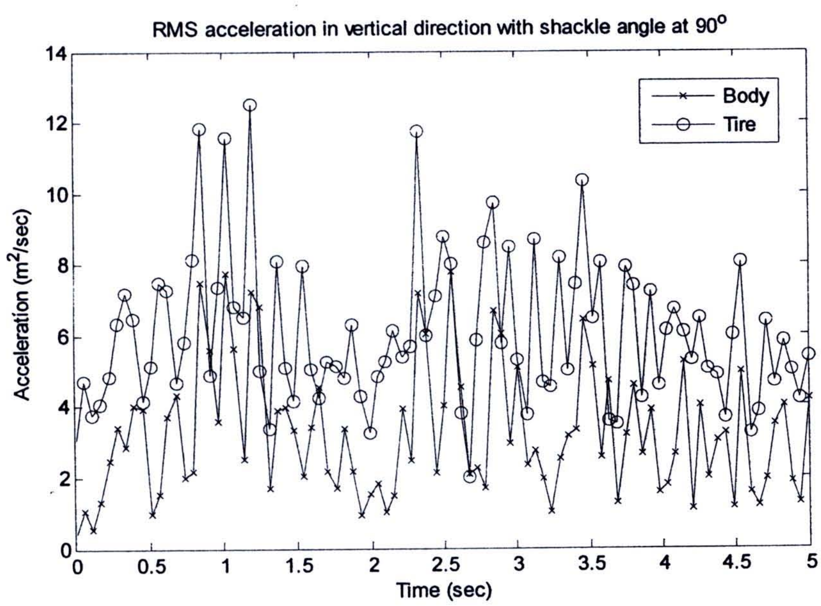

Fig.6-9(b) The plots of RMS vertical acceleration of vehicle body (cross line) and tire (circle line) at 90° of shackle angle

The results of weighted RMS acceleration obtained from nonlinear and linear model are also compared to each other as shown in Figs 6-10 – 6-15. The RMS acceleration values were plotted against different types of roads.

When the initial installation shackle angle is at 60 degree, the shape of curves are similar to each other which tends to increase when the road is rougher but when the road changes from "poor" to "very poor" the value obtained from nonlinear model seems to increase rapidly.

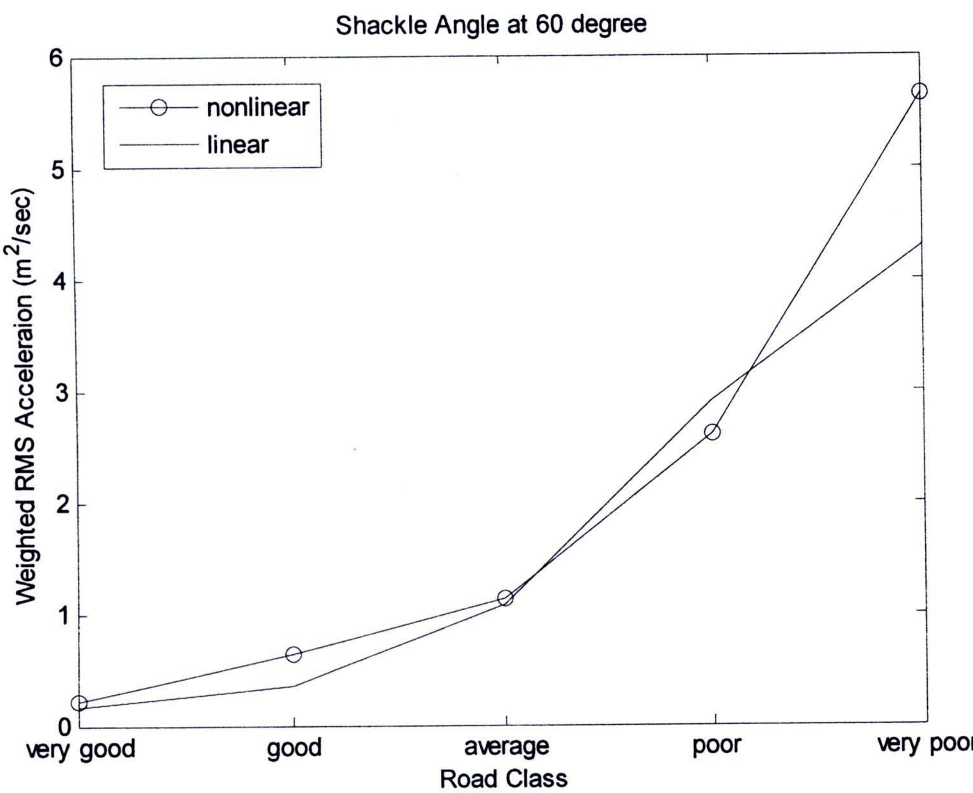

Fig.6-10 Ride comfort results of nonlinear and linear model at shackle angle 60°

At shackle angle of 70 degree, there's the significant difference between results of nonlinear and linear models as the values obtained from linear model changes faster than the nonlinear's, resulting in higher values of RMS weighted acceleration for almost every type of roads. When the shackle angle increased up to 80 degree, the results from both models are closer although the linear model seems to produce higher accelerations. At 90 degree, the curves are almost the same except for the very poor road as the value obtained from nonlinear model is higher.

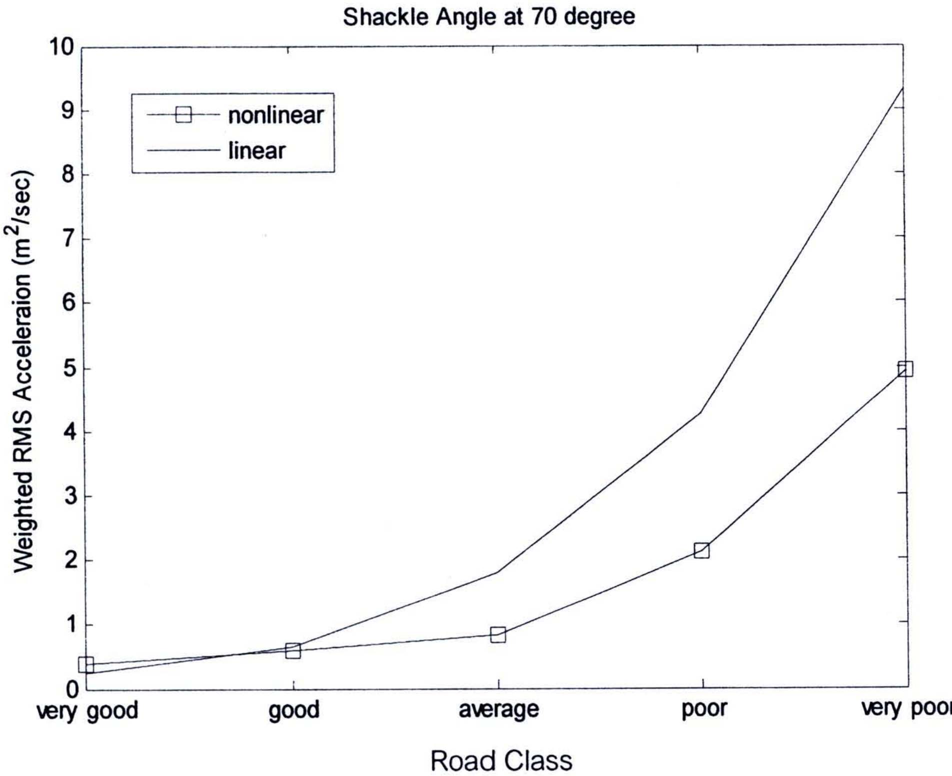

Fig.6-11 Ride comfort results of nonlinear and linear model at shackle angle 70°

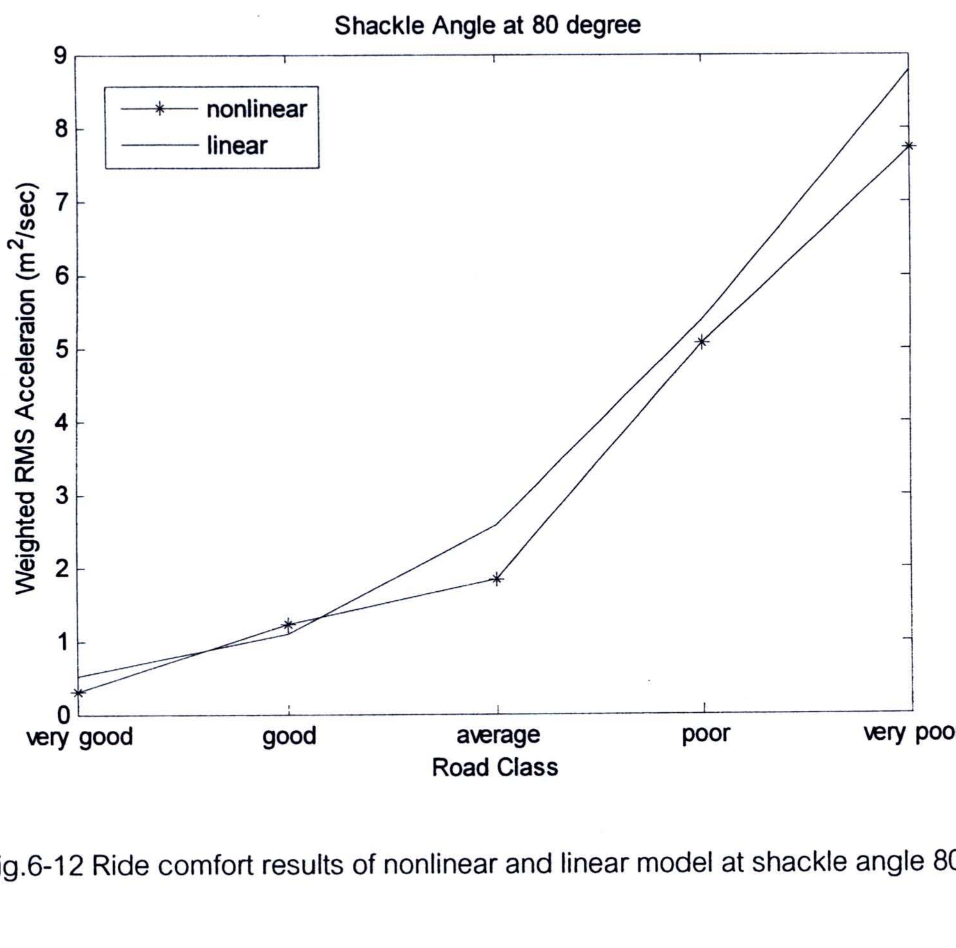

Fig.6-12 Ride comfort results of nonlinear and linear model at shackle angle 80°

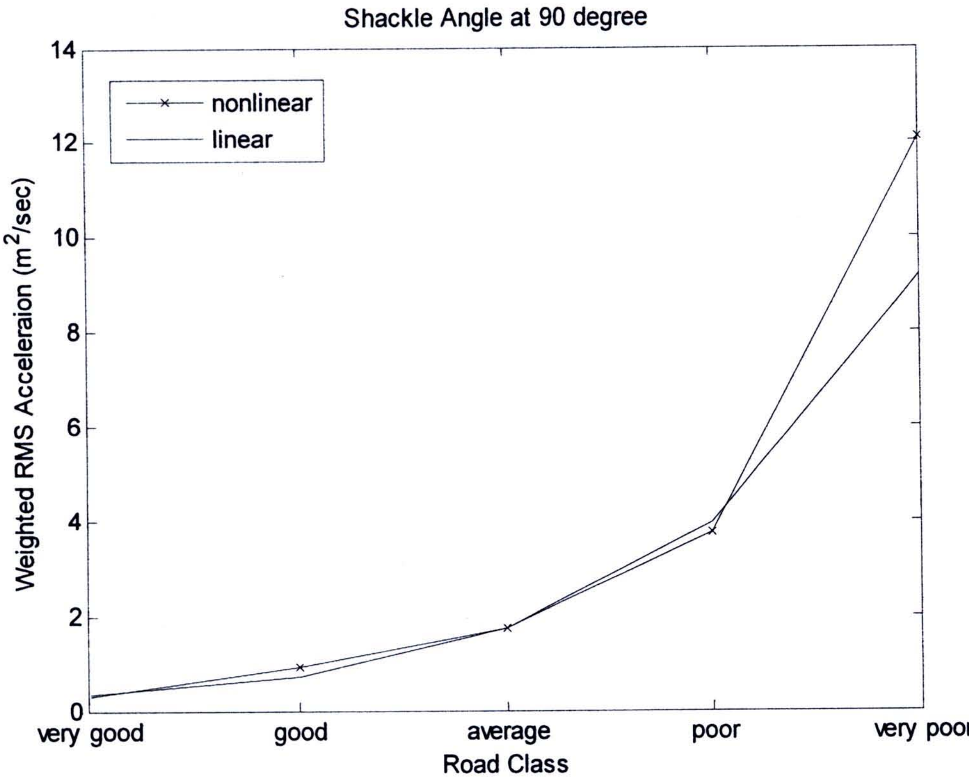

Fig.6-13 Ride comfort results of nonlinear and linear model at shackle angle 90°

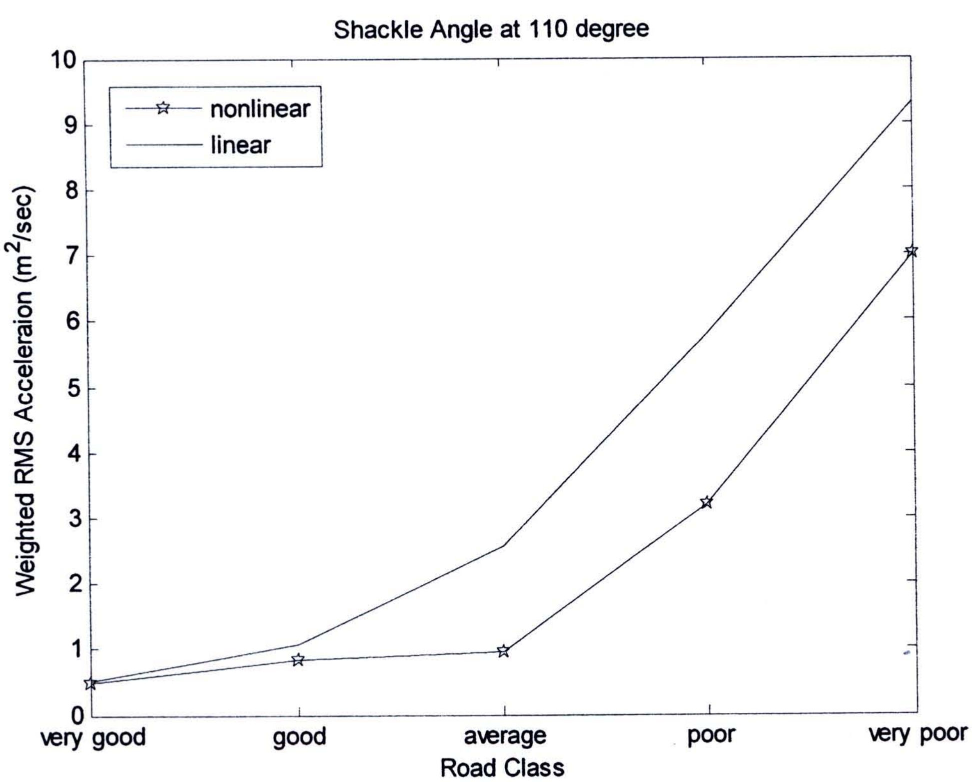

Fig.6-14 Ride comfort results of nonlinear and linear model at shackle angle 110°

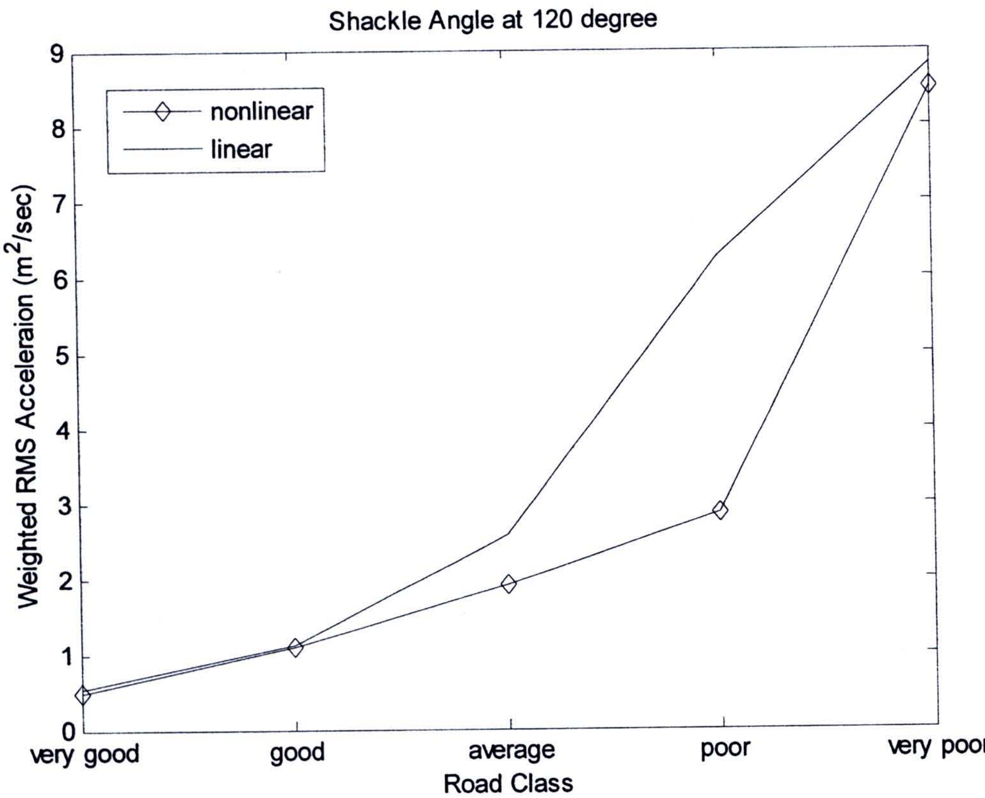

Fig.6-15 Ride comfort results of nonlinear and linear model at shackle angle 120°

The shapes of two curves are different again at 110° with the higher rate of change from the linear model's results and even faster when increasing to be at 120°. It can be said that for the overall results, ride comfort is better on smooth road with gradual changes from "very good" to "poor" road and sudden increase for the "very poor" road. When considering only for the results of nonlinear model, the RMS acceleration values were plotted against shackle angles for every type of road. It can be seen from the graph that for the "good", "average", and "poor" road, the RMS acceleration values are lowest when shackle angle is 70° while the lowest for the "very good road" appears at shackle angle of 60°. However, the obtained value of 70 ° is just slightly different from that of 60° and can be acceptable as it falls in to a "fairy uncomfortable" criteria. It is also found that at 80°, the weighted RMS acceleration value is highest for the "good" and "poor" road while the highest values for the "very good", "average", and "very poor" road are found at shackle angle of 110°/120°, 120°, and 90°, respectively.

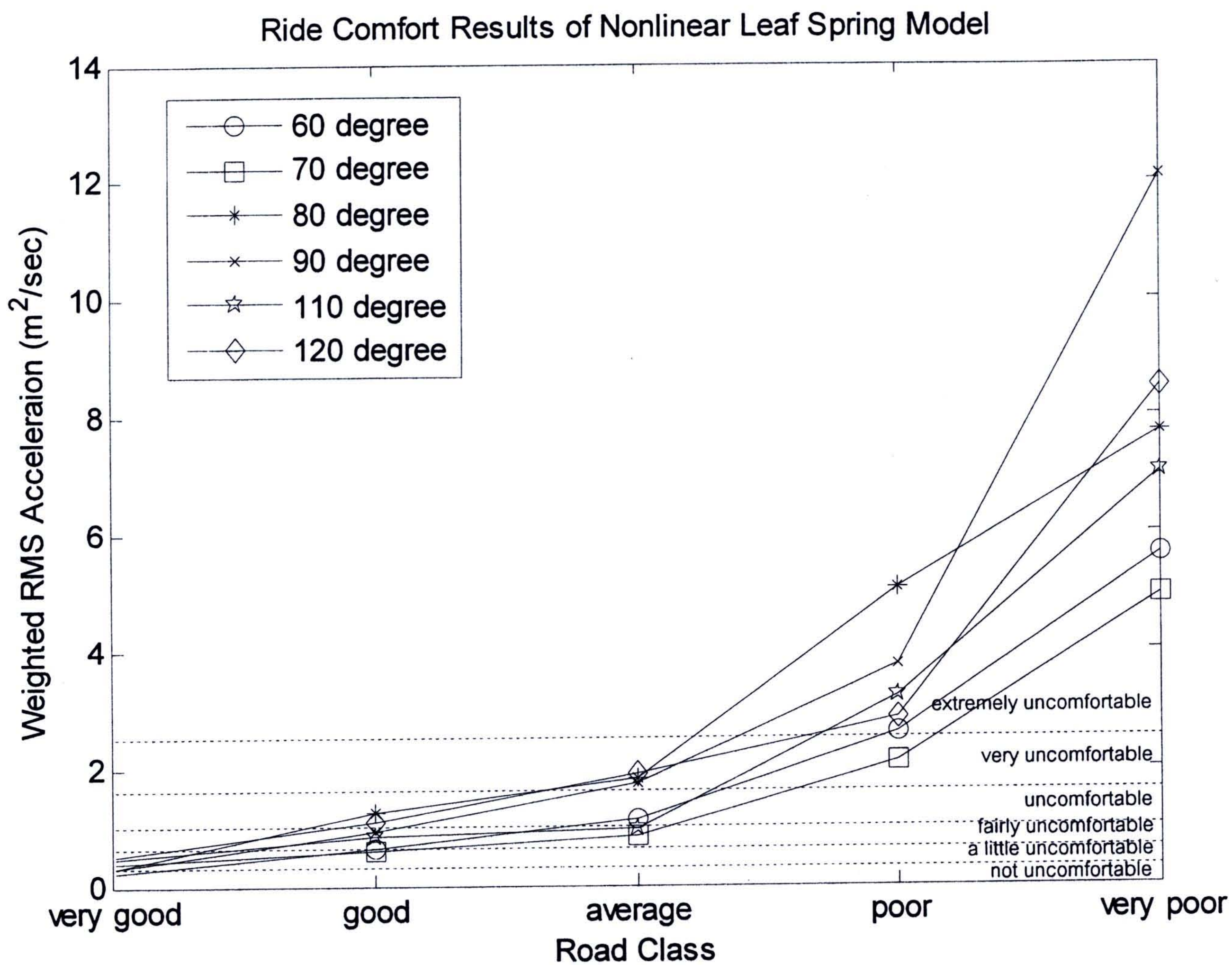

Fig.6-16(a) Ride comfort results of nonlinear leaf spring model at different shackle angles

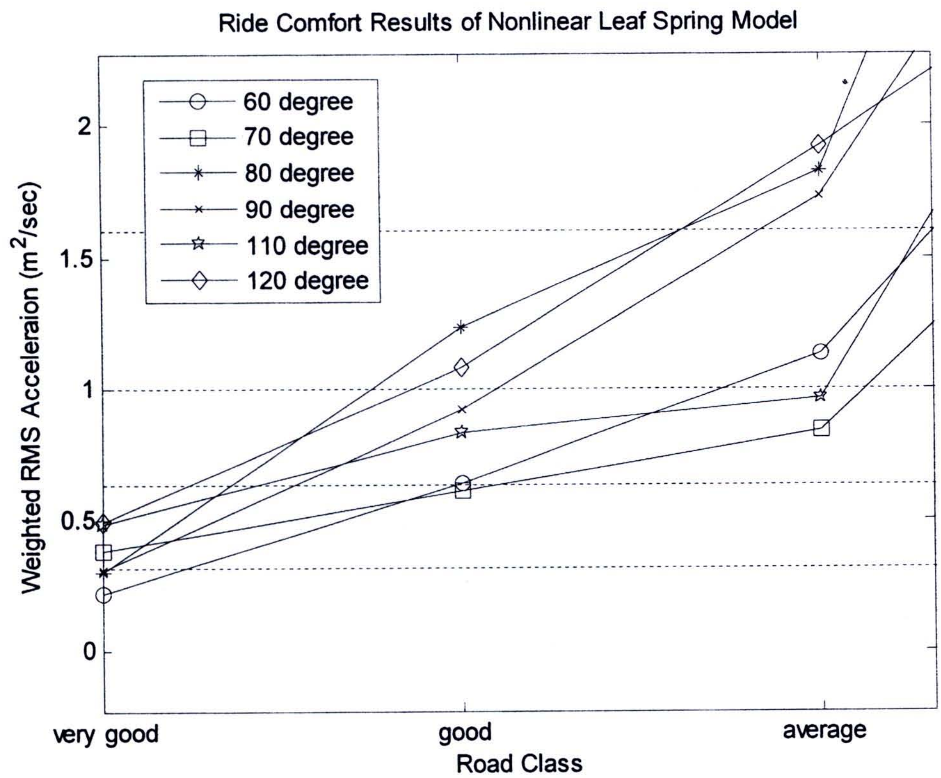

Fig.6-16(b) Ride comfort results of nonlinear leaf spring model for "very good" to "average" road

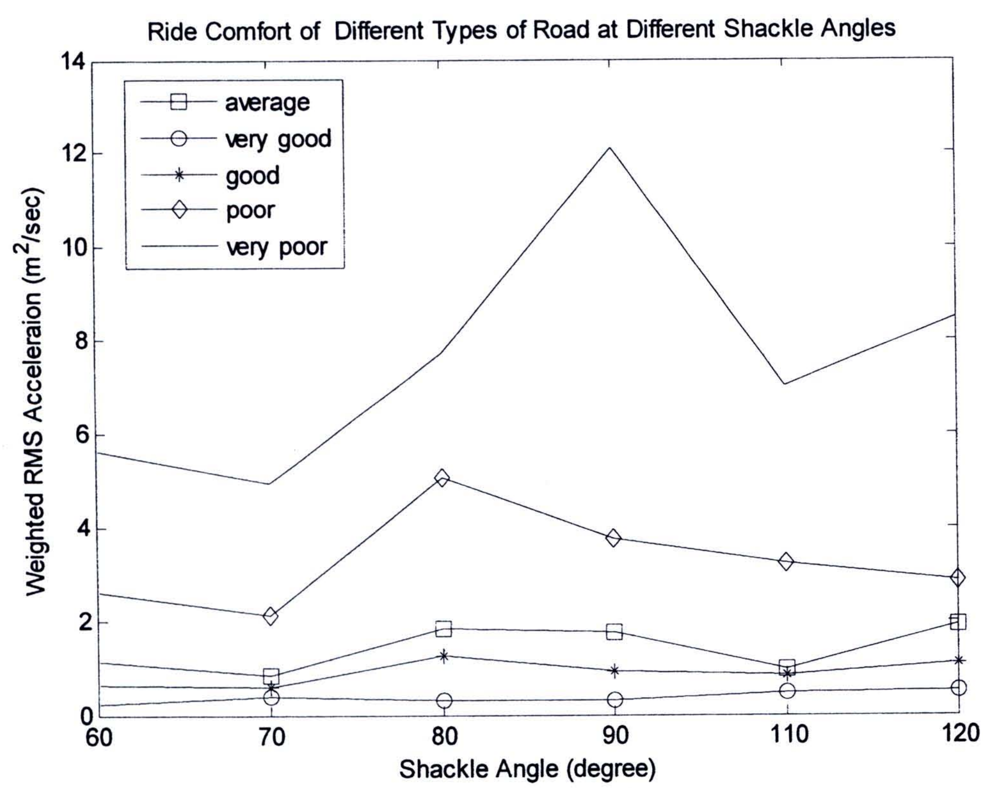

Fig.6-17 Ride comfort results of nonlinear leaf spring model at different classes of roads

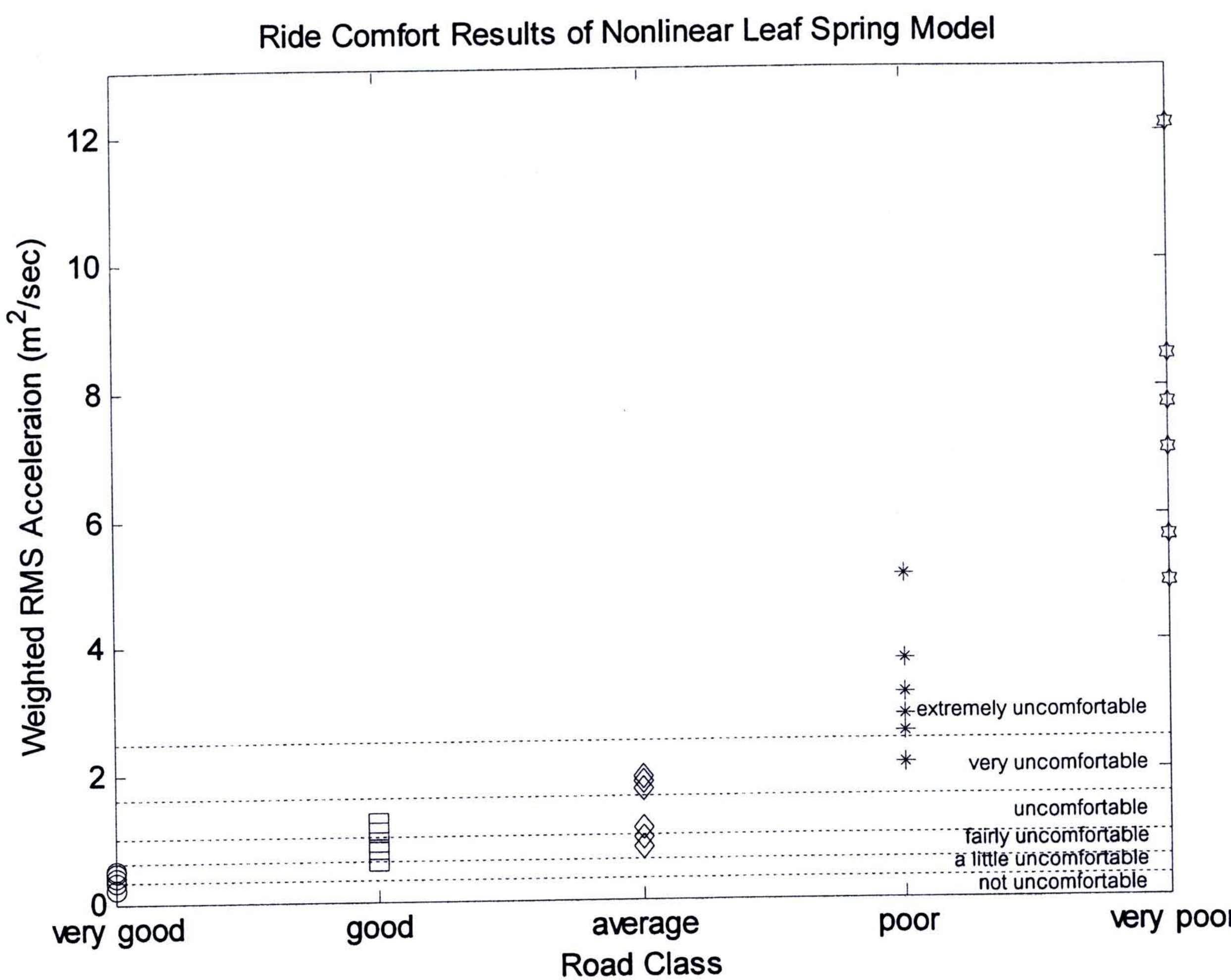

Fig.6-18 Ride comfort level of nonlinear leaf spring model at different classes of roads

6.6. Discussion of Results and Conclusion

From the results presented in the previous section, the overall results show that ride comfort is better on smooth road with gradual changes from "very good" to "poor" road and sudden increase for the "very poor" road. The values obtained from the nonlinear leaf spring suspension model mostly agrees with the suggested ride comfort/discomfort level proposed by ISO. The ride values lie within "not uncomfortable" and "a little uncomfortable" categories for simulation of "very good" road while those of the "good" road lie within "a little uncomfortable" to "uncomfortable" zone. For the "average" road, the values lie within two separate zones. The lower zone includes the results obtained from simulation of shackle angles at 60°, 70°, and 110° which lie within the "fairly uncomfortable" and "uncomfortable" categories while the higher zone includes the results obtained from simulation of shackle angles at 80°, 90°, and 120° which lie mostly in the "very uncomfortable" zone. For the "poor" road, the values lie in the "very uncomfortable" and mostly in the "extremely uncomfortable" zones. The worst case of ride comfort is in the "extremely uncomfortable" area for all values of shackle angles which is highest at 90° and lowest at 70°. From the comparison of the linear and nonlinear results, the linear model tends to produce higher value in degree of discomfort. This is because of the nonlinear leaf spring suspension model contains hysteresis and friction components that resist leaf spring movement and behave as a natural damper, resulting in less displacements. It is noticeable that at the position 90° of shackle angle at initial installation, the nonlinear leaf spring behaves similarly to the linear leaf spring. This may be explained by the knowledge of leaf spring installation effect from section 2.1.2.3.

When a leaf spring is not perpendicular to the datum line, the shackle load possesses its component in longitudinal axis (force P shown in Fig.6-19) which can be either compression or tension, depending on the direction of the swinging motion. If the shackle swings towards the fixed end of the leaf spring, it results in compressive load while it tends to produce tensile load as the shackle swings in the opposite direction. Under shackle tension, spring rate is increased while it is decreased when subjected to shackle compression. At 90°, there's no effect from the longitudinal load so that the leaf spring's behavior of linear and nonlinear models are almost the same.

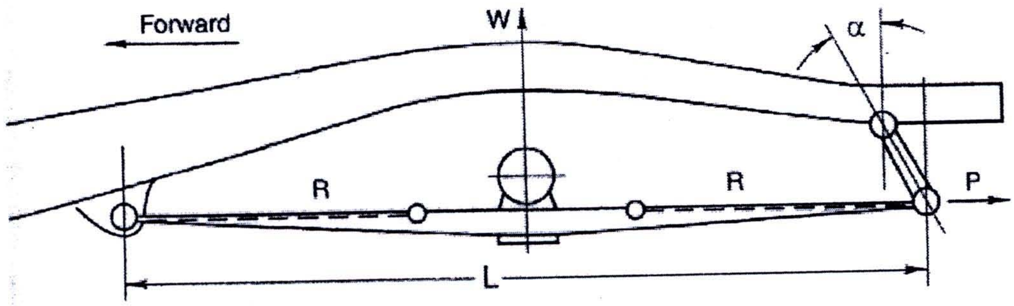

Fig.6-19 Symmetrical leaf spring with shackle [15]