Dynamic Behavior of Three-wheeled Vehicle Under Random Road Characteristics Using FEM

R. Koonan, M.K. Naidu and B. Satyanarayana

1. Associate Professor, Mechanical Engineering Department, College of Engineering, Andhra University, Visakhapatnam – 530 003, Andhra Pradesh, India, e-mail: ramjidme@yahoo.co.in., Tel: +91 9440670584.
2. Assistant Professor, MVGR college of engineering, Chintalavalasa, JNTU, Vizianagaram.
3. Professor and HOD, Mechanical Engineering Department, College of Engineering, Andhra University, Visakhapatnam – 530 003, Andhra Pradesh, India.

ABSTRACT

In general three wheelers are one of the most popular city transport vehicles in the world and particularly in developing countries especially for short hauls. In the present work finite element model of three wheeled motor vehicle chassis has been developed using ANSYS software by taking into account the chassis longitudinal/cross members, the suspensions, the axles and the tires. Modal analysis is carried on the three wheeled motor vehicle chassis using Block Lanczo’s technique to obtain the natural frequencies and mode shapes. Vertical dynamic response of the three wheeled motor vehicle chassis has been carried in the frequency domain by performing the spectrum analysis using ANSYS software. When the vehicle moving at 45 kmph on different Indian and International roads, and the response is obtained at four significant points i.e. at node 2 (rear part of the chassis sprung mass), at node 8 (passengers seat position), at node 657 (rear axle) and at node 34 (steering position). The response i.e. power spectral density of acceleration is compared between the Indian and International roads and different modes which affect the passenger comfort are also discussed using spectrum analysis. Parametric study has also been carried out using the finite element model to study the effect of various parameters and to find the optimum values both quantitatively and qualitatively, in the ride response of the vehicle. Finally based on the above analysis certain recommendations are made about manufacturer point of view.

INTRODUCTION

The automotive industry is facing tremendous engineering challenges over the several years to meet the requirements generated by the fuel shortage, fuel cost pollution control, raw material cost, and consumer safety. In meeting this challenge a wide variety of new models has emerged or entered into the market in search for better fuel efficiency. As a result, three wheeled motor vehicles have been developed for the purpose of public and private transport, all over the world and is particular in developing countries. Three wheeled motor vehicles, typically used in India and most of the developing countries have their front steering with one wheel similar to those of motor cycles and motor scooters and the two rear wheels are driving wheels with a differential and suspension, which are similar to those of automobiles. The dynamics of three-wheeler, like any other vehicle is characterized by its performance (acceleration and braking), ride (vertical and pitch model) and handling (lateral, yaw and roll model) characteristics. Performance characteristics of road vehicle are primarily concerned with its capability to accelerate, to decelerate and to negotiate grades in straight line motion. The tractive or braking effect and the resisting forces determine the performance potential of the vehicle. Ride characteristics are related to the vibrations of the vehicle excited by surface irregularities and its effects on passengers and goods. Handling characteristics of a road, vehicle is concerned with its response to steering and to environmental inputs effecting the direction of motion of the vehicles such as wind and road surface irregularities. The effect of vehicle vibration response on human comfort and safety is of current importance in ride quality studies. Ride comfort problem
mainly arises due to the surface irregularities, aerodynamic forces, vibration of the engine and drive line and the non uniformities of the tire/wheel assembly.

![Excitation sources](image)

**Fig. 1 Ride dynamic system**

The dynamic response of a passenger vehicle in terms of acceleration and strain has been computed at all nodes by giving PSD of acceleration as input to the tires of a passenger vehicle [1, 2 and 3] using finite element modeling and random vibrations concept. The finite element bus model [3] takes into account the chassis members, the suspensions, and the axle members have been modeled as 3-D beam elements with 6 D.O.F at each node. Frequency response of automotive vehicle structure [4] using random vibration analysis concept with NASTRAN computer program has been studied. The response of a 9 D.O.F dynamic model was obtained when it is moving over rough road. Locomotive dynamic responses [5] to the random excitation from the track irregularities have been obtained using theory of stochastic process and the dependence between input power spectral densities and output power spectral density are also studied here. The response of vehicle [6] has been studied by varying the suspension spring and damping characteristics, tire spring characteristics and vehicle speed. Vehicle response [7] to road irregularities has been studied by assuming two different road surfaces for loaded and unloaded cases. The Eigen values of a typical Indian transport bus [8] has been obtained by using simultaneous iteration method and Lanczo's scheme for two different finite element models i.e. model with chassis and superstructure and model with chassis alone. The finite element technique to the solution of the dynamic behavior of an engine chassis mount bracket [9] has been applied. The finite element stress analysis of three wheeler chassis [10, 11 and 12] has been obtained under critical loads, simulating Indian road conditions by considering dead weight of vehicle, passengers, driver and ground excitations. Static and dynamic analysis has been carried out and suitable modifications have been suggested to prevent failure of chassis frame. The ride dynamic characteristics of typical medium weight high speed, military tracked vehicle [13] subjected to rough cross-country terrain has been studied. The vehicle is modeled using finite element simulation method with beam and shell elements. An eigen value analysis has been done using accelerated subspace iteration method to estimate the natural modes of vibration of the vehicle. The dynamic response of certain salient locations is obtained by carrying out transient dynamic analysis. A finite element idealization of a truck chassis [14] frame was carried using beam elements. The static analysis of chassis is carried under types of loads such as out of plane bending, torsional loads and lateral loads using a computer program. Stress and vibration response of passenger bus [15] has been measured experimentally on typical Indian roads. The analytical results obtained in [3] using FEM and random vibration concepts are compared with the experimental values of [15].

**MODELING**

The model consists of chassis elements, suspension and tires. The chassis frame is modeled as 3-D beam elements of different cross section with 6-DOF at each node. The suspensions and tires are modeled as linear springs with stiffness and damping in the vertical direction alone. The stiffness and damping values of suspension and tires which are used in analysis are shown in Table 1. The mass of the components like engine, gear box, propeller shaft etc. is considered as 3-D mass elements. Mass of these components is lumped at appropriate nodes in y-direction. The individual assemblies are attached to each other by relevant displacement constraints, which specify the interconnection with the adjacent assembly. Following displacement constraints are applied in the present modeling.

1. Revolute joint at the front and rear suspensions.
2. The bottom points of tires are fully arrested except in vertical direction.
3. Nodal coupling is given to the center of the tires and to the end of arms.
Any problem becomes complex, if the real situations are considered, it becomes very difficult to analyze the problem with such complexities. In order to simplify the problem, some assumptions have to be made. In the present analysis the following assumptions are considered.

- The curvature of the chassis members where cross section changes takes place is neglected
- The tires are considered as linear springs with stiffness in vertical direction alone
- The mass of the engine gear box and other components are lumped at appropriate nodes
- Some small members which do not have any significant load carrying function are neglected

<table>
<thead>
<tr>
<th>Description</th>
<th>Stiffness(K) N/m</th>
<th>Damping Coefficient(C) N-s/m</th>
</tr>
</thead>
<tbody>
<tr>
<td>Front suspension</td>
<td>32,700</td>
<td>3500</td>
</tr>
<tr>
<td>Rear right suspension</td>
<td>50,400</td>
<td>2207.5</td>
</tr>
<tr>
<td>Rear left suspension</td>
<td>49,400</td>
<td>2207.5</td>
</tr>
<tr>
<td>Front tire</td>
<td>2,38,260</td>
<td>557</td>
</tr>
<tr>
<td>Rear left and right tire</td>
<td>2,50,490</td>
<td>436</td>
</tr>
</tbody>
</table>

ANALYSIS

After modeling the three wheeler chassis using 3-D beam, combination 14, structural mass elements by defining real constants for each element, material properties and boundary conditions, the response at salient points of the vehicle chassis is obtained by performing spectrum analysis when the vehicle moving at 45kmph on different roads. A spectrum analysis is one where the results of modal analysis are used with a known spectrum to calculate displacements and stresses in the model. It is mainly used in place of time-history analysis to determine the response of structures to random loading conditions such as earthquakes, wind loads, ocean wave loads, jet engine thrust, rocket motor vibrations and so on. In the present work the analysis is performed using power spectral density analysis. A PSD spectrum is a statistical measure of the response of a structure to random dynamic loading conditions. It is a graph of power spectral density versus frequency, where the PSD may be a displacement PSD, velocity PSD, acceleration PSD or force PSD. Similar to response spectrum analysis random vibration analysis may be single point or multi point. In a single point random vibration analysis, only one PSD spectrum is specified at different points in the model. In the present paper single point response analysis has been used.

The power spectral density (PSD) of random process provides the frequency composition of the data in terms of the spectral density of its mean square value. The track input PSD describes the frequency content of track roughness. The relation between the PSD and spatial frequency in general is represented as

$$S_{PP} (\Omega) = C_{SP} \Omega^{-N}$$

Where $S_{PP} (\Omega)$ → PSD in $m^2$/cycles/m

$C_{SP}, N$ → constants and depend upon type of road

$\Omega$ → spatial frequency in cycles/m

The above relation in terms of actual frequency is given by

$$S_{PP} (f) = \frac{C_{SP} f^{-N}}{V^{1-N}}$$

Where $S_{PP}(f)$ → PSD of displacement and $f$ → frequency in cycles/s

$V$ → vehicle speed in m/s
Power spectral density of acceleration which is given as input to the tire contact points is obtained by

\[ S_{PP}(f) = f^4 S_{PP}(f) \]

In the present problem PSD of acceleration is applied as input to the tire contact points. Analysis has been carried to obtain the response of the vehicle, when the vehicle moving at 45kmph on different roads. \( C_{SP} \) and \( N \) values for different roads are as shown in Table 2.

### Table 2 \( C_{SP} \) and \( N \) values for power spectral density for Indian and International roads

<table>
<thead>
<tr>
<th>Indian road data</th>
<th>International road data</th>
</tr>
</thead>
<tbody>
<tr>
<td>( C_{SP} )</td>
<td>( C_{SP} )</td>
</tr>
<tr>
<td>4.9x10^-6</td>
<td>4.4x10^-6</td>
</tr>
<tr>
<td>Highway with gravel</td>
<td>Highway with gravel</td>
</tr>
<tr>
<td>2.2</td>
<td>2.1</td>
</tr>
</tbody>
</table>

**RESULTS AND DISCUSSION**

The objective of the present work is to develop finite element model of three – wheeled motor vehicle for modal and vertical dynamic spectral response (ride) analysis in the frequency domain. Finally parametric analysis is carried out to study the effect of different design parameters on the ride characteristics. The eigen pairs have been determined by using Lanczo’s scheme and mode shapes analyzed. Dynamic response studies in the frequency domain were carried out for the above said three – wheeled model with power spectral density (PSD) of Indian and International Highway With Gavel road profiles is fed as input by treating the road as an ergodic stationary random process. The first few significant natural frequencies and maximum displacements obtained from modal analysis using Block Lanczo’s method for laden condition are tabulated in Table 3 which shows the predominant bounce, pitch, roll, twist and hop mode frequencies. A few mode shapes derived from the FE model are shown in Fig. 2 to Fig. 7 below.

### Table 3 Natural Frequencies of Significant Modes

<table>
<thead>
<tr>
<th>S.No</th>
<th>Type of mode</th>
<th>Frequency in Hz</th>
<th>Maximum Displacement in meters</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Bounce mode</td>
<td>1.958</td>
<td>0.049</td>
</tr>
<tr>
<td>2</td>
<td>Pitch</td>
<td>5.608</td>
<td>0.111</td>
</tr>
<tr>
<td>3</td>
<td>Roll</td>
<td>13.194</td>
<td>0.048</td>
</tr>
<tr>
<td>4</td>
<td>Front hop mode</td>
<td>20.081</td>
<td>0.214</td>
</tr>
<tr>
<td>5</td>
<td>Twist mode of chassis</td>
<td>20.532</td>
<td>0.147</td>
</tr>
<tr>
<td>6</td>
<td>Rear hop</td>
<td>21.326</td>
<td>0.170</td>
</tr>
</tbody>
</table>

In the present work response of the vehicle, i.e. power spectral density of acceleration is obtained at node 2 (rear part of the chassis), at node 8 (passenger seat position), at node 657 (rear axle) and at node 34 (steering position) as shown in Fig.2, when the vehicle is moving at 45 kmph on different roads by performing spectrum analysis. The response of the vehicle i.e. P.S.D of acceleration is compared between road data related to Indian and International Highway with Gravel roads are shown in Figures 8 to 15. From these Figures it is observed that P.S.D of acceleration is high at all nodes for Indian roads as compared with the similar international road data.
Fig. 2  Bounce mode of the three wheeler chassis

Fig. 3  Front hop mode of the three wheeler chassis

Fig. 5  Twist mode of the three wheeler chassis

Fig. 4  Pitch mode of the three wheeler chassis

Fig. 6  Roll mode of the three wheeler chassis

Fig. 7  Rear hop mode of the three wheeler chassis
Fig. 8 P.S.D of acceleration at node 2 for highway with gravel road

Fig. 9 PSD of acceleration at node 2 for Highway with gravel road
**Fig. 10** P.S.D of acceleration at node 8 for Highway with gravel road

**Fig. 11** P.S.D of acceleration at node 8 Highway with gravel road
Fig. 12  P.S.D of acceleration at node 657 for Highway with gravel road

Fig. 13  P.S.D of acceleration at node 657 for Highway with gravel road
Fig. 14 P.S.D of acceleration at node 34 for Highway with gravel road

Fig. 15 P.S.D of acceleration at node 34 for Highway with gravel road
At node 2 (Rear part of chassis): There is abnormal increase in P.S.D of acceleration at bounce mode frequency as shown in Fig. 8 and Fig. 9. Considerable increase in P.S.D of acceleration is observed at pitch and front hop mode frequencies. There is negligible increase in P.S.D of acceleration at roll and rear tramp mode frequencies. Rear hop mode and roll mode have very low magnitudes of PSD of acceleration to the front hop mode. Front hop mode i.e. pitching of the front wheel is predominant when compared to other modes at this node. High values of PSD of acceleration at front hop mode leads to the passenger as well as driver discomfort and therefore in general this type of behavior is not recommended.

At node 8 (Passenger seating position): Considerable increase in P.S.D of acceleration of Indian road is observed at bounce mode frequency which is shown Figures 10 and 11. There is negligible variation in P.S.D of acceleration at front hop, rear hop and rear tramp mode frequencies. From Fig 10 and Fig. 11 it is observed that bounce mode has higher magnitudes compared to other modes at this node. Pitch and roll modes are suppressed at this node. As the pitch mode is suppressed and bounce mode is predominant the passenger will feel comfort since pitch mode is more annoying than the bounce mode.

At node 657 (Rear axle): Figures 12 and 13 shows that there is considerable increase in P.S.D of acceleration at rear hop mode frequency. P.S.D of acceleration increased slightly at rear tramp mode frequency and negligible increase in P.S.D of acceleration at bounce, pitch, and roll mode frequencies. Rear hop mode is highly dominating the other modes at this node. Compared with the rear hop and rear tramp modes, bounce, pitch and roll modes have very low magnitudes of PSD of acceleration. Front hop mode is suppressed at this node. The high values of PSD of acceleration are due to the low stiffness and high damping of rear tire. By increasing the stiffness or by decreasing the damping of rear tire, the PSD of acceleration values can be minimized at rear axle.

At node 34 (Steering position): From figures 14 and 15 it is observed that P.S.D of acceleration increased sharply at pitch mode frequency and considerable increase at front hop mode frequency. There is negligible increase in P.S.D of acceleration at rear hop and rear tramp mode frequencies. Bounce mode and roll modes are suppressed at this node. As the pitch mode is more annoying, it causes driver discomfort. So the PSD of acceleration values has to be minimized. The high values of PSD of acceleration values at this node are due to the low values of stiffness and high values of damping compared with the rear portion of the chassis. Rear portion has two suspensions where as front portion has only one. Increasing the stiffness and or reducing the damping of front suspension can minimize the high values of PSD of acceleration. Pitch mode is the predominant mode at this node.

Parametric Analysis: Parametric study has also been carried out using the finite element model to study the effect of various parameters and to find the optimum values of various parameters, both quantitatively and qualitatively, in the ride response of the vehicle. In order to conduct parametric study, the parameters like suspension stiffness and damping of tire, stiffness and damping of suspension, lumped mass and moment of inertia and vehicle speed, which play a major role in the design of the system, are selected. Then by decreasing or increasing each parameter by 10% and 20% calculated the response of the system at significant points.

SUMMARY AND CONCLUSIONS

A finite element model of the chassis with 57 nodes, 80 elements and 51 D.O.F for the three wheeled motor vehicle has been developed for vertical dynamic behavior study by performing spectrum analysis. The suspension and tires have been modeled as linear springs with stiffness and damping in the vertical direction alone. The eigen pairs have been determined by using Block Lanczo’s method. The various mode shapes such as bounce, pitch, roll, front hop, twist mode of chassis and rear hop mode are analyzed. The response of the three wheeled vehicle has been determined along the length of chassis on different road surfaces. A maximum increase in PSD of acceleration values is approximately 5.83% at rear part of the chassis sprung mass, 25.41% at passenger seating position, 5.31% at rear axle and 17.13% at steering position for Indian Highway with gravel road is observed as compared with the similar international road data.

The parameters that were considered for the study are stiffness and damping of tires, stiffness and damping of suspensions, lumped masses and moment of inertia and vehicle speed. The following conclusions are made from the parametric analysis in order to reduce the PSD of acceleration as it causes the passenger discomfort.
By increasing the front suspension stiffness, PSD of acceleration is decreased in bounce mode frequency for rear part of the chassis sprung mass, passenger seating position and at pitch mode frequency for steering position. So Front suspension stiffness must be reduced for better passenger comfort.

The effect of PSD of acceleration on passenger and driver is reduced by increasing the front tire stiffness. In order to reduce the PSD of acceleration at rear part of chassis and passenger seating position rear tire stiffness must be reduced.

By increasing rear suspension stiffness PSD of acceleration at rear part of chassis and passenger seating position can be minimized.

Passenger comfort can be increased by decreasing lumped masses and moment of inertia acting on the chassis.

Maintaining the medium vehicle speed results in the reduction of PSD of acceleration up to 27.22% at bounce mode frequency for rear part of the chassis sprung mass, 27.21% at bounce mode frequency for passenger seating position, and 27.25% at pitch mode frequency for steering position.

REFERENCES