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DYNAMIC BEHAVIOUR OF THREE-WHEELER PASSENGER VEHICLE USING FINITE ELEMENT METHOD, RIGID BODY MODELLING AND COMPARISON WITH INTELLIGENT DESIGN AUTOMATION

Dr. L. V. Venugopal Rao[1] and Prasanna Priya Chinta[2]

[1]Associate Professor, MVGR College of Engineering, Vizayanagaram, Andhra Pradesh, India
Email: venugopal_l@rediffmail.com
[2]Student, M. Tech., MVGR College of Engineering, Vizayanagaram, Andhra Pradesh, India
Email: prasannapriya84@gmail.com

ABSTRACT

The study of vibration in vehicles was almost exclusively based on test results until a few years ago. Then came the practice of using rigid body model with two parameters namely viz., the sprung mass and unsprung mass. Recently, the vehicle models based on finite element method have become a practical alternative to rigid body models. This approach enables the analyst to generate a model based on structural blue print dimensions and continuously varying displacements impose at the tyre contact points of the vehicle traversing on a road may be considered to be an input excitation of a dynamic system of masses, stiffness and dampers.

The vehicle was descritized by taking into account the chassis element, suspension and tyres. Dynamic behaviour of the three-wheeler vehicle was analyzed by using model, harmonic and transient analysis. The vehicle chassis is descritized by using mass, beam and combination elements. In the rigid body modelling the three-wheeler light passenger vehicle is modelled as a rigid body using sprung mass, unsprung, stiffness and damping elements as vibration elements. The part above the suspensions is modelled as sprung mass; suspension system is modelled using spring and damper. Tyre is modelled using spring and damper and axle including the rim is modelled as unsprung mass.

The method, intelligent design automation (IDA) system for seeks to provide the designer the facility of quickly generating an optimum design, using an artificial intelligence based approach, by inputting requirements thus minimising laborious design computations and iterations. The objective of this work is to conduct a comparative study on three wheeler chassis and its performance based on various performance metrics. In order to execute this task a forward mapping model has been developed using Support Vector

Machines (SVM) for the prediction of chassis design performance metrics. Finally the results of Finite Element Method is compared with rigid body model and Intelligent design automation

INTRODUCTION

The power driven three-wheeled road vehicles, typically used in India on a large scale, are important part of transportation system in major cities and also becoming increasingly popular in smaller towns. The next two decades are likely to witness a sharp rise in the use of three-wheeler. The main compelling reasons for this are scarcity of energy resources and space. Three wheelers also have the advantage of being a compromise between two wheeled and four wheeled vehicles in various aspects like cost, load carrying capacity, fuel consumption, space occupied, weight etc. Any efforts in solving the above problems will directly and/or vehicle dynamics is of great significance and increasing importance. The three-wheeled vehicles operating in India have their front steering with one wheel similar to those of motor cycles and motor scooters, the two rear wheels are the driving wheels with a differential and a suspension, which are similar to those of automobiles.

MATERIALS AND METHODS

Modelling in ANSYS

The model consists of chassis elements, suspension, and tyres

1. The chassis frame is modelled as 3-D beam elements of different cross-section with 6 D.O.F at each node.
2. The suspensions and tyres are modelled as linear springs with stiffness in the vertical direction alone. The stiffness and damping values are taken as mentioned in table.

Table 1. Stiffness and Damping value

Description	**Stiffness** (K) (N/m)	**Damping Coefficient** (C) (N-s/m)
Front Suspension	32,700	3,500
Rear Right Suspension	50,400	2,207.5
Rear Left Suspension	49,800	2,207.5
Front Tyre	2,38,260	557
Rear Right Tyre	2,50,490	436
Rear Left Tyre	2,50,490	436

Table 2. Sheet Properties

Young's modulus N/m^2	**Density Kg/m^3**	**Thickness (m)**
196.2E09	7890	0.015

Table 3. Mass Distribution

S.No	**Node Number**	**Lumped Mass** (kg)
1)	1,2	16.275
2)	28,29	56.675
3)	8,9	32.127
4)	38,39	10.98 (5.49+5.49)

Table 3. Mass Distribution (Contd….)

S.No	Node Number	Lumped Mass (kg)
5)	5	42.695
6)	37	8.46(4.3+4.3)
7)	1	35.525
8)	27	22.8 (7.6+7.6+7.6)

Modelling of Rigid Body Model

The 6-D.O.F of the three-wheeler vehicle system is modelled according to the idealizations quoted.[11].. Front and rear tyres are represented by springs with damping. The independent suspensions of the vehicle are represented by the unsprung masses "murl" and "murr". "muf. " denotes the front unsprung mass. All the 6-D.O.F's are clearly represented in the fig (5.14) It includes the bounce "z_s", pitch "ϕ"and roll "θ" of sprung mass and bounce of front unsprung mass "ξ_f", bounce of the rear left unsprung mass"ζ_1"and bounce of rear right unsprung mass "ζ_2". The body of the vehicle is treated as a sprung mass "ms",which is supported by front and independent rear suspension springs with damping.

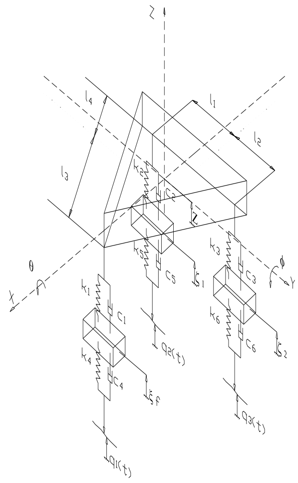


fig 4. 6 D.O.F MODEL

Intelligent Design Automation Method

The estimation of performance metrics for three wheeler chassis, requires representation of the influence of design parameters on design performance metrics and estimation of the optimum design parameters that would yield a required performance. This involves a combination of regression mapping and optimisation. SVM and Genetic Algorithms (GA) are used in this concept to build a forward-reverse mapping model for optimisation chassis design.

The chassis design parameters are sent to the software-based forward mapping model that has been built to map the relationship between the design parameters and performance metrics. For the given operating conditions, the forward mapping model predicts the performance. The desired optimum performance is defined so that the designer always operates under these conditions. This is achieved by linking the artificial intelligence based reverse mapping model with the forward mapping model. The reverse mapping model estimates the optimum performance metrics from a range of candidate design parameter solutions that satisfy the requirement.

Support Vector Machine (SVM)

The foundations of Support Vector Machines (SVM) have been developed by Vapnik (1995) and are gaining popularity due to many attractive features, and promising empirical performance. This is a consequence of the optimization algorithms used for parameter selection and the statistical measures used to select the 'best' model

SVMs were developed to solve the classification problem, but recently they have been extended to the domain of regression problems (Vapnik et al., 1997). Support Vector Machines (SVM) is a technique that performs classification by constructing an N-dimensional hyper plane that optimally separates the data into two categories. SVM models are closely related to neural networks. In fact, a SVM model using a sigmoid kernel function is equivalent to a two-layer, feed-forward neural network.

PROCEDURE

Analysis of model in ANSYS

In the model analysis different mode shapes like bounce, roll, front hop, and yaw were found and observed the obtained displacement using the model analysis. In the harmonic analysis the change in displacement with frequency over the entire range (0 – 80 Hz) was obtained. For transient analysis a semi circular bump was taken as excitation force and noticed the variations in displacement.

Rigid Body Model

Once the original system is approximated to a mathematical model, the principles of dynamics are used to derive the equations of motion that describe the vibration of the system. The equations of motion can be derived conveniently by drawing the free body-diagram of all the masses involved. The free body diagram of a mass can be obtained by isolating the mass and indicating all the externally applied forces, the reactive forces and the inertia forces. The equations of motion of a vibrating system are usually in the form of a set of differential equations for a discrete system and partial differential equations for a continuous system. The derived equations can be represented in the matrix from

$$[M][\ddot{q}]+[C][\dot{q}]+[K][q] = [F]$$

In the present analysis matrix method is applied for solving the equations of motion. In the present case 6-D.O.F, model is subjected to frequency domain simulations it's response is obtained in terms displacement or dynamic tyre force.

Determination of Transfer Function for Displacement Response:

Transfer function is defined as the ratio of output response for a given input. In the present analysis it is defined as the ratio of output displacement for a given input road undulation. The expression for the transfer function is derived for the 6 – D.O.F model.

The basic assumption is that the displacement characteristics including the input road profile are all harmonic functions. Mass, stiffness, damping, displacement and force matrices for a six–degree of freedom model are given below.

This equation reduces to the following form

B. H (ω) = $\overline{F}$

where B (ω) $= -\omega^2 [m] + i\omega [c] + [k]$

and $H(\omega) = \frac{[\overline{q}]}{\overline{u}}$

H (ω) is the transfer function which is a non – dimensional parameter. For various frequencies ranging from 0 to 81, the transfer function is computed and a graph is plotted with frequency on x-axis and transfer function on y – axis.

Vehicle Simulation for Calculation of Dynamic Tyre Force Response

Dynamic tyre force is defined as the response of the tyre for the road input (i.e.,) The unspruing mass which is in contact with the spring damper first experience the force. The force experienced by the spring and damper which is in contact with the road is given by

$$F_t = (k_t + i\omega \, . \, c_t) \, .(\overline{z}_u - \overline{u}). \, e^{i\omega t}$$
$$= \overline{F}_t \; e^{i\omega t}$$

$\overline{F}_t$ is known as vector of complex dynamic tyre force .Also the actual force or generalised force expression from equation Can be expressed as

$$\overline{F} = [D] \; [R] \; \overline{u}$$

The expression can also be expressed in terms of D, R, q and u as

$$\frac{\overline{F}_t}{\overline{u}} = H(\omega) \; or \overline{F}_t = H(\omega).\overline{u}$$

$$\therefore \; \frac{\overline{F}_t}{\overline{u}} = [R]\,(\,[D]^T [B^{-1}]\,[D]\,[R] - I\,)\,\overline{u}$$

$$H(\omega) = [R]\,(\,[D]^T [B^{-1}]\,[D]\,[R] - I\,)$$

Where I - Identity matrix and

H(ω) – Transfer function

Transfer-function is thus defined here as the ratio of dynamic tyre force vector to the input road undulation. Transfer function in this case is a dimensional parameter having the dimensions of force / displacement (i.e.,) N/m

If the dynamic tyre force is to be evaluated, transfer function is multiplied with the input rector (i.e)

$$\overline{F}_t = H(\omega)\,.\overline{u}$$

The transfer function is calculated for different frequencies ranging from 0 to 81 Hz and a graph is plotted with x – axis frequency and y – axis transfer - function.

Development of Prediction Model in SVM

In this thesis, an attempt is made to develop a forward mapping model using SVM to predict the design of chassis. The SVM model is developed by assigning a few parameters namely the kernel function, the cost function etc. The goal of SVM is to find out a function that gives a deviation of error from the actual given output and at the same time is as flat as possible. This is achieved by mapping the training patterns from the input space to a high dimensional feature space in such a way that the data which could not be separated by a linear function in the input space can be separated in the feature space. Thus, in arriving at a suitable SVM Model for the given sparse data, the only parameters that the user deals with and has to specify are the kernel function, type of loss function, the error goal, the constant and the width of the radial basis function. This makes it convenient to use SVM in the prediction of the output variable for a given combination of input variables especially in a situation where collection of data for training the model is difficult.

In order to develop a prediction model a program has been developed and executed in Matlab. Input parameters for the SVM model were taken as stiffness of material, damping coefficient, displacement, frequency in harmonic analysis, time in transient analysis and the output parameters were taken as response of the chassis.

RESULTS AND DISCUSSION

Modal Analysis

The first few significant frequencies obtained by the "Block Lanczos" method are tabulated in the table 5.1 Which shows the predominant bounce, front hop, roll, yaw and pitch modal frequencies for the three-wheeler vehicle model.

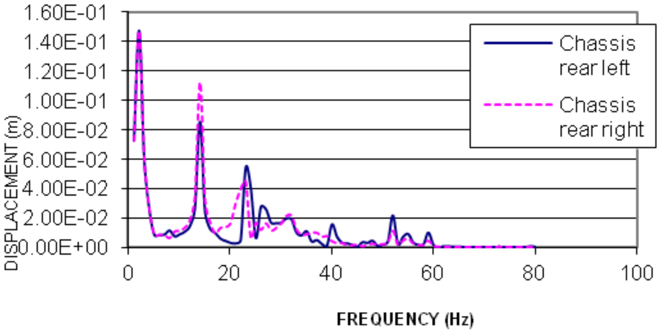
Table 4. Natural frequencies of significant modes

S.No	Type of Mode	Frequency (Hz)	Maximum displacement (m)
1	Bounce mode	1.99	0.054
2	Front hop mode	3.48	0.11
3	Roll mode	8.33	0.13
4	Yaw mode	16.55	0.20
5	Pitch mode	21.45	0.47

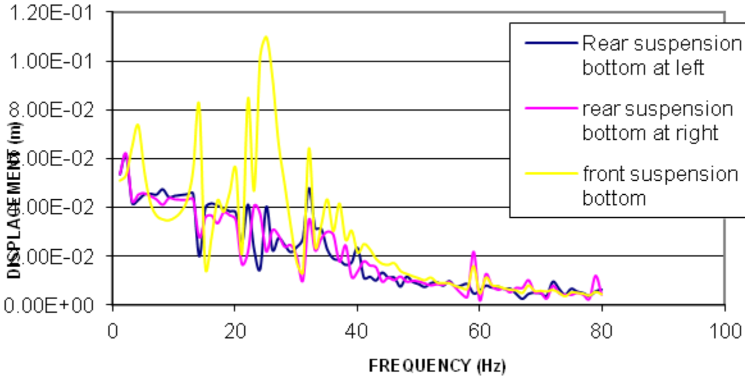
Harmonic Analysis

From this analysis, the displacements of various nodes and stresses for different elements over the entire frequency range 0 – 80Hz with amplitude of 0.05m were obtained. The displacements at various nodes were plotted against frequency and are shown in figures 4.1.2(a) and 4.1.2(b).

The variation in displacement at the left and right end of the rear chassis was represented in Fig 4.1.2(a). Maximum displacement was obtained at the right end of the rear chassis. The variation in the displacement values at the suspension bottoms at the rear and front suspensions were represented in Fig 4.1.2(b). The maximum displacement was observed at the front suspension bottom at the third peak (25Hz).



4.1.2(a) VARIATION IN DISPLACEMENT AT LEFT AND RIGHT ENDS OF THE REAR CHASSIS



4.1.2(b) VARIATION OF DISPLACEMENT AT SUSPENSION BOTTOM

Transient Analysis

For transient analysis the model is simulated when the vehicle is moving at 40 kmph over a semi-circular bump of 100mm diameter. The total time taken by the vehicle to cross over the bump is 0.295sec. This total time is divided into 112 sub steps in this analysis.

The result that was obtained from the analysis is presented in the form of displacements at different nodes on the chassis. The displacement at the right end of the rear chassis is on the higher side compared to the rear chassis as shown in Fig. 4.1.3 (a). The maximum displacement occurred at 0.38sec with a magnitude of 0.5916m.

The vertical displacement experienced by the suspension bottoms at all the three wheels are shown in Fig.4.1.3(b) which shows the maximum displacement value was experienced by the front suspension bottom which is having a displacement value of 0.54m at 0.16sec.

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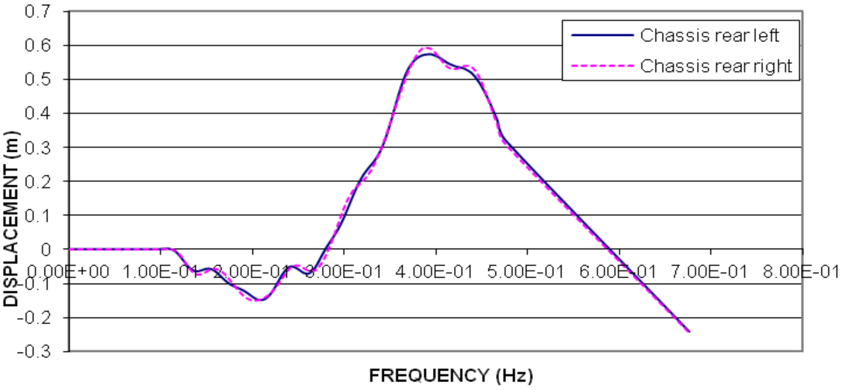
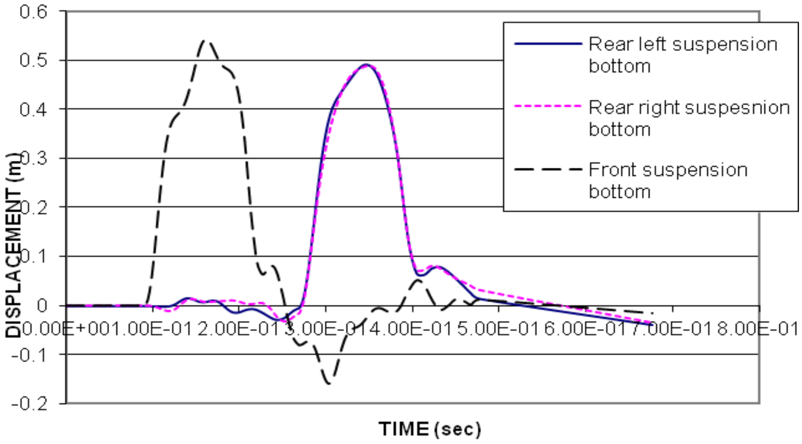


4.1.3(a) VARIATION IN DISPLACEMENT AT LEFT AND RIGHT ENDS OF THE REAR CHASSIS



4.1.3(b) VARIATION IN DIPLACEMENT AT SUSPENSION BOTTOM

Rigid Body Model

Displacement response transfer function

At the extreme refinement of the present analysis, the 6-D.O.F model gives 6 mode i.e., sprung mass-bounce, sprung-mass roll, sprung mass pitch, unsprung mass bounce for rear left and right, front unsprung mass bounce as represented in graph. The highest peak quoted in the figure is obtained for Sprung mass-bounce at a frequency of 2 Hz and amplitude of 1.74. Sprung mass-rolls with a maximum value of 0.02 at 2 Hz. Amplitude of Sprung mass-pitch is 0.50 at 3 Hz. Unsprung mass bounce for rear left and rear right are equal and hence is considered as rear hop which occurs at a frequency of 2Hz and amplitude of 0.50. Front unsprung mass bounce also known as front hop occurs at a frequency of 7 Hz and amplitude of 1.14.

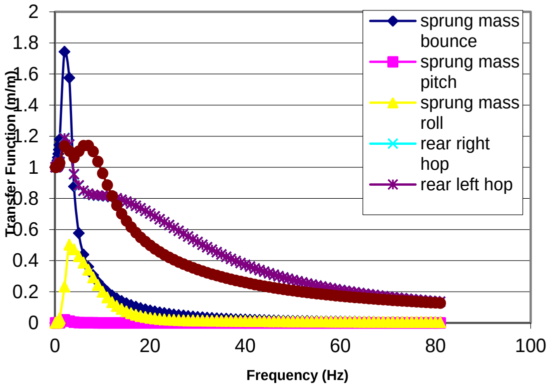


fig.4.2.1 Overall Displacement Response For 6 D.O.F

Dynamic tyre force response transfer function The value of dynamic tyre force response increases rapidly up to a frequency of 10 Hz. The increment in the dynamic tyre force is nearly linear up to a frequency of 3 Hz. Beyond 10 Hz of frequency the variation is not rapid.

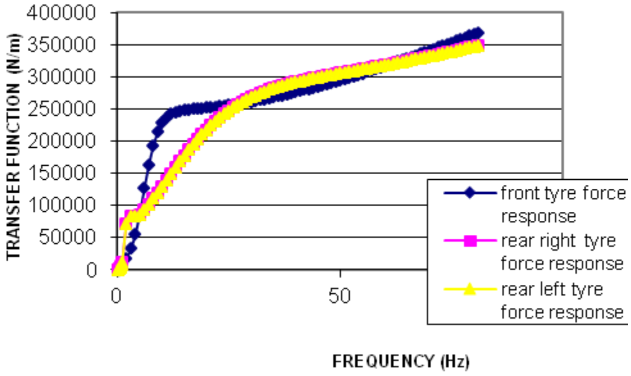


fig.4.2.2 DYNAMIC TYRE FORCE RESPONSE FOR 6-D.O.F.

Response using SUPPORT VECTOR MACHINE

The variation in displacement at the left and right end of the rear chassis was represented in Fig A. Maximum displacement 0.15m was obtained at the right end of the rear chassis at frequency 2Hz. The variation in the displacement values at the suspension bottoms at the rear and front suspensions were represented in Fig B. The maximum displacement 0.12m was observed at the front suspension bottom at the third peak (25Hz).

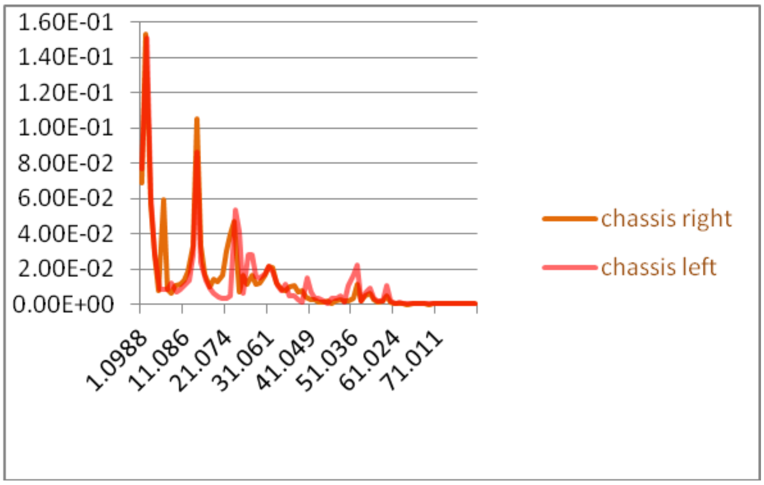


Fig A. Variation In Displacement In Rear Chassis

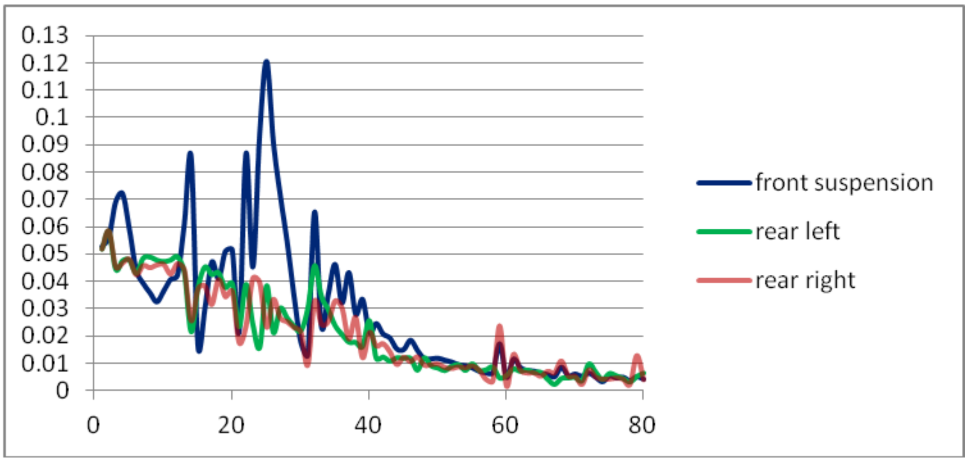


Fig B. Variation in Displacement In Front And Rear Suspensions

CONCLUSIONS

1. The peak value for both rear left and rear right suspension for 6-D.O.F model obtained at a frequency of 2Hz is 0.059m. When this value is compared with finite element model the error is 4.5%
2. The maximum displacement 0.1474m was observed in rear right chassis in harmonic analysis. Also in the SVM method maximum displacement is observed in rear right chassis. The displacement is 0.153m, when compared with finite element model the error is 3.6%
3. The maximum displacement was observed in front suspension system in both finite element method and SVM method with an error 3.49%
4. The dynamic tyre force response increases sharply and then gradually no peaks were accounted.

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