Dynamic Behaviour of Three-Wheeler Light Passenger Vehicle Using Rigid Body Modeling

K.RAMJI*, S. SRINIVASA RAO and M. KANNAM NAIDU*****

*Associate professor, Dept. of Mech. Engineering, Andhra University, Visakhapatnam-3, A.P.
Ph: 0891 - 2797897 (R), 0891 - 2844816 (O), E - Mail: ramjidme@yahoo.co.in
** Asst Prof., Dept. of Mech. Engineering, Sir CRR College of Engg,, Eluru, W.G.Dist.
*** Asst Prof., Dept. of Mech. Engg., MVGR College of Engg,, Vizyanagar, VZM (Dist.).

ABSTRACT

In the present paper, a three-wheeler light passenger vehicle was considered and five simple mathematical models v.i.z, 2 Degrees of Freedom (D.O.F.) model, 4-D.O.F model, 6-D.O.F model for the entire vehicle, 3-D.O.F model for rear portion and 2-D.O.F.model for front portion were developed to study the dynamic behaviour of the vehicle systems. These models will allow the simulation of many vehicle structures like light, medium or heavy passenger vehicles. The vibration characteristics of the vehicle systems i.e., displacement response of sprung mass and complex dynamic-tyre force response are computed in frequency domain using the rigid body models. Here, the input to the system is the base excitation i.e., road undulation. The inputs and corresponding outputs are assumed as harmonic functions. The variation of the response of the system with change in frequency is studied for all the models under consideration. Displacement Response of 6 D.O.F. model is on the higher side of all the remaining models, except for 4-D.O.F.model. Dynamic-tyre force response increases with increase in tyre stiffness.

1. 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 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.

Ride problems mainly arise from vibrations of the vehicle body, which may be induced by a variety of sources, including surface irregularities, aerodynamic forces, vibrations of the engine and drive line, non-uniformities of the tyre-wheel assembly. Usually, surface irregularities acts as a major that enriches the vibrations of the vehicle body through tyre-wheel assembly and the suspension system. In surveying the Literature very few publications on three wheelers are found, which mostly deal with the stability and handling characteristics. Various works presented in the dynamic behaviour of the different vehicle systems [2,4,8] were studied. Also several researches [5,6,7] dealing with the three-wheeler vehicle system is also studied.

2. MODELING

As a first step of idealization the entire model is approximated using simple vibration elements. The present system under consideration is a three-wheeled Bajaj-rear engine model, which is approximated to a mathematical model by using spring, damper and mass elements. The original system is modeled as 2-D.O.F. system in the initial stage. Further refinement of the original system has been done by modeling the entire system as 3-D.O.F.model for rear part and 2-D.O.F.model for front part, 4-D.O.F.system and 6-D.O.F.system for the entire vehicle. Idealization of the actual vehicle model to a corresponding mathematical model for 2-D.O.F and 3-D.O.F is given by David Cebon [1] and it is extended here. Several researches [3,7] done in the development of mathematical model for 4-D.O.F.model and 6-D.O.F. system are extended to the present vehicle.

❖ 2-D.O.F.model of three-wheeler vehicle

2-D.O.F rigid body model has two masses, sprung mass and unsprung mass, which were constrained to move in vertical direction. 2-D.O.F associated with the system are sprung mass bounce and unsprung mass bounce. Diagrammatic representation of this model is shown in the Fig 1.

❖ 4-D.O.F. model of the entire vehicle system

Rigid body model for 4-D.O.F. presented in paper [1] is extended for the vehicle system under consideration. The model has a sprung mass, which moves with vertical displacement and pitch of rotation. The remaining two displacement characteristics are bounce of the rear effective unsprung mass and front unsprung mass bounce. Front unsprung mass is connected to the suspension spring at the top and tyre spring at the bottom. Whereas for the rear part the two rear tyres stiffness, damping and unsprung masses are combined to form effective tyre stiffness, damping and mass respectively. This effective stiffness, damping and mass are respectively considered as rear unsprung stiffness, damping and mass. Similarly the rear suspension system is modeled and schematic representation is given in Fig 3.

❖ 6-D.O.F. model

The 6-D.O.F of the three-wheeler vehicle system is modelled according to the idealizations quoted in the paper [11]. Front and rear tyres are represented by springs with damping. All the 6-D.O.F's are clearly represented in the Fig 4. It includes the bounce, pitch and roll of sprung mass and bounce of front unsprung mass, bounce of the rear left unsprung mass and bounce of rear right unsprung mass. The body of the vehicle is treated as a sprung mass, which is supported by front and independent rear suspension springs with damping. Various parameters related to tyres like dynamic rolling tyre stiffness, damping, mass and inertia parameter of the above vehicle are taken from papers [5,6,10].

❖ 2-D.O.F.model & 3-D.O.F. model of three-wheeler system

The entire system is modeled as 3-D.O.F.model for rear portion with 2/3rd part of total sprung-mass and 2-D.O.F.model for front portion with 1/3rd part of the total sprung-mass. The schematic representation of the 3-D.O.F rigid body model showing sprung-mass bounce, bounce and pitch of rotation of the unsprung mass is presented in the Fig 2.

3. ANALYSIS

The equations of motion of a each 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 equation may be linear or non-linear, depending on the behaviour of the components of the system. Newton's second law of motion is here used for deriving equations of motion. The derived equations represented in the matrix form is given in the following equation.

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

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 two - degrees of freedom model and the same can be extended to apply for 3-D. O.F, 4- D.O.F and 6 - D.O.F models. Dynamic tyre force is defined as the response of the tyre for the given road input (i.e.) the unsprung mass which is in contact with the spring damper first experience the force. Transfer-function is thus defined as the ratio of dynamic tyre force vector to the input road undulation.

4. RESULTS AND DISCUSSION

4.1 Displacement response transfer function

The rigid body modeling is simple and can be used for preliminary analysis of ride and force characteristics of various vehicle systems. Vibration analysis of a three-wheeled light passenger vehicle has been done using five rigid body models. The vibration response has been computed in frequency domain for harmonic input, given for five three-wheeled vehicle models. The results are presented in terms of displacement and force transfer function of the sprung mass and tyres. Two-degree of freedom model gives two rigid modes, one is the sprung mass bounce and the other is the unsprung mass bounce and is shown in Fig 5. Maximum peak for bounce of the sprung mass occur at a frequency of 2 Hz with a magnitude of 1.70 and the corresponding maximum amplitude for the unsprung mass bounce is 1.17.

In the second case, 2-D.O.F.model for front part gives the magnitude of the sprung mass bounce at a frequency of 2 Hz as 1.40 and unsprung mass bounce or front hop occurs at 3 Hz with amplitude of 1.18. In the next case i.e., 3-D.O.F model for rear part, the peak of sprung mass bounce occurred at a frequency of 2Hz is 1.84. Also unsprung mass bounce occurs at the same frequency but of lower magnitude 1.19. The unsprung mass pitch has no significant role, as its value is negligibly small. Schematic representation for the above stated models is shown in Figs. 5 & 6 respectively. The next stage of the refinement for the entire system is 4-D.O.F model and it's displacement responses are represented in Fig 7. From the figure the sprung mass bounce occurs at a frequency of 2 Hz and 2.28 magnitude. And the amplitude of the pitch of the sprung mass occurs is 1.6 at a frequency of 4 Hz. Rear unsprung mass bounce occurs at a frequency of 2 Hz and amplitude of 1.25 and it's also known to be as rear hop mode. Front hop/ front unsprung mass bounce is obtained at 7 Hz with a peak of 2.97.

At the extreme refinement of the present analysis, the 6-D.O.F model gives 6 modes 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 Fig 8. 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. It is observed that the sprung mass bounce obtained in six degrees of freedom model is more accurate due to refinement in mathematical modeling. Also the sprung mass bounce for two degrees of freedom model is in good agreement with this model. Due to the varied contribution of sprung mass given for the two degrees of freedom front wheel and 3-D.O.F rear wheel model, the sprung mass bounce varies largely as compared with 2-D.O.F and 6-D.O.F. If the respective contributions are taken from the frequencies and amplitudes, the bounce occurs at same frequency but amplitude shifts to 1.69 and it is just nearer to 2-D.O.F. model value. Sprung-mass bounce mode for 4-D.O.F is occurred at the same frequency but of higher amplitude. The reason is due to the combining of the rear independent suspension system. Unsprung mass bounce of rear axle for 3-D.O.F rear-part model is in good agreement with that of the 6-D.O. Fmodel.

4.2 Dynamic tyre force response transfer function

The force vs. frequency graph for the five models is shown in the Figs. 9,10,11,12,13. For all the models, 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 slow and gradual for all the cases. The front tyre responses for 4-D.O.F. and 6-D.O.F are approximately same at the frequencies of 3 Hz, 10 Hz, 20 Hz and 30 Hz. The rear dynamic tyre force response for the 6-D.O.F model is nearly one half of the rear dynamic tyre force of 4-D.O.F.model. The reason is due to the combining the two rear-tyre stiffness and damping. When the dynamic tyre force value of 3-D.O.F. rear-part model is compared with the 6-D.O.F. full model, the earlier shows the variation with the later at the mean position.

5. CONCLUSIONS

The displacement response for all the models shows a peak and it indicates dominance of the corresponding displacement. Bounce for every model occurred at the same frequency. Except 4 D.O.F, all models are in good correlation, in the case of displacement response. The dynamic tyre force response increases sharply in the initial phase and then raises gradually but no peaks were accounted. In all the cases, it is observed that as stiffness of the tyre increases dynamic tyre force response increases.

7. REFERENCES

1) **David Cebon,** "Hand Book of vehicle - road Interaction", Swets & Zeitlinger publishers, 2000.

2) **Hisanori Tachibana, Kazuhiko Gotoh, and Hitomi Sakai.** "Vibration Analysis of drive line and suspension using finite element models", Toyota motor corp., paper no. 931306.

3) **Karuppaiah.N, Deshpande.P.S, C.Sujatha and Ramamurti.V(2003).** "Vibration Analysis in a Light Passenger Vehicle by Rigid Body / Finite Element Modelling", pp 106-120.

4) **Lawrence Selna and David Salinas.** " Dynamic analysis of automotive structural systems ", school of engineering and applied science, university of California, Los Angeles, paper no.700844.

5) **Mukul Shukla, Saravana kumar.G, Siva Prasad.K, Santosh kulkarni CAD-P Laboratory, Mechanical Engg Department IIT Kanpur** (2003). "Development of a new chassis for vikram 3 wheeler", pp 1- 12.

6) **Mukul Shukla, Saravana kumar.G, Siva Prasad.K, Santosh kulkarni.** "Finite Element Analysis of 3 Wheeler Chassis".

7) **Murali Krishna.M, Ramji.K, Goel.V.k, Deep.K.** "Optimization of three-wheeled motor vehicle characteristics from the aspect of ride comfort using random search technique", presented and published at 18th National convention of Mechanical Engineers, Institute of Engineers, (India) local center, Rourkela-2002.

8) **Ramamurti.V and Sujatha.C** (1990), " Bus vibration study-experimental response to road undulations", International Journal of Vehicle Design, vol.11, No.4/5.pp 390-400.

9) **Ramji.K, Goel.V.K and Saran.V.H,**" stiffness properties of small-sized pneumatic tyres ",International multidisciplinary Journal, proc. Institution of Mechanical Engineers, Journal of the automobile engineering. (part-D), U.K.,Vol.216, 107-114.

10) **Ramji.K, Goel.V.K, Murali Krishna.M,** (2002). "An assessment of ride quality of three-wheeled vehicles", presented National seminar on modern trends in automobiles, Islampur,228-235,R.I.T., Rajaramnagar.

11) **Rao.S.S,** "Mechanical Vibrations ", published by Pearson Education (Singapore) pte. L td. 2004.

12) **Saran.V.H. and Goek.V.K.** (2000), "Rolling dynamic stiffness and damping characteristics of small sized pneumatic tyres ", *proc.* . Institution of Mechanical Engineers, (part-D), 214(3), 243-248.

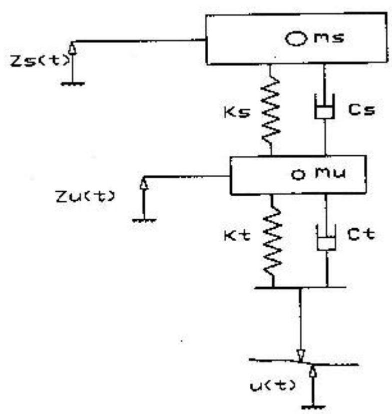

fig 1. 2-D.O.F. MODEL

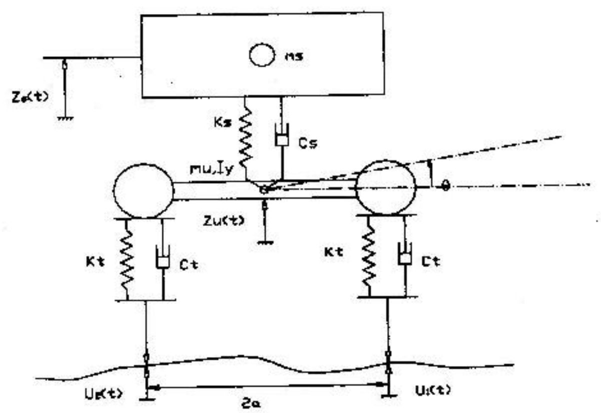

fig 2. 3 D.O.F MODEL

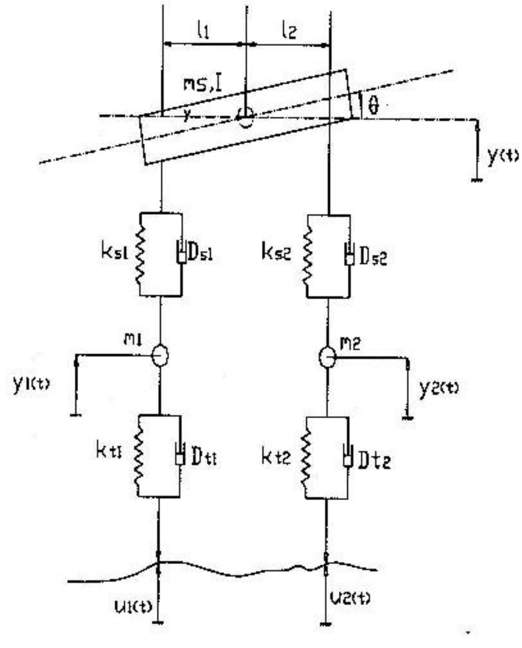

fig 3. 4 D.O.F MODEL

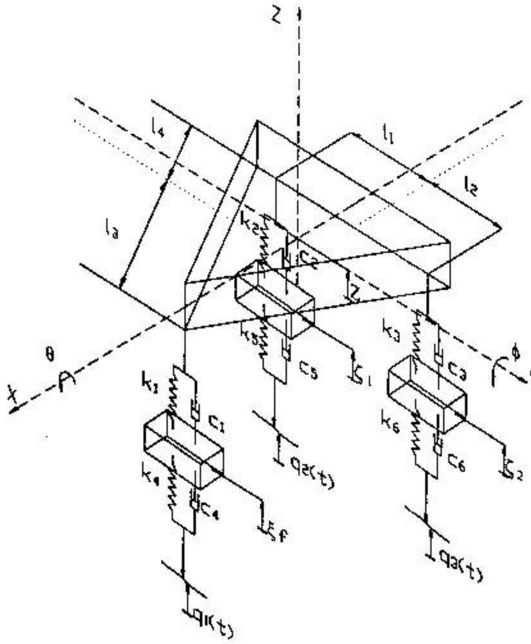

fig 4. 6 D.O.F MODEL

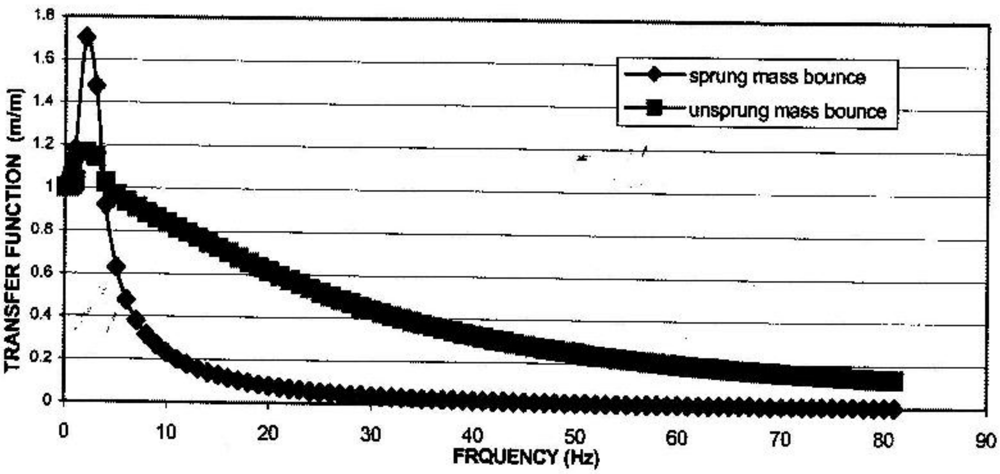

Fig. 5 DISPLACEMENT RESPONSE FOR 2 D.O.F

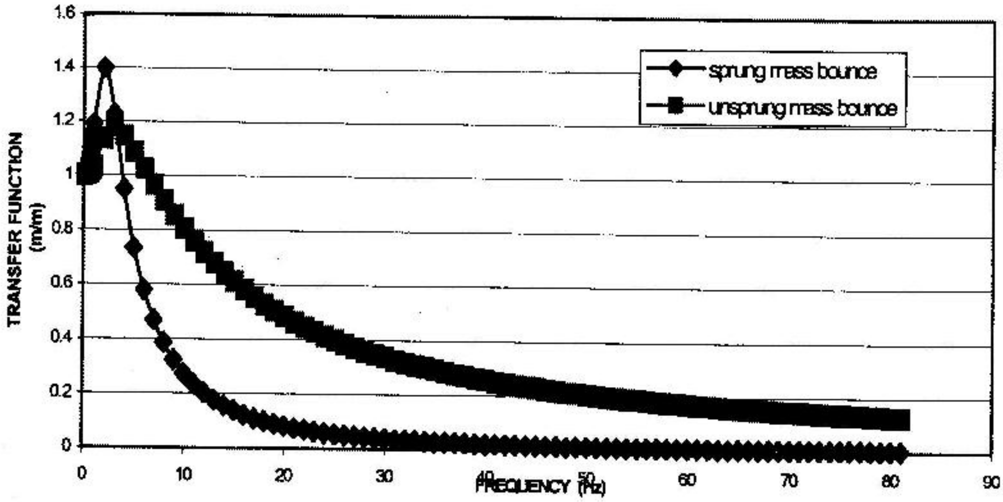

Fig. 6 DISPLACEMENT RESPONSE FOR 2 D.O.F.model FOR FRONT PART

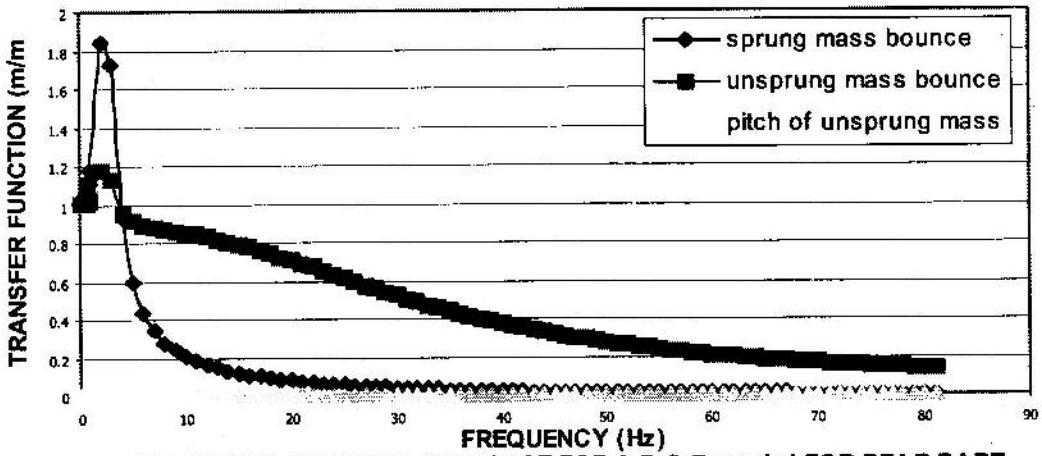

Fig. 7 DISPLACEMENT RESPONSE FOR 3-D.O.F.model FOR REAR PART

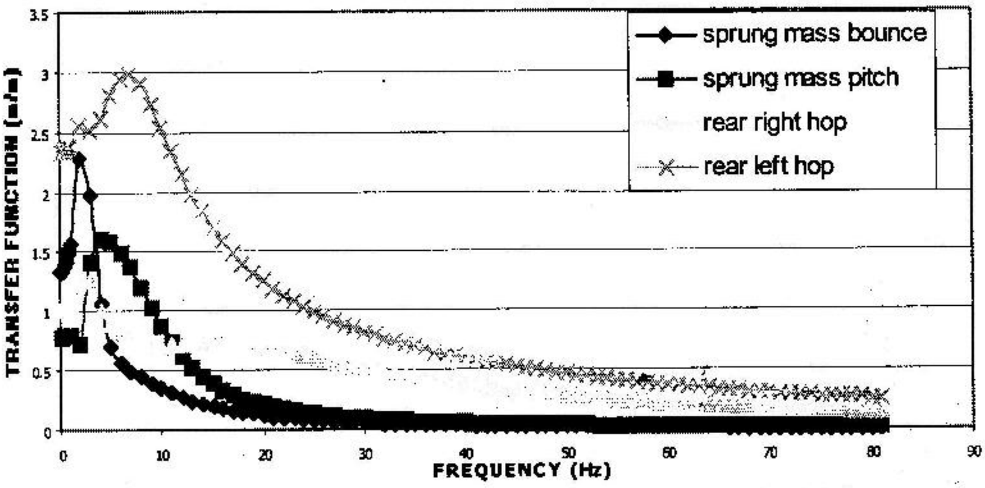

Fig. 8 OVERALL DISPLACEMENT RESPONSE FOR 4 D.O.F model

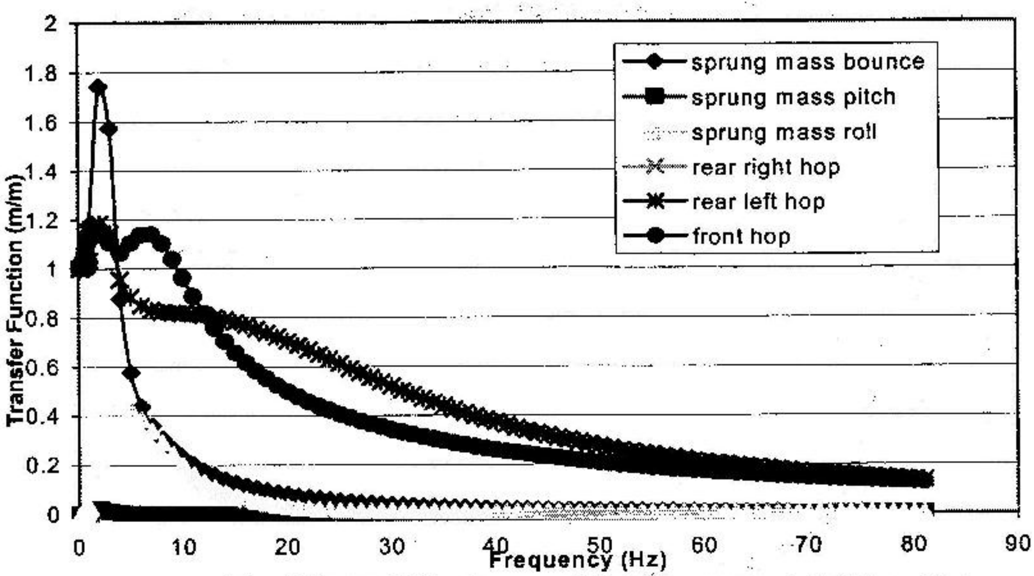

Fig. 9 Overall Displacement Response For 6 D.O.F model

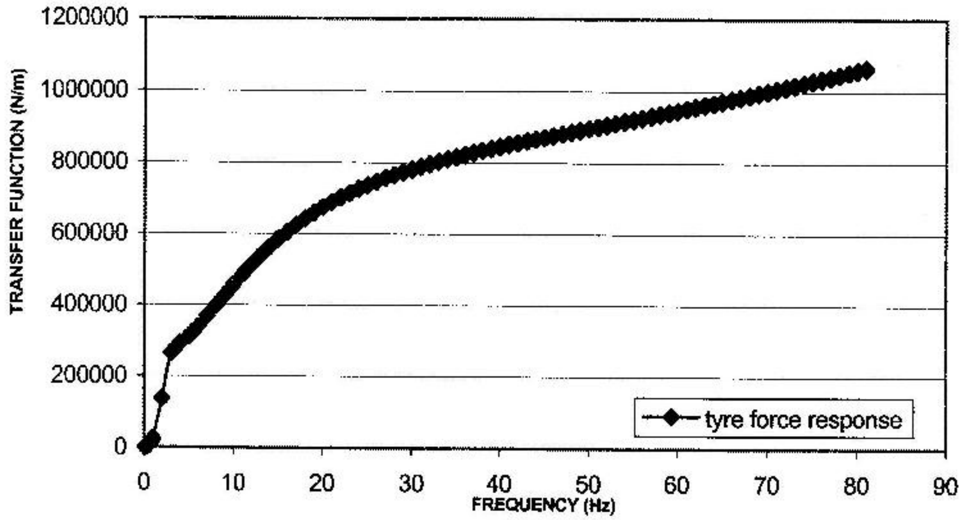

Fig.10 DYNAMIC TYRE RESPONSE FOR 2 D.O.F

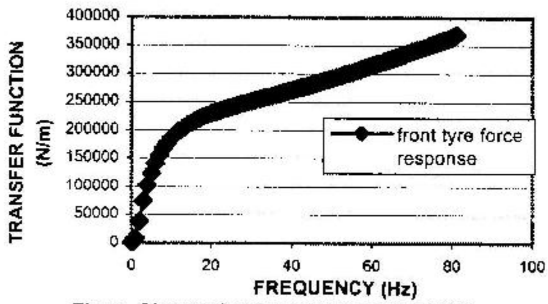

Fig.11 DYNAMIC TYRE FORCE RESPONSE

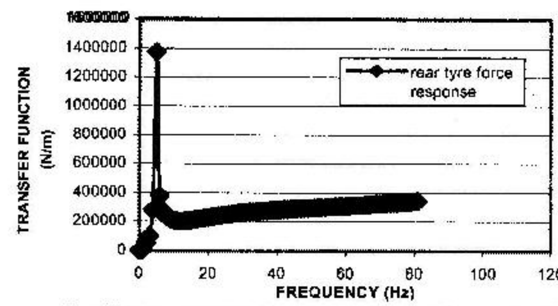

Fig. 12 DYNAMIC TYRE FORCE RESPONSE FOR 3- D.O.F REAR PART

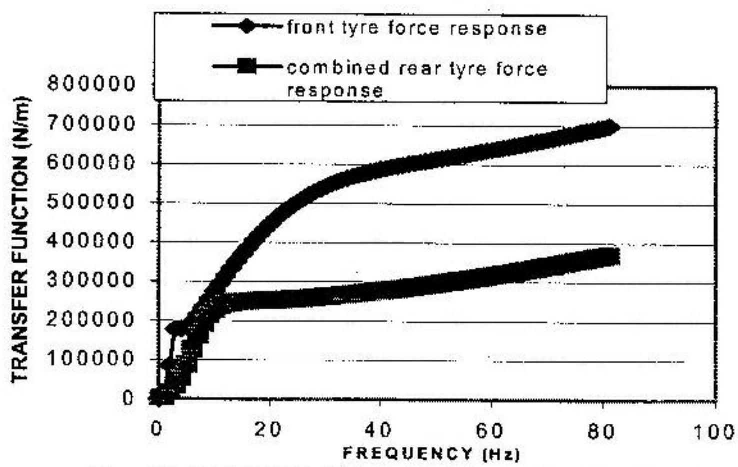

Fig. 13 DYNAMIC TYRE FORCE RESPONSE FOR 4-D.

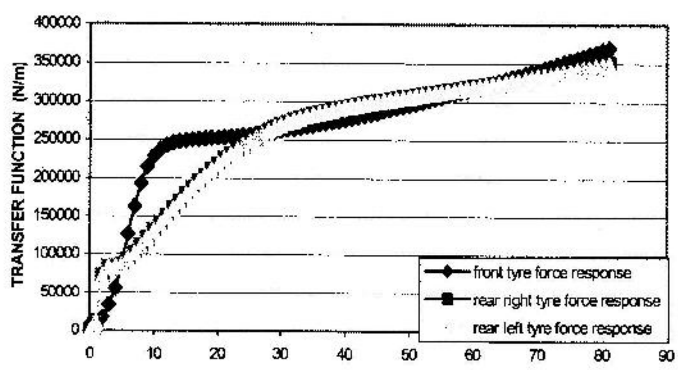

Fig.14 DYNAMIC TYRE FORCE RESPONSE FOR 6-D.O.F.