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Dynamic Response Analysis of a Three Wheeler Chassis Frame using Finite Element Analysis

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Abstract: A three-wheeler is essentially a motion disturbance vibrating system with the ground providing input. The tyres which are in contact with the ground, convert this input displacement into a forcing function that force acts on the un sprung masses, linkages will constrain these masse so that they follow certain paths. Ride response will vary with variation in the tyres, suspension characteristics, the ground profile, physical dimensions and the inertial properties of the sprung and unsprung masses of the vehicle. Analyzing the dynamic behavior of vehicle using Finite Element modeling is the main objective of this thesis .This project also includes the response of the three wheeler chassis frame to road surface inputs and to provide the best vehicle in vibratory motions. In Finite element modeling, analysis is done to determine the dynamic behavior of the vehicle in Harmonic and Transient excitations. Parametric study and modal analysis and has also been carried out. In order to get the maximum stresses, as the vehicle running on the road is subjected to the random vibrations spectrum analysis is carried out. We studied the results obtained from Finite element modeling in modal analysis and are compared with those of rigid body modeling. In Transient analysis excitation at the key nodes magnitude and acceleration values are analyzed. Maximum stress point is noted from spectrum analysis. To improve the performance of vehicle conclusions were derived from the study and suggestions are made, thus stability is established.

Keywords: AGVs, FE, RE.

I. INTRODUCTION

A. Introduction to Three Wheeled Vehicle

With the growing fuel costs, significant pressure has been put on automotive industries to develop more fuel-efficient passenger vehicles. Due to safety, emissions and economy requirements, automotive transportation will be undergoing vast changes in the next few decades. Obviously, the automobile will become smaller and lighter in the search for better fuel efficiency. As a result, the designs based on three-wheeled motor vehicles are likely to be the most popular mode of public and private transport not only in India but in other countries as well. The power driven three-wheeled road vehicles, typically used in India on a large scale, in major cities these are important part of transportation system and in smaller towns also becoming increasingly popular. Some of these are designed for commercial use with little engineering considerations and are marked as product in great demand. Where as, others fabricated more as improvisation. There is a sharp rise in the use of three-wheeler in next two decades. Scarcity of energy resources and space is the main compelling reasons for this. Three wheelers also have the advantage of being a compromise between two wheeled and four vehicles in various aspects like cost, load carrying capacity, fuel consumption, space occupied, weight etc. any efforts in solving the 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 to those motor cycles and motor scooters, the two rear wheels are the driving wheels with a differential and a suspension, which are similar to those automobiles.

A three-wheeler with the ground providing input is essentially a motion disturbance vibrating system. Converting this input displacement into a forcing function which acts on the unsprung masses is done as the tyres are in contact with the ground, these masses are constrained by linkages so that they follow certain paths. Springs and dampers which are constrained by linkages so that they follow certain paths. Springs and dampers which are the coupling between unprung mass (vehicle body) transform this forcing function and transmit the resultant force to a sprung mass. The sprung mass being a rigid body its response to the acting forces is decided by its physical properties (i.e., moment of inertia, mass, centre of mass location etc.)Summarizing the ride response is dependent on the ground profile, tyre and suspension characteristics, physical dimensions and the inertial properties of the sprung and unprung masses of the vehicle. The Fig. 1 shoes the basic components of terrain vehicle simulation model. The vibration environment is one of the most important criteria by which people judge the design and consideration qualities of a vehicle. Being a judgment, it is a subjective in nature, from which arises one of the greatest difficulties in developing objective engineering methods for dealing with ride as a performance mode of the vibration. There are primarily two sources of vibrations, one external to the vehicle and other within the vehicle. The Fig.1 shows the over all picture of ride behavior of the vehicle.

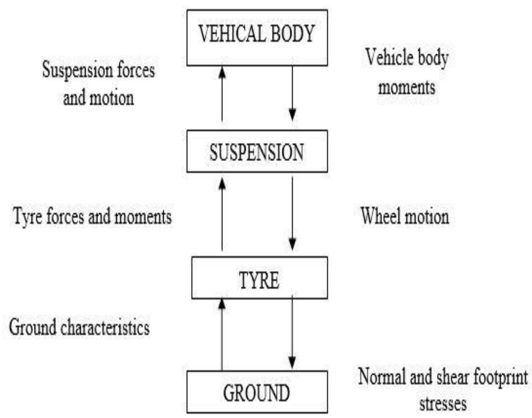

Fig.1. Basic components of terrain-vehicle simulation model.

B. Types

The classification of all kinds of vehicles may be made by support and propulsion principles. The support mechanism has to balance the gravity acting on vehicles and the propulsion generates the forward speed. Reaction forces generated by wheels, air cushion or magnets support ground vehicles. They are driven by friction flow or magnetic forces. Static and dynamic lift forces generated by water or air support fluid vehicles and flow forces propel them generally. Dynamic lift or inertia forces generated by air, jet propulsion or orbital motion support inertia vehicles and inertia forces accelerate them only. Ground vehicles are broadly classified as guided and non-guided. Guided ground vehicles are constrained to move along a fixed path, guide way (e.g. railway vehicle, tracked-levitated vehicles). Non-guided ground vehicles can move by choice in various directions on the ground such as road or off-road vehicles. The non-guided vehicles may be single-track vehicles (two wheeled vehicles like pedal cycles, scooter, motor cycles etc.,), two track vehicles (e.g. four wheeled vehicles like cars, trucks etc.,) and the track vehicles (e.g. three wheeled motor vehicles like pedal rickshaw, auto-rickshaw, delivery van etc). Therefore, three-wheeled vehicles can be classified as non-guided three-track ground vehicles and it consists of following type vehicles.

- Auto-rickshaws (Front engine (FE) and Rear engine (RE)).
- Delivery vans.
- Pick-up vans.
- Auto trailers.
- Unactuated variable length three-wheeled vehicles.
- “Dakota sun” the next generation solar powered racing vehicles.
- Three-wheeled mobile robots i.e., automated guided vehicles (AGVs).
- Three-wheeled road rollers etc.,

C. Advantages and Applications

Advantages:

- The advantage of the three-wheeled motor vehicles is a three wheel chassis is inherently lighter because it does not have to resist one-wheel bump twisting loads as compare with four-wheels.
- A three-wheel vehicle allows a more efficient aerodynamic enclosure in that either the front or rear may be tapered more gradually.
- In mass production, a three-wheeler is less expensive because of the elimination of the redundant wheel, tyre, brake, and suspension components and reduced assembly time.
- The steering mechanism of a three-wheeled vehicle is quite easy to design and manufacture, eliminating the steering and optimum gear mechanism as in four wheeled vehicles.
- The three wheeled motor vehicles are being increasingly used as a popular of public transport and are likely to be developed as a means of private transport as well.
- The three-wheeled motor vehicles are being smaller and lighter and also better fuel-efficient passenger vehicles compare with four-wheeled vehicles.
- Three-wheelers also have the advantage of being a compromise between two-wheelers and four wheelers in various aspects like cost, load carrying capacity, fuel consumption, space occupied and weight etc.
- The work done in three-wheeler dynamics is very less as compared to other types of vehicles and hence, there are many areas, which require much better attention, research, analysis and study.

Application:

- In India the three-wheeled motor vehicles are being increasingly used as a popular means of public transport in intercity applications are likely to be developed as a means of private transport as well for small article transportation.
- Three-wheelers are used in India for plant transportation of goods in industries and local transportation in intercity applications.
- At present, three wheelers are predominantly being used in major cities and smaller towns for movement of goods and materials for personal use.

C. Vehicle Dynamics

Vehicle dynamics concerned with the movements of vehicles, automobiles, trucks, buses and special purpose vehicles on a road surface, the forces imposed on the vehicle from tyres, gravity and aerodynamics determine the dynamic behavior. The vehicle and its components are studied to determine what forces will be produced by each of these sources at a particular trim condition and how vehicle responds to these forces. Like any other vehicle, the three-wheeler vehicle is characterized by its performance, handling and the ride as shown in the Fig.2. Performance characteristics of the vehicle are concerned with the acceleration, deceleration and negotiation of grades in straight-line longitudinal motion or x-translation. The tractive or braking effort and resisting forces determine the performance potential of the vehicle. Handling characteristics of road vehicles are concerned with its response to steering commands and to environmental inputs effecting the direction of motion of the vehicle such as wind and road disturbance. The vehicle motions associated with sideslip (motion along y-axis), yaw (rotation about z-axis) and roll (rotation about x-axis) are usually referred to as lateral motions of the vehicle. The behavior of the vehicle in these modes determines to a great extend its handling characteristics.

The handling requirements are:

- **Directional Stability:** It is ability to stabilize its direction of motion against disturbances. A vehicle is considered be directionally stable if it returns to steady state regime with in a finite time after the disturbances are removed.
- **Wind Stability:** The vehicle must not wander in gusty side winds.
- **Cornering Ability:** That means the vehicle must be capable of withstanding a maximum amount of lateral acceleration per degree of slip angle.
- **Response:** Proper response to driver's commands is essential for good control.

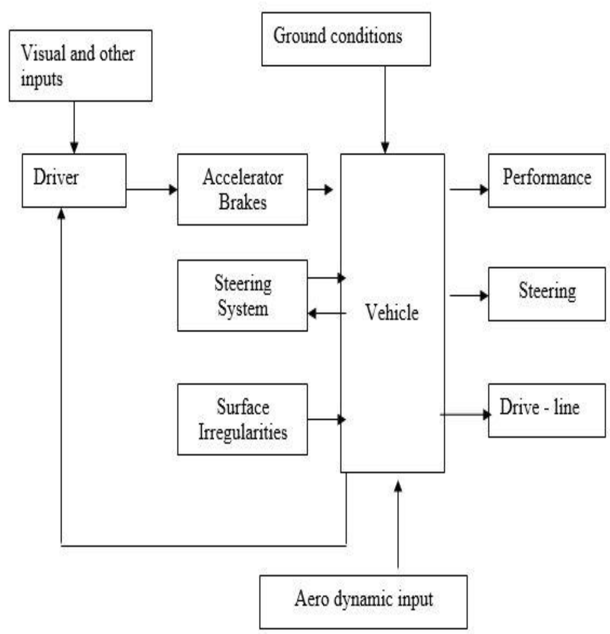

Fig.2. The driver-vehicle ground system.

E. Requirements of The Body Structure Of The Vehicle

Strength: The body must be strong enough to withstand all types of forces which the car is subjected, which include

- weight of the car
- weight of passengers
- inertia
- braking
- side forces
- impact loads

Stiffness: The body must be stiff enough to withstand to prevent bending.

Torsional Stiffness: To resist twisting on bad roads.

Space: To accommodate passengers and luggage.

Air Drag: Resistance if air during running depends upon the body shape and increases directly as the square of the vehicle speed. The shape of the body should be such that the air drag is minimum.

Protection against Weather: The design of the body must be such that the occupants and the luggage are protected from bad weather.

Resistance to Corrosion: The body should be so designed that no moisture is accumulated which would produce rust. The materials used should also be such that no corrosion takes place.

Protection in accident.

F. Ride Dynamics

Ride quality is concerned with the sensation or feel of the passenger in the environmental of a vehicle. The effect of vehicle vibration response on human comfort and safety is of current importance in ride quality studies. The overall ride quality of a vehicle is affected by its vertical, lateral and longitudinal vibration in case of vehicle traveling at a constant speed over roadways having varying degrees of freedom. Ride comfort problem mainly arises by a variety of source i.e., surface irregularities, aerodynamic forces, tyre vibration and from engine and transmission. The given Fig.3 shows schematically the ride dynamics system. The ride comfort is assessed by the acceleration experienced by the passengers as a disturbance. Two major factors must be considered in evaluating by the ride performance of a vehicle, the first is "ride safety" and the second is "ride comfort" (both of the driver and passengers as well as cargo). The objectives therefore, of the studies of ride dynamics are to assess the response of the vehicle systems to road surface inputs and provide the best vehicle in vibratory motions, therefore the road surface irregularities induce vibration of tyre and by suspension system. These vibrations are transmitted to vehicle body.

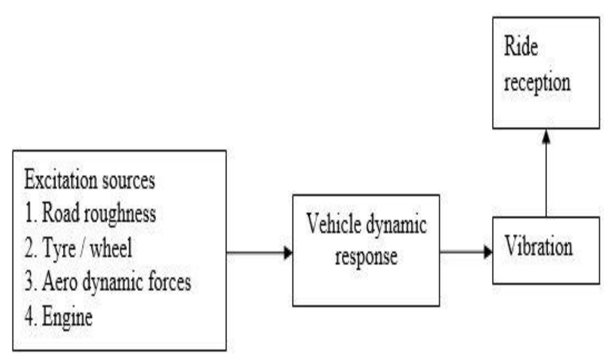

Fig.3. Ride –dynamics system.

G. Objective of The Thesis

The objective of this thesis is to analyze the dynamic behavior of vehicle using Finite Element modeling. This project also includes the response of the three wheeler chassis frame to road surface inputs and to provide the best vehicle in vibratory motions. In Finite element modeling, analysis is done to determine the dynamic behavior of the vehicle in Harmonic and Transient excitations. Modal analysis and parametric study has also been carried out. Spectrum analysis is carried out in order to obtain the maximum stresses, as the vehicle running on the road is subjected to the random vibrations. The results obtained from Finite element modeling in modal analysis were studied and are compared with those of rigid body modeling. In Transient analysis excitation magnitude and acceleration values at the key nodes are analyzed. From spectrum analysis maximum stress point is noted. Conclusions were derived from the study and some suggestions are made to improve the performance of vehicle.

H. Organization of the Thesis

In chapter two, literature survey has been made to get an idea which described the basic ideas about modeling of the body, idealizations to be considered for static as well as dynamic analysis and solution technique which leads to effective results for the analysis is obtained from literature survey. In chapter three, the basic concept of finite elements analysis was presented. It includes the element selection, general procedure adapted for the analysis. In chapter four, multipurpose finite element software that is ANSYS features were presented. It includes analysis capabilities, program overview. Types of elements available, solution techniques were detailed. The entire chassis of the vehicle is modeled using beam, mass, combination elements. Modal, harmonic, transient analysis, spectrum analysis and parametric study have been carried out and results were presented. In chapter five, conclusions were drawn from the analysis done by the finite element method and parametric analysis.

II. DYNAMIC ANALYSIS OF 3 WHEELER CHASSIS FRAME

A. Model Description

Bajaj Rear Engine (Petrol) is selected for analyzation purpose. The specifications of the vehicle are:

TABLE I: Specifications Of The Vehicle

COMPONENTS	SPECIFICATION
Engine type	Single cylinder 4-stroke petrol engine
Power	5.04KW, 3000rpm
Dimensions	Length:2650mm Width:1300mm Height:1710mm Wheel base:2000mm Ground clearance:1700
Weights	Gross vehicleweight:680kg Max. payload:330kg

B. Modeling

The model consists of chassis elements, suspension and tyres. In this analysis, for beams of the chassis BEAM 189 is taken. Suspensions and tyres are modeled as COMBN14, keeping the option as 2D longitudinal i.e., linear spring. Isometric view is shown in fig.4.

- The chassis frame is modeled as 3-D beam elements of different cross-section with 6 D.O.F at each node.
- The suspension and tyres are modeled as linear springs with stiffness in the vertical direction alone. The stiffness and damping values are taken as mentioned in table.
- The material used for the chassis frame of the vehicle is cast iron, having an young's modulus of 196.2E09 N/m^2 and the density as 7890 kg/m^3.
- The mass distribution is taken as lumped mass i.e., the mass of the passenger, driver, and certain key loads like engine gear box are lumped at the appropriate nodes.

TABLE II: Stiffness and Damping Values of Suspensions and Tyres

Description	Stiffness(K) (N/m)	Damping coefficient(C) (N-s/m)
Front suspension	32700	3500
Rear right suspension	50400	2207.5
Rear left suspension	49800	2207.5
Front tyre	238260	557
Rear right tyre	250490	436
Rear left tyre	250490	436

1. Element Description: When we use certain elements in the problem, it is important to know the element description in detail. Descriptions of the elements which are used here are given below:

COMBIN 14: It has longitudinal or torsional capability in one, two, or three-dimensional applications. The longitudinal spring-damper option is a Uni-axial tension-compression element with up to three degrees of freedom at each node i.e., translations in the nodal x, y, and z directions. No bending or torsion is considered. The torsional spring-damper option is a purely rotational element with three degrees of freedom at each node: rotations about the nodal x, y, and z-axis. No bending or axial loads are considered.

Assumptions and restrictions:-

- The length of the spring-damper element must not be zero, i.e., nodes I and J should not be coincident.
- The longitudinal spring element stiffness acts only its length.

- The torsion spring element stiffness acts only about its length, as in a torsion bar.
- The element allows only a uniform stress in the spring.

TABLE III: COMBIN 14 Element Input Table

Element name	COMBIN 14
Nodes	I, J
Degree of freedom	Ux, Uy, Uz, ROT X,ROT Y, ROT Z
Real constants	K,CV1,CV2
Material properties	None
Surface loads	None
Body loads	None
Special features	Nonlinear, stress stiffening, large deflections

TABLE IV: COMBIN 14 Element Output Table

Name	Type	Definition
Force or torque	SMISC	Spring force or moment
Strength or twist	NMISC	Stretch of spring or twist of spring
Velocity	NMISC	Velocity
Damping force or torque	NMISC	Damping force or moment

MASS 21: MASS21 is a point element having up to six degrees of freedom i.e., translations in the nodal x, y, and z directions and rotations about the nodal x, y, and z axis. A different mass and rotary inertia may be assigned to each coordinate direction.

TABLE V: MASS 21 Element Input Table

Element name	MASS21
Nodes	I
Degree of freedom	Ux, Uy, Uz, ROT X,ROT Y, ROT Z
Real constants	Mx, My, Mz, Ix, Iy, Iz
Material properties	Density
Surface loads	None
Body loads	None
Special features	Large deflections

Assumptions and Restrictions:

- The mass element has no effect on the static analysis solution unless acceleration or rotation is present, or inertial relief is selected.
- The standard mass summary input is based only on the X mass term, if directional mass is input.
- In an inertial relief analysis, the full matrix is used.

Beam 189: BEAM189 is a quadratic (3-node) beam element in 3-D. BEAM189 has six or seven degrees of freedom at each node, with the number of degrees of freedom depending on the value of KEYOPT(1). When KEYOPT (1) = 0 (the default), six degrees of freedom occur at each node. These include translations in the x, y, and z directions and rotations about the x, y, and z directions. When KEYOPT (1) = 1, a seventh degree of freedom (warping magnitude) is also considered. This element is well-suited for linear, large rotation, and/or large strain nonlinear applications.

TABLE VI: BEAM 189 Input Data

Element name	Beam 189
Nodes	I,J,K,L(the orientation node, is optional but recommended)
Degree of freedom	Ux, Uy, Uz, ROT X,ROT Y, ROT Z,WARP
Real constants	Mx, My, Mz, Ix, Iy, Iz
Material properties	Density, Young's modulus, shear modulus, damp
Surface loads	None
Body loads	Temperatures
Special features	Creep

BEAM189 Assumptions and Restrictions:

- The beam must not have zero length.
- By default (KEYOPT (1) = 0), the effect of warping restraint is assumed to be negligible.
- Cross-section failure or folding is not accounted for.
- Rotational degrees of freedom are not included in the lumped mass matrix if offsets are present.
- BEAM189 includes the effects of transverse shear and accounts for the initial curvature of the beams.

2. Description of the Model: Three dimensional modal of the chassis frame is drawn in the ANSYS using key points and lines. Five different types of sections are used in the modal. They are C-section, rod of smaller diameter, a circular cross-section of high diameter, rectangular c/s, and tapered cross-section. The cross-sections are appropriately assigned by picking an orientation key point. Real constants defined for the suspensions and tyres are properly assigned to the respectively. Then the model is meshed. This meshed model is given the boundary conditions, which are specified in the next section.

C. Boundary Conditions

Following displacement constraints are applied in the present modeling.

- The top and bottom points of tyres are fully arrested except in vertical direction.
- Nodal coupling is defined between the top nodes of the suspensions and the chassis frame, so as to provide proper connectivity between the elements in the present model.

III. RESULTS AND DISCUSSIONS

The results should be consistent and assist the designer to design the vehicle to satisfy customer needs, safety, ride comfortness, service and economic conditions. In present analysis the results of modal, harmonic, transient are presented in the form of figures and graphs by selecting the effective software like ANSYS.

A. Modal Analysis

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

TABLE VII: Natural Frequencies of Significant Modes

S. No	Type of mode	Frequency(Hz)
1	Bounce mode	7.70
2	Front hope mode	10.17
3	Roll mode	11.46
4	Yaw mode	15.53
5	Pitch mode	30.81

After MODAL ANALYSIS of the chassis model, we obtained a series of nature frequencies within 1Hz---100Hz (7.7072 Hz, 10.177Hz, 11.466Hz, 15.530Hz, 30.811Hz, 32.099Hz, 45.691Hz and 51.841Hz etc). The mode shapes of the modal are shown in the figs from 5 to 8. The lower nature frequencies (7.7072 Hz, 10.177Hz, and 11.466Hz) represent the nature characteristics of the suspension system and the others represent the nature characteristics of the body. Taking the 1st nature frequencies as an example, it represents the nature characteristics of a simplified vibrating system where the suspension mass of the vehicle is simplified as a mass point and the suspension system as spring and damp. The nature characteristics of a simplified vibrating system can be calculated directly as following:

$$f = \sqrt{\left(k / M\right)} / 2\pi \quad \textbf{(1)}$$

Where f is natural frequency, k is spring coefficient, and M is Mass. If k = (32700 +50400 +49800 +238260 +2*250490) = 647140(N/m) and M = 1010Kg, f = 6.454Hz.

Comparing the calculated result from different methods, i.e., like rigid body modeling, the relative error is 11.45%, which may be mainly caused by simplification of the model, and it primarily proved that the whole FE Model of chassis is reliable. The calculated nature mode can be displayed by ANSYS. If there are some malformations on any nature mode, the reason may be: on one hand, there is really design problem on local part of vehicle (especially during development period), the analysis engineer should discuss the problem with the design engineer about how to solve the problem at this time. On the other hand, the malformation may be caused by improperly simplification during creation of the model, and then the model should be modified. We have removed the malformations that may effect the analysis result based. A few typical mode shapes of the vehicle model such as bounce, front, hop, roll, yaw and pitch are shown in Fig. from 5 to 48. The bounce mode is shown in fig.5 and the frequency for this mode is 7.70Hz. During this mode the maximum displacement is 0.09m. The maximum displacement is occurred in rear left portion of the chassis. The front hop mode is shown in fig 6 and the frequency for this mode is 10.17Hz. During this mode the maximum displacement is 0.10m. Here the maximum displacement is observed at left side of the passenger seating. During the roll mode the frequency and the maximum displacement 11.46Hz and 0.11m. This mode shape is shown in fig 7. The steering portion has the maximum displacement especially the bottom portion of the steer has the maximum. The yaw mode shape is shown in fig 8. Frequency for this mode shape is 15.53Hz and the maximum displacement is 0.10m. During pitch mode the frequency and maximum displacement are 30.81Hz. This mode shape is shown in fig 9.

B. Spectrum Analysis

For spectrum analysis, road load input data is considered. Displacements at different frequencies are given as input. Modal analysis is carried out and then spectrum is done. Then by expanding the modes in modal analysis, we obtain the spectrum results.

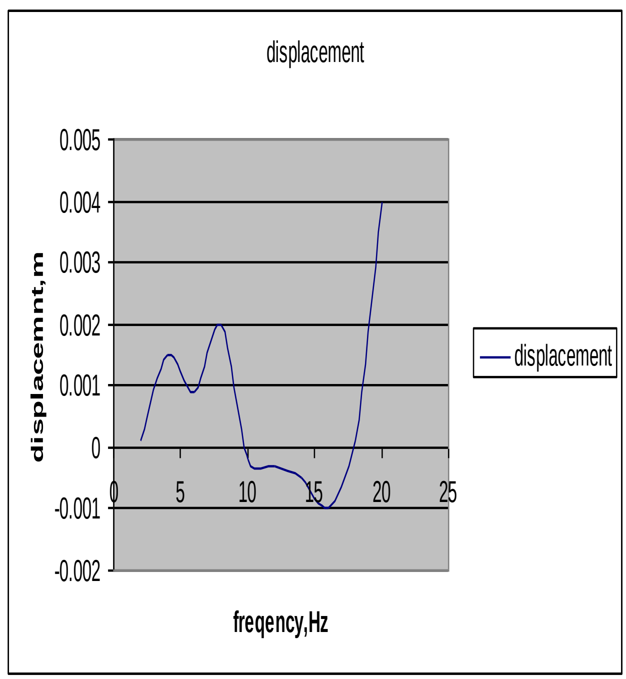

Fig.4. spectrum input as displacement versus frequency graph.

Output of the spectrum analysis is stress verses frequency plot which is shown in the figs.10 to 25. From the graph we can observe the variations of the maximum stress. The value of the maximum of stress, 277.7Mpa is observed near the first fundamental frequency i.e., around 7 Hz. The maximum stress at this frequency occurs near the right passenger seating.

IV. CONCLUSION

- In this case, frequencies obtained from modal analysis lies between the working frequencies range, hence resonance conditions exists. Resonance can be avoided by reinforcing the cross members by another high stiffness members. Hence natural frequency may be improved and less chance to resonance.
- Transient displacements are quite high in the front portion of the driver seat compared with passenger seat.
- Maximum stress is obtained from the spectrum analysis, and the value is less than the allowable stress for the material considered. So, we can conclude that the design is in safe region.
- Spectrum analysis result indicates that, there is a peak stress that occurs near 8Hz because there is a frequency overlap between road load and suspension system.
- Based on the parametric analysis, it is observed that the by changing front suspension damping and rear suspension stiffness, the displacements at the key nodes are negligible.
- The vertical displacement at the passenger seating is not effected by the increasing and decreasing the values of front suspension damping, rear suspension stiffness.
- From the parametric analysis, it is observed that that by decreasing the values of rear suspension damping and rear tyre stiffness, most displacements are on the lower side i.e., which results in comfortness for the passenger.

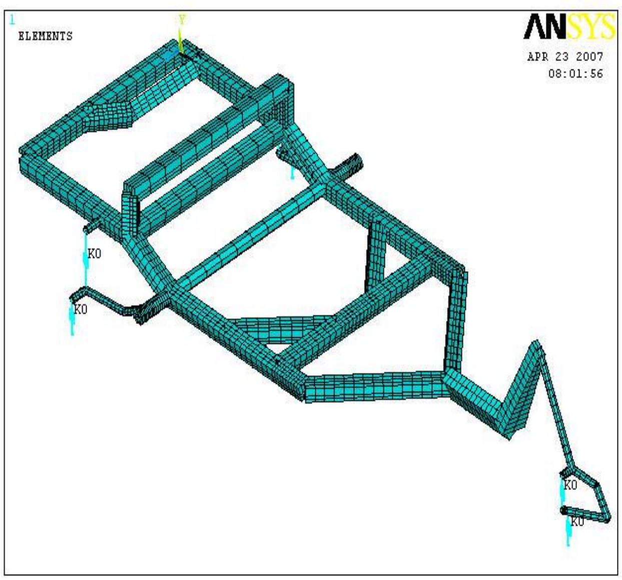

Fig.5. Isometric view of model having different cross-sections.

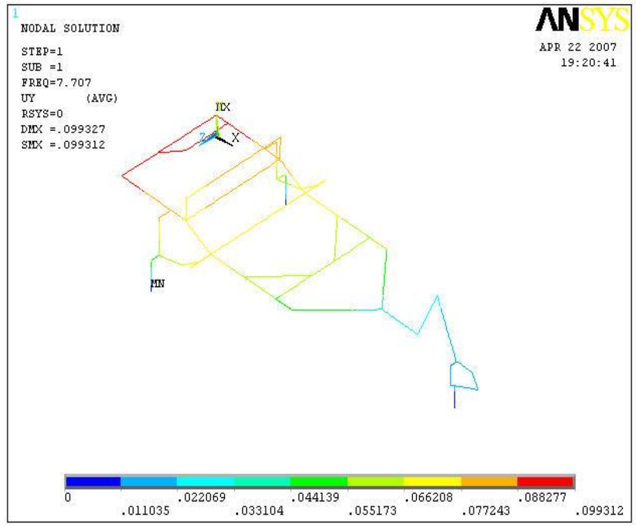

Fig.6. Bounce mode of the chassis modal.

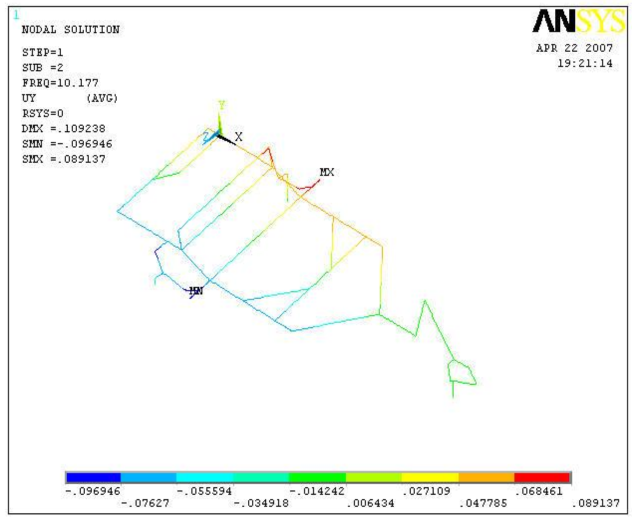

Fig.7. Front mode of the chassis modal.

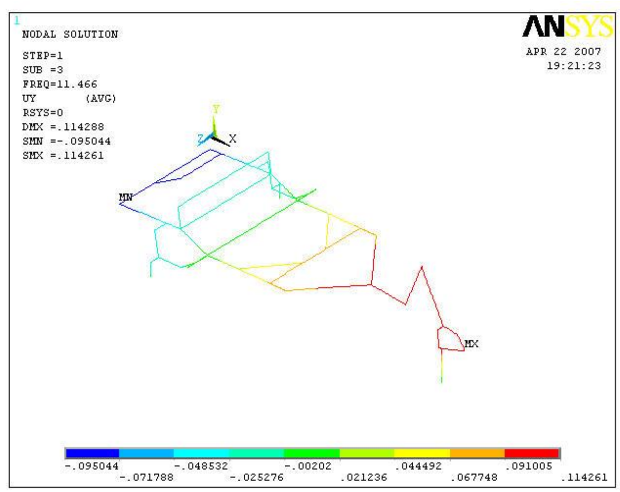

Fig.8. Roll mode of the chassis modal.

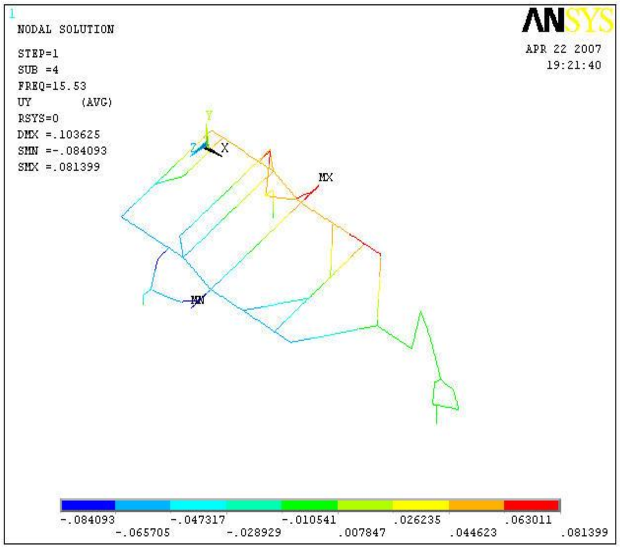

Fig.9. Yaw mode of the chassis modal.

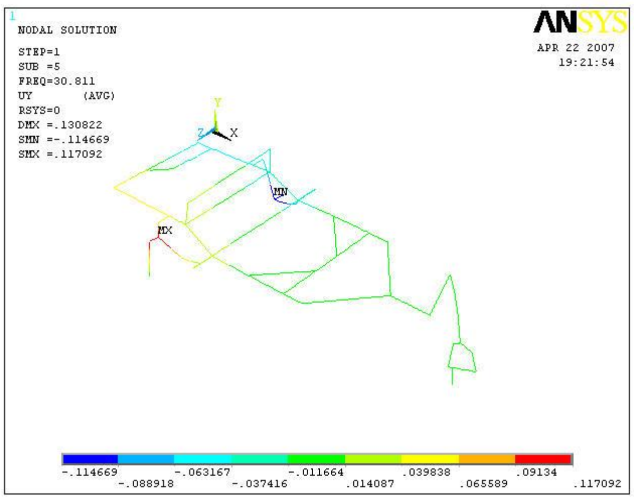

Fig.10. Pitch mode of the chassis modal.

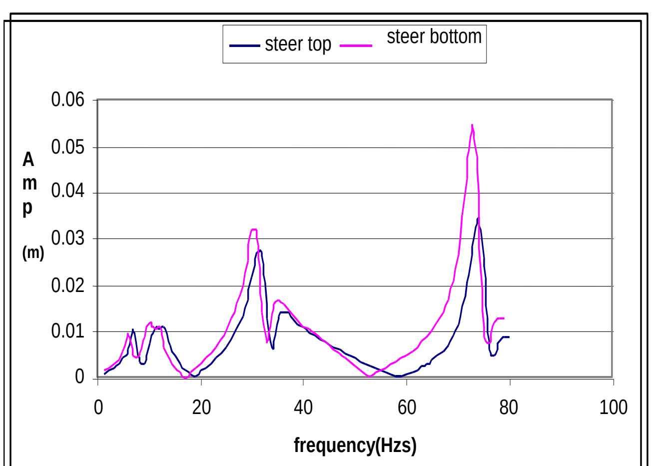

Fig.11. variation in displacement at the top and bottom portions of the steering column.

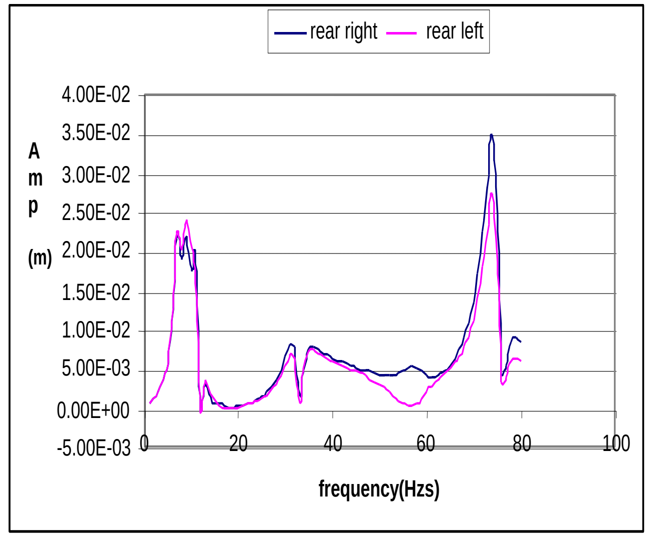

Fig.12. variation in displacement at the rear end left and right side nodes.

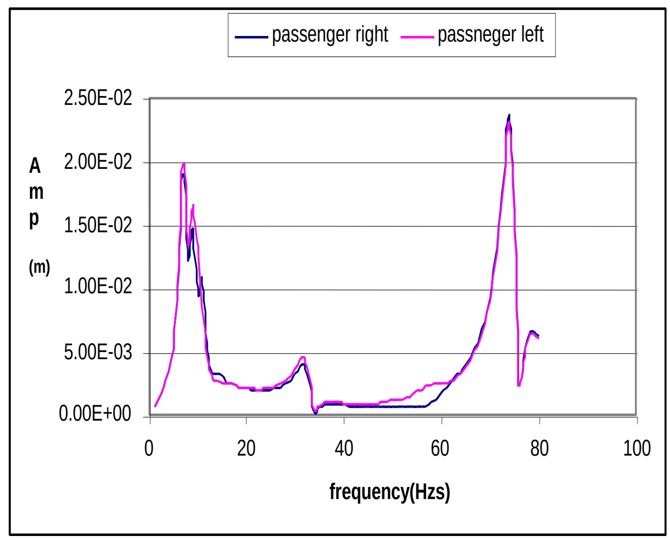

Fig.13.variation in displacement at the passenger right and left nodes.

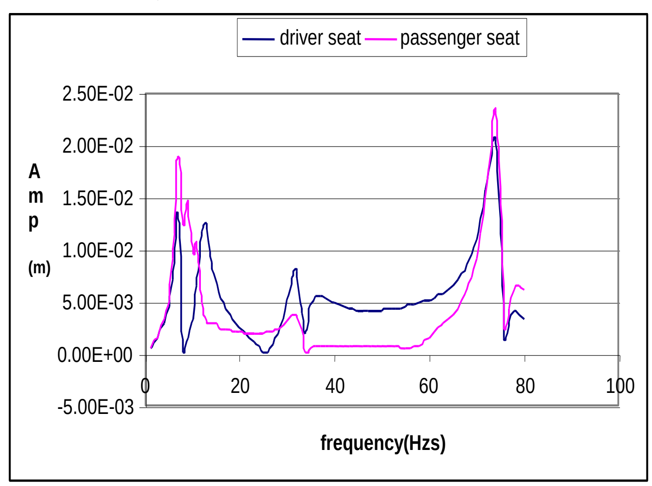

Fig.14.variation in displacement at nodes of the driver seat and passenger seat.

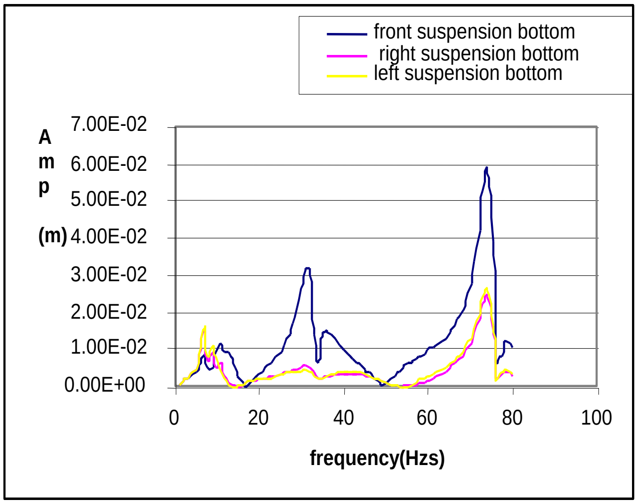

Fig.15. variation in displacement at nodes of the front suspension bottom, rear right suspension bottom and rear left suspension bottom.

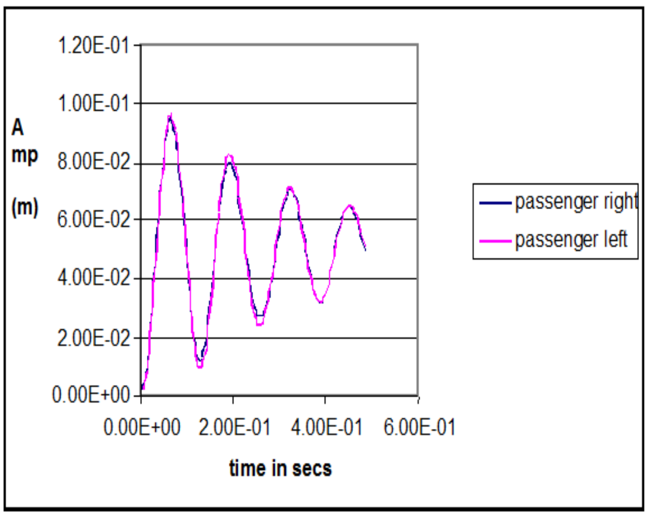

Fig.16. variation in displacement at nodes of the passenger right and left seating (Transient response).

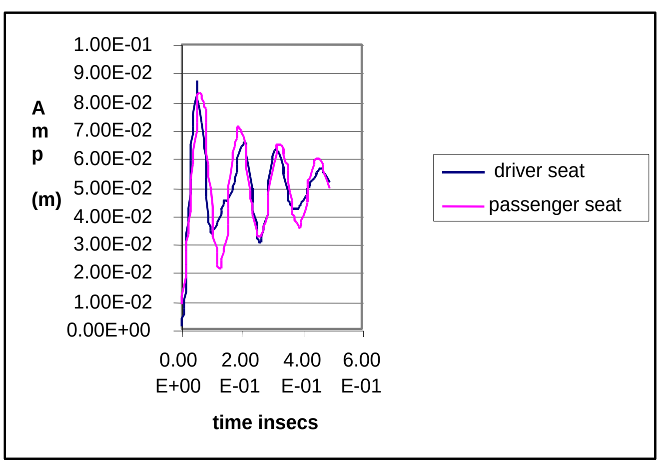

Fig.17.variation in displacement at nodes of the driver seat and passenger seat (Transient response).

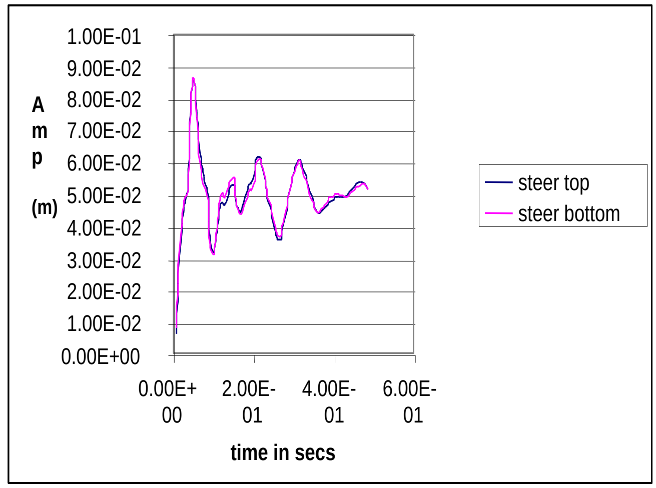

Fig.18.variation in displacement at nodes at the top and bottom portions of the steering column (Transient response).

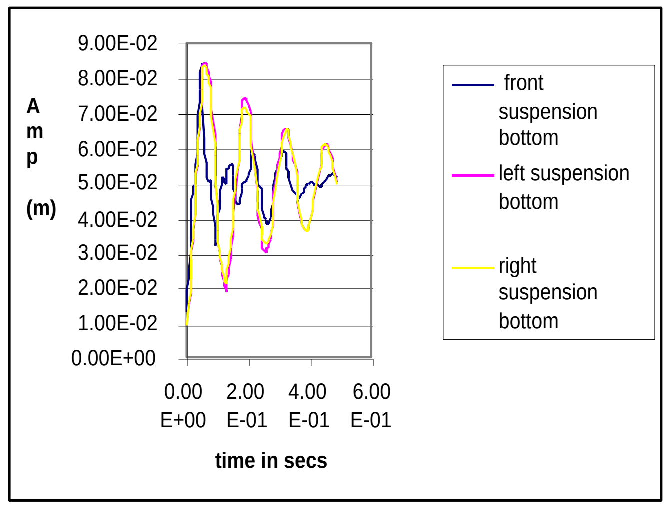

Fig.19. Variation in displacement at nodes of the front suspension bottom, rear right suspension bottom and rear left suspension bottom (Transient response).

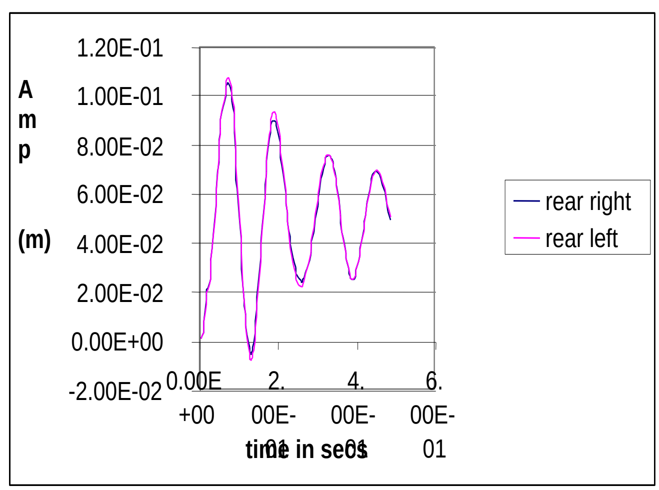

Fig.20.variation in displacement at nodes at the rear right and left ends of chassis.

(Transient response)

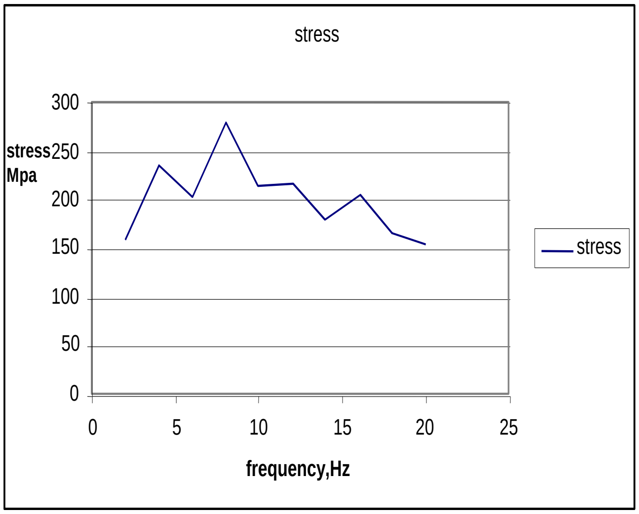

Fig.21. spectrum analysis result showing the peak stresses at different frequencies. (Spectrum analysis)

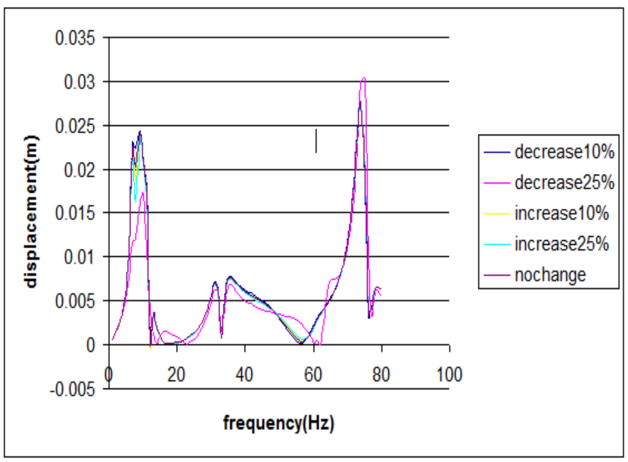

Fig.22.effect of change in rear suspension damping at node 1. (Parametric Analysis).

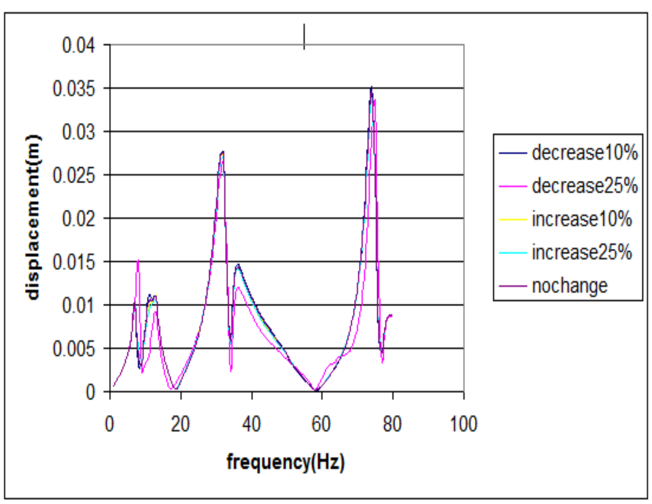

Fig.23. effect of change in rear suspension damping at node 169. (Parametric Analysis).

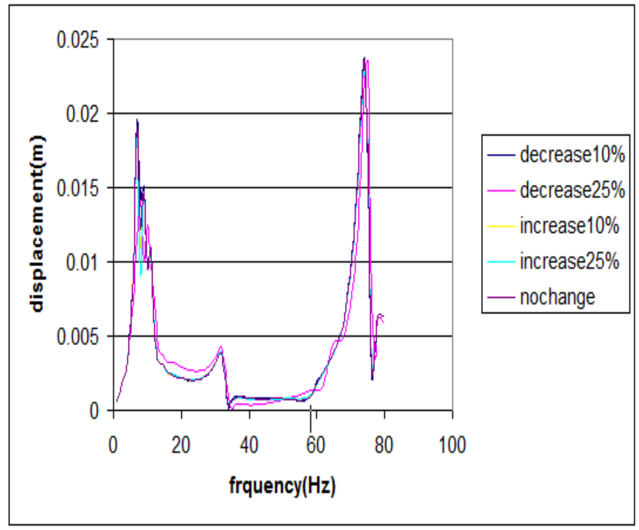

Fig.24. effect of change in rear suspension damping at node 562. (Parametric Analysis).

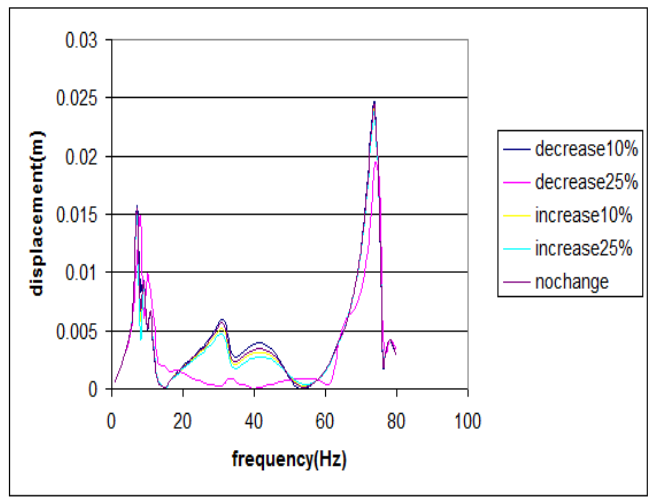

Fig.25. effect of change in rear suspension damping at node 799. (Parametric Analysis).

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