Dynamic Analysis Of A Car Chassis

Pravin A Renuke*[1]
Asst.Professor in Mechanical Engg.GHRIEM.Jalgaon

Abstract-

The paper investigates into the vibrational characteristics of the car chassis including the natural frequencies and mode shapes. Car chassis forms the structural backbone of a passenger vehicle. When the car travels along the road, the car chassis is excited by dynamic forces caused by the road roughness, engine, transmission and more. Modal analysis using Finite Element Method (FEM) can be used to determine natural frequencies and mode shapes. In this study, the modal analysis has been accomplished by the Commercial finite element packaged ANSYS. The model has been simulated with appropriate accuracy and with considering the effect of bolted and riveted joints. The chassis has been altered by some companies for using in After constructing finite element model of chassis and appropriate meshing with shell elements, model has been analyzed and first 6 frequencies that play important role in dynamic behavior of chassis, have been expanded. In addition, the relationship between natural frequencies and engine operating speed has been explained. The results show that the road excitation is the main disturbance to the car chassis as the chassis natural frequencies lie within the road excitation frequency range. Finally advantages of the modified chassis which leads to the increase of the natural frequencies and placing them in the appropriate range, has been discussed.

Keywords: Vibration, car chassis, Modal analysis, Dynamic, Finite elements

1. INTRODCTION

The dynamic response of simple structures, such as uniform beams, plates and cylindrical shells, may be obtained by solving their equations of motion. However, in many practical situations either the geometrical or material properties vary, or the shape of the boundaries cannot be described in terms of known mathematical functions. Also, practical structures consist of an assemblage of components of different types, namely beams, plates, shells and solids. In these situations it is impossible to obtain analytical solutions to the equations of motion [1]. This difficulty is overcome by seeking some form of numerical solutions and finite element methods.

Automotive industry is one of the biggest users of the technology of modal analysis. The modal behavior of car chassis is a part of most necessary information for the inspection into car's dynamic behavior. In this essay the modal analysis of car chassis has been studied.

As a car travels along the road, the car chassis is excited by dynamic forces induced by the road roughness, engine, transmission and more. Under such various dynamic excitations, the car chassis tends to vibrate [2]. Whenever the natural frequency of vibration of a machine or structure coincides with the frequency of the external excitation, there occurs a phenomenon known as resonance, which leads to excessive deflections and failure. The literature is full of accounts of system failures brought about by resonance and excessive vibration of components and systems [3].

The global vibrational characteristic of a vehicle is related to both its stiffness and mass distribution. The frequencies of the global bending and torsional vibration modes are commonly used as benchmarks for vehicle structural performance. Bending and torsion stiffness influence the vibrational behavior of the structure, particularly its first natural frequency [4].

The mode shapes of the car chassis at certain natural frequencies are very important to determine the mounting point of the components like engine, suspension, transmission and more. Therefore it is important to include the dynamic effect in designing the chassis [2].

Many researchers carried out study on truck chassis. Vasek and his cooperators (1998) have analysed a truck dynamically. In their method in addition to simulating ctruck with finite element packaged ANSYS and being sure that structure vibrational modes are in appropriate range, they vibrationally analyzed it [5]. Yuan Zhang and Arthur Tang (1998), compare natural frequencies of a ladder chassis with finite element and experimental methods [6]. Guo and chen (2008), research into dynamic and modal analysis of a space chassis (complex 3-dimensional chassis) and analyse transient response using the principal of superposition [7].This paper deals with a car chassis that includes natural frequencies and mode shapes. In the studied model unlike the most previous models,rivets and bolts have been modeled completely. Also shell element has been used for analysis. This element has better and more disciplined meshing in comparison with other elements and has the capability of gaining moreaccurate results with the same meshing containing the related 3-dimensional elements. It is mentionable that validity of the results has

(IJERA) ISSN: 2248-9622 www.ijera.com
Vol. 2, Issue 6, November- December 2012, pp.955-959

been verified by comparing the results from a similar model with the model proposed by Mr Foui and his cooperator [2].

2. CAR CHASSIS

In this article, a wagonR car chassis has been studied. This car chassis is a ladder chassis and it's longitudinal and cross connecting sections are channel
shaped.Automotive view with dimensional plan and dimensions has been illustrated in Fig.1 and Fig.2 and table 1.Chassis material is structural steel with
7850 kg/m3 density and 250 MPa yield strength and 590 MPa tensile strength.

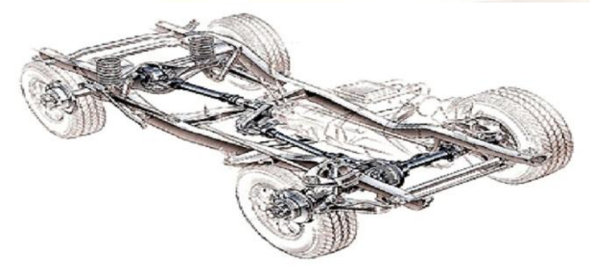
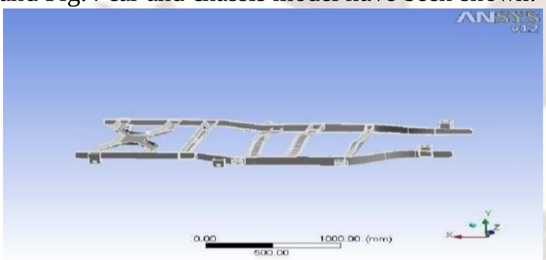


Fig. 1 View of car chassis

3. FINITE ELEMENT

Car chassis has been modeled with 10-noded tetrahedral (Tet-10) solid element in ANSYS. Numerical studies on simple rectangular beam show that this element is suitable for creating and meshing the model and it yields accurate results. The element used has 4 nodes with 6 degrees of freedom and is appropriate for linear and nonlinear deformations and also large deflections. There are approximately 70000 elements in the model that has proved suitable in comparison with other cases, so that the error in each case is less than one percent. In Fig.3 and Fig.4 car and chassis model have been shown.



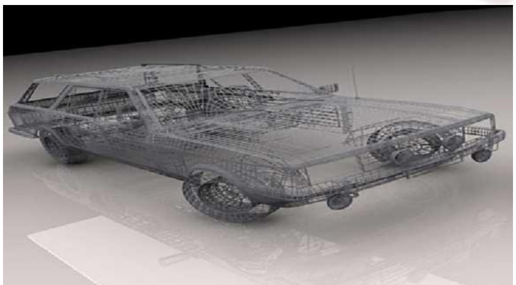
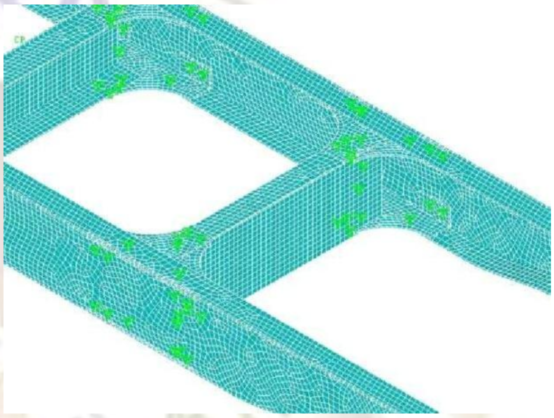


Fig. 3 Geometric chassis model

Model with appropriate accuracy and with considering bolts and riveted joints effects has been simulated. Meshes and constraints have been shown in Fig.6.

The boundary conditions are different for each analysis. In normal mode analysis, free-free boundary condition will be applied to the car chassis model, with no constraint applied to the chassis model [2].

Fig. 4 Geometrical car model



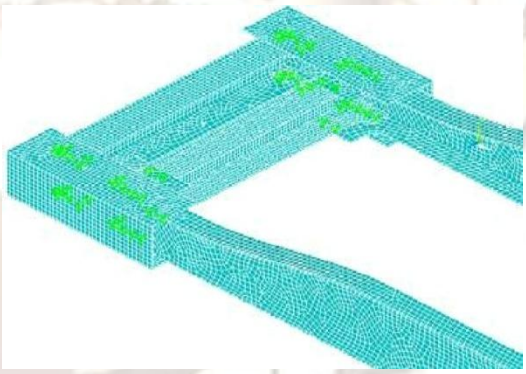


Fig. 5 Meshed chassis

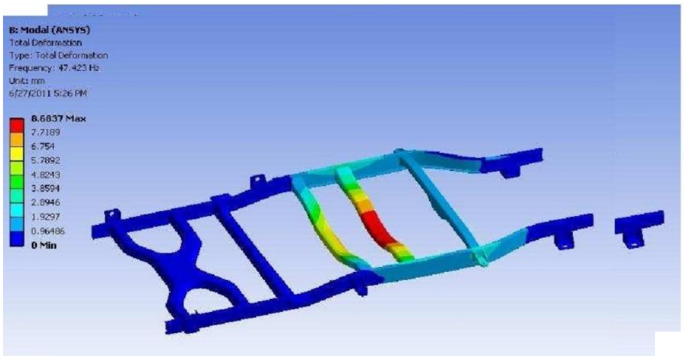
A free-free boundary condition has been chosen as it is much simpler to test experimentally in this condition, if required.

4. MODAL ANALYSIS AND RESULTS

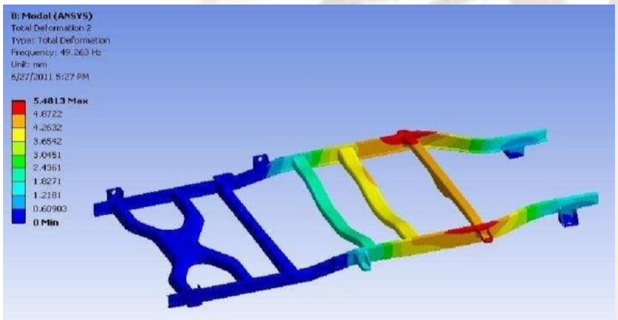
Modal analysis has been performed after creating the chassis finite element model and meshing in free-free state and with no constraints. The results have been calculated for the first 15 frequency modes and show that road simulations are the most important problematic for car chassis. In this analysis we have made use of subspace method in ANSYS.

Since chassis has no constraints, the first 6 frequency modes are vanished. 3 modes are related to the chassis displacement in x, y and z directions

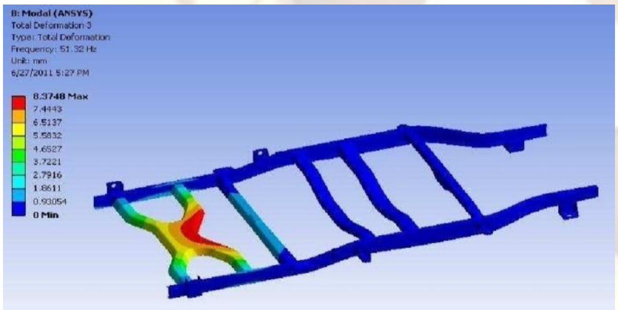
and 3 modes are related to chassis rotation about x, y and z axes. In Fig.6
elated natural frequencies and mode shapes for chassis with maximum displacement in y direction in each mode, have been shown.



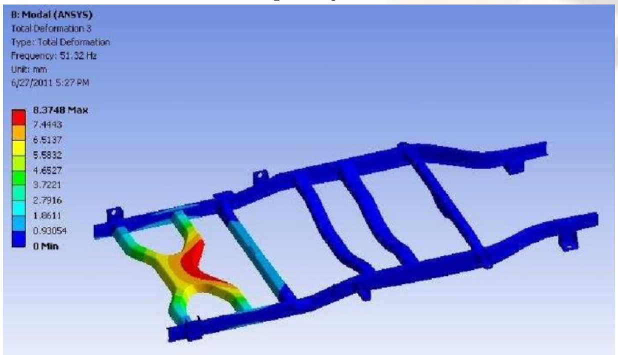
1st mode of frequency



2nd mode of frequency

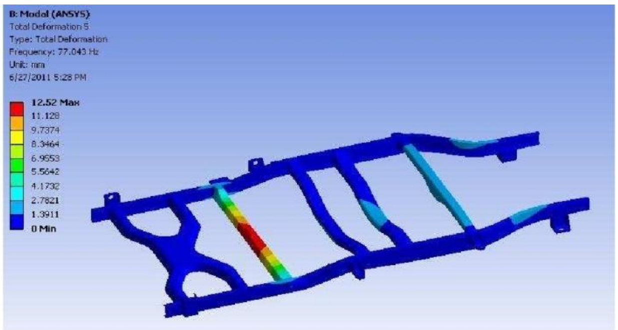


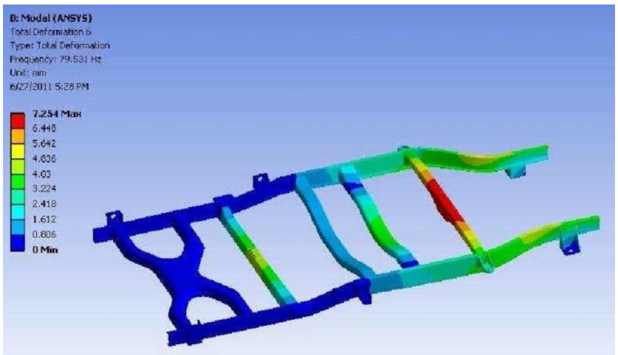
3rd mode of frequency



4th mode of frequency

5th mode of frequency





6th mode of frequency

Fig. 6 Natural frequencies of car chassis

There are two types of vibration, which are global and local vibrations. The global vibration means that the whole chassis structure is vibrating while local vibration means the vibration is localized and only part of the car chassis is vibrating. The first mode shape of the car chassis at 47.423 Hz. The chassis experienced first twisting mode about x-axis (longitudinal) and fall under global vibration as the whole chassis follow to vibrate. The maximum translation was at the front end of the car chassis.The second mode shape was 2nd twisting about axis-x at 49.263 Hz The maximum translation was again at the front end of the car chassis.The third mode shape was at 51.32 Hz that experienced first bending about axis-Z. The maximum translation was at the rear end of the chassis as shown in figure. The fourth mode shape at 66.114 Hz. It experienced second twisting mode about axis-X and maximum translation occurred at front end of the chassis.The fifth mode shape was 77.043 Hz that experienced combined torsion and twisting mode. The sixth mode shape is shown in figure 6. It was found at 79.531 Hz where the chassis experienced the second bending mode. The maximum translation occurs at front end of the chassis. Results of first 15 analyzed modes for chassis are shown in table 2.

Mode Number	Natural Frequency
1	0
2	0
3	0
4	0
5	0
6	0
7	42.423
8	49.263
9	51.32
10	66.114
11	77.043
12	79.531
13	82.553
14	88.042
15	93.700

In general the natural frequency can be calculated using the equation (1)

$$Wn = \sqrt{k/m}$$

where K and m stand for stiffness and mass respectively Diesel engine is known to have the operating speed varying from 8 to 33 revolutions per second (rps) [8]. In low speed idling condition, the speed range is about 8 to 10 rps. This translates into excitation frequencies varying from 24 to 30 Hz [2]. The main excitations are at low speeds, when the car is in the first gear. At higher gear or speed, the excitations to the chassis are much less. The natural frequency of the car chassis should not coincide with the frequency range of the axles, because this can cause resonance which may give rise to high deflection and stresses and poor ride comfort. Excitation from the road is the main disturbance to the car chassis when the car travels along the road. In practice, the road excitation has typical values varying from 0 to 100 Hz. At high speed cruising, the excitation is about 3000 rpm or 50 Hz. Mounting of vibration components of the car on the nodal point of the chassis is one of the vibration attenuation methods to reduce the transmission of vibration to the car chassis [2].

The mounting location of the engine and transmission system is along the symmetrical axis of the chassis's first torsion mode where the effect of the first mode is less. However, the mounting of the suspension system on the car chassis is slightly away from the nodal point of the first vertical bending mode. This might due to the configuration of the static loading on the car chassis.

Regarding the previous discussions about the diesel engine speed, we can say that natural frequencies are in critical range. Hence with decreasing the chassis length which increases the chassis stiffness, we increase the natural frequencies to place them in the appropriate range.In addition, with changing the gasoline tank situation and performing similar changes, we can prevent coinciding the simulation force frequencies and natural frequencies. Otherwise resonance phenomenon occurs and if these two frequencies coincide, this phenomenon destroys the chassis. As a reminder it is mentionable that validity of the results has been verified by comparing the results from a similar model with the model proposed by Mr Foui and his cooperator [2]

Because the analysis is in free-free state, the first 6 modes that have zero frequency aren't considered and mode numbers 7 to 12 in table 2 represent the first 6 modes of frequency. It is necessary to notice that in usual, the first 6 modes of frequency (mode numbers 7 to 12 in table 2 that have been shown in Fig.6) play the main role in dynamic behavior of chassis and limited energy of motor to generate these frequencies can be ignored.

5. CONCLUSION

The article has looked to changes of chassis dynamic behavior caused by change in usage with finite element method. First six frequency modes of the modified chassis that determine its dynamic behavior are below100 Hz and vary from 42.423 to 79.531 Hz. For the first two modes and sixth mode, the car chassis experienced global vibration. The global vibrations of the car chassis include torsion and vertical bending with 2 nodal points. The local bending vibration occurs at the top hat cross member where the gearbox is mounted on it. Since chassis mass increases due to the installed equipment, the natural frequencies fall out of the natural range that can be compensated with increasing the chassis stiffness. Decreasing the chassis length, can increase the chassis stiffness. Using this method, we can prevent resonance phenomenon and unusual chassis vibration and place the natural frequencies in natural range.

REFERENCES

[1] Fahi, F., and Walker, J., Advanced Applications in Acoustics, Noise and Vibration, 1st ed, spon press, London, 2004.

[2] Fui, T. H., Rahman, R. A., "Statics and

Dynamics Structural Analysis of a 4.5 Ton Car Chassis," Jurnal Mekanikal, No. 24, 2007, pp. 56-67.

[3] Rao, S. S., Mechanical Vibration, 4 th Edition, pearson education press, 2004.

[4] Jason, C., Brown, A., John Robertson, Stan, T., Serpento,Motor vehicle structures, Butterworth-Heinemann, the University of Michigan, 2002.

[5] Vasek, M., Stejskal, V., Sika, Z., Vacalin, O., and, Kovanda, J., "Dynamic Model of Car for Suspension Control", Vehicle System Dynamic, Supplement 28,1998, pp.496-505.

[6] Zhang, Y., and Tang, A., "New Approach for Vehicle System NVH Analysis," SAE International, Paper Number 981668 Detroit, MI, USA, 1998.

[7] Guo Y. Q., chen, W. Q., Pao, Y. H. ,"Dynamic analysis of space frames," The method of Reverberation-ray matrix and the orthogonality of normal modes ,2008.

[8] Johansson, I., Edlund, S., Optimization of Vehicle Dynamics in Cars by Use of Full Vehicle FE- Models, Göteborg, Sweden, Department of Vehicle Dynamics & Chassis Technology, Volvo Car Corporation, 1993.

[9]. Dave Anderson and Grey Schede. Development of a Multi- Body Dynamic Modal of a Tractor – Semi trailer for Ride Quality Prediction. International Truck and Engine Corp. 2001.

[10] I.M. Ibrahim, D.A.Crolla and D.C. Barton. Effect Of Frame Flexibility On The Ride Vibration Of Trucks. Department of Mechanical Engineering, University of Leeds LS2 9JT, U.K. August 1994.

[11] Pomulo Rossi Pinto Filho. Automotive Frame Optimization. Universidade Federal de Uberlandia. November 2003.

[12] Zaman@Bujang, Izzudin and Abd. Rahman, Roslan (2006). Application of Dynamic Correlation Technique and Model Updating on Truck Chassis. 1st Regional Conference on Vehicle Engineering & Technology, July 2006.

[13] Lonny L. Thomson, Jon K. Lampert and E. Harry Law. Design of a Twist Fixture to measure the Torsional Stiffness of a Winston Cup Chassis. Department of Mechanical Engineering, Clemson Univ.1998.

[14] Murali M.R. Krisna. Chassis Cross-Member Design Using Shape Optimization. International Congress and Exposition Detroit, Michigan. February 23-26,1998.

[15] Wesley Linton. Analysis of Torsional Stiffness and Design Improvement Study of a Kit Car Prototype. Cranfield University, September 2002.

[16] Marco Antonio Alves Jr, Helio Kitagawa and Celso Nogueira. Avoiding Structural Failure via Fault Tolerant Control – An Application on a Truck Frame. Detroit, Michigan November 18-20, 2002.

[17] http://www-5.jeep/vensuite/vehiclecompare.jsp

[18] Jeroen Deweer and Tom Van Langenhove. Identification of the Best Modal Parameters and Strategies for FE Model Updating. Society of Automotive Engineers, 2001.

[19]. Wesley Linton. Analysis of Torsion Stiffness and Design Improvement Study of A Kit Car Chassis Prototype. Cranfield University 2001-2.

[20]. Lonny L. Thomson, Pipasu H. Soni, Srikanth Raju, E. Harry Law. The Effects of Chassis Flexibility on Roll Stiffness of a Winston Cup Race car. Departmental of Mechanical Engineering, Clemson University, 1999

[21] "ME'sope VES. Demo Guide. Vibrant Technology Inc., Central of America, 2003.