Analysis of a Tanker Truck Chassis

1Mukesh Patil, 2Rohit Thakare, 3 Aniket Bam.
1,2,3Pillai’s Institute of Information Technology & Engineering

Abstract: Tanker truck chassis forms the structural backbone of a commercial vehicle. Tanker truck chassis' main purpose it to support the body and various other components associated with it and their loads acting. The chassis experiences a stress whether when it is moving or in a static condition. This report presents an analysis of the static stress acting on the upper surface of the truck chassis. Critical parts that will lead to failure are also observed. The method used in this numerical analysis is finite element analysis (FEA). Finite element analysis helps in accelerating design and development process by minimizing number of physical tests, thereby reducing the cost and time for analysis. 3D model of the truck chassis is made using Creo Parametric and analyzed in ANSYS Workbench. The results show that the road excitation is the main disturbance to the truck chassis as the chassis natural frequencies lie within the road excitation frequency range. The mode shape results determine the suitable mounting locations of components like engine and suspension system. Some modifications are also suggested to reduce the vibration and to improve the strength of the truck chassis.

I. INTRODUCTION

Automobile chassis usually refers to the lower body of the vehicle including the tires, engine, frame, drive line and suspension. Out of these, the frame provides necessary support to the vehicle components placed on it. Also the frame must be strong enough to withstand the shock, twist, vibrations and other stresses.

The chassis frame consists of side members attached with a series of cross members. If any of the excitation frequencies coincides with the natural frequencies of the truck chassis, then resonance phenomenon occurs. The chassis will undergo dangerously large oscillations, which may lead to excessive deflection and failure. The vibration of the chassis will also cause high stress concentrations at certain locations, fatigue of the structure, loosening of mechanical joints, and creation of noise and vehicle discomfort. To solve these problems, study on the truck chassis static and dynamic analysis is essential. Thus stress analysis and Modal analysis of TATA LPT-1613 is done using Finite Element Method (FEM) to locate the critical point with the highest stress and vibrational characteristics like natural frequency and mode shape.

II. OBJECTIVE

• Performing the structural analysis only the pay load, engine load, cabin load are taken into consideration while the other miscellaneous loads are neglected for the ease of analysis.

• Modal analysis is used to determine the vibration characteristics and natural frequencies for first 10 modes only.

III. LITERATURE REVIEW


2. Hirak Patel, Chetan Jadhav, 2013 [8] has undergone a research regarding structural analysis of a truck chassis frame and design optimization for weight reduction. The paper highlights that maximum stress; maximum equilateral stress and deflection are important criteria for design of chassis frame. This report is the work performed towards the optimization of the automotive chassis with above constraints. A sensitivity analysis is also carried out using ANSYS workbench for weight reduction.

3. P.K.SharmaandNilesh Parikh, February 2014 [10] have gone through linear finite element analysis to identify critical parts that will lead to failure as a result of stress distribution acting on the chassis. This paper presents analysis of static stress acting on the upper surface of truck chassis.

4. Yatin RaturiandAmit Joshi, August 2013 [9] have done a thorough FEA analysis on TATA SUPER ACE chassis and its verification using solid mechanics. The chassis has been modeled in CATIA v5R18 using the most of actual dimensions and FEM analysis was done using ANSYSv14 WORKBENCH.

5. A paper by Teo Han Fui and Roslan Abd Rahman,[3] December 2007 published on the modal analysis of chassis using FEA presents work
on dynamic load characteristics. Location of high stress area, natural frequency of vibration and mode shape are analyzed using finite element method. The results show that the road excitation is the main disturbance to the truck chassis as the chassis natural frequencies lie within the road excitation frequency range. Some modifications are also suggested to reduce the vibration and improve strength of truck chassis.

6. Dr. R Rajappan and Mr. Vivekanandhan, 2013 [6] have done a thorough research on the modal analysis of chassis published in the International Journal of Engineering and Science determines the torsion stiffness, static and dynamic mode shape of truck chassis by using torsion testing. Improvement in the dynamic behavior of truck chassis is suggested by changing the geometrical dimension and structural properties. Modal updating of truck chassis model is done by adjusting selective properties such as mass density and Poisson’s ratio. Predicted natural frequency and mode shape are validated. The modifications of updated FE truck chassis model are proposed to reduce vibration and optimize weight of chassis.

IV. METHODOLOGY:

- Dimensions of the chassis are taken from the data provided from the company.
- From the data obtained, solid model of each component is made and assembled in PTC Creo Parametric v2.0.
- The assembly model is imported in simulation and analysis tool ANSYS Workbench v14.5.
- The model is assigned material properties.
- Meshing of the assembly is done in ANSYS.
- Boundary conditions and loads are applied to the assembled meshed model.
- Deflection and modal analysis is performed.
- Post processing is carried out and the report is generated.
- Result is concluded on the basis of the reports generated.

V. PROCESSING

Modal analysis has been performed after creating the chassis solid model i.e. finite element model and meshing in ANSYS. The calculation of results is done for the first 10 frequency modes and show that road simulations are the most important hurdle for truck chassis.

There are two types of vibration, which are global and local vibrations. The global vibration indicates that the whole chassis structure is vibrating while the local means the vibration is localized and only part of the truck chassis is vibrating.

Diesel engine is known to have the operating speed varying from 8 to 33 revolutions per second (RPS). 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.

The main excitations are at low speeds, when the truck is in the first gear. At higher gear or speed, the excitations to the chassis are much less. The natural frequency of the truck 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 truck chassis when the truck 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 truck on the nodal point of the chassis is one of the vibration attenuation methods to reduce the transmission of vibration to the truck chassis. The mounting location of the engine and transmission system is along the symmetrical axis of the chassis first torsion mode where the effect of the first mode is less. However, the mounting of the suspension system on the truck chassis is slightly away from the nodal point of the first vertical bending mode. This might be due to the configuration of the static loading on the truck 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 have been verified by comparing the results from a similar model with the model proposed by Mr. Fouil[3] and his cooperator.

Assumptions made in the ANSYS software:

- Point load is taken into consideration
- The material used is St-37 which has the following properties
  a) Compressive Yield Strength – 250MPa
  b) Tensile Yield Strength – 250MPa
  c) Tensile Ultimate Strength – 490MPa
  - Poisson’s ratio is taken as 0.3
  - Young’s modulus is taken as 2.1x10^5
Material assumed is linear isotropic and homogeneous

The effect of heat is neglected.

The analysis performed is linear and independent of time and displacement.

Fig.1: Imported assembly of chassis in ANSYS

VI. RESULT & DISCUSSION

- Linear static structural analysis

- In a Static Analysis, the calculation of the effect of steady loading conditions on a structure is performed while the effect of inertia and damping effects which are caused by time and varying loads are neglected. The determination of the displacements, stresses and forces in structures or components caused by loads is done.

- Static Structural analysis

Modal analysis

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There are two types of vibration, which are global and local vibrations. The global vibration indicates that the whole chassis structure is vibrating while the local means the vibration is localized and only part of the truck chassis is vibrating. The first mode shape of the truck chassis is at 21.35 Hz as shown in fig.4. The chassis experienced first bending mode about Y-axis (longitudinal). The maximum translation was at the center of the truck chassis.

The second mode shape was twisting about axis-Z at 24.305 Hz as shown in fig.5. The maximum translation was at the rear end of the truck chassis.

The third mode shape was at 26.109 Hz that experienced first bending about axis-Y. The maximum translation was at the rear end of the chassis as shown in fig.6.

The fourth mode shape at 32.908 Hz experienced second twisting mode about axis-Z and maximum translation occurred at rear end of the chassis.

The fifth mode shape was 43.202 Hz that experienced combined torsion and twisting mode.

The sixth, seventh, eighth, ninth and tenth mode shape are shown in Fig.9, Fig.10, Fig.11, Fig.12 and Fig.13 respectively. It was found at 43.303 Hz, 50.001 Hz, 53.57 Hz, 67.843 Hz, and 70.181 Hz respectively where the chassis experienced the global vibrations in terms of bending and twisting.

It is known that the operating speed of Diesel Engine varies from 8 to 33 RPS and 8 to 10 RPS in idling conditions. Thus translating into excitation frequencies varying from 24 to 30 Hz.

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Fig.4: First Mode Shape of Deformation at frequency 21.35 Hz

Fig.5: Second Mode Shape of Deformation at frequency 24.305 Hz

Fig.6: Third Mode Shape of Deformation at frequency 26.109 Hz

Fig.7: Fourth Mode Shape of Deformation at frequency 32.908 Hz

Fig.8: Fifth Mode Shape of Deformation at frequency 43.202 Hz

Fig.9: Sixth Mode Shape of Deformation at frequency 43.303 Hz

Fig.10: Seventh Mode Shape of Deformation at frequency 50.001 Hz
VII. CONCLUSION

The maximum shear stress occurred by FE Analysis in ANSYS is 672.07 MPa.

The maximum displacement by numerical simulation is 14.01 mm.

The report has looked into changes of chassis dynamic behaviour caused by change in usage with finite element method. First ten frequency modes of the chassis that determine its dynamic behaviour are below 100 Hz and vary from 21.35 Hz to 70.181 Hz. For the last five modes the truck chassis experienced global vibration. The global vibrations of the truck chassis include torsion and vertical bending with 4 nodal points. 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 the natural range.

REFERENCES


