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Static and Modal Analysis of Chassis by Using Fea

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Abstract

Truck chassis is a major component in a vehicle. In truck chassis different type of failures are occur due to static and dynamic loading condition. In this present work static and dynamic load characteristics are analysed using FE models from this work. It is found that indtifying location of high stress area, analyzing vibration, natural frequency and mode shape by using finite element method. Modal updating of truck chassis model will be done by adjusting the selective properties such as mass density and Poisson's ratio. Predicted natural frequency and mode shape will be validated against previously published result. Finally, the modifications of the updated FE truck chassis model will be proposed to reduce the vibration, improve the strength, and optimize the weight of the truck chassis.

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I. Introduction

There are many industrial sectors using this truck for their transportations such as the logistics, agricultures, factories and other industries. 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, creation of noise and vehicle discomfort. To solve these problems, study on the truck chassis dynamic characteristics is thus essential. The torsion stiffness and modal parameters were determined experimentally and then used to validate the finite element model and finally the chassis was optimized to increase the structural stiffness. It was noted that the torsion mode dominated the natural frequency.

1.1 Objectives

The objectives of this study are:

i) To determine the torsion stiffness and static and dynamic mode shape of the truck chassis by using torsion testing, modal analysis and finite
ii) To improve the static and dynamic behaviour of the truck chassis by changing the geometrical dimension and structural properties.
iii) To develop a new truck chassis.

II. Chassis

The truck chassis used for the study has a narrow body with a gross weight of 4.5 ton and a payload of 2500 kg. It consists of two C-channel side rails and have 5 cross members along the 2 side rails as shown in Figure 1. There are some additional members like flat and gusset brackets located at the joint between side rails and cross members to strengthen the joints. Towards the middle is a top hat cross member to provide space for mounting of the gear box. The final 2 cross members are C-channel and top hat member. These are located exactly at the location where the rear suspension is mounted at the side rails. It is to strengthen the chassis frame as the suspension mounting point is a highly stressed area. The material of the truck chassis is AISI 4130 alloy with quenched and tempered treatment.

2.1 The properties of the material are listed below:

Chemical Composition by weight, % = 0.30 C, 1.0 Cr, 0.90 Mn, 0.20 Mo
Modulus of Elasticity, E = 207 GPa
Mass Density,ρ = 7798 kg/m^3
Yield Strength = 910 MPa
Tensile Strength = 1030 Mpa

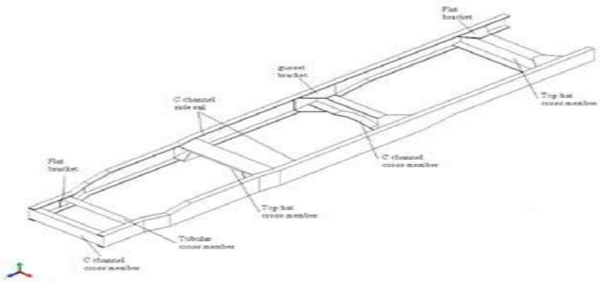

Figure1 (a) Without mounting bracket

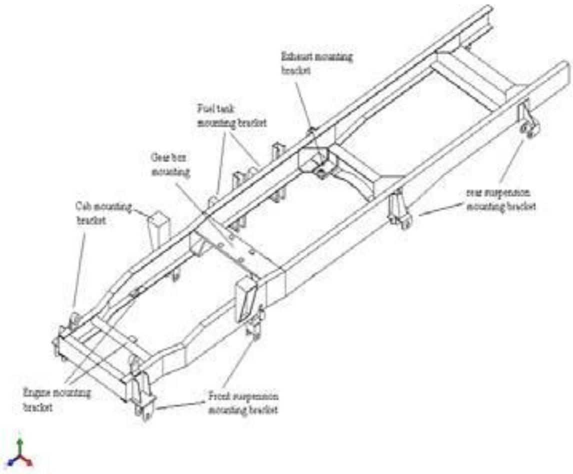

Figure 2 (b) With mounting bracket

2.2 Dynamic Analysis of a Modified Truck Chassis

Truck chassis forms the structural backbone of a commercial vehicle. When the truck travels along the road, the truck 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 municipal service (street sweepers) and it raises the question: Are natural frequencies of the modified chassis in suitable range? 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 truck chassis as the chassis natural frequencies lie within the road excitation frequency range. Therefore it is important to include the dynamic effect in designing the chassis. Many researchers carried out study on truck chassis. In their method in addition to simulating truck with finite element packaged ANSYS and being sure that structure vibrational modes are in appropriate range, they vibrationally analyzed it. This paper deals with a 6 ton truck chassis that includes natural frequencies and mode shapes. This chassis has been shortened by related Companies for using in municipal service (street sweeper) and here we face the challenge and it raises the question: Are natural frequencies of the modified chassis in suitable range? 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 more accurate results with the same meshing containing the related 3-dimensional elements. Truck chassis has been modeled with 4-node shell element in ANSYS. Numerical studies on simple hollow 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.

III. Various Impacts On The Trucks

3.1 Frontal impact

For cases of frontal impact, there has to be a frontal crush zone that will absorb and distribute the forces when the truck crashes head-on, whether into another truck, a large tree, or a bridge abutment. Structurally, the frontal shroud should not be shoved rearward and impinge into the front-door hinges and door-latch system. There must not be any penetration of frontal structures rearward through the windshield, which ought to be a tri-laminate of glass-plastic glass, retained with both adhesives and mechanical-retaining frames. The driver and front passengers should have the protection of a seatbelt-restraint system, preferably integrated within tough seats, with strong anchorages. The driver and passengers also need the protection of airbags, designed with internal tethers to control the shape when inflated. The windshield pillars, roof siderail, instrument panel, and under-dash must all be padded with energy-absorbing foam materials.

3.2 Side impact

For side impacts, the truck must have strong frame members and rocker sections that are at the outermost periphery of the vehicle body, or if the main structural frame members are further inboard, serve as extensions. These members should be internally reinforced with a baffle plate, or with rigid-foam filling, or both. Such reinforcement measures will typically triple the compressive and bending strength of the hollow member that it is filling. The doors should have internal reinforcement beams, with strong hinges and door-latch system. The remote rod must not be a tension-type, where flexing of the remote rod can cause the latch to open. The driver’s seat needs a wrap-around contour for additional protection in side impacts. The cab must have side-curtain airbags that inflate in side impacts, and in rollovers. The side windows must be of laminate glass, typically of enhanced protective-glass design that is a tri-laminate of glass-plastic-glass.

3.3 Rear impact

For cases of rear impact, the seats should be designed with high backrests and integrated head restraints to help keep the head, neck, and upper torso in safe alignment during the dynamics of a crash.

3.4 Rollover

For incidences of rollover accidents, the cab needs to incorporate a strong safety-cage construction with

No.	Components	Weight (kg)	Load (N)	Position from origin (mm)
1	Cab	125	1226	4183
2	Engine	50	490	3875
3	Engine	100	981	3523
4	Cab	125	1226	3216
5	Gear box	50	490	2873
6	Pay load	417	4088	2873
7	Fuel tank	40	392	2433
8	Pay load	417	4088	2150
9	Chassis weight	200	1962	2150
10	Fuel tank	40	392	2023
11	Exhaust	20	196	1805
12	Pay load	417	4088	1710
13	Pay load	417	4088	1080
14	Pay load	417	4088	450
15	Pay load	417	4088	0

roof pillars, windshield header, side rails, and cross-members, all internally reinforced with baffle plates and lightweight-rigid foam, thereby increasing each member's compressive and bending strength by at least a factor of three. The roof's structural integrity should be capable of being maintained in a dynamic lateral rollover test, at a speed of at least 50mph. Side-curtain airbags should inflate when the sensors determine that a lateral rollover sequence is beginning, and should stay crash test Technology International Safe liner trucks, which provide under ride protection around the entire truck.

3.5 Weights and forces of components and positions along the chassis

IV. Mounting Location Of Components On The Chassis

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. Figure 3 shows the mounting location of the engine and transmission system on the chassis which is along the symmetry axis of the chassis first torsion mode. At such location, the excitation from the engine can be reduced for the first torsion mode.

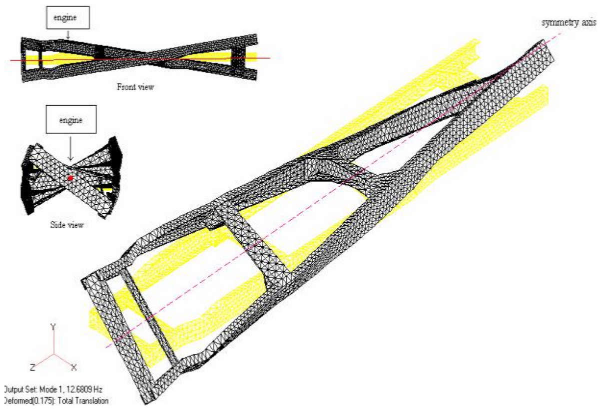

Figure 4.1: Mounting location of engine and transmission on the chassis

The first vertical bending mode with the location of nodal points and the truck wheel center location is shown in Figure 4.1. It is apparent that the wheel center is located not on the nodal point, but close to the nodal point. The distance between the nodal point and the wheel center is about 306 mm for front wheel and 127 mm for rear wheel. This is mainly due to the configuration of the static loading on chassis which determines the mounting of the suspension system. The mounting location of the suspension is suitable because the excitation from the suspension input motion can be reduced for the vertical bending mode.

4.1 STATIC ANALYSIS TRUCK COMPONENTS LOADING

This simulation is based on the condition of the truck being stationary. The ladder frame chassis was treated as a simply supported beam and loads were due to the weight of components applied to the beam. The support loads from the axles were distributed through spring hangers. The axle's reaction loads were obtained by resolving forces and taking moments from the weights and positions of the components. For practical calculations, it was recommended that the load on the chassis frame, including its own weight, is concentrated at a small number of points. These point loads were statically equivalent to the actual distributed load truckried by the vehicle. The weight of components mounted was considered as point loads acting on the chassis. The pay load of 2500 kg was divided into six equivalent forces acting on positions where the truck go is mounted. The weights and forces of the components and their positions along the chassis.The axle reaction forces acted vertically upward and applied at the center position between the suspension mounting brackets. The result shows that the front axle reaction force is 854 kg (8376N) and the rear axle reaction force is 2396 kg (23507 N)

V. Stress Analysis Of Truck Chassis

5.1 Asymmetrical Loading

This simulation is based on the condition whereby one of the truck's front wheel rest on a hump, thus causing torsion to the chassis. Figure 19 illustrates the stress contour and deformation pattern of the chassis under asymmetrical loading. The front suspension brackets with constraint experience highest stress, about 16 KN. The other region has less stress value, below 12KN Figure 16 shows the deformation contour and deformation pattern of the chassis under asymmetrical loading. The front part of the chassis where the asymmetry load is applied experiences the highest translation, whose magnitude depends on the height of hump. For the two types of loading, it was shown that the suspension mounting brackets experienced the highest stress when; the truck components loading 128MPa and asymmetrical loading 490MPa The maximum stress of 16 KN is below the yield strength of the chassis material which is 910 MPa. The additional hump loading on the chassis increased the stress. As for the other regions of the chassis, low stresses were obtained. Thus, the chassis structure and joints are sufficiently strong to withstand the loading applied to it.

5.2 Modal analysis

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 30 frequency modes and show that road simulations are the most important problematic for truck 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 and3 modes are related to chassis rotation about x, y and z axes. In Fig.8 related natural frequencies and mode shapes for chassis with maximum displacement in y direction in each mode, have been shown. The first, second and sixth modes are the global vibrations, while the others are local vibrations. Local vibration starts at the third mode at 10.661Hz. The dominant mode is a torsion which occurred at 7.219Hz with maximum translation experienced by both ends of the chassis. The second mode is a vertical bending at 7.49 Hz. At this mode, the maximum translation is at the front part of the chassis. The third and fourth modes are localized bending modes at29.612 Hz and 33.517 Hz. The maximum translation is experienced by the top hat cross member. the member also experienced big translation at fifth mode which is a localized torsion mode. The top hat cross member is the mounting location of the truck gear box. The sixth mode is the torsion mode at 38.475 Hz with maximum translation at both ends of the chassis. Found natural frequencies from modal analysis of truck chassis, are used for determining the suitable situations for truck parts in working conditions.

5.3 SPECIFICATION OF TRUCK CHASSIS

Truck chassis outer length = 85mm

1. Truck chassis inner length = 84mm
2. Truck chassis outer length = 16mm
3. Truck chassis inner length = 14mm
4. Chassis thickness = 7mm
5. Cross bar length = 22mm
6. Total number of cross bar = 5

5.4 MODELING AND OPTIMIZATION ANALYSIS

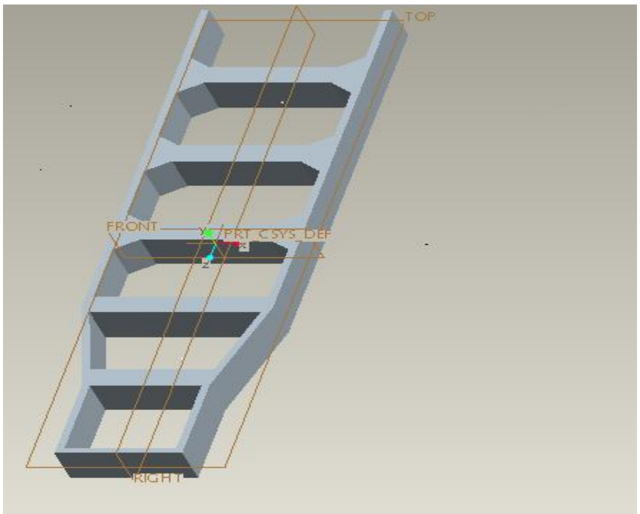

Figure 4: TRUCK CHASSIS MODELING

11.1 DEFORMATION OF THE TRUCK CHASSIS

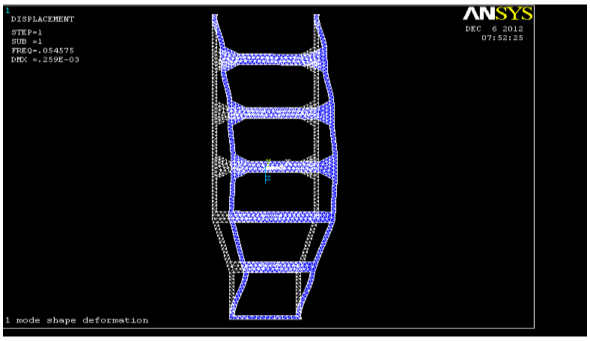

Figure 5: FIRST MODE SHAPE OF DEFORMATION

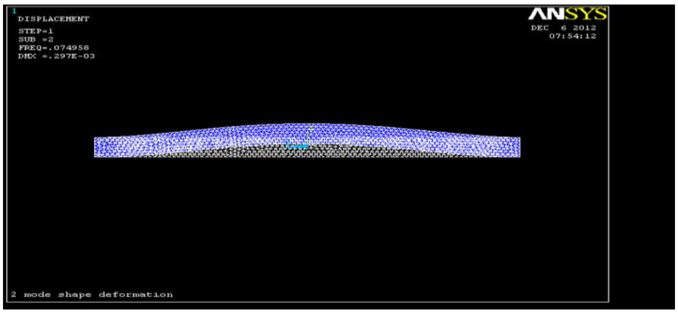

Figure 6 : SECOND MODE SHAPE OF DEFORMATION

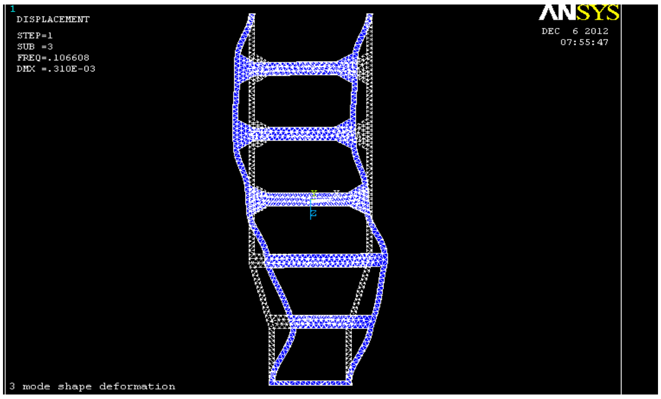

Figure 7 : THIRED MODE SHAPE OF DEFORMATION

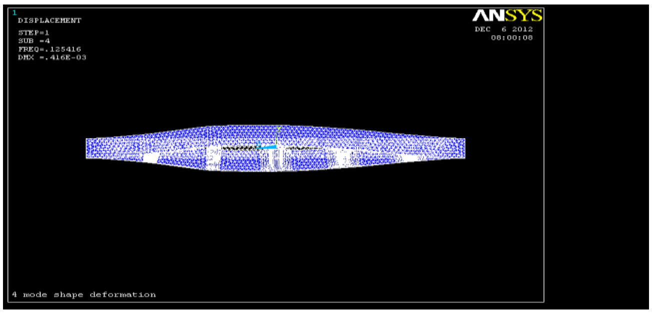

Figure 8 : FOURTH MODE SHAPE OF DEFORMATION

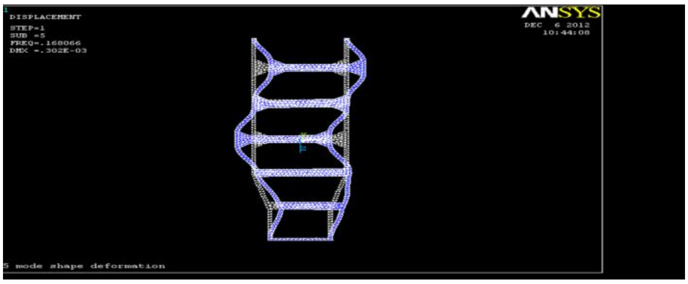

Figure 9: FIFTH MODE SHAPE OF DEFORMATION

11.2 DISPLACEMENT OF THE TRUCK CHASSIS

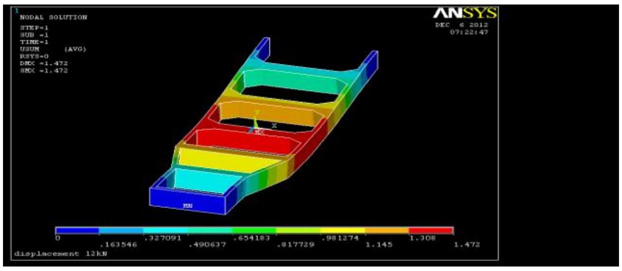

Figure 10: Total displacement results behavior for the Load of 12KN sectioned truck chassis.

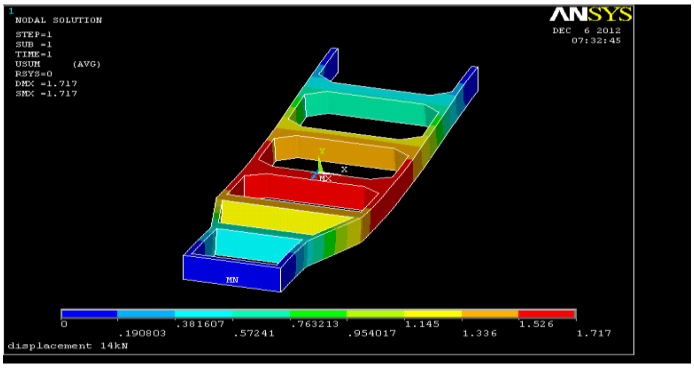

Figure 11: Total displacement results behavior for the Load of 14 KN sectioned truck chassis

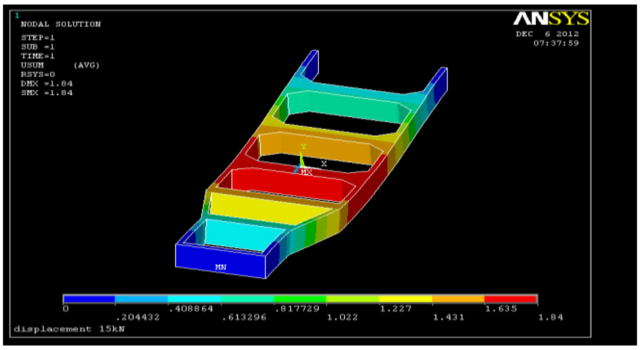

Figure 12: Total displacement results behavior for the Load of 15KN sectioned truck chassis

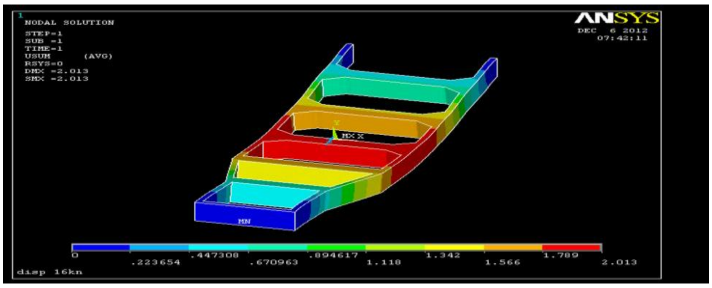

Figure 13: Total displacement results behaviour for the Load of 16KN sectioned truck chassis

11.3 STRESS ANALYSIS OF THE TRUCK CHASSIS

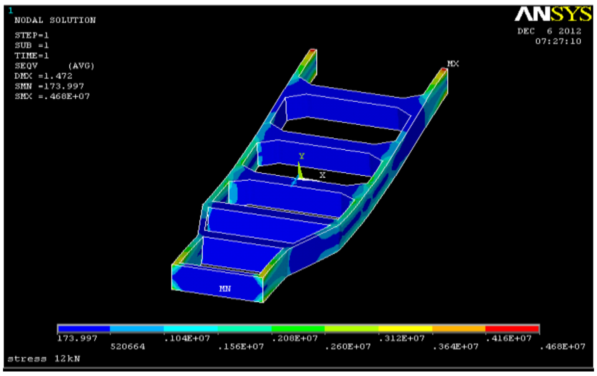

Figure 14 ;Stress contour and deformation pattern of the chassis under asymmetrical 12 KN loading

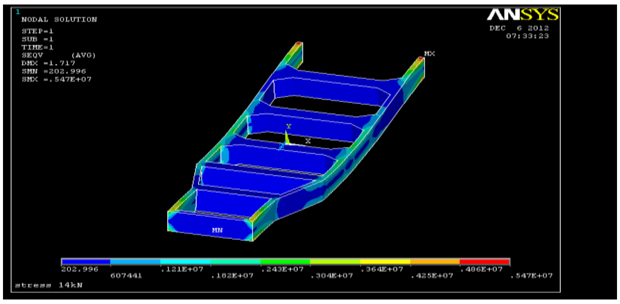

Figure 15;Stress contour and deformation pattern of the chassis under asymmetrical 14 KN loading

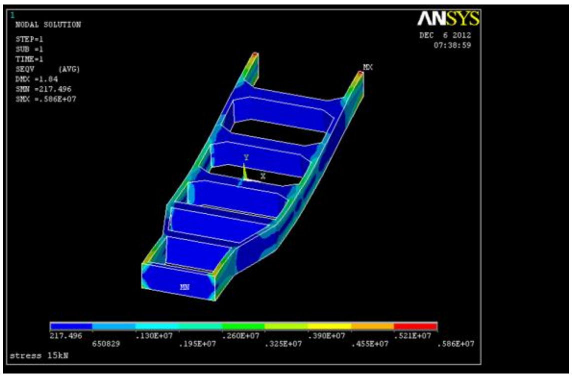

Figure 16 Stress contour and deformation pattern of the chassis under asymmetrical 15 KN loading

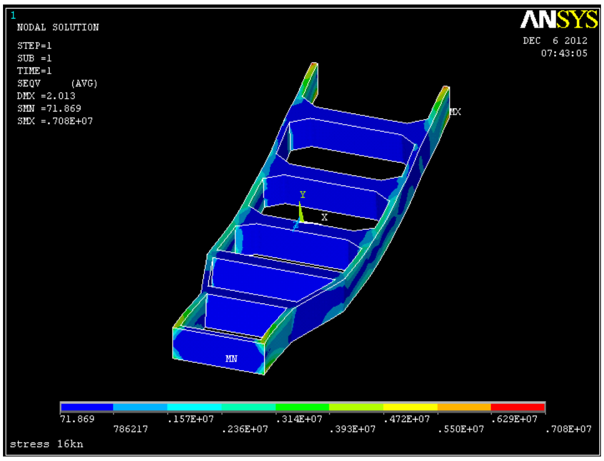

Figure 17;Stress contour and deformation pattern of the chassis under asymmetrical 16 KN loading

11.4 STRAIN ANALYSIS OF THE TRUCK CHASSIS

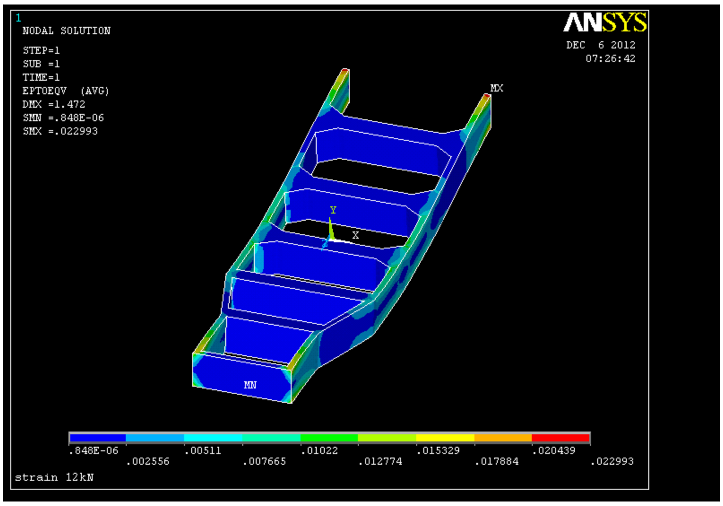

Figure 18 Total Strain results behaviour for the Load of 12KN sectioned truck chassis

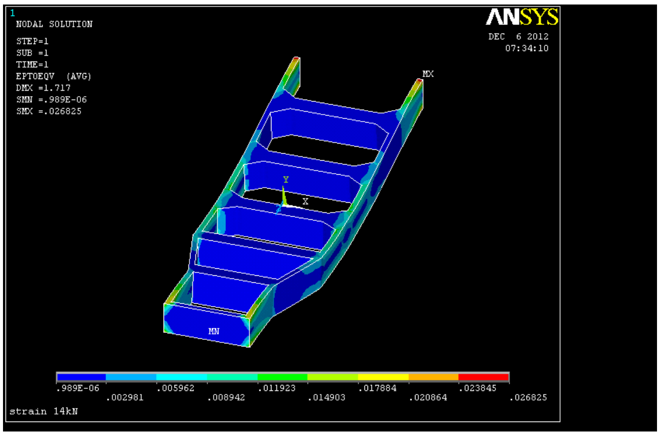

Figure 19 Total Strain results behaviour for the Load of 14 KN ectioned truck chassis

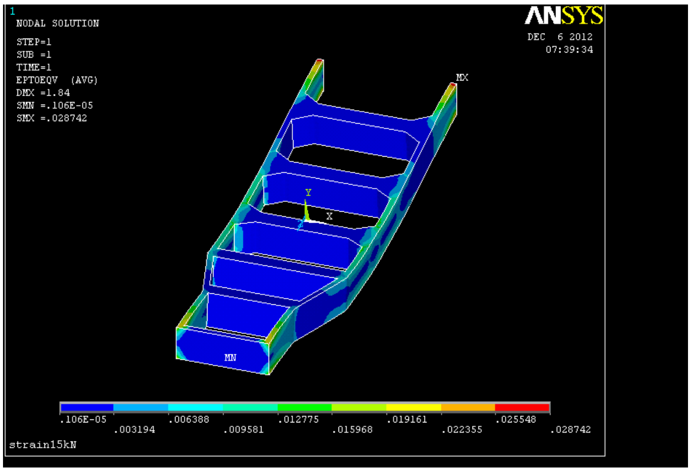

Figure 20 Total Strain results behaviour for the Load of 15 KN sectioned truck chassis

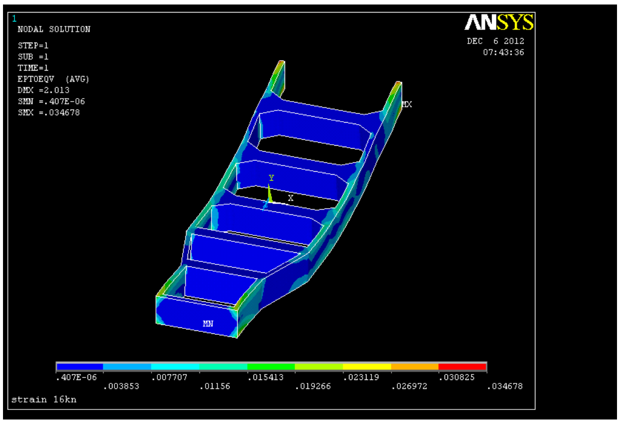

Figure 21 Total Strain results behaviour for the Load of 16 KN sectioned truck chassis

VI. Conclusion

The paper has looked into the determination of the dynamic characteristic the natural frequencies and the mode shapes of the truck chassis, investigating the mounting locations of components on the truck chassis and observing the response of the truck chassis under static loading conditions. The first six natural frequencies of the truck chassis are below 100 Hz and vary from 16.24 to 61.64Hz. For the first four modes, the truck chassis experienced global vibration except for the fifth mode. The global vibrations of the truck chassis include torsion, lateral bending and vertical bending with 2 and 3 nodal points. The local bending vibration occurs at the top hat cross member where the gearbox is mounted on it.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 truck 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 truck chassis. For the linear static analysis, the stress distribution and deformation profile of the truck chassis subjected to two loading conditions: truck components loading and asymmetrical loading had been determined. Maximum stress occurred at the mounting brackets of the suspension system while the maximum translation occurred at the location where the symmetry and asymmetry load is acting. The maximum stress of the truck chassis is 16KN while the maximum translation is 2.013mm. These values are acceptable as compared to the yield strength of the chassis material and the tolerance allowed for the chassis.

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