Design and Analysis of A Ladder Frame Chassis for Static and Dynamic Characteristics

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Abstract- The research work deals with the investigation of the static and dynamic characteristics of the considered prototype ladder frame chassis of a sports utility vehicle. The analysis has been completed using the Finite Element Method (FEM) approach. Modifications have been suggested on the current chassis, based on the suggestions two new chassis have been modeled as Modified chassis 1(modi 2) and Modified chassis 2(modi 11). The modified chassis are the made to undergo static structural analysis, modal analysis and harmonic analysis. The original and modified chassis had been modeled using the modeling software Dassault Systèmes SolidWorks and then imported into the commercial finite element package ANSYS 14.0 for further analysis. The meshing of the chassis is completed by using the auto meshing feature. In the modal analysis the first six natural frequencies of chassis are found and further process of harmonic analysis is carried out. The results obtained from the two modified chassis are compared with the original chassis, and the most optimized chassis is presented. From the results obtained from the analysis it has been concluded in the research that the modified1 chassis is the most compatible to the objectives of the research.

Keywords- Ladder frame chassis, structural analysis, modal analysis, harmonic analysis, FEM, SolidWorks, ANSYS 14.0, weight reduction

I. INTRODUCTION

The recent trend that has been adapted by the automobile industries is that of focusing on ways to deliver high quality products to the market at a faster rate and at low cost. The consumer demands products at a relatively cheaper cost with no sacrifices being made on the quality of the product being delivered. Chassis is one of the fundamental portions of the vehicles structure and requires providing enough stiffness to ensure assemble and to provide support to the whole automobile structure [17]. While the chassis is subjected to various mechanical loads in the form of shock loads and mechanical vibrations that may result in resonance phenomenon to occur, causing the mechanical structure to fail completely Even if the external vibrations are below the resonance frequency the external vibration can get transmitted to the vehicle body causing the chassis to vibrate leading to ride discomfort, jeopardizing ride safety and stability of the vehicle. Structural resonance is a phenomenon which occurs when the external or internal excitation frequency of an applied force onto the structure is equal to the natural frequency of the system. The results of resonance can be summarized as occurrence of vibrations of very high amplitude thus causing material fatigue and fracture, ensuing failure of the system. Thus, it is of utmost importance to determine the natural frequency of the system in order to avoid the occurrence of such a phenomenon. Thus considering the various effects vibration problems can have on a machine, analysis of the dynamic characteristics is very important from an engineering point of view [14].

A perfect chassis can be defined as a large diameter thin walled tube. The word chassis is of French terminology and was in the beginning used to denote the frame parts and basic structure of the vehicle [1]. In layman's terms a vehicle without a body is called a chassis. The automotive chassis can be defined to have two basic goals:

- Hold the weight of the components
- To rigidly fix the suspension components together while moving [1].

The chassis can be classified roughly based on the frame type as;

- Ladder frame chassis
- Space frame chassis
- Backbone chassis
- Tub design chassis
- Monocoque chassis

A. Ladder frame chassis-

Figure 1. Ladder frame chassis

The ladder frame chassis is the simplest and the oldest form of chassis frame ever used in the modern vehicular construction. Body on frame is an auto mobile construction method where a separate body is mounted on a rigid frame. The frame generally comprise of two longitudinal beams (usually C-sections) that run the entire length of the vehicle with provided cross members provided to hold the rails in place. The motor may be placed in the front or the rear and supported at suspension points. Once we add a passenger compartment and a trunk with a load and it becomes a simple indeterminate beam. This type of chassis provides little support for a performance automobile. The cross members are provided to prevent torsional deflection and maintain geometry. Sometimes X bracing is also done to increase torsional stiffness of main frame [1].

II. PROBLEM STATEMENT

The paper focuses on reducing the current weight of the chassis and presenting a new chassis with increased structural reliability and performance standards. The need of the hour being reduced fuel consumption and low weight. The reduced weight can be achieved by:

- Modifications in the design of the chassis
- Experimenting with new materials for the chassis

The former is utilized in this research. Two modified designs have been presented in this paper; both modified chassis have been tested against the current existing chassis. The chassis that is seen to have the better overall performance characteristics is chosen as a replacement for the formerly existing chassis. Various assumptions to be listed later have been made on the part of the researcher in the completion of this project. The results obtained from the research are from simulation and experimental physical testing is not a part of the research.

III. EXPERIMENTAL OBJECTIVE

There are several objectives regarding the computational stress and modal analysis of the car chassis. The main objective of the research being that of the reduction of weight of the chassis by; method of implementation of modification in the former existing chassis without sacrifice in strength and if possible improvement of structural properties. The various sub objectives are listed as below:

- Static Structural Analysis:
  - Finding the Von Misses stresses and its position
  - Finding the Maximum principal stresses
  - Finding Normal and Shear stresses
  - Finding maximum deflection and its position
- Modal Analysis:
  - Find the mode shapes of the chassis for the first six natural frequencies
  - Determination natural frequencies
- Harmonic Analysis:
  - Obtaining the resonance frequency for the chassis by plotting the amplitude vs. frequency graph.

IV. METHODOLOGY

The finite element method is a numerical technique for solving engineering problems. It is a powerful analysis tool used to solve simple as well as complicated problem.

- Create 3D CAD model: The 3D modeling of the chassis is done using Dassault Systèmes SolidWorks software and saved in a neutral format such as .igs format.
- Importing: Import the CAD geometry into the FEA package ANSYS 14.0.
- Material properties: The material properties are defined in the FEA package. The material to be used is structural steel.
- Meshing: In this operation the CAD geometry is divided into large number of small pieces called mesh. The auto generate mesh feature of ANSYS 14.0 is used in this research due to the limited resources at the disposal of the researcher.
- Defining boundary conditions: The loads are applied and the position of the load is defined. The constraints and the supports provided are used as input.
- Solve: The FEA package ANSYS 14.0 solves the model with the given mesh and loads for static, dynamic and harmonic analysis.
- Post processing: The reviewing of the results and the solutions are carried out in ANSYS 14.0 itself. The results are viewed in various formats such as: graph, values and animations.

A. Material specification-

The material that has been used in the manufacture of the chassis frame is Structural Steel. The physical and mechanical properties of the steel are listed below.

1. Density = 7850 kg/m^3
2. Coefficient of thermal expansion = $1.2E^{-05}/^oC$
3. Young's modulus = $2E^{+11}$ MPa
4. Poisson's ration = 0.3
5. Bulk Modulus = $1.6667E^{+5}$ MPa
6. Shear Modulus = $7.6923E^{+4}$ MPa
7. Ultimate tensile strength = 460 MPa
8. Yield tensile strength = 250 MPa
9. Yield compressive strength = 250 MPa
10. Specific heat = 434 J/Kg/K

B. Load Calculation, boundary conditions and assumptions employed for the analysis-

Due to the tedious nature of the manual measurements of all the components of the Scorpio chassis and unavailability of resources at the researcher's disposal; various assumptions were made regarding the application of loads and boundary conditions.

- The mounting brackets for the engine are not considered, and the load on the chassis is presumed to have acted on the upper surface of the two longitudinal members (Rails).
- Fixed supports are provided beneath the chassis to act as the support from axles/suspension.
- Effects of rivets and welded joints are ignored and the chassis is considered to be uniformly joined.
- The approximate kerb weight of the vehicle is 1800 Kg and about 65% of this weight approximately 1200 Kg is considered as the load acting on the chassis. The load eliminating the tires, axles, suspension, etc. the 1200 Kg is converted into Newton’s giving 11772 N. This value is rounded up and a value of 12000 N is considered for the Structural analysis of the chassis.
- The self weight of the engine is considered as the external excitation for the Harmonic analysis. The self weight of the engine is taken at 100 Kg. When converted to Newton is taken as 1000 N which is assumed to act on the centre of the cross member on the front.

V. DESIGN OF CHASSIS

Model of the existing chassis is done on the SolidWorks platform and then the .igs file is exported to ANSYS 14.0 for analysis. Two modifications are made to the original namely modified 1 and modified 2 chassis.

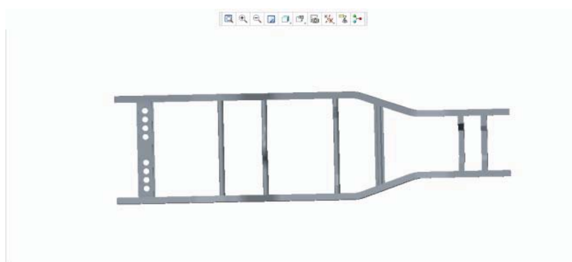
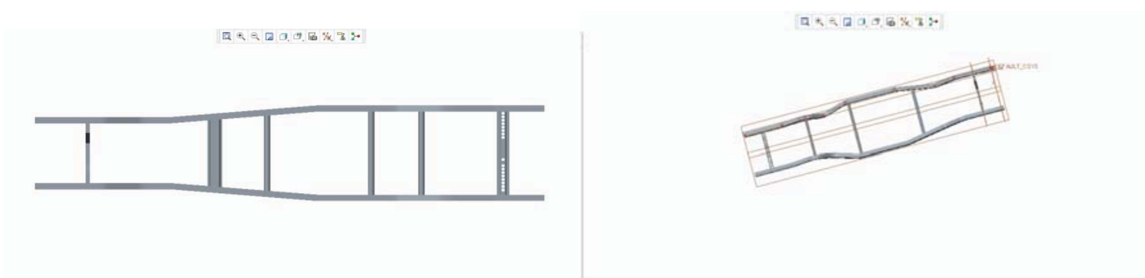


Figure 2. Model of existing chassis



(a) (b)

Figure 3. (a) Model of modified 1 chassis (b) Model of modified 2 chassis

Table – 1 Dimension of the Original Chassis

Part	Dimensions
Chassis weight	**111.84 Kg**
Total length	**4004 mm**
Total no. of cross members	**7**
No. of main rails	**2**
No. of tubular cross member	**2**
No. of cross member at front	**2**

Breadth (Front section)	**778 mm**
Breadth (Middle section)	**1237 mm**
Breadth (Rear section)	**1237 mm**
Breadth of main rails	**70 mm**
Height of main rails	**85 mm**
Cross section type	**C section**
Thickness of main rails	**5 mm**
42 inch Dropped tube cross member	**Yes**
Open cross member	**Yes**

Table- 2 Dimension of the Modified 1 Chassis

Part	**Dimensions**
Chassis weight	**94.817 Kg**
Total length	**4848 mm**
Total no. of cross members	**6**
No. of main rails	**2**
No. of tubular cross member	**2**
No. of cross member at front	**1**
Breadth (Front section)	**730 mm**
Breadth (Middle section)	**966 mm**
Breadth (Rear section)	**966 mm**
Breadth of main rails	**64 mm**
Height of main rails	**76 mm**
Cross section type	**C section**
Thickness of main rails	**5 mm**
42 inch Dropped tube cross member	**No**
Open cross member	**Yes**

Table- 3 Dimension of the Modified 2 Chassis

Part	**Dimensions**
Chassis weight	**95.562 Kg**

Total length	**4316 mm**
Total no. of cross members	**5**
No. of main rails	**2**
No. of tubular cross member	**2**
No. of cross member at front	**1**
Breadth (Front section)	**785 mm**
Breadth (Middle section)	**1017 mm**
Breadth (Rear section)	**767 mm**
Breadth of main rails	**65 mm**
Height of main rails	**76 mm**
Cross section type	**C section**
Thickness of main rails	**5 mm**
42 inch Dropped tube cross member	**No**
Open cross member	**Yes**

VI. MESHING, LOAD APPLICATIONS AND BOUNDARY CONDITIONS OF THE CHASSIS

A. Analysis of the Existing Chassis-

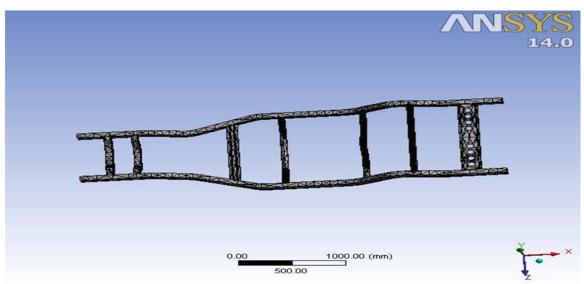
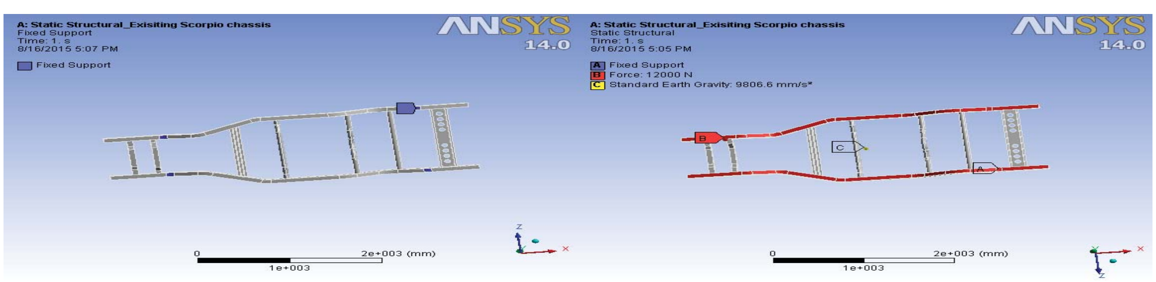
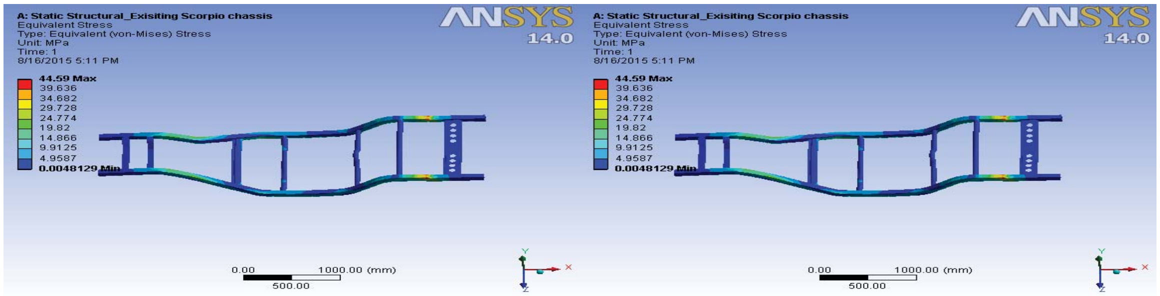


Figure 3. FEM model of existing chassis

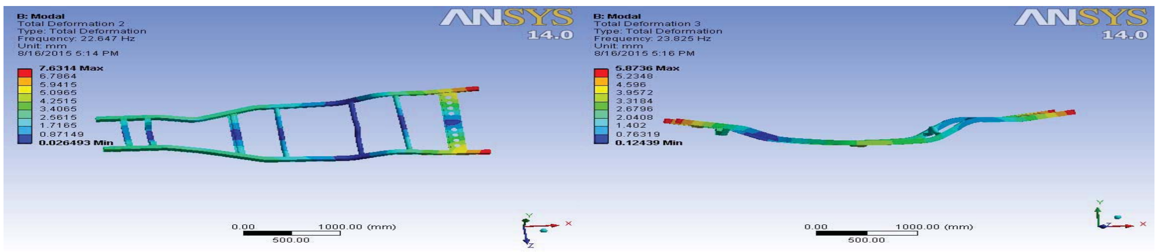


(a) (b)

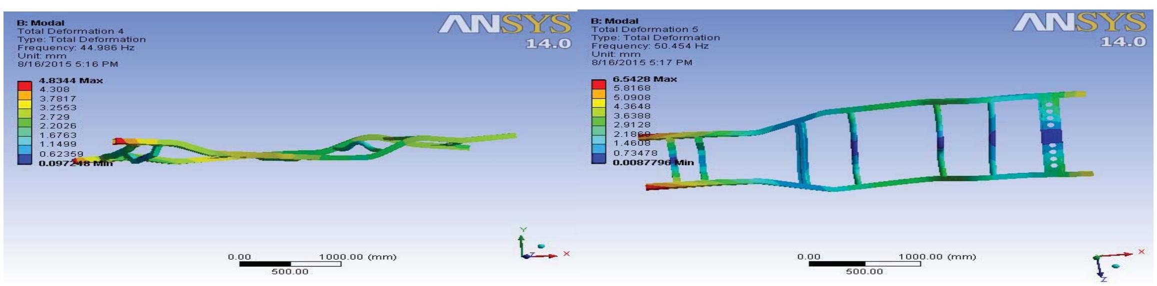


(c) (d)

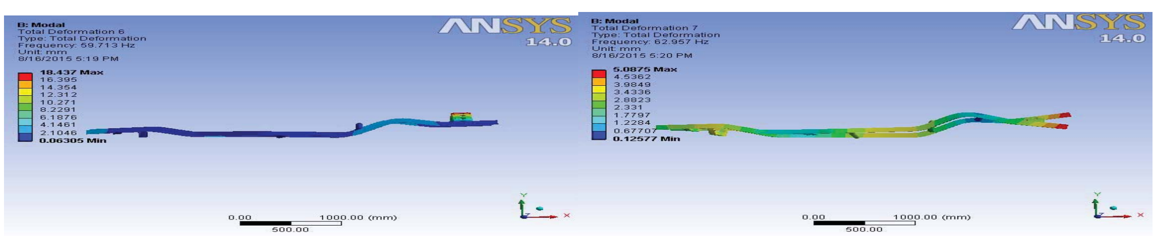
Figure 4 (a) Rigid supports (b) Loads applied (c) Equivalent stress (d) Total deformation



(a) (b)



(c) (d)



(e) (f)

Figure 5. (a) mode shape 1 (b) mode shape 2 (c) mode shape 3 (d) mode shape 4 (e) mode shape 5 (f) mode shape 6

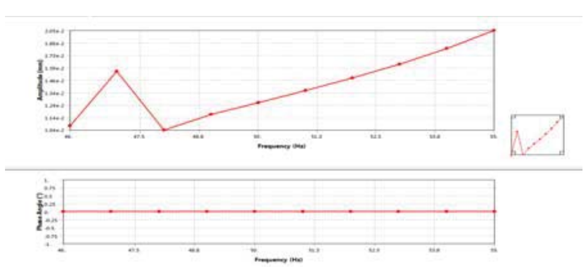


Figure 6. Frequency vs. amplitude graph

B. Analysis of the Modified 1 Chassis-

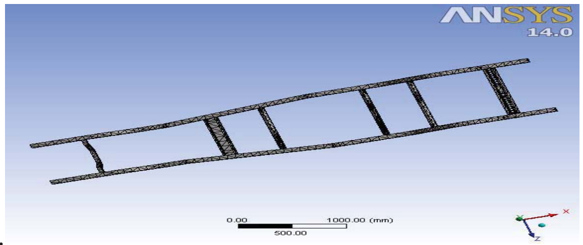
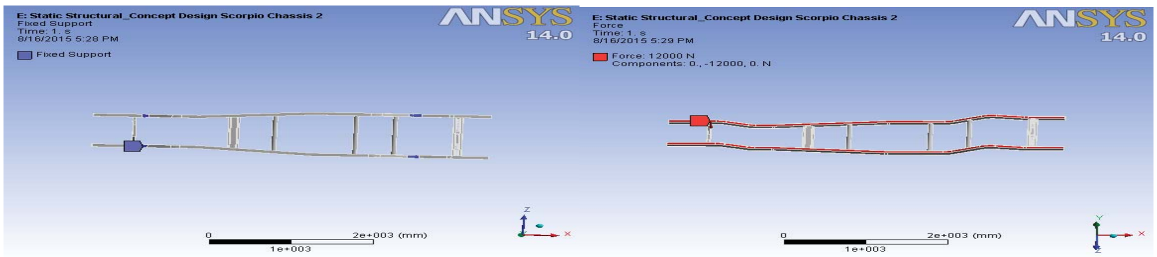
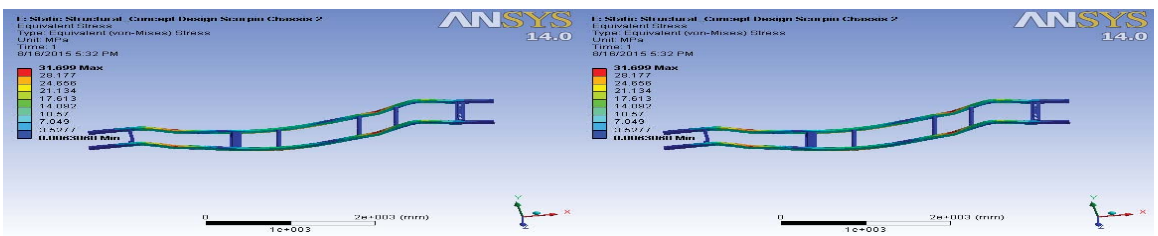


Figure 7. FEM model of the modified 1 chassis

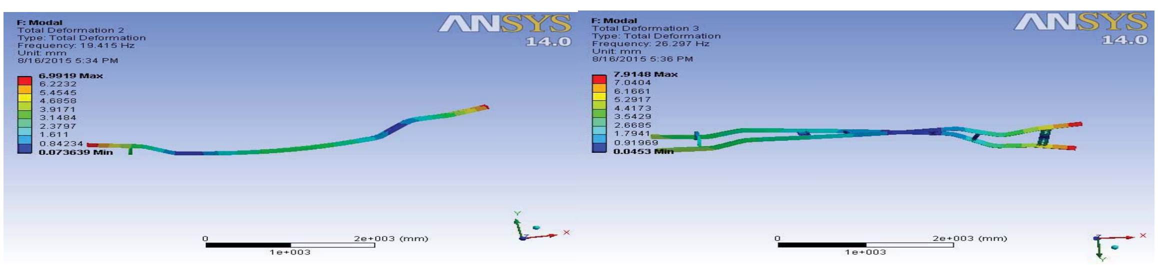


(a) (b)

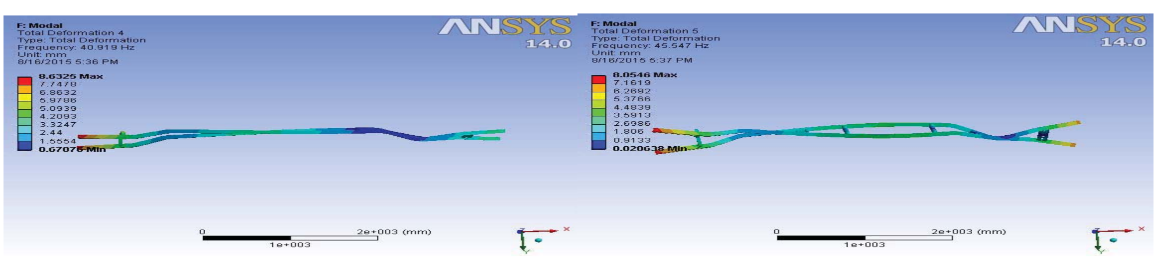


(c) (d)

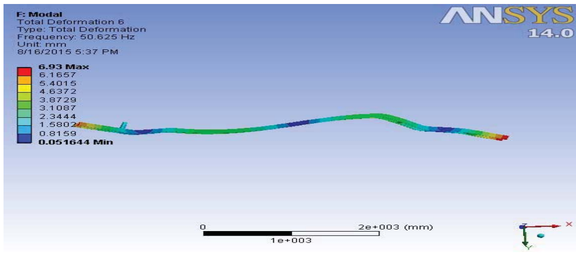
Figure 8 (a) Rigid supports (b) Loads applied (c) Equivalent stress (d) Total deformation



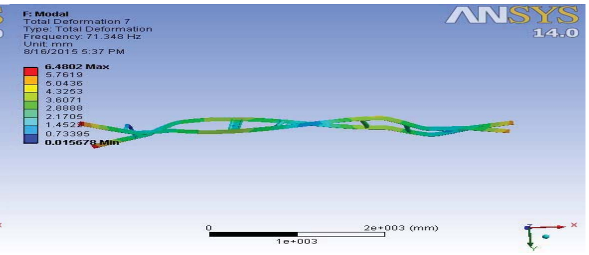
(a) (b)



(c) (d)



(e)



(f)

Figure 9. (a) mode shape 1 (b) mode shape 2 (c) mode shape 3 (d) mode shape 4 (e) mode shape 5 (f) mode shape 6

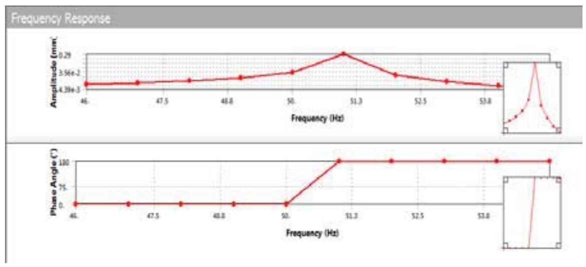


Figure 10. Frequency vs. amplitude graph

C. Analysis of the Modified 2 Chassis-

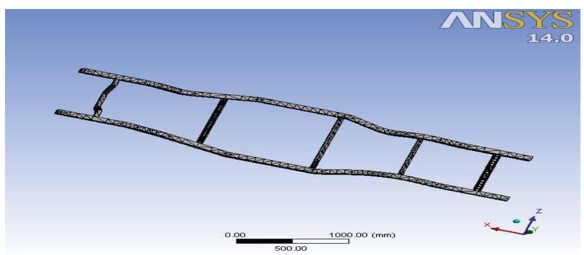
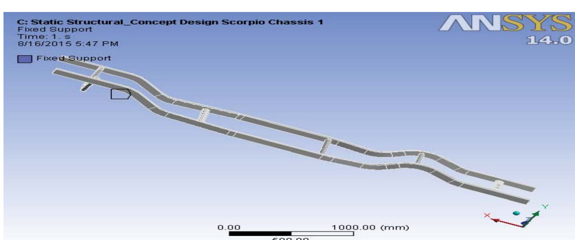
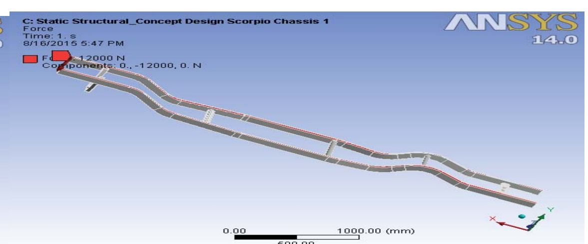


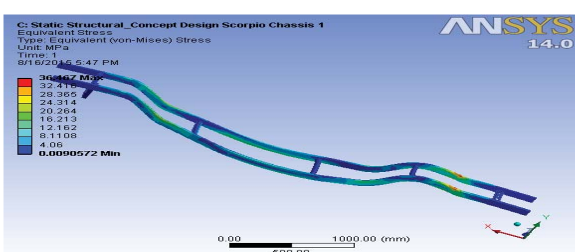
Figure 11. FEM model of the modified 2 chassis



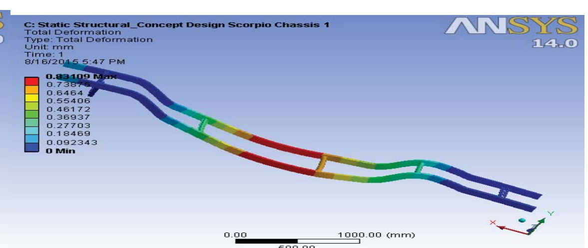
(a)



(b)



(c)



(d)

Figure 12 (a) Rigid supports (b) Loads applied (c) Equivalent stress (d) Total deformation

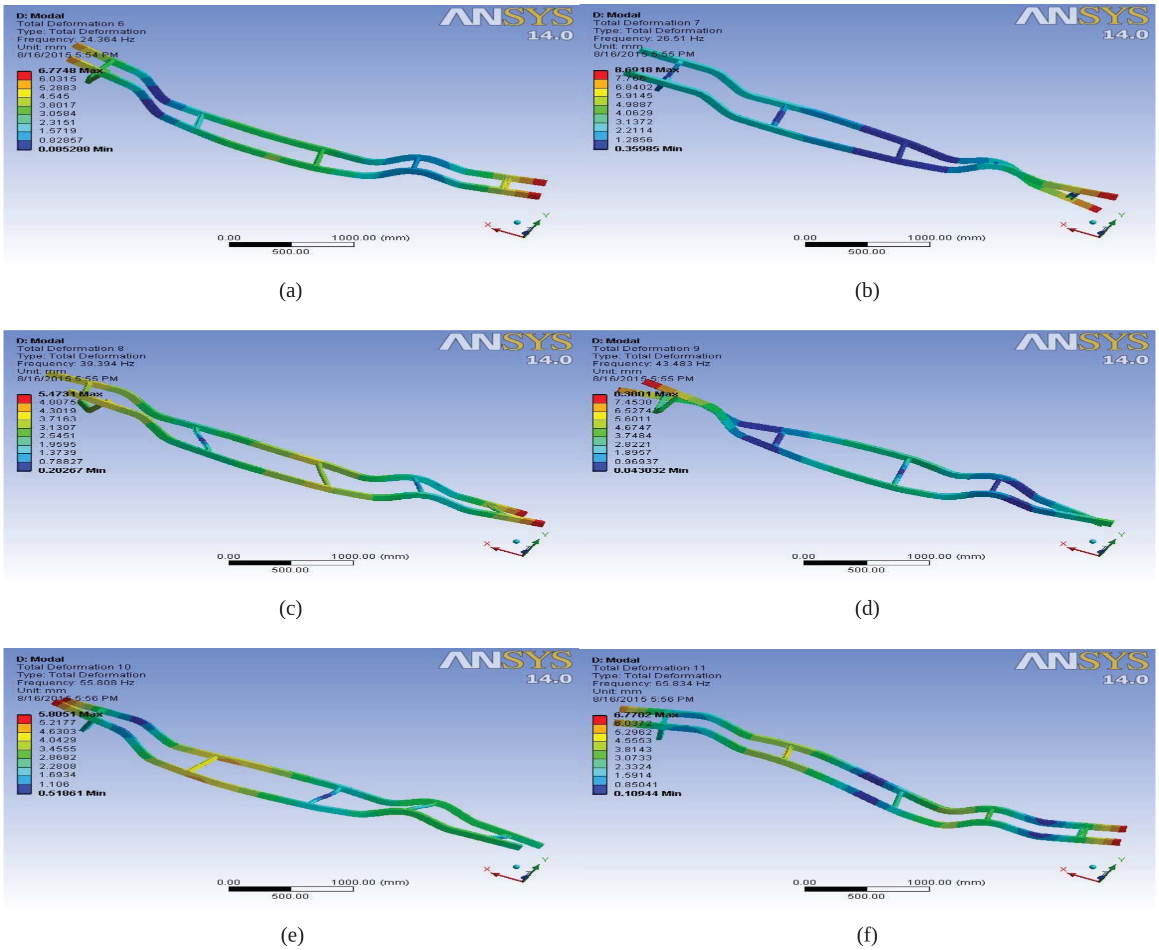


Figure 13. (a) mode shape 1 (b) mode shape 2 (c) mode shape 3 (d) mode shape 4 (e) mode shape 5 (f) mode shape 6

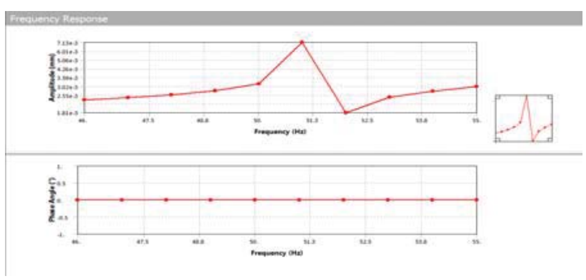


Figure 14. Frequency vs. amplitude graph

VII. RESULTS AND DISCUSSIONS

A. *Results for the Existing Chassis-*

Table- 4 Modal Analysis Results

Mode no.	Frequency (Hz)	Displacement (mm)
1	22.647	7.6314
2	23.825	5.8736
3	44.986	4.8344

4	**50.454**	**6.5428**
5	**59.713**	**18.437**
6	**62.957**	**5.0875**

Table- 5 Mode Shape Characteristics

Mode no.	Mode shape characteristics
1	Twisting motion along the longitudinal X axis
2	Translation along the Y axis
3	Lateral distortion on the X plane and twisting motion along X axis
4	Bending along Y direction and twisting motion along X axis
5	Translation of cross member along Y axis
6	Lateral bending along X plane and twisting motion

Table- 6 Harmonic Response Analysis Results

Serial no.	Frequency (Hz)	Amplitude (mm)
1	46	1.0766e-002
2	47	1.5538e-002
3	48	1.0443e-002
4	49	1.1615e-002
5	50	1.2585e-002
6	51	1.3635e-002
7	52	1.4855e-002
8	53	1.6325e-002
9	54	1.815e-002
10	55	2.0489e-002

B. Results for the Modified 1 Chassis-

Table- 7 Modal Analysis Results

Mode no.	Frequency (Hz)	Displacement (mm)
1	19.415	6.9919
2	26.297	7.9148

3	**40.919**	**8.6325**
4	**45.547**	**8.0546**
5	**50.625**	**6.93**
6	**71.348**	**6.48**

Table- 8 Mode Shape Characteristics

Mode no.	Mode shape characteristics
1	**Translation along the Y axis**
2	**Twist along longitudinal axis**
3	**Lateral distortion on the X plane and twisting motion along X axis**
4	**Lateral distortion on the X plane and twisting motion along X axis**
5	**Translation along Y axis**
6	**Lateral bending along X plane and twisting motion**

Table- 9 Harmonic Response Analysis Results

Serial no.	Frequency (Hz)	Amplitude (mm)
1	**46**	**7.655e-003**
2	**47**	**9.1469e-003**
3	**48**	**1.1554e-002**
4	**49**	**1.6504e-002**
5	**50**	**3.2349e-002**
6	**51**	**0.28952**
7	**52**	**2.278e-002**
8	**53**	**1.0817e-002**
9	**54**	**6.5713e-003**
10	**55**	**4.3864e-003**

C. Results for the Modified 2 Chassis-

Table- 10 Modal Analysis Results

Mode no.	Frequency (Hz)	Displacement (mm)
1	**24.364**	**6.7748**
2	**26.51**	**8.6918**

3	**39.394**	**5.4731**
4	**43.483**	**8.3801**
5	**55.808**	**5.8051**
6	**65.834**	**6.7782**

Table- 11 Mode Shape Characteristics

Mode no.	Mode shape characteristics
1	**Translation along the Y axis**
2	**Twist along longitudinal axis**
3	**Lateral distortion on the X plane**
4	**twisting motion along X axis**
5	**Lateral bending on X plane and twisting along X axis**
6	**Translation along the Y axis**

Table- 12 Harmonic Response Analysis Results

Serial no.	Frequency (Hz)	Amplitude (mm)
1	**46**	**2.318e-003**
2	**47**	**2.4281e-003**
3	**48**	**2.5679e-003**
4	**49**	**2.7696e-003**
5	**50**	**3.1726e-003**
6	**51**	**7.1296e-003**
7	**52**	**1.8071e-003**
8	**53**	**2.4553e-003**
9	**54**	**2.759e-003**
10	**55**	**3.0053e-003**

VIII. CONCLUSION

The modification of the chassis in contrast with the existing chassis has been carried out by iterating on various design modifications. A total of 11 modifications were carried out, from which the two modified designs were chosen: modified 1 (modi2) and modified 2 (modi11) which were examined with respect to the existing chassis. The modified 1 (modi2) chassis is presented as the more ideal chassis with a reduced weight of 94.817 Kg, reduced induced Von-Mises stress of 31.699 MPa and from the modal analysis results present a favorable result as well. The 1st (7th) mode frequency 19.415 Hz is the most critical of all the frequency and as found from the harmonic analysis the resonance frequency is at 51 Hz is well far away from it. The deflection through the prominent first six mode shapes is also found to be lesser in value when compared to the existing chassis. A reduction in weight of about 15.22% and a valuable reduction in the induced stress is also seen at 28.75%; thus demonstrating a superior chassis.

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