ISSN 2395-1621

Modeling, Static Analysis and Optimization of chassis by Using CAE Tools

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ABSTRACT

The chassis is the framework to which is everything in a vehicle is attached. For a chassis to reach these functions, it should be light enough to reduce inertia and offer satisfactory performance. It also should also be tough enough to resist fatigue loads produced due to interaction between the driver, engine, power transmission and road conditions. The aim of this work is to achieve good strength of automotive ladder chassis, so engineering solution to the component addressing functionality during the service life of the component. for reducing the overall cost as well as for smoothing the performance of chassis, optimization is needed. FEA approach for performing structural weight optimization of chassis using trial and error method to reduce the mass.

Keywords: chassis, computer aided designing, static analysis, mass reduction, optimization, etc

ARTICLE INFO

Article History

Received: 20th June 2016

Received in revised form : 20th June 2016

Accepted: 25th June 2016

Published online : 28th June 2016

I. INTRODUCTION

The design of chassis is fully on conceptual basis and the objectives are to create conceptual design for an automotive chassis which will utilize standard components. And then develop cad drawings for this concept design of chassis. We have chosen one automotive chassis for the analysis which is to be done by using finite element analysis (FEA) of the chassis design. The component should withstand all the forces acting on it without rupture or failure that might render the component incapable during its service life because of a mishap due to sudden failure during operation. An attempt to evolve an improved design resisting the failure and in turn enhancing the life.

The best design is the one that satisfies the stress and displacement constraints, and results in the least cost of construction. Although there are many factors that may affect the construction cost, the first and most obvious one is the amount of material used to build the structure. Therefore, minimizing the weight of the structure is usually the goal of structural optimization. The overall recommendation is to study the structural analysis and should be covered on the overall truck system and after that focus on the specific area such as chassis. This analysis will help to make full body refinement and improvement because it can be related to actual running condition.

It is observed that strength is prime important point for vehicle chassis design if possible with reduced weight. So for reducing the overall cost as well as for smoothing the performance of chassis, optimization is needed. FEA approach for performing structural weight optimization

II. DESIGN AND CALCULATIONS

The machine design is defined as use of scientific principles, technical information and imagination in the description of a machine or a mechanical system to perform specific functions with maximum economy and efficiency. It is also a decision making process. Decision to be made for certain condition when less information is available. The designer uses basic principles of engineering sciences such as physics, mathematics, statics & dynamics.

Front track	1930mm
Rear track	1809mm
Overall length	7660mm
Front Overhang	1130mm

Front track	1930mm
Complete chassis kerb Mass	4195 Kg.
Max. gross vehicle mass	16160 Kg.
mass of the Engine	411 Kg.
Total mass acted on the chassis	16160 Kg.

Capacity of Truck = 16.16ton
=16160 kg
=158529.6 N
Capacity of Truck with 1.25%= 158529.6 x 1.25 N
= 198162 N

All parts of the chassis are made from "C" Channels with 228.60mm x 76.20mm

Each Truck chassis has two beams. So load acting on each beam is half of the Total load acting on the chassis.

Load acting on the

$$\text{Chassis}=\frac{\text{Total load acting on the chassis}}{2}$$

$$=\frac{198162}{2}$$

= 99081 N / Beam

CALCULATIONS OF TERMS

Calculations of mass of chassis

$M = \rho v$ ...(2.1)

$\rho = 0.00785$ kg/mm^2
$v = 4.756*10^7$ mm^3
$M = 373.346*10^3$ kg

Here,
M= Mass in Kg, V= volume in mm^3, I= mass moment of inertia kg-mm^2

Calculations of mass moment of inertia

$I = Mk^2$
...(2.2)

$k = h/2$
...(2.3)

h = 228.6 mm
k = 114.3 mm
$I = 4.8775*10^9$ kg-mm^2

Where,
k= Radius of gyration mm, h= height mm, I= mass moment of inertia kg-mm^2

Calculation for Reaction

Beam is simply clamp with Shock Absorber and Leaf Spring. So Beam is a Simply Supported Beam with uniformly distributed load.

1. Load acting on Entire span of the beam is 99081 N.
2. Length of the Beam is 7660 mm.
3. Uniformly Distributed Load is 99081 / 7660 = 12.93 N / mm

Now, taking the reaction around the support "A".

According to loading condition of the beam, beams has a support of three axle means by three wheel axles but among these three wheels one wheel / axle are working as a supporting only.

Total load reaction generated on the beam is as under:-

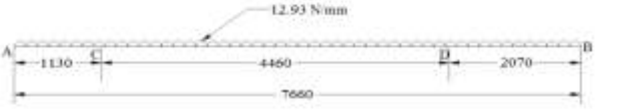

Figure 1 load reaction generated on the beam

For getting the load at reaction C and D,
Taking the moment about C and we get the reaction load generate at the support D.
Calculation of the moment is as under.

Momentum about C

Total load acting on the beam is 99081 N.
So load acting on the reaction is as under

$$\frac{wL_1^2}{2} = \left(\frac{wL_2^2}{2}\right) - [L_2D + wL_3(L_1 + L_2)] \quad \text{... (2.4)}$$

$$\frac{12.93(1130^2)}{2} = \left(\frac{12.93(4460^2)}{2}\right) - [4460 \times D + 12.93 \times 2070(1035 + 4460)]$$

D=59959.25 N
$C+D = 100.3368*10^9$
C = 40377.53 N

Where, w= load acting on side member, D=reaction at D, L_1=length between AC, L_2=length between CD, L_3=length between DB

Calculation for Shear Force Diagram and Bending Moment Diagram

Shear Force Diagram

F_A =0 N

F_{CL} $= wL_1$
$=14.6109*10^3$

F_{CR} $= -(wL_1) + C$
$= -(12.93*1130)+40377.53$ N
$= -25.7663*10^3$N

F_{DL} $= -[w(L_1+L_2)]+C$
$= -[12.93(1130+4460)]+40377.53$
$= -31.54517*10^3$N

F_{DL} $= -[w(L_1+L_2)]+C+D$
$= -[12.93(1130+4460)+40377.53+59959.25$N
$= -28.05803*10^3$N

F_B
=0N

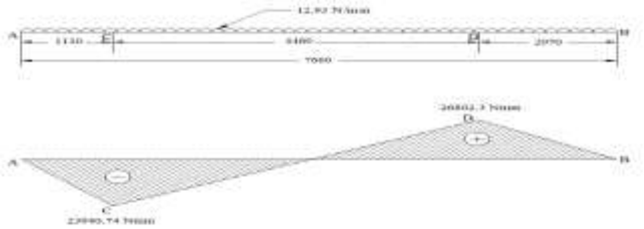

Figure 2 Shear force diagram

Bending Moment of chassis

M_A = 0 N.mm

$M_C = -\left(\frac{wL_1^2}{2}\right)$

$= -\left(\frac{12.93*1130*1130}{2}\right)$

$= -8255158.5$ N.mm

$M_D = -\left[\frac{w(L_1+L_2)^2}{2}\right]+(C*L_2)$

$= -\left[\frac{12.93(1130+4460)^2}{2}\right]+(40733.53*4460)$

$= -21937422.7$ N.mm

$M_B = 0$ N.mm

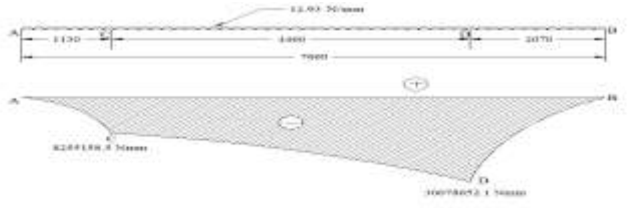

Figure 3. Bending Moment Diagram

Calculation for the deflection

Material of the Chassis is as per IS: - 9345 standard is Structural Steel with St37.

Material Property of St37

E = 2.10 x 105 N / mm2

Poisson Ratio = 0.3

Density=7800 Kg/m^3

Radius of Gyration R = (228.60 / 2) =114.30 mm

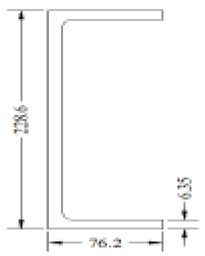

Figure 4 Main "C" Section

Moment Of Inertia around The X – X Axis

$I_{XX} = \frac{bh^3-(b_1h_1^3)}{12}$ ...(2.5)

$= \frac{(76.20*228.60^3)-(69.85*215.90^3)}{12}$

$= 17279023.84$ mm^4

Where, I_{XX}=mass moment of inertia about x-axis, b=width, h=height

Section of Modules Around The X – X Axis

$Z_{XX} = \frac{I}{Y}$ ...(2.6)

$= \frac{17279023.84}{228.6/2}$

$=151172.56$ mm^3

Where, Z_{XX}=section modulus in kg-mm, I=mass moment of inertia in mm^3, Y=distance from C.G. to base in mm

Basic Bending Equations Are As Follows

$\frac{M}{I} = \frac{E}{R} = \frac{б}{Y}$ ...(2.7)

Here, M: maximum bending moment, I: Mass moment of inertia, E: Modulus of elasticity, R: radius of curvature,б: direct stress

Maximum Bending Moment acting on the Beam:

M_{max} = 30078652.1 Nmm

I = 17279023.84 mm4

Y = 114.30 mm

Stress Produced On the Beam Is As Under

$б = \left(\frac{M}{Z}\right)$ ...(2.8)

$= \frac{45.6254*e6}{151172.56}$

= 301.81061 N/mm^2

Where,б=bending stress in N/mm^2, M= bending moment in N-mm, Z= section modulus mm^3

Slope Produced On the Beam

$EI\frac{d^2y}{dx^2} = \frac{W*X^3}{6} - \frac{x^2}{2}(c+d) - x(1130c+5590d)+A$ ...(2.9)

$= \frac{1.007*e12-6.04199e12-2.9549e12+1.811540e12}{2.10*10^5*4.875*e9}$

= 0.060326 rad

=0.356035^0 slope of the deflection

Maximum Deflection Produced On the Beam

$Y = \frac{1}{(4.8775*e9*2.10*10^5)} \times \left[\left[\frac{12.93*7760^4}{24} - \frac{7760^3}{6}(40.3775e3 + 59.959e3) - \frac{7760^2}{2}(1130*40.3775e3 + 5590*59.959e3) + 1.81154e12*7760 - 1.803586e15\right]\right] \frac{1}{E*I}\left[\frac{wL^4}{24} - \frac{L^3}{6}(c+d) - \frac{L^2}{2}(1130c + 5590d) + A*L + B\right]$

...(2.10)

= -4.9520mm(negative means downword)

But allowable deflection for simply supported beam is 25.73 mm according to deflection span ratio.

Calculation For Shear Stress Generated In Chassis

Reaction generated on Beam at the centre of wheel alignment = 12.93 x 4460

= 576678 N

With the consideration of at the rate of angle of twist = 1°

$\Theta = \frac{1*3.1412}{180}$

= 0.017452 radian

By considering the whole system as a one rotational body as per following data when in twist from its support.

Width of the chassis = 900mm

Length of chassis = 7660mm

Distance between two reaction = 4460mm

Modulus of rigidity for structural steel= 80000 N/mm2

Now basic rule for Twisting Moment is:-

$$\frac{T}{J} = \frac{G*\Theta}{L} = \frac{\tau}{R}$$ ....(Torsional equation)...(2.11)

Now equation

$$\frac{T}{J} = \frac{G*\Theta}{L}$$ ...(2.12)

Now, Equation For the rotational shaft J is 2 times higher the Mass Moment of inertia, Mass moment of Inertia for Chassis body = 61142913.44 mm4

So,

J= 2 x I J

J= 2 x 181483130.4 mm^4

J= 3629662660.8 mm^4

$$T = \frac{G*\Theta*J}{L}$$ ...(2.13)

$$T = \frac{80000*0.017452*3629662660.8}{7660}$$

= 114766295.2 Nmm

Shear Stress Generated In Chassis Body

Take width of body as a radius o rotational body R = 900mm

$$\frac{T}{J} = \frac{\tau}{R}$$ ...(2.14)

$$\tau = \frac{T*R}{J}$$

$$= \frac{114766295.2*900}{3629662660.8} = 28.457 \text{ N/mm}^2$$

The Results Of Mathematical Calculations

SR.NO	TERMS	VALUES
1	Displacement	4.952mm
2	Moment Of Inertia	$4.8755*10^9$ Kg-Mm^2
3	Slope	0.017452 Radian
4	Max. Bending Stress	301.81N/Mm^2
5	Shear Stress	28.457 N/Mm^2

II. MODELLING

Finite element analysis is performed on cad model. Though modelling is not part of fea but it is considered as prerequisite for finite element analysis. Accuracy of results produced by fea tool depends on accuracy of cad model created by cad software package cad model should represent the actual geometry of object to be analyzed. Modeling of backhoe attachments of excavator is done in cad software package catia v5.

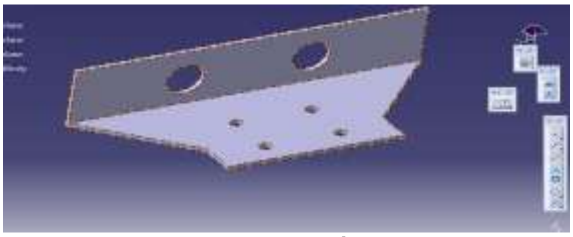

Fig.3.1 Upper Bracket

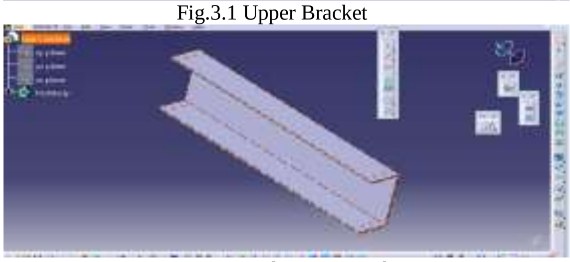

Fig.3.2 Rear end Cross-Member

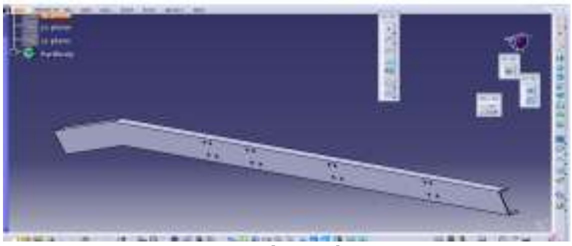

Fig.3.3 Side Member

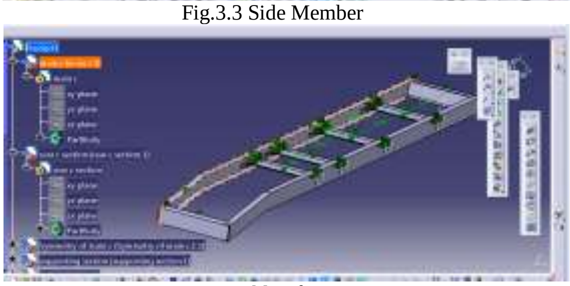

Fig.3.5 Assembly of Components

III. FINITE ELEMENT ANALYSIS

Pre-processing of FEA

1.Geometry cleans up

As we know only a perfect error free geometry can produce correct result. Any of the above mentioned error can take away results from accuracy. So it is important to remove all these errors, to remove these errors we can use quickly edit option and also some surfaces are created according to requirement.

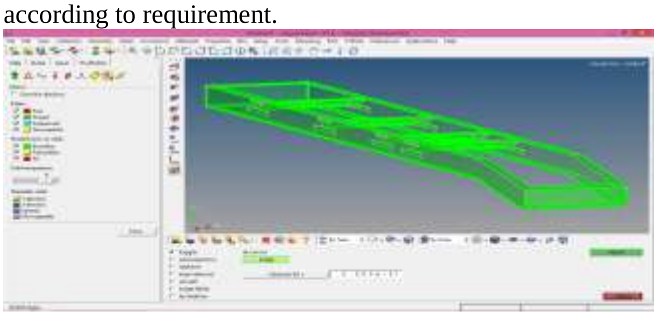

Fig no 3.1 Geometry clean up

2.Meshing

There are three types of elements used in meshing wiz 1d element, 2d elements, and 3d elements. These elements were used for meshing various components according to complexity of component and function of component. It is general practice that parts which are subjected to high stress should meshed using 3d elements.

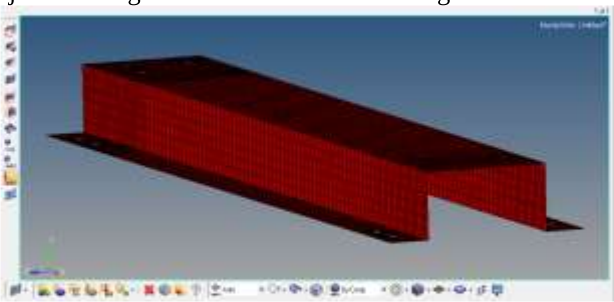

Fig no 3.2 Cross Member

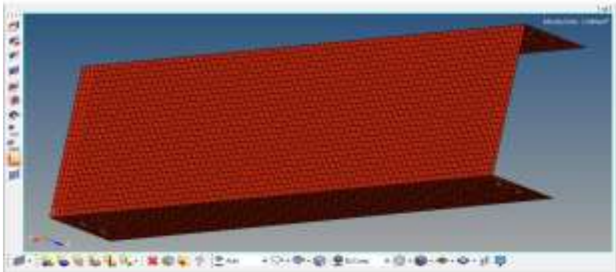

Fig no 3.3 Front & Rear End

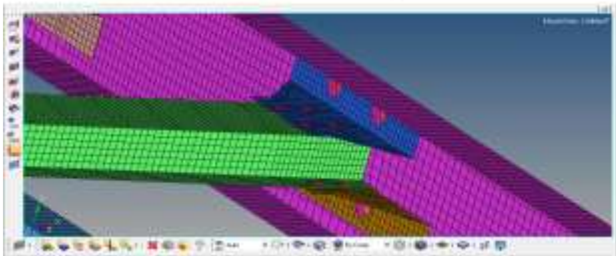

Fig no 3.4 Connectivity of Meshed Component

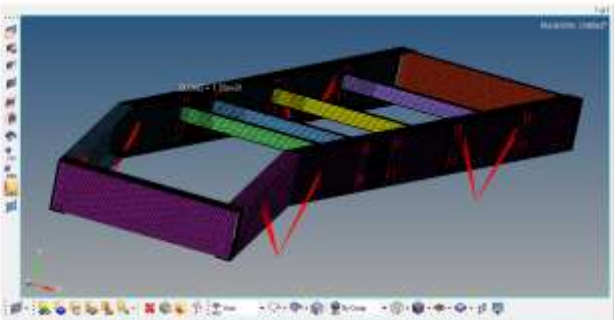

Fig no 3.5 Meshed Assembly of Chassis

3.Quality Check

Element quality is a subject often talked about and never fully under fact that quality is relative and the solution, by definition, is approximate coordinate system is assumed for each element type and how matches the parametric dictates element quality.

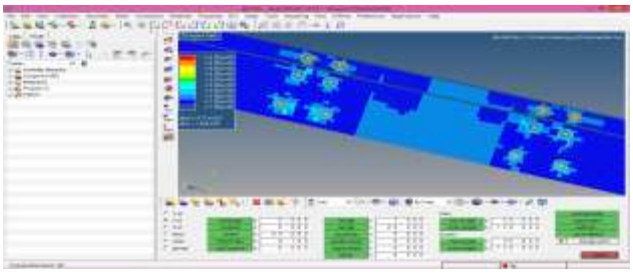

Fig.No 3.6 Refined Mesh

4.Boundary Condition

In static analysis one configuration of the mechanism has to be decided first for which the analysis is to be carried out.

4.1 Constraint

The chassis is fixed at the point of connection with leaf spring so that the degree of freedom for constraint is zero. It is neither allowed to move transverse nor rotated.

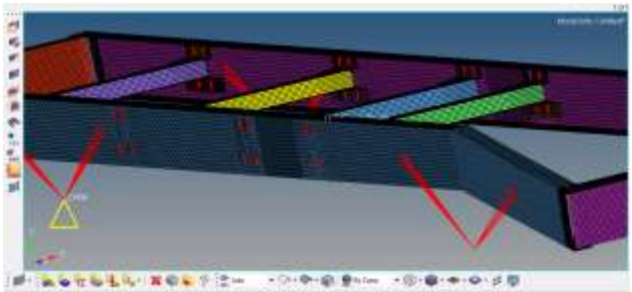

Fig no 4.1.1 Constraint Boundary Condition

4.2 Load

The load applied on chassis is uniformly distributed over a upper span of chassis. The condition of load is of static type.

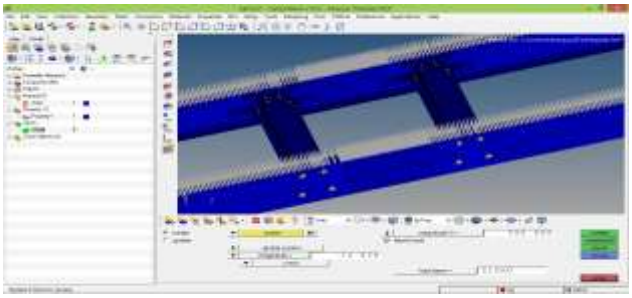

Fig no 4.1.2 Load Boundary Conditions

IV. RESULTS & DISCUSSION

1. Displacement

Deflection is the degree to which a structural element is displaced under a load. The deflection distance of a member under a load is directly related to the slope of the deflected shape of the member under that load and can be calculated by using the slope of the member under that load.

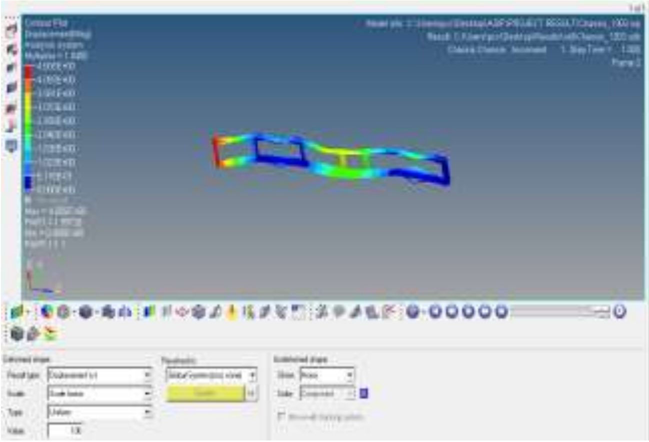

Fig no 4.1 Maximum Displacement

1.The maximum deflection is occurring at rear end of the chassis.
2.The reason of max deflection is overhang at rear end.
3.The value of maximum displacement is 4.600 mm

2. Von mises stresses

The concept of Von mises stress comes from the distortion energy failure theory. Distortion energy failure theory is energy in the actual case & Distortion energy in a simple tension case at the time of failure. According to this theory, The failure occurs when the distortion energy in actual case is more than the distortion energy in a simple tension case at the time of failure.

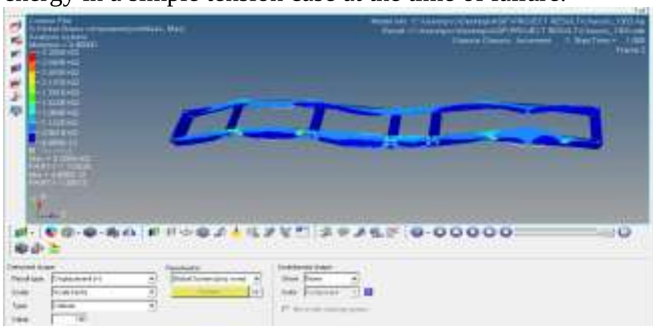

Fig no 4.2 Von Mises Stresses

1. The max von-mises stresses in chassis are occurring in longitudional members at the points of connection with leaf springs.
2. The max value of the von-mises stresses generated in a chassis is 320.5N/mm^2
3. The stresses are minimum in cross members.

The value of the displacement is less than displacement evaluated by span ratio equation, also the stresses generated in cross members are less than allowable stresses. So that there is an opportunity for optimization of chassis for weight reduction.

V. OPTIMIZATION

Structural optimization methods are specific ways of applying traditional optimization algorithms to structural problems solved by means of finite element analysis.

1. **Von Mises Stresses in Optimum Chassis**

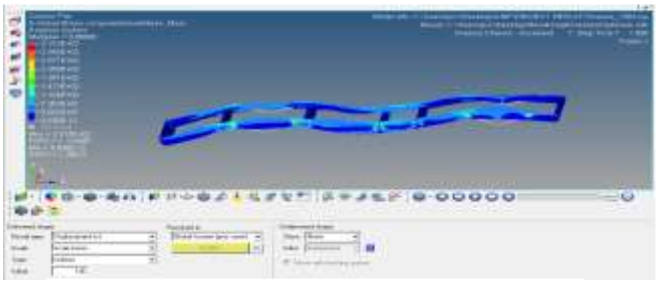

Fig no 5.1 Von Mises Stresses in Optimum Model

Above figure shows von-mises stresses in chassis, we can see that maximum stresses induced in chassis are 331.1N/mm^2. These stresses only seen in longitudinal support. Excepting longitudinal support member stresses induced in remaining boom body are far below yield stresses. This triggers the scope for optimization of chassis.

2. Displacement in Optimum Chassis

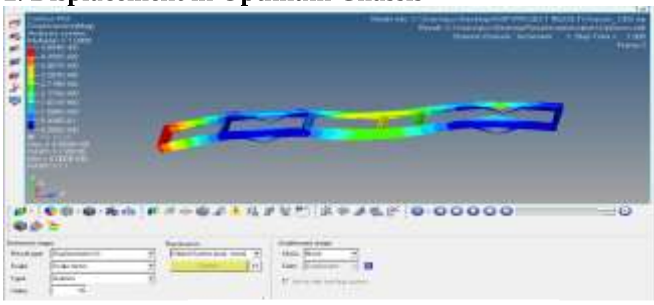

Fig no 5.2 Displacement in Optimum Chassis

The maximum deflection is occurring at rear end of the chassis. The reason of max deflection is overhang at rear end. The value of maximum displacement is 4.890 mm. As we can see maximum von-mises stresses increased from 320.5MPa to 331.2MPa and Maximum displacement increased from 4.06 mm to 4.89.

Discussion on optimization results

The thickness of existing model of main members & cross members is reduced for purpose of optimization. The main aim of reduction thickness is for minimizing weight of material required. The value of thickness of side member is reduced from 6.35mm to 6.00mm & thickness of cross member is reduced from 4.00mm to 3.50mm.The respective change in dimensions resulting to the reduction in a mass of chassis

The change in mass of both models is shown below

Model	Mass
Existing model	377.9 kg
Optimum model	353.3 kg

3. Mass Variation

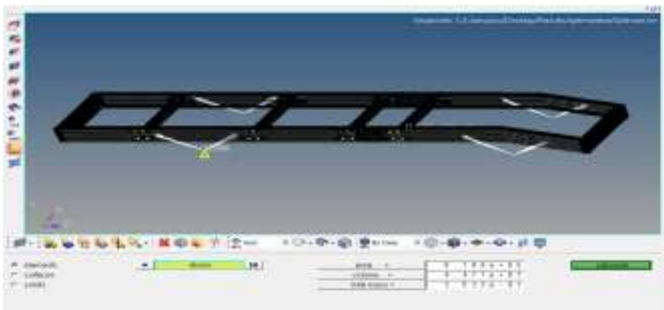

Fig no 5.3Mass Of Optimum Model

The reduction in the mass of chassis after optimization is 24.6 kg. cost of material is Rs 41/kg so that saving of cost per unit is approximately Rs 1000/-

VI. CONCLUSION

Based on the mathematical calculations & finite analysis techniques, we can conclude various aspects related to the displacement, stresses, mass of model etc. These are as follows,

1. The stress and displacement values are calculated by using mathematical formulation. The value for maximum permissible stress is 301.8 N/mm^2. The value for maximum displacement is 4.952mm.
2.The stress and displacement values are calculated by using finite element formulation. The value for maximum permissible stress is 320.5N/mm^2. The value for maximum displacement is 4.60mm.
3. By comparing the values for results from both analyses, we can conclude that the variation in stresses is 5.83% & variation in displacement is 7.1%. The reason for variation in results is approximations used in finite element analysis.
4. By analysing the results from FE analysis, the area of optimization is specified. Depending upon analysis, the thickness of cross member & main members are reduced by 0.5mm & 0.35mm respectively.
5. The corresponding reduction in mass is 4.83%. So that we reduced 24.6 kg of material.
6. The market cost of material is Rs 40/kg. According to this, saving of cost per unit is approximately Rs 1000/-

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