# Weight Reduction and Fatigue Life Assessment of Light Commercial Vehicle Chassis

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## Abstract

It is common practice in the automotive industry to employ extensive experimental measuring technique and durability study to perform structural analysis of chassis in its working environment. In this paper a few simpler methods were used to perform static and dynamic analysis of an existing light commercial vehicle chassis and solved through finite element analysis software Ansys. The study shows that when the existing 'inverted hat section' of the chassis is changed to 'C-channel' there is considerable reduction in weight. Certain local stress concentration areas were added with gusset plates along the rail of the chassis in a measure to increase the endurance strength of the chassis which leads to increased life.
**Keywords: Light Commercial Vehicle, Structural, Weight, Endurance, Life**

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## I. INTRODUCTION

Present automotive developments are much directed and encouraging towards weight reduction of components without much effect towards their life time use. To meet the specific need an engineer works mostly on material characterization, shape modification or minimize the nature of loads acting on the component. The frame in conjunction with the vehicles suspensions, axles, wheels and tires makeup the principal load-carrying components of a vehicle. The chassis must be rigid enough to support or carry all the loads and forces that the vehicle is subjected to in operation. In order to satisfy various customer needs and payload applications, designers have wide variety of chassis design selections, however there are certain standards or regulations that need to be conformed in designing[5].

Johann Wannenburg[2] proposed fatigue equivalent static load (FESL) methodology for the numerical durability assessment of heavy vehicle structures, where fatigue load requirements are derived from experimental measurements, the responses to which are considered as stress ranges applied a number of times during the lifetime of the structure. In order to support these experimental methods certain numerical techniques such as FEM can be employed to verify the theoretical results obtained. In conditions when prototype testing turns to be expensive and time consuming, finite element analysis of several different models lead to a reduction of physical and expensive tests [9], V. Veloso proposed reinforced stringer through numerical analysis which minimized plastic deformation and offered high durability.

Mehdi Mahmoodi-k[7] proposed that in order to improve static and dynamic characteristic of chassis to endure equipment loads, cross section and mass distribution of the chassis were optimized. The simulation results indicate that the chassis weight optimization and cross-section have considerable effect on ride comfort, handling, stability and prevention of vehicle rollover. N. Sefa Kuralay [4] proposed through numerical results that stresses on the side member can be reduced by increasing the side member thickness by adding local plates only in the joint area to increase side member thickness and therefore the excessive weight of the chassis frame is prevented.

Therefore in order to support the extensive experimental measurement exercises to conduct analytical fatigue life assessments, certain numerical procedures and simulation techniques can be used to verify the results and predict the life by enabling few modifications to the existing.

## II. STRESS LIFE APPROACH

The stress life approach is preferred since the number of cycles for this application is more than 1000 cycles where the component undergoes yielding for a few periods before failure. The main objective of this approach is to predict the total number of cycles to failure. The endurance strength is influenced by certain limiting factors as mentioned in [1] such as

$$S_e = k_a\, k_b\, k_c\, k_d\, k_e\, S'_e$$

$k_a$ = Surface factor, $k_b$ = Size factor , $k_c$ = Loading factor, $k_d$ = Temperature factor, $k_e$ = Miscellaneous effects factor $S'_e$ = Endurance strength obtained from material property. The Size factor ($k_b$) is obtained from the sectional dimension of the chassis.

## III. SIZE FACTOR

The LCV chassis taken for the study is TATA ACE (LT) with inverted hat section at the side rails and its dimensions are

Overall Length = 3875mm, Wheel Base = 2005mm

Cross Sectional Dimension =122 x64mm, Overall Width = 1405mm, Length of the front Cabin= 1310mm, Length of Loading Deck = 2140mm, Gross Vehicle weight = 1550 kg

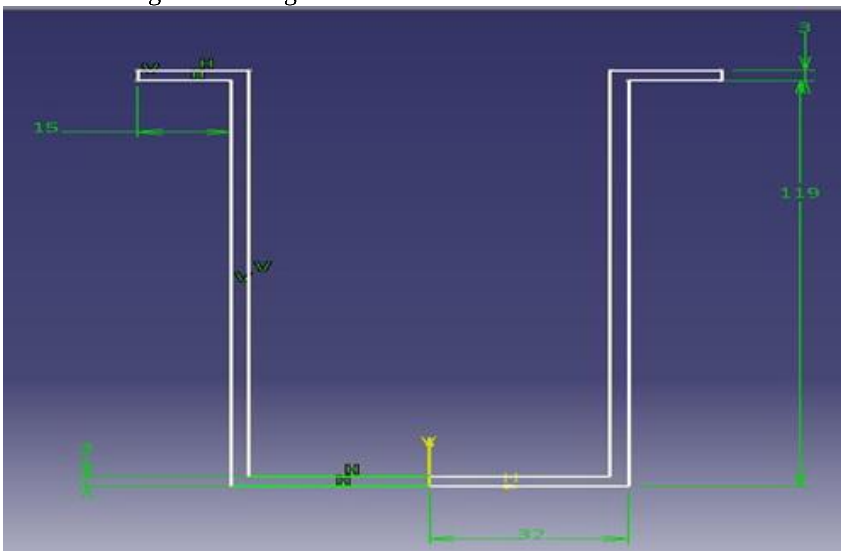


Fig. 1: Inverted Hat Section (122x64)

The size factor for bending & torsion from [1] is given by

$$k_b = 0.859-0.000839\ d_e \quad (2)$$

Calculating for the above section $d_e$ = 71.39mm

Substituting in equation(2) $k_b$ = 0.7992

The endurance strength ($S'_e$) calculation from material property is obtained from the relation[1]

Density = 7.85g/cm$^3$, Yield Stress ($S_y$) = 250MPa, Ultimate Stress ($S_{ut}$) = 460MPa, $S'_e = 0.506\ S_{ut}$ when $S_{ut}$ <=1460MPa $S'_e$ = 232.76MPa

Thus the endurance strength obtained for the inverted hat section from material property is given by

$$S_e = k_b\ S'_e$$

$$S_e = 186.02\text{MPa}$$

$$\sigma'_f = 805\text{MPa}, b = -0.1009,\ f = 0.8127,\ a = 751.42\text{MPa}$$

Where $\sigma'_f$ = fatigue strength coefficient, f = fraction of ultimate strength, $d_e$ = effective diameter, $N_e$ = number of cycles to endurance limit ($10^6$ cycles).The fatigue strength relation thus obtained is

$$S_f = a\ N^b$$

$$S_f = 751.422\ N^{-0.1009} (3)$$

## IV. STATIC ANALYSIS

The shear force & bending moment calculations are made taking the following loads into consideration.

Table – 1
Load Data

| *Loads* | *N* | *On a single rail* | *N/mm* |
|---|---|---|---|
| *Cabin* | *981* | *490.5* | *.33367* |
| *Deck* | *7375.5* | *3678.75* | *1.719* |
| *Engine* | *981* | *490.5* | *-* |

 

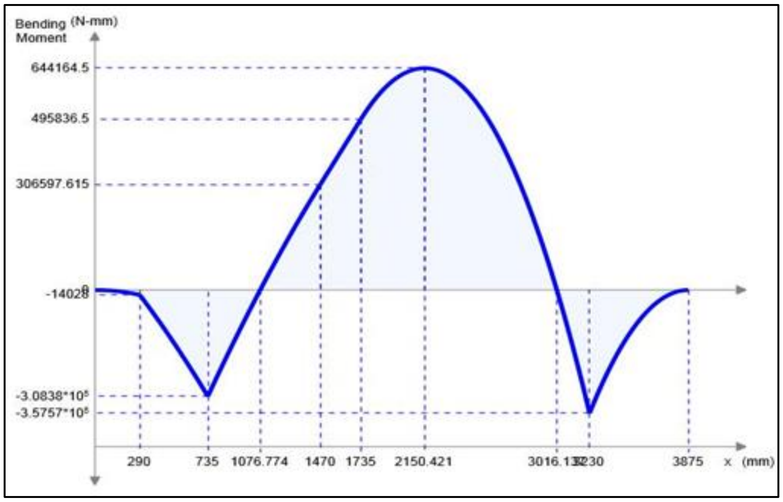


Fig. 2: Bending moment plot

The maximum bending moment occurs at a distance of 2150mm along the length and its magnitude is 644164.5 N-mm.

The moment of inertia for the above section is given by $I=I_{xx}= 2704447mm^4$

y = d/2=61mm

The maximum bending stress thus calculated as σ = (M*d) / (I *2) =14.529 $N/mm^2$

The symmetric model of the beam with same sectional dimensions was modelled using Ansys APDL. Beam 3-D Element is chosen and its 6 degrees of freedom along the translational and rotational axes are arrested at four leaf spring suspension points on the rail. Meshing was done starting from edge length of 10mm with descending order in steps of 0.5 and the variations were minimal from 4 to1mm. Edge length of 1mm is chosen thus making 3875 elements along the beam.

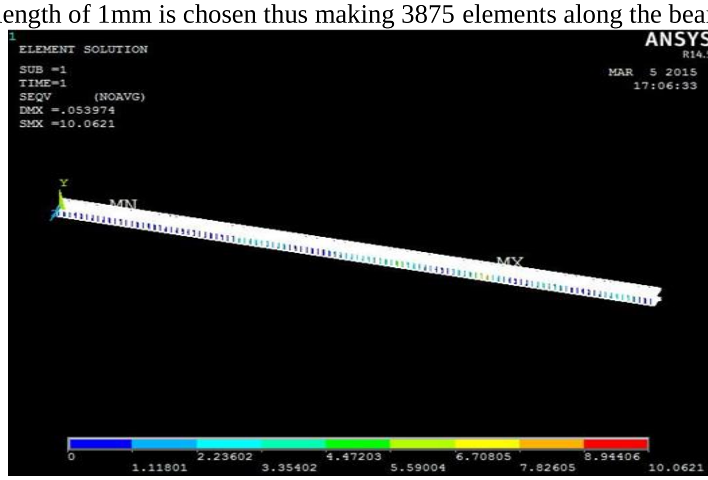


Fig. 3: Stress plot for symmetric section

A maximum stress of 10.06$N/mm^2$ was obtained for the symmetric section with a deformation of 0.053794mm.

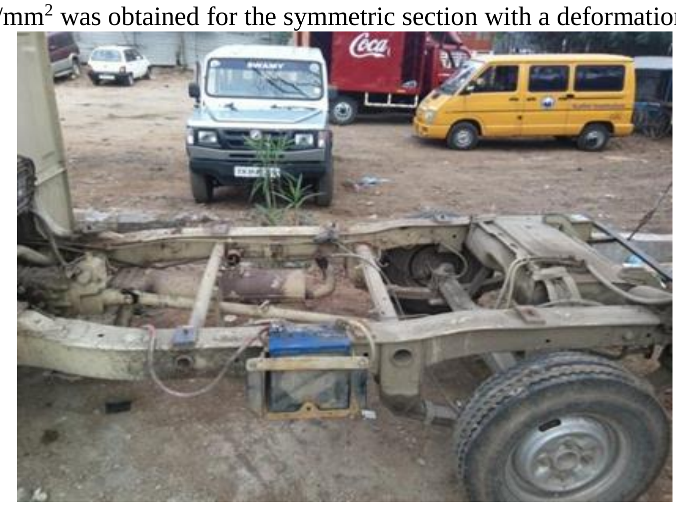

Fig. 4: Tata ACE LT Chassis

 

Symmetric model analysis is done to find the location of stress regions along the length in the form of elements A 3-D model of the entire chassis was made using Catia V5 and it was analyzed using Ansys Workbench 14.5. Eight suspension points were fixed and loads were applied in the form of pressure. Tetrahedral element was chosen for meshing with a transition ratio of 0.272(default).

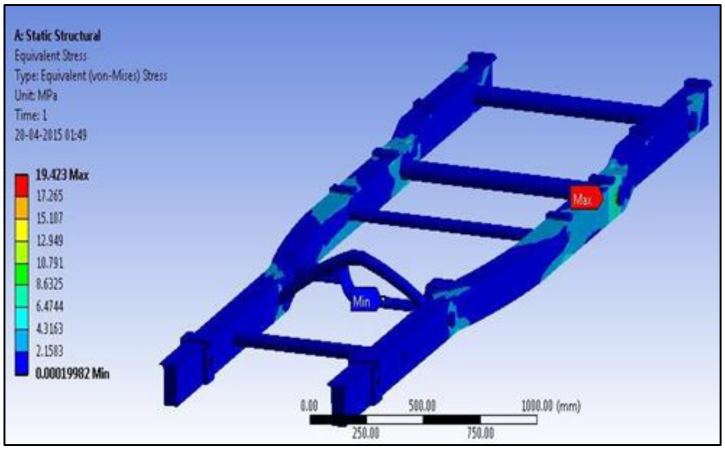

Fig. 5: 3-D Model Stress Plot

A maximum stress of 19.423 $N/mm^2$ with a deformation of 0.068972mm is obtained. The maximum stress occurs in the same region as the symmetric section plot, thus material in the form of gusset plates can be added in these regions to minimize the stress concentrations.

The volume of the entire chassis model is $1.6094x\ 10^7\ mm^3= 0.016094m^3$ and the density of steel is $\rho = 7850\ kg/\ m^3$ . Thus the weight of the chassis is

$$m = \rho \times v = 126.34kg$$

## V. Dynamic Analysis

As a vehicle travels along the road, depending on the road profile, the chassis is excited with dynamic forces induced by road roughness. Under such various dynamic excitations, the chassis vibrates. Several methods such as modal analysis, multi body dynamics are employed to determine the stresses acting on the chassis regions under such condition. However the dynamic forces are mainly experienced during stages of acceleration, braking & road induced vibrations.

In this paper two steps of dynamic loads are taken in consideration

- Inertial load transfer during acceleration & deceleration from front and rear axle loads while moving on a plane road surface.
- Road induced vibration due to road profile.

### *A. Inertial Loads*

Modified Indian driving cycle is used for emission testing of LCV vehicles with a help of a chassis dynamometer. The vehicle velocity data's at individual time instant are obtained from [8], through which the inertial loads during acceleration and deceleration values are calculated. This test is carried for a period of 1180sec covering 10.467 km with the maximum speed of 90km/hr.

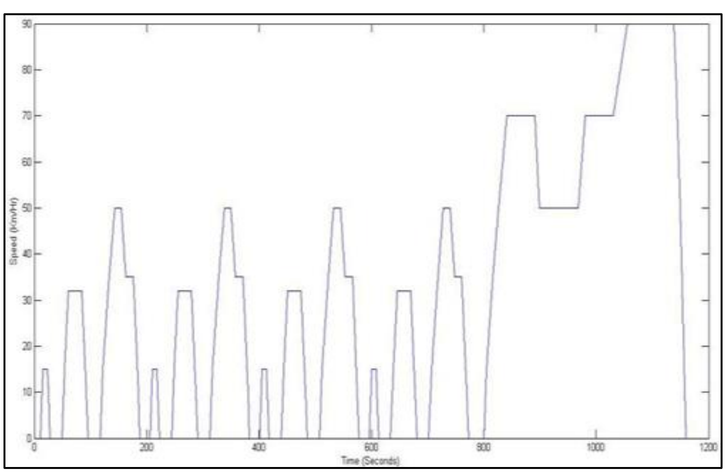

Fig. 6: Modified Indian Driving Cycle Matlab Plot

The front axle and rear axle loads are 750kg & 880 kg respectively in loaded condition. When travelling from 0 - 15km/hr for a period of four seconds

$$\text{Acceleration} = 1.04\ m/s^2$$

Load transfer $W_L$ = (W x a x h)/ (g x L) (4) where W = axle loads h=height of center of gravity (assumed as L/4) L= wheel base.

$$W_L = 41.081\ N$$

The load transfer at front & rear axles during acceleration

$$W_F = 7316.42\ N \quad W_R = 8673.8\ N$$

In this manner similarly the inertial loads are calculated at each time interval and the following graph is plotted.It can be seen from the (fig 8 & 9) that during deceleration from 90 to 0 km/hr there is a sharp peak rise in the front axle plot and a fall in the rear axle plot due to inertial loading. Transient structural analysis is carried out from the above load data at individual time steps. Boundary conditions are given in such a way that the cabin and deck positions are fixed and load from the axle at different time instances are applied at the suspension points.

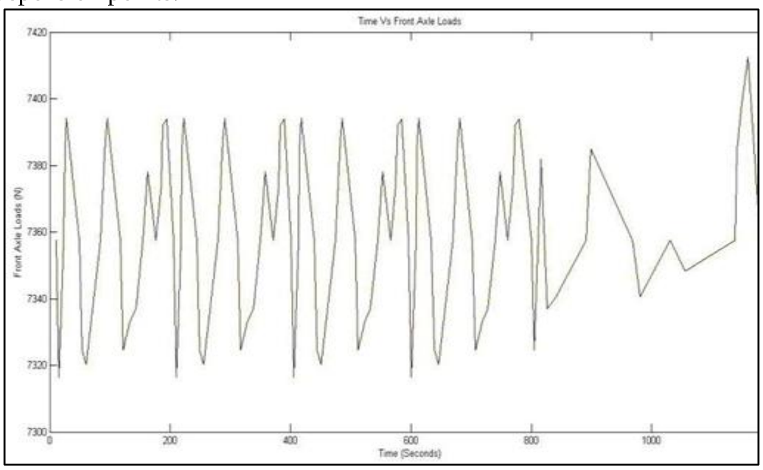


Fig. 7: Time Vs Front axle loads

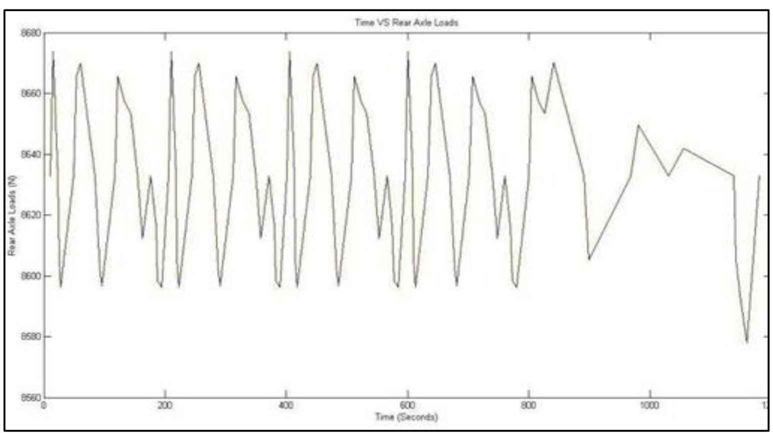


Fig. 8: Time Vs Rear axle loads

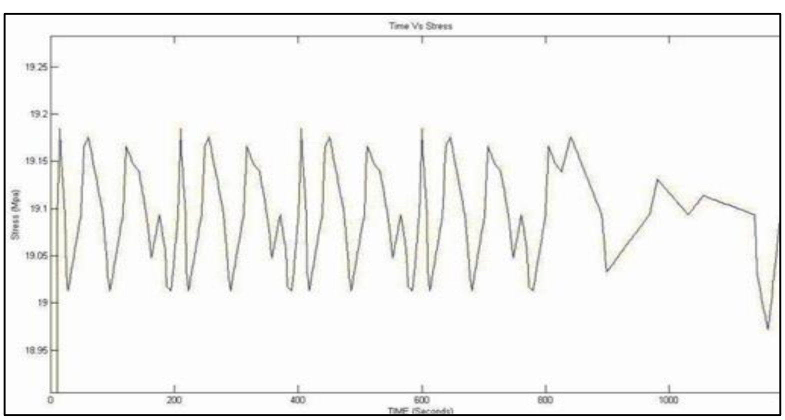


Fig. 9: Time Vs Stress

Maximum stress of 19.18 N/mm$^2$ occurs near the end of the chassis with the maximum deformation of 0.03521mm. Stress due to inertial loading does not vary much from static loading.

### *B. Loads Due To Road Profile*

According to IRC 99-1988 for a highway road condition, the vehicle is tested with road bumps of 10cms height at a wavelength of 3.7m. From the modified Indian driving cycle the vehicle is tested with a standard speed of 40km/hr. A quarter car model is developed from equation (5)&(6) to determine the sprung mass acceleration from which the force transmitted to the chassis can be calculated. A sine wave input with an amplitude of 0.1m and frequency of 18.868rad/sec as calculated from above standards is given. The equations are given by

$$m_s x''_s + k_s x_s - k_s x_{us} + c_s x_s' - c_s x_{us}' = 0 \quad (5)$$

$$m_{us} x_{us}'' + k_s x_{us} - k_s x_s + k_t x_{us} - k_t x + c_s x_{us}' - c_s x_s' + c_t x_{us}' - c_t x' = 0 \quad (6)$$

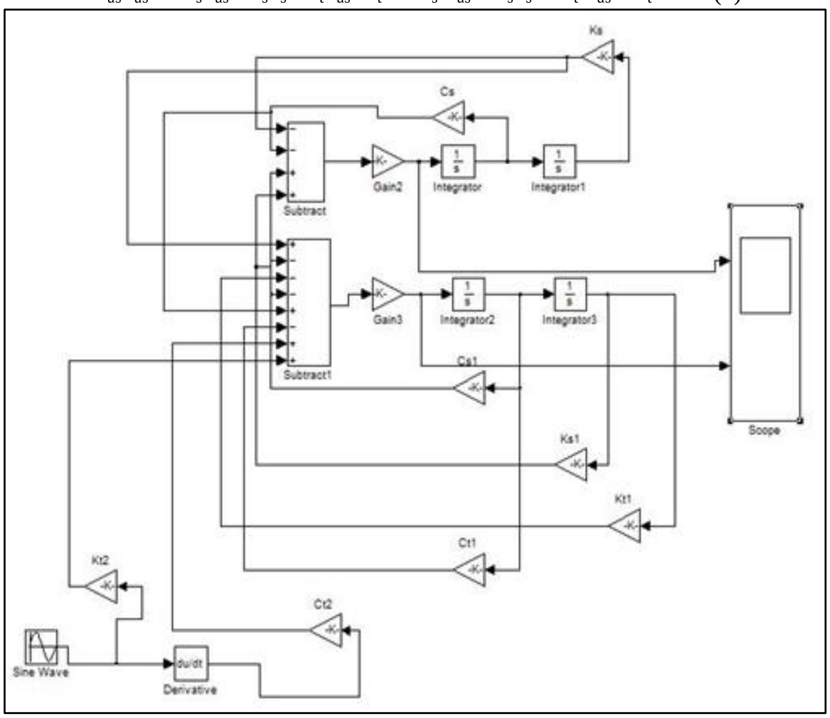


Fig. 10: Quarter-Car Simulink model

Where $m_s$ = sprung mass = 1170kg, $m_{us}$ = unsprung mass = 380kg, damping ratio = $\zeta$ = 0.4(assumed) $x_s$=sprung mass displacement, $x_{us}$ = unsprung mass displacement x= road displacement. The suspension stiffness is calculated by

$$F/\delta = (E*n*b*t^3)/6*l^3$$

Effective stiffness $k_s$= 87732.85 N/m

Tire Stiffness is assumed to be ten times of spring stiffness $k_t$ = 877328.5N/m

Where E= young's modulus, n= no.of leaf, b= width of leaves t= thickness of a leaf, l = length from eye to eye.

$$\zeta = c/2\sqrt{(mk)}$$

Calculating, damping coefficient of spring $c_s$ =8105.20Ns/m damping coefficient tyre $c_t$ =14607.06 Ns/m.

From fig 11 & 12, the unsprung mass acceleration reaches a peak value of 70.5m/s$^2$ and sprung mass acceleration reaches 19.5m/s$^2$. The unsprung mass acceleration is transmitted to the chassis, through the suspension. The force transmitted to the chassis is given by

Force transmitted from unsprung mass to individual front suspension points = $W_f$ x70.5=750x70.5=52875/4 = 13218.75N Force transmitted from unsprung mass to individual rear suspension points = $W_r$x70.5 = 800x70.5=56400/4=14100N

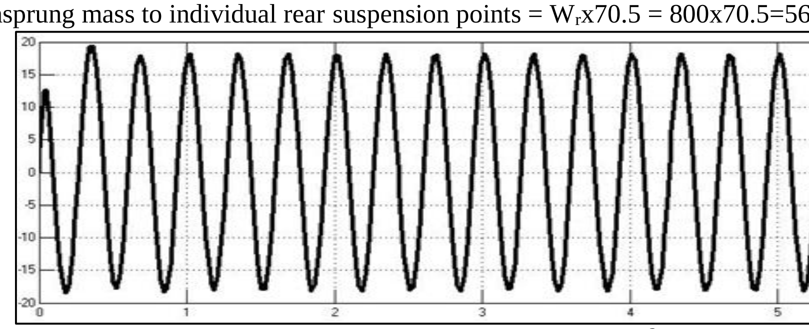


Fig. 11: Time(s) vs Sprung mass acceleration (m/s$^2$)

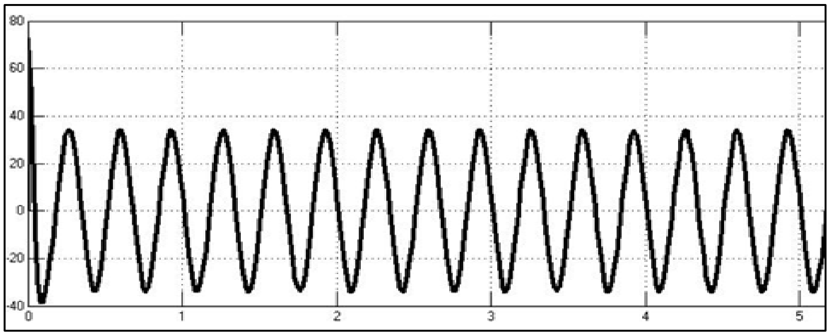


Fig. 12: Time(s) vs Unsprung mass acceleration (m/s$^2$)

The calculated forces are applied in upward direction at suspension points by fixing the cabin and deck positions.The maximum stress due to excitation forces from the road profile is 115.87N/mm$^2$ which is beyond the yield & ultimate stress levels, with the maximum deformation of 0.21075mm.

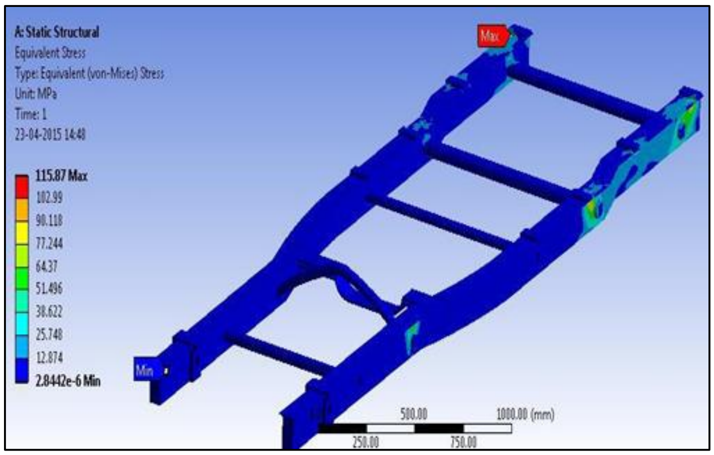


Fig. 13: Stress due to road profile

## VI. Modified C Section

The Stresses in inverted hat section are well below the yield and ultimate stress levels. Other section such as box channel was attempted, but did not produce efficient size factor & weight reduction percentage as that of the C-section. In order enhance weight reduction and to increase the size factor a modified C section of cross-section 125x65 is chosen. Calculating for the C-section effective diameter $d_e$ = 53.71,therefore from (2)

$$k_b = 0.814$$

The endurance strength is increased by $S_e$ = 189.47 N/mm$^2$ Thus the fatigue strength equation is given by

$$S_f = 751.322\ N^{-0.0997} \qquad (7)$$

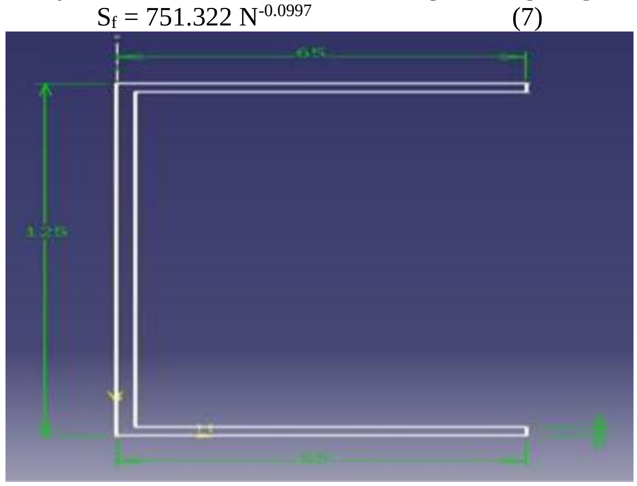


Fig. 14: C section (125 x 65)

The chassis was replaced with C section at the side members and modeled again. This modified chassis has a volume of 0.012809 mm$^3$. Thus the weight is given by

$$m = \rho \times v = 100.55\text{kg}$$

However certain gusset plates needed to be added in areas of stress concentration and its location can be found using the stress result data's from the symmetric beam plot fig(3) which gives the location along the rail. These gusset plates may increase certain amount of weight.

Fig. 15: Modified C section chassis

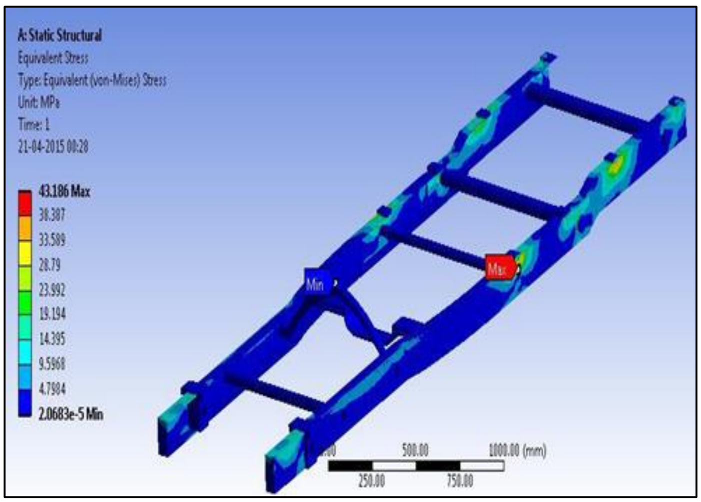


Fig. 16: Static Load on C-Section

The maximum stress value is 43.106 N/mm$^2$ which is well below the yield and ultimate stress levels with the deformation of 0.51485mm.

Fatigue strength of the chassis can be approximated from Soderberg and Goodman relation through which the lifecycle for individual chassis can be approximated for the loading conditions.

## VII. CONCLUSION

Thus C- section offers an increase in the endurance strength from 186.02 to 189.476Mpa and a percentage reduction in weight is given below % reduction in weight = ((126.34 -100.55)/126.34 )*100 = 20.41%

However certain gusset plates needed to be added on areas of stress concentration obtained from FEA analysis , which may increase the weight by three or four kilograms. Further the life cycle of both the chassis are to be approximated which are considered as future works.

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