Stress Analysis and Design Optimization of an Automotive Frame Structure

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ABSTRACT

The cost and quality of the products are very important in the present competitive market to sustain the business. The engineering team is continuously working towards improving their products in terms of durability life, lower cost and better performance. The cost reduction can be achieved by changing material like from alloy to composites, changing design parameters like gauge thickness, cross section changes etc. Automotive frame structures (chassis) are subjected to high stresses under 3 loading scenarios viz. 3g vertical ditch, under turning and 1g braking load cases. A finite element analysis is performed for all these load cases. The loads are applied at the tire-ground contact location. The welding between the different components in the chassis structure is captured accurately. The thickness of the chassis plates can be optimized to achieve the required stress levels. The stresses developed in the chassis for the 3 load cases are utilized to run a fatigue analysis. The fatigue envelope thus developed shows the presence of any "hot spots" in chassis that may still have to be improved.

Keywords *- Automobile chassis, frequency, Stress, Modal analysis, Structural analysis.*

1. INTRODUCTION

Automotive chassis is a skeletal frame on which various mechanical parts like engine, tires, axle Assemblies, brakes, steering etc. are bolted. The chassis is considered to be the most significant Component of an automobile. It is the most crucial element that gives strength and stability to the vehicle under different conditions. Automobile frames provide strength and flexibility to the automobile. The backbone of any automobile, it is the supporting frame to which the body of an engine, axle assemblies are affixed. Tie bars, that are essential parts of automotive frames, are fasteners that bind different auto parts together. [1]

Chassis should be rigid enough to withstand the shock, twist, vibration and other stresses. Along the strength, an important consideration is chassis design is to have adequate bending and torsional stiffness for better handling characteristics. So, strength and stiffness are two important criteria for the design of chassis. [2]

2. THEORETICAL CALCULATION

2.1 Calculating Position of the Car CG

When making an analysis of the forces applied on the car, the CG is the point to place the car weight, and the centrifugal forces when the car is turning or when accelerating or decelerating. Any force that acts through the CG has no tendency to make the car rotate.

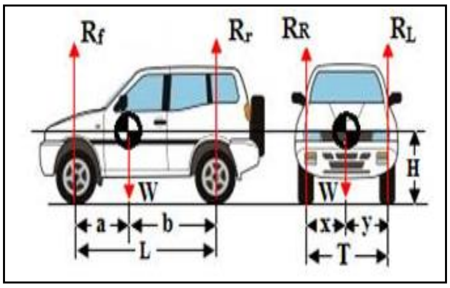


Fig 1: The CG location in the Side and Front View.

The center of mass height, relative to the track, determines load transfer, (related to, but not exactly weight transfer), from side to side and causes body lean. When tires of a vehicle provide a centripetal force to pull it around a turn, the momentum of the vehicle actuates load transfer in a direction going from the vehicle's current position to a point on a path tangent to the vehicle's path. This load transfer presents itself in the form of body lean. Body lean can be controlled by lowering the center of weight or the widening the car track, it can also be controlled by the springs, anti-roll bars or the roll center heights.

A vehicle is not symmetrical in shape or mass from front to rear. Most vehicles are symmetrical left to right in shape but not in mass, especially front wheel drive vehicles.

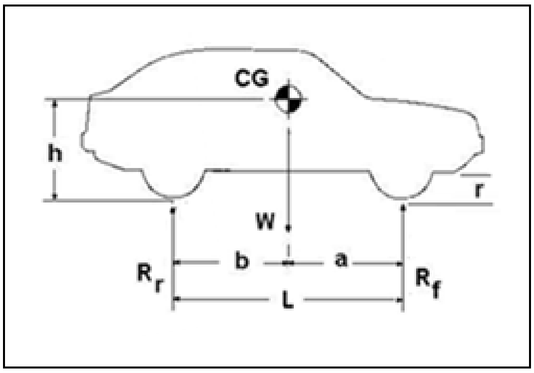
From the above Figure, W is the weight of the tire (always vertical and downward), while R is the reaction from the ground (always perpendicular to the ground surface and away from it). The tire is balanced in the vertical direction under its weight (W) and the ground reaction (R), that means the summation of forces in that direction is zero (W – R = 0), which gives W = R.
In our analysis we measure the weights, but when studding the car balance we use the reactions supporting the car from the ground (where: $W_f = R_f$, $W_r = R_r$, $W_w = R_w$ and $W_L = R_L$).

W is the car weight.
R_f is the ground reaction of the frontal axle weight.
R_r is ground reaction of the rear axle weight.
R_R is ground reaction of the car right wheels weight.
R_L is ground reaction of the car left wheels weight.
L is the car wheel base (distance between the front and rear car wheels/axles).
T is the car track (distance between the centers of the wheels on the same axle).
a is the location of the CG behind the front axle.
b is the location of the CG in front of the rear axle.
x is the location of the CG away from the right wheels.
y is the location of the CG away from the left wheels.

Weights	Kgs	Units
Tare Weight (Curb Weight)	1460	Kg
Payload (85*5+ 50Kg)	475	Kg
Gross Weight	1935	Kg
Axle rate	967.5	Kg/axle
Wheel Base	2.85	m
Tire size	0.381	m
Gravity	9.81	m/s
Correction Load factor	1.2-1.4	

Table 1: Input data.

2.1.1 The Longitudinal Location of CG



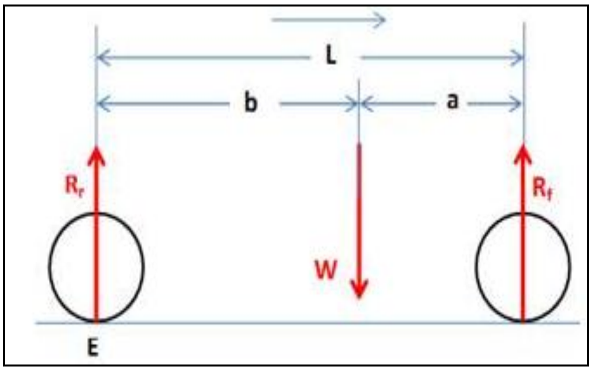


Fig 2: The CG Location in the Side View.

*Measure the wheel base, **L** (the distance between the center of the front and rear wheels), from Figure from the table, we have,

$L = a + b$ ………………………………………………. (1)

$a = L – b$ ………………………………………………. (2)

$\Sigma F_y = 0$

$W – (R_f + R_r) = 0$ ………………………………………………. (3)

Then $W = (R_f + R_r)$ ……………………………………………… (4)
We know that, Gross weight =18982.35 N

$$= \frac{18982.35}{4} = 4745.6 \text{ N (per wheel)}$$

Hence, $W = (4745.6+4745.6) = 9491.2\text{N}$
* Measure the car front axle weight W_f = **R_f**, and the car rear axle weight W_r = **Rr**, from Figure.
Use equation (4) to find the car weight **W**.
Take moment around point E.
$\Sigma M_E = 0$
$R_f L - W b = 0$
$R_f L = W b$, then
$b = L (R_f/W)$ …………………………………………… (5)
$b = 2.85*(4745.6/9491.2)$
$= 1.425\text{m}$
Substitute the value of b in eq. (2) to get the distance **a**.
$a = L - b$
$= 2.85-1.425 = 1.425\text{m}$

2.1.2The Side Location of CG

The right and left wheels weights (**W_R, W_L**) will be measured and the car track **T**.

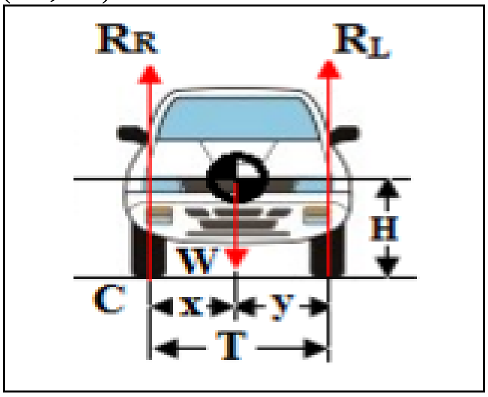


Fig 3: The CG position in the frontal view.

The car load balance in vertical direction shows:
$W = W_R + W_L$
Taking moment about point C will get the distance x
$\Sigma M_C = 0$,
$W (x) - R_L (T) = 0$
$x = (R_L/W) T$,
$= 1.55* (4745.6/9491.2) = 0.775\text{m}$
$y = T - x$
$= 1.55-0.775 = 0.775\text{m}$
Most cars except the front wheel drive are symmetric in weight distribution in the front view ($R_R = R_L$). So that the CG position will be in the middle ($x = y = T/2$). In most cars it is a fair approximation to assume that.

2.1.3 The Height above Ground of CG (H)

The center of gravity height is found using the rules of trigonometry and right triangles. Specifically, we are using the Law of Tangents, and the Pythagorean Theorem. The following diagrams are greatly exaggerated for illustration purposes.

Tan q = opposite / adjacent

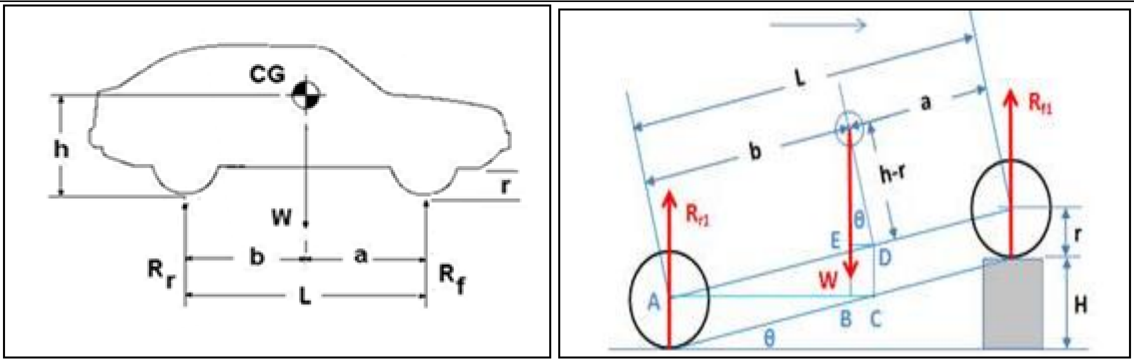


Fig 4: h is the Unknown Distance of the CG Height.

The summation of vertical forces in the y direction is equal 0.

$\Sigma F_y = 0$

$R_{f1} + R_{r1} – W = 0$

Then: $R_{f1} = W – R_{r1}$

The summation of moments about any point is equal 0, then:

$\Sigma M_A = 0$

$R_{f1} (L \cos \theta) - W (AB) = 0$

$R_{f1} (L \cos \theta) = W (AB)$ ………………………………. (6)

From the figure:

AB = AC – BC,

Where:

$AC = b \cos \theta$, and

$BC = ED = (h-r) \sin \theta$,

Then:

$AB = AC – BC = b \cos \theta – (h-r) \sin \theta$

Substitute the value of AB form the above equation in Eq. (6), then

$R_{f1} (L \cos \theta) = W (b \cos \theta – (h-r) \sin \theta)$

$R_{f1} (L \cos \theta) = W b \cos \theta – W (h-r) \sin \theta$

$W (h-r) \sin \theta = W b \cos \theta – R_{f1} (L \cos \theta)$

$h– r = [b – L (R_{f1}/W)] \cot \theta$ …...………………………….. (7)

$h = [b – L (R_{f1}/W)] \cot \theta + r$ ………………………………. (8)

Where:

$\boldsymbol{\theta} = \sin^{-1} (H/L)$

In this case, additional height H is 0.250m and Wheel base L = 2.85m

r =0.1905m

b = 1.425m

R_{f1} = 9580N

W= 18982.35N

$\boldsymbol{\theta} = 0.087^0$

h = [1.425 -2.85 (10741.95/18982.35) * 11.35+0.1905

= 0.9945+.1905 = 1.1847m.

3. MODELLING AND BOUNDARY CONDITIONS

3.1 Geometry Modelling

Initially we were provided the conventional chassis geometry and the criteria is to create a new design by maintaining the same length and wheel base as it is. As we have the reference model, it was way easy task for us to create the new geometry, as half the critical work (choosing the right length and width of the chassis) of the model is carried out using the initial geometry. The geometry of the model is carried out in modelling software called Solidworks. Initially frames are modelled and then cross bar are assembled to the main chassis frame along the spring seat and load carrier points. Geometry modelling plots of conventional and new design are shown in the below fig 5.

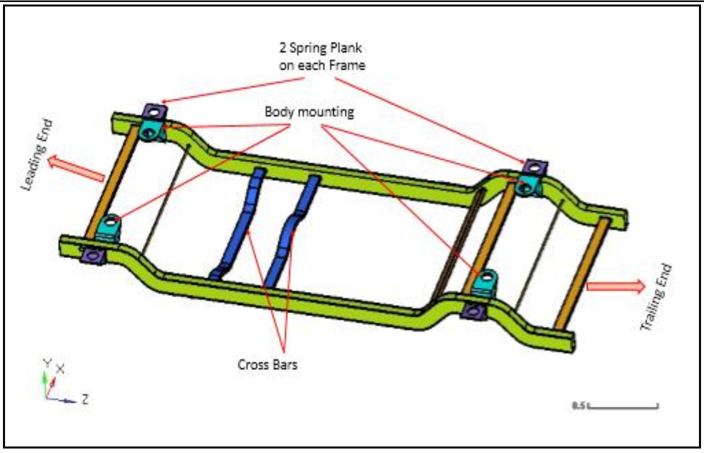


Fig 5: Geometry of the Wave Type Chassis.

3.2 Finite Element Modelling

FE Model is nothing but the discretization of the geometry into smaller section called elements and nodes. Hence elements and nodes play an important role in FE analysis, accuracy of the results is directly depended on the shape, size and quality of the mesh. Hence, suitable mesh size is chosen to mesh the chassis frame and the complete model is meshed with quadrilateral elements. The FE model is comprised of 12006 no of nodes and 11545 no of elements. FE model of the chassis is shown in the below fig 6.

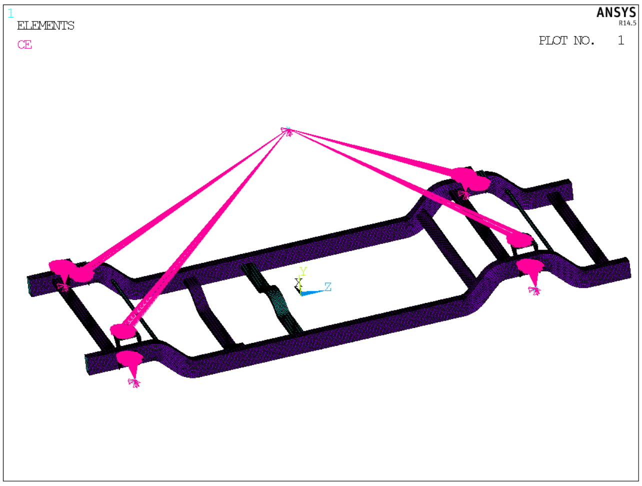


Fig 6: FE model of Car Frame.

3.2 Boundary Conditions

The car chassis is validated for modal and structural load cases and the no of load cases is listed below.

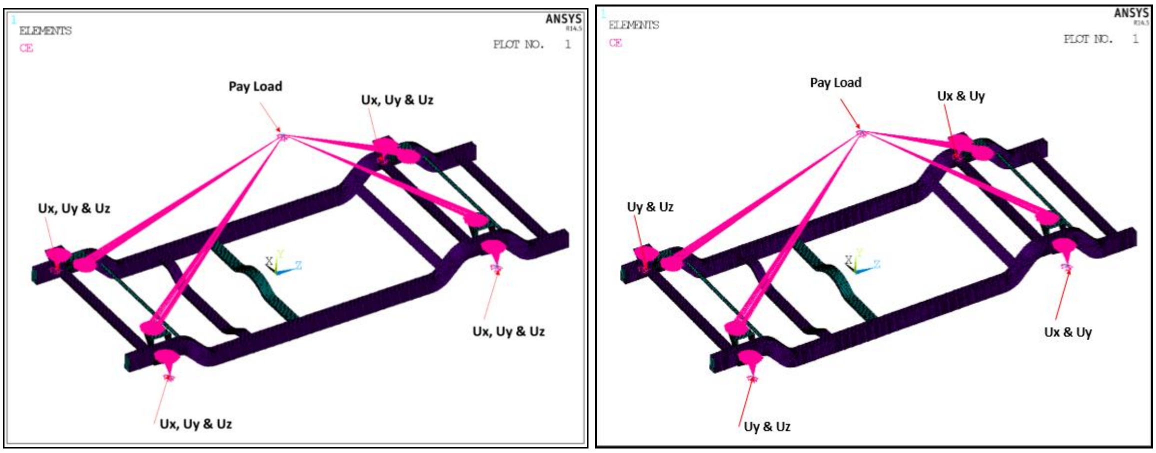


Fig 7: Boundary Conditions used in Modal and Structural Analysis.

1. Modal Analysis
2. Structural Analysis
    a. Vertical load: Empty Car
    b. Vertical load: Loaded Car
    c. Cornering: Loaded Car
    d. Braking: Loaded Car
    e. Front wheel in ditch: Both Wheels
    f. Rear wheel in ditch: Both Wheel
    g. Front LH and Rear LH in ditch
    h. Front RH and Rear RH in ditch

Out of this many load cases only the first four load cases are considered in this project, as this are the first stage critical loads and two type of boundary conditions are used for modal and structural analysis which is shown in the above fig 7.

4. RESULTS AND DISCUSSIONS

4.1 Modal Analysis

This load case is comprised of dynamic analysis (modal analysis), form this load case we will check the minimum frequency of the model. In order to check the electronic components attached to in the car to be safe the car body frequency is kept higher than the electronic component's frequency which is usually maintained at less than or equal to 20Hz. Hence, our first task is to maintain the frequency of the car chassis above 20Hz. The boundary condition used for the modal analysis is as shown in the below fig 7. For our understanding, we will extract 10 frequency and the respective mode shape is shown in the below fig 8.

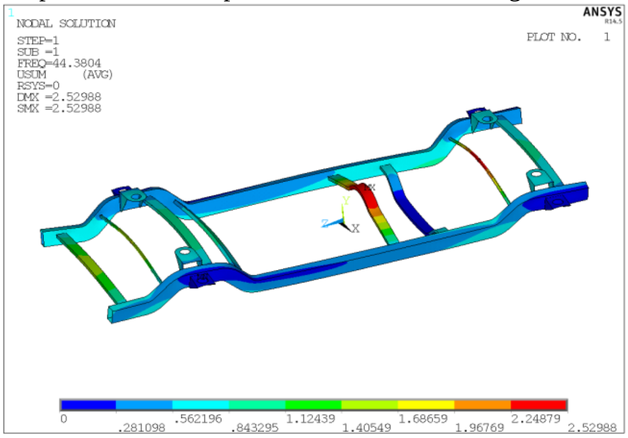


Fig 8: 1st Mode Shape with Frequency 44.38 Hz.

From the above pic it is clear that the first frequency obtained for the chassis is 44.38Hz, which is way higher than the minimum required frequency of 20Hz. Hence this design can be used further.

4.2 Structural Analysis

4.2.1 Vertical Loads (Empty Car)

In this load case the chassis model is validated for 1.4g vertical with empty car i.e. the payload are not considered in this iteration and the detailed boundary condition is shown in the below fig 7.

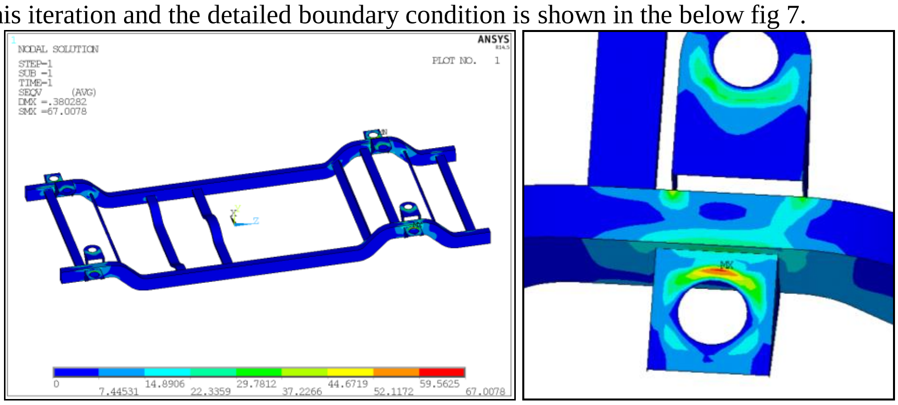


Fig 9: Maximum von-Mises Stress with Zoom in View Plots.

4.2.2 Vertical Loads (Loaded Car)

In this load case the chassis model is validated for 1.4g vertical acceleration with full loaded car i.e. the payload is considered in this iteration. Hence the total loads acting on the car chassis is equal to tare load plus pay load and the detailed boundary condition is shown in the fig 7 in the above section 3.2.

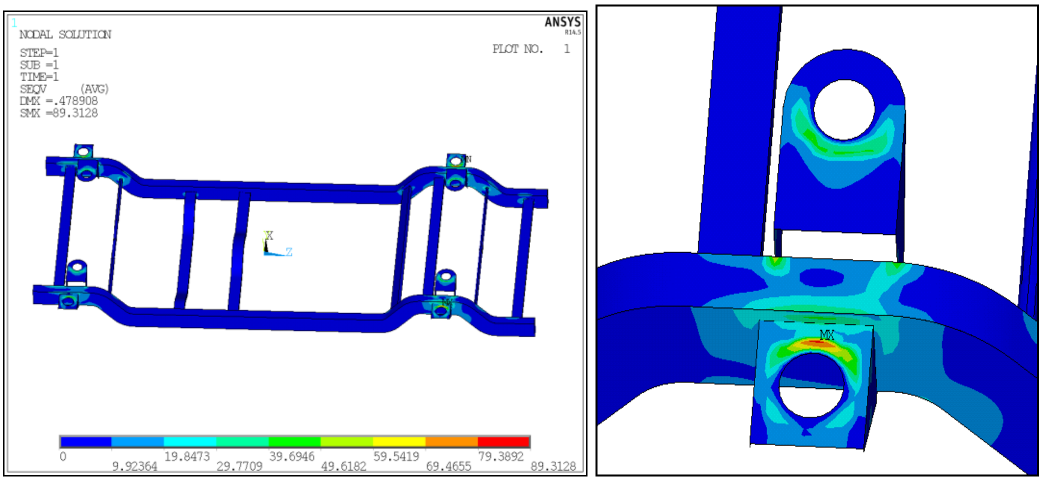


Fig 10: Maximum von-Mises Stress with Zoom in View Plots.

4.2.3 Cornering (Loaded Car)

This load case is comprised of vertical gravitational load of 1.4g which will be acting vertically downwards to the chassis frame and cornering (Lateral) load of 1g i.e. along x axis. Hence the total loads acting on the car chassis is equal to tare load plus pay load along with the lateral load of 1g and the detailed boundary condition is shown in the fig 7 in the above section 3.2.

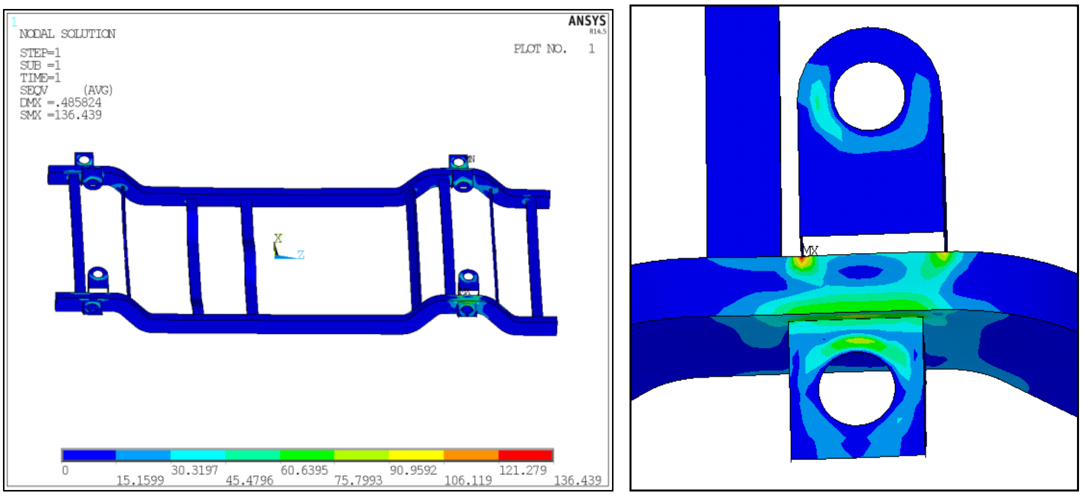


Fig 11: Maximum von-Mises Stress with Zoom in View Plots.

4.2.4 Braking

This load case is comprised of vertical gravitational load of 1.4g which will be acting vertically downwards to the chassis frame and loads induced in the model due to braking (longitudinal) load of 1g i.e. along z axis. Hence the total loads acting on the car chassis is equal to tare load plus pay load along with the longitudinal load of 1g and the detailed boundary condition is shown in the fig 7 in the above section 3.2.

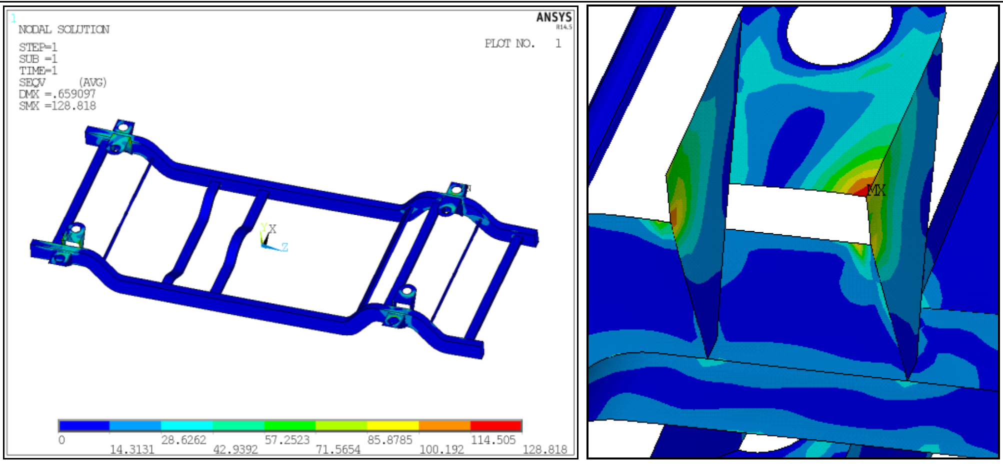


Fig 12: Maximum von-Mises Stress with Zoom in View Plots.

5. CONCLUSION

From the detailed Finite Element analysis we can draw the fallowing conclusions.

It is clear that the induced stresses are well below the yield stress of the material and the minimum factor of safety considering all the load is about 2.3, this shows that the new design is safe the given load condition and can take much higher loads.

Since the FOS is more than 1.25 the much cheaper material can be used, which further save the cost of manufacturing and production.

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