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# STRUCTURAL PERFORMANCE ANALYSIS OF FORMULA SAE CAR

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## ABSTRACT

*Formula SAE competitions take place every year and challenge teams of engineering students to design and build a small single-seater racing car. Among many other key components, chassis is an indispensable structural backbone of an automobile especially in a racing car. Good designs allow a light, stiff and extremely safe chassis to be produced at a reasonable manufacturing cost. The work shown in this research paper was taken from second international participation by Chitkara FSAE team. This paper introduces several concepts of frame's load distributions and consequent deformation modes. Design model was prepared using anthropometric parameters of tallest driver (95$^{th}$ percentile male), SAE rules book and previous design knowledge. Static and dynamic load distributions were calculated analytically followed by extensive study of various boundary conditions to be applied during diverse FEA tests. Stress distributions, lateral displacements during static, dynamic and frequency modes were analyzed and found considerable factor of safety as required. Torsional rigidity was calculated to be 615.98 Nm/deg which was 2.46 times the torsional rigidity of older design (250 Nm/deg). Weight of the chassis was measured to be approximately 32 kg which was 1.125 times less than the previous chassis (36 kg). In nutshell, ratio of percentage increase in torsional rigidity to percentage decrease in weight was calculated to be 13.15:1.*

**Keyword:** Chassis, FEA, Stress, Displacement, Torsional rigidity.

## 1.0 INTRODUCTION

Formula SAE® Series competitions challenge teams of university undergraduate and graduate students to conceive, design, fabricate and compete with small, formula style, autocross vehicles [1].The basis of the competition is that a fictitious company has contracted a group of engineers to build a small formula car which can sell in the market. Cars are expected to perform very high in acceleration, braking, handling, aesthetics, ergonomics, manufacturing and maintenance etc. within minimum manufacturing cost with no compromise on driver safety. Vehicle must accommodate drivers having statures ranging from 5$^{th}$ percentile female to 95$^{th}$ percentile male. The car must also satisfy safety requirements such as side impact protection and impact attenuator [1]. Finally the cars are judged on the basis of performances during static and dynamic events including technical inspection, business presentation, cost, design, endurance tests etc. This research paper is casted from the work done for second international participation in FSAE competition by Chitkara, India team which took place in USA in June 2010. Team had already represented country in Australia in November 2008 and also adjudged as 2$^{nd}$ overall best team in SAE Chennai, India in December 2009. So on the basis of past experience

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and knowledge, whole of the car was re-modeled and re-fabricated for international participation in USA as per the SAE rules. This time reduction of chassis weight minimally by 10% using new materials and efficient design was decided as one of the main objective of car development. New materials were deemed to use to widen the structure's strength and re-modeled the design to inculcate more driver comfort, safety, structure triangulation and reduced inertial properties etc. The work started from the review of technical reports of several winning universities. Their main points regarding materials, design and load estimations were noted and discussed. Along with it, orthographic drawings, finite element analysis (FEA) reports of existing car's chassis were also brainstormed and reasons of high stresses and displacements were tried to discover. Modes of load distribution and their deformation concepts were taken care of from various reference books. Some of the concepts are also enunciated in this paper to help other universities while preparing the design of their car.

## 2.0 CHASSIS LOADING

Frame is defined as a fabricated structural assembly that supports all functional vehicle systems. This assembly may be a single welded structure, multiple welded structures or a combination of composite and welded structures [1]. Depending upon application of loads and their direction, chassis is deformed in respective manner briefed as follows [2]:

i. Longitudinal Torsion
ii. Vertical Bending
iii. Lateral Bending
iv. Horizontal Lozenging

### 2.1) Longitudinal Torsion

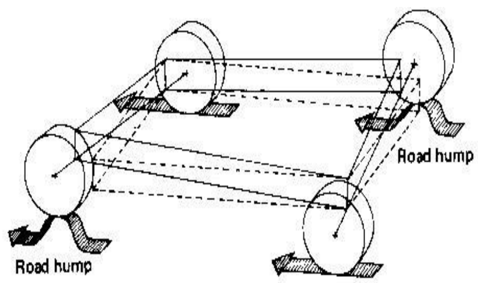


Figure 1: Longitudinal Torsion [3]

Application of equal and opposite forces act at a certain distance from an axis tends to rotate the body about the same axis. Automobiles also experience torsion while moving on road subjected to forces of different magnitudes acting on one or two oppositely opposed corners of the cars as shown in Figure 1. The frame can be thought as a torsion spring connecting the two ends where suspension loads act [2]. Torsional loading and resultant momentary elastic or permanent plastic deformation and subsequent unwanted deflections of suspension springs can affect the handling as well as performance of car. The resistance to torsional deformation is called as stiffness and it is expressed in Nm/degree in SI units. Torsional rigidity is a foremost and primary determinant of frame performance of cars.

## 2.2 Vertical Bending

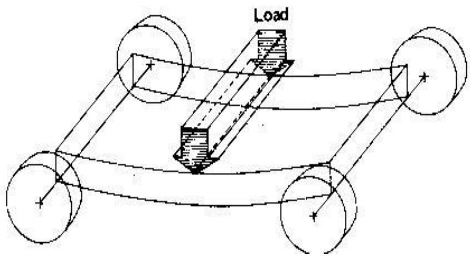


Figure 2: Vertical Bending [3]

Weight of driver, engine, drive-train, radiator and shell etc. under an effect of gravity produce sag in the frame as shown in Figure 2. Frame is assumed to act as simply supported beam and four wheels as supports tend to produce reactions vertically upward at the axles. Vertical dynamic forces due to acceleration/deceleration further increase the vertical deflections, hence stresses in chassis.

## 2.3 Lateral Bending

Lateral bending deformation occurs mainly due to the centrifugal forces caused during cornering and wind forces to some extent. Lateral forces act along the length of chassis and is resisted by axles, tires and frame members viz. hoops, side impact members and diagonal hoops etc as shown in Figure 3.

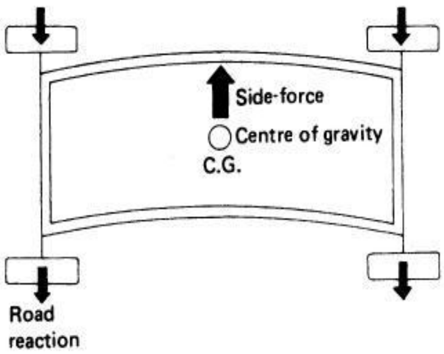


Figure 3: Lateral Bending [3]

## 2.4 Horizontal Lozenging

This deformation is caused by forward and backward forces applied at opposite wheels [3]. These forces may be caused by vertical variations in the pavement or the reaction from the road driving the car forward. These forces tend to distort the frame into a parallelogram shape as shown in the Figure 4. The magnitude of these loads changes with the operating mode of the car.

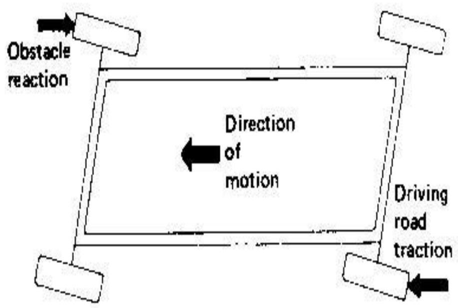


Figure 4: Horizontal Lozenging [3]

It is generally thought that if torsional and vertical bending stiffness is satisfactory, then the chassis structure is expected to perform well. But torsional stiffness is given more weight-age as the total cornering traction is the function of lateral weight transfers [2].

## 3.0 LOAD ESTIMATION

After literature review, it was brought in view that normally FSAE car parts are designed to withstand 3.5 g bump, 1.5 g braking and 1.5 g lateral forces [4]. These loads have to be considered individually and combined. Determination of magnitudes, types and center of gravity (cg) of loads is obligatory for optimum frame structure which is likewise a repetitive task. An understanding of different loads in respective directions is shown in Figure 5 in reference to Formula cars.

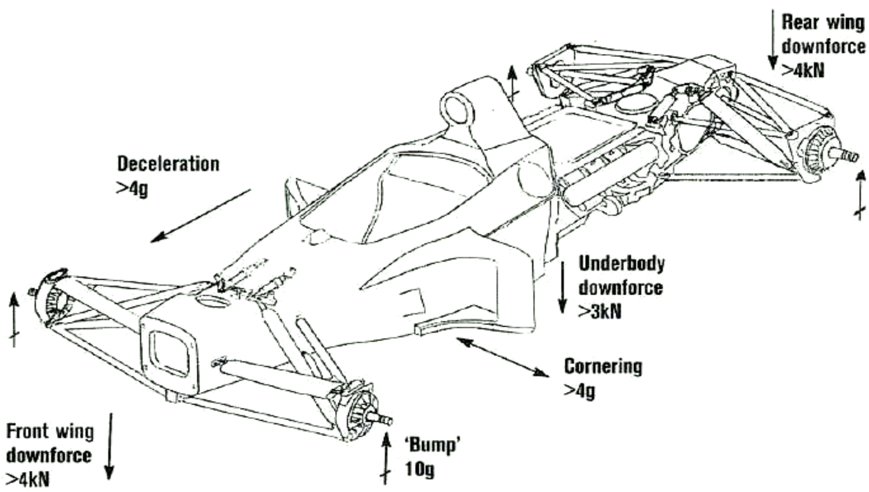


Figure 5: Forces acting on a formula one car [7]

To estimate an individual and total load of various components and car as a whole, a block diagram showing estimated position of components was created as shown in Figure 6. This schematic diagram simplified the understanding of different loads and their respective positions.

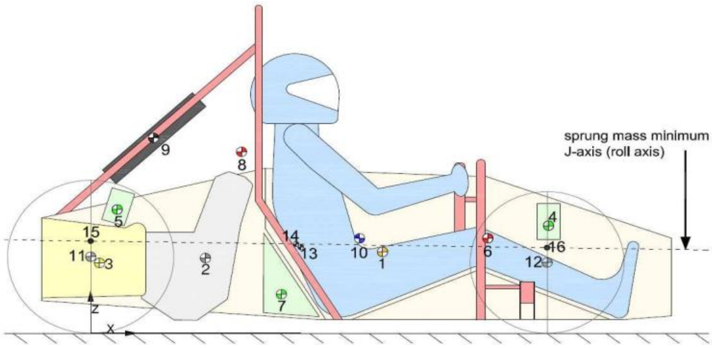

Figure 6: Car side view with all parts

Table 1: Approximate masses of main components

| C.G | Components | Mass (kg) |
|---|---|---|
| 1 | Driver | 80 |
| 2 | Engine | 70 |
| 3 | Drive-train | 20 |
| 4 | Steering | 10 |
| 5 | Battery | 03 |
| 6 | Chassis | 32 (Later calculated from mass properties) |
| | Total | 215 |

To consider mass of other components also, estimated mass of 250 kg was considered instead of 215 kg. This includes mass of wishbones (front and rear), petrol tank as well as radiator etc. Different forces viz. cornering, acceleration forces were computed from masses using Newton's second law of motion.

## 4.0 MATERIAL SELECTION

After load approximation, next step was the selection of material to construct a chassis. Availability is one of the factors which dominate the material selection process. Working on this single aspect, list of different desirable and available materials was prepared. Steel and aluminum alloys are always the choice of most of the teams. After reviewing mechanical properties, availability, cost and other significant factors, following material was selected.

Table 2: Mechanical Properties of Chassis Material

| **STEEL GRADE: IS 3074** | | |
|---|---|---|
| **S.No.** | **Properties** | **Values** |
| 1 | Young's modulus | 2e+011 N/m$^2$ |
| 2 | Poisson ratio | 0.266 |
| 3 | Density | 7860 kg/m$^3$ |
| 4 | Yield Strength | 3.73e+008 N/m$^2$ |

## 5.0 SOLID MODELLING

After load approximation and material selection, preparing CAD model of chassis was a next step. Based on past design knowledge, anthropometric data of tallest driver was taken and previous 3-D chassis model was modified. CATIA V5 software tool was used for designing as well as Finite Element analyses (FEA). SAE rules were taken care of while designing. Mankin was created in same software on the basis of anthropometric data and checked it under different realistic conditions to suit chassis design. It was a two way process as firstly creating model and checking clashes with mankin and vice versa was a repetitive task. After much iteration, CAD model was proposed as shown in Figure 7.

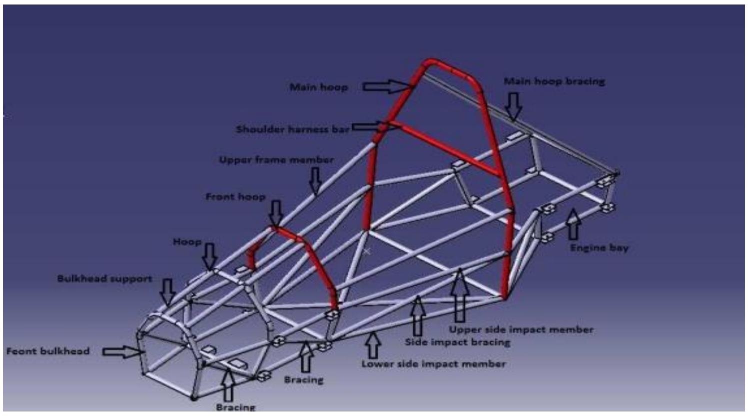


Figure 7: 3D Chassis Structure

Whole of the chassis model was made up of round hollow cross section tubes of IS 3074 steel throughout chassis. Tubes of two different sizes were used in the design. Whole of the structure comprises of tube 1" (outer diameter) and 1.6 mm wall thickness except main hoop and front hoop. Both (front and main) hoops are made up of 1" (Outer diameter) and 2.5 mm wall thickness as shown in red color in Figure 7. Mass properties showed the mass of chassis was to be 32 kg.

## 6.0 FINITE ELEMENT ANALYSIS (FEA)

Structure designing was followed by its testing and consequent validity. To determine the stiffness of a proposed frame design before construction, finite element analysis could serve the purpose. While the process of solving Finite Element problems is a science, creating the models is quite an art [2].

Conventionally in FEA, the frame is subdivided into elements. Nodes are placed where tubes of frame join. There are many types of elements possible for a structure and every choice the analyst makes can affect the results. The number, orientation and size of elements as well as loads and boundary conditions are all critical to obtain meaningful values of chassis stiffness [2]. Beam elements are normally used to represent tubes. The assumption made in using beam elements is that the welded tubes have stiffness in bending and torsion [2]. If a truss or link elements were used, the assumption being made would be that the connections do not offer substantial resistance to bending or torsion [2]. Another aspect of beam elements is the possibility of including transverse shearing effects.

While modeling the stiffness contribution from each part of the frame, method to apply the loads and constrain the frame plays significant role for an accurate analysis. Accurate analysis means to predict the stiffness of frame close to actual stiffness as the frame operates in real conditions. The problem here has normally been how to constrain and load a frame, so to receive multiple load inputs from a suspension, while it has been separated from that suspension and many other such problems. For practical reasons, it is recommended that the load on the chassis frame, including its own weight should be applied at the joints (nodes) of structural members. These point loads were statistically equivalent to the actual distributed load carried by the vehicle [5].

Another thing to consider while modeling the frame is how to represent an engine. For the engine, the first step is to locate a node at each position where there is an engine mount. These mounts then need to be connected to the frame by an element. Engine was assumed to be very stiff relative to the car frame. Thus assuming an engine to be infinitely rigid, it can be modeled by connecting each engine mount node to every other engine mount node by a beam element of high stiffness.

One of the few things that could be done to reduce the number of elements was to replace the engine model with a solid block of aluminum connected to the frame by the engine mounts. As most of the parts of an engine are made up of aluminum alloy, so it was assumed that an engine as a whole will behave in the similar manner as can be behaved by a solid block of same material. This greatly reduced the complexity of the meshed model and produced satisfactory results. Various elements used in the present paper to mesh the different parts of chassis are shown in Table 3.

Table 3: Elements used for meshing

| S. No. | Element | Purpose |
|---|---|---|
| 1 | Linear Tetrahedral | Round hollow tubes of frame, |
| 2 | Linear Tetrahedral | Engine and suspension mounts |

Applying static loads on model is comparatively easier than ascertaining a frequency range at which frame needs to be tested. In idle conditions, the speed range of Honda VFR engine which was used in this project is 12 to 14 revolutions per second. This translates into excitation frequency range of 13-15 Hz. The excitation from transmission is about 0-100 Hz. [6]. The main excitation is at low speeds, when the vehicle is in the first gear. At higher gear or speed, the excitation to the chassis is much less [5]. The natural frequency of the vehicle chassis should not coincide with the frequency range of the axles because this can cause resonance which may give rise to high deflection and stresses and poor ride comfort. Excitation from the road is the main disturbance to the chassis when the vehicle travels along the road. In practice, the road excitation has typical values varying from 0 to 100 Hz [5]. At high cruising speed, the excitation is about 9000 rpm or 150 Hz. Various boundary conditions and force/moments applied during various FEA tests are enunciated in the Table 4

Table 4: Boundary conditions used during various tests

| S. No. | Test | Boundary condition | Force Moments |
|---|---|---|---|
| 1 | Static Shear | Clamp- rear suspension mounts | Downward force at front bulkhead |
| 2 | Static overall bending | Clamp- front and rear suspension mounts | Uniformly distributed loading |
| 3 | Static torsional loading | Clamp- rear suspension mounts | Clockwise Moment at bulkhead side |
| 4 | Acceleration Analysis | Clamp- front and rear suspension mounts | Force applied towards rear |
| 5 | Frequency analysis | Clamp- front and rear suspension mounts | Frequency range- 69.12 Hz to 204.79 Hz |

## 7.0) RESULTS AND DISCUSSION

### 7.1 Static Shear

In static shear, it is assumed that frame acts like a cantilever beam and its one end is made fixed and other end is subjected to vertical downward force as shown in Figure 8. Shear force and bending moment diagrams were drawn and maximum bending moment was calculated analytically at the fixed end of frame. Blue color shows clamping and yellow color shows vertically downward forces acting at the front bulkhead as shown in figure 8. The rear suspension mounts were clamped in this case. Force of 1440 N was applied at the bulkhead which is the sum of weight of impact attenuator, driver legs and steering weight etc. Maximum bending moment of 2081 Nm calculated to act about the Y-axis.

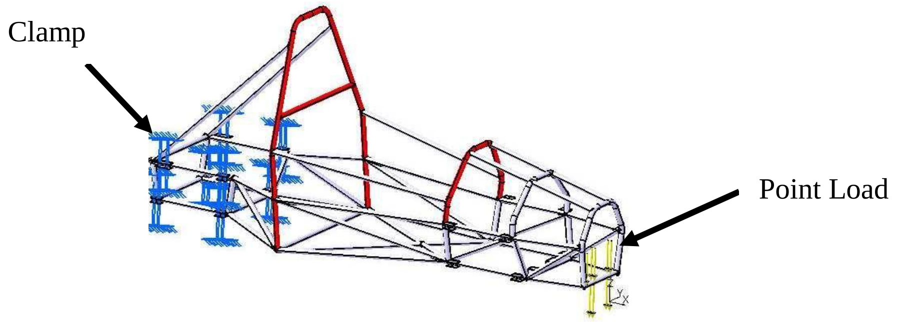


Figure 8: Boundary conditions during static shear

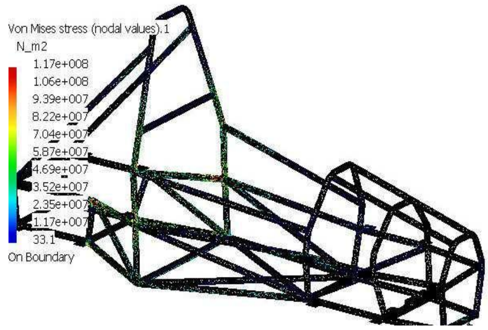


Figure 9: Von Misses stresses during static shearing

Results showed that maximum Von Misses stress was to be $1.17 \times 10^8$ N/m$^2$. Maximum strain energy (Proof resilience) capability of 4.345 Joules was observed from this analysis. Elements in red color show the maximum stress areas and corresponding maximum stress is shown in red color in stress tree in the left of Figure 9.

### 7.2 Static Vertical Bending

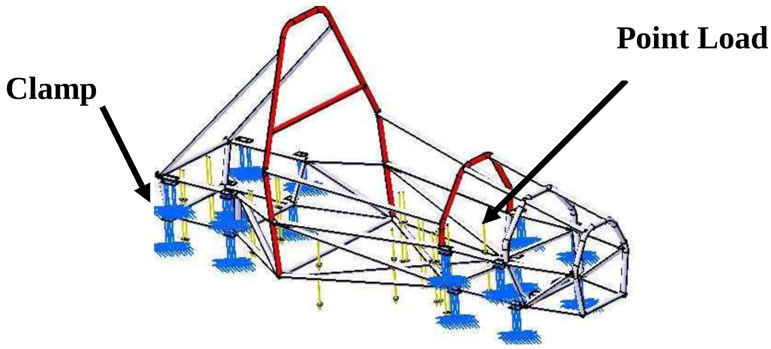


Figure 10: Boundary conditions during static vertical bending

Both front and rear suspension mounts were clamped and vertically downward point forces of 1550 N were applied equally in driver cabin, engine bay and drive-train section as shown in Figure 10. Frame is assumed to be a fixed beam with both ends clamped and subjected to point shear forces acting downward. Bending moment and shear force diagrams were drawn and values were calculated analytically. Blue color shows the clamping restraint and yellow color shows the point forces acting downward.

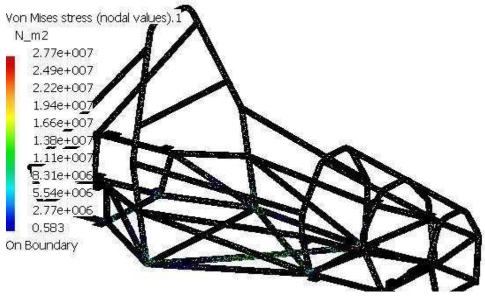


Figure 11: Von Misses stresses during static vertical bending

Maximum bending moment of 4.482 Nm about Y-axis with maximum vertical downward displacement of 0.369 mm was noted. Maximum Von Misses stress of $2.77 \times 10^7$ N/m$^2$ was observed at one or two places shown in red color in Figure 11. Most of the areas throughout chassis were observed to be subjected to minimum value of stress as shown in stress distribution tree in Figure 11. Maximum deflection was observed in the center of driver cabin floor and noted down its position to strengthen it. Strain energy of 0.087 J was noted.

### 7.3 Lateral Bending

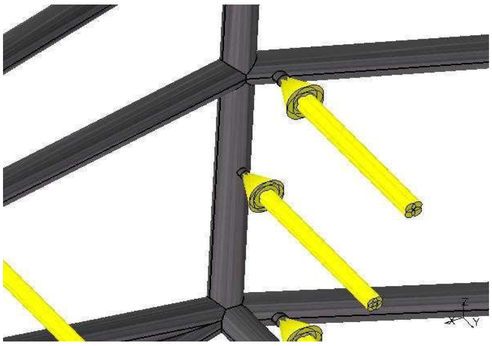

Figure 12: Application of lateral forces acting on roll hoop in driver cabin

Clamping restraint was applied at both front and rear suspension mounts as in previous cases. Lateral cornering point forces of 2325 N (Sum of engine and driver forces) was applied on side impact bracings of driver cabin, engine mounts and drive-train side braces. Yellow color arrows are depicted application of forces acting outwards.

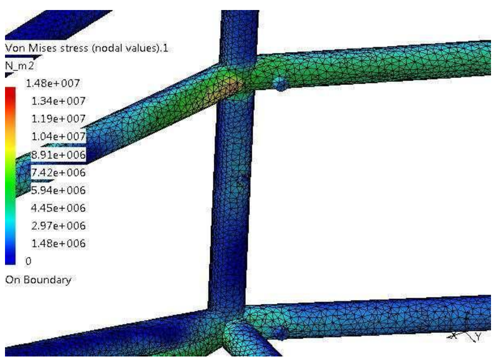


Figure 13: Von Mises stresses during lateral bending

Maximum principal stress of $1.48 \times 10^7$ N/m$^2$ (Figure 13) was observed with maximum translational displacement of 0.142 mm after post processing which are within the permissible limit of stresses. Strain energy of 0.015 J was observed.

### 7.4 Static Torsional Loading

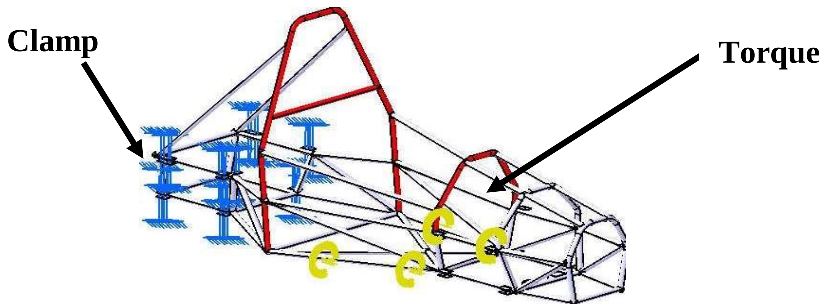


Figure 14: Boundary conditions during torsion test

Torsional rigidity test is one of the most important tests which validates/rejects the chassis structure. In this case, chassis is assumed to act as a cantilever with one end fixed and other end free and subjected to torque about its longitudinal axis as shown in Figure 14. A chassis should be able to resist angular deformation and resultant shear stresses. Again clamping is shown by the blue color and clockwise torque is shown in yellow color. Clockwise moment 316 Nm about longitudinal X-axis was applied.

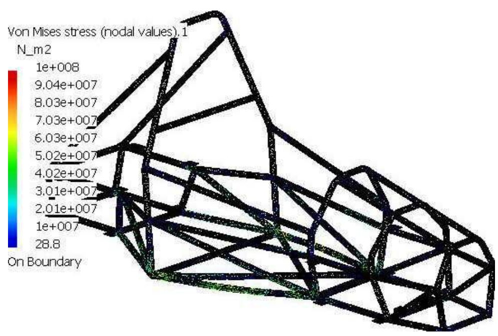


Figure 15: Von Misses stresses during torsion

Uniform stress of $5.02 \times 10^7$ N/m$^2$ (Figure 15) was observed with maximum stress of $01 \times 10^8$ N/m$^2$ at few points as shown in red color. Maximum translational displacement of 2.24 mm was noted in front bulkhead supports and lowers side impact members. Almost all other areas were found to be safe with approximately no stress and displacement. Strain energy of 2.303 J was observed from the results.

### 7.5 Accelaration Test

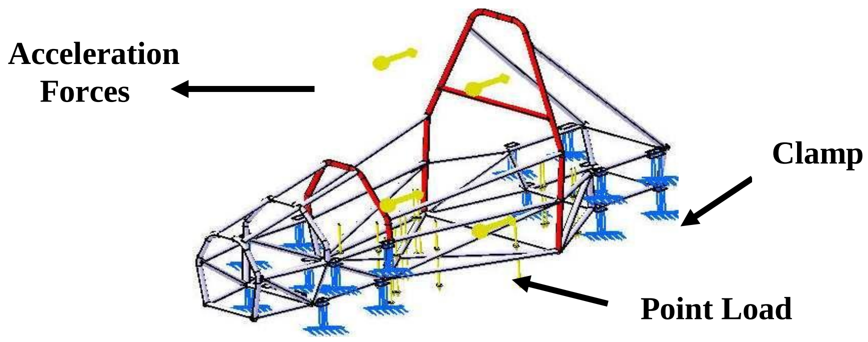


Figure 16: Boundary conditions during acceleration test

Due to inertia effect, acceleration forces tend to act in opposite direction to the motion of body. Forces due to acceleration were calculated considering masses of driver (80 kg) and engine (70 kg) respectively. Engine acceleration of 6.61 m/s$^2$ was taken from the manual of Honda VFR engine to be used in this vehicle and calculated an acceleration force using Newton second law of motion (F=m*a). Total acceleration force of 992.2 N was applied on the structure in backward direction shown in thick yellow color arrows in Figure 16. Load of 1550 N was applied throughout in driver cabin, engine bay and drive-train section as to simulate realistic conditions.

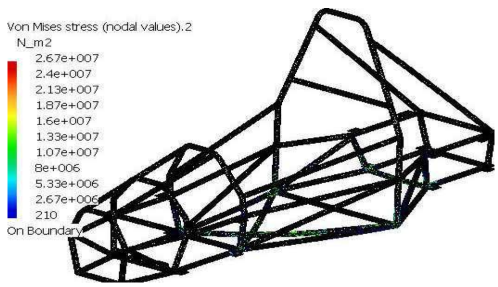


Figure 17: Von Misses stresses during acceleration

Chassis experienced maximum bending moment of 171.4 Nm about Y-axis (anticlockwise) at cross-sections shown in red color. Whole of the chassis was found to be within permissible stress limit with maximum stress was observed to be $2.67 \times 10^7$ N/m$^2$ as shown in red color in Figure 17. Strain energy of 0.102 J was given by the FEA results.

### 7.6 Frequency Test

Engine is a source of vibrations in any vehicle. Chassis as with any structure has an infinite number of resonant frequencies [8]. A resonant frequency, also known as natural frequency, is a preferred frequency of vibration and results when the inertial and stiffness forces cancel. For each of the infinite natural frequencies of vibrations which exist, a different shape that the chassis deform during vibration also exists [8]. The deformed shape that chassis will vibrate is also known as modes of vibration [8]. So, frequency analysis is mandatory to check structural behavior during a set range of frequencies. Indirect actuated suspension system was used at front and rear of the vehicle. A component called as rocker used to transmit the forces and motion from the tires through tie rods to springs of shock absorbers. The rocker was clamped and hinged with the chassis. Hence chassis was clamped at both front and rear due to suspension construction with no external load applied. Only structural mass was taken into account for analysis as shown in Figure 18.

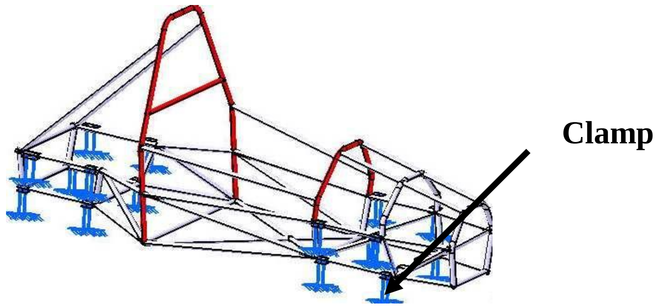


Figure 18: Boundary conditions during frequency test

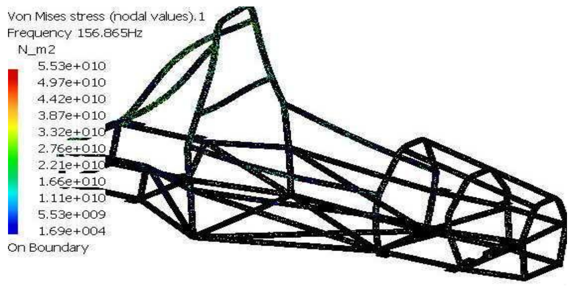


Figure 19: Von Misses stresses and modal analysis during frequency test

Software checked the structure from 69 Hz to 205 Hz automatically. No stresses were observed below 156.86 Hz and chassis was found to be safe (high factor of safety). At frequency of 156.86 Hz, $5.53\text{x}10^{10}$ N/m$^2$ (elements shown in red color in figure 19) was observed to be more than the permissible stress of material (3.73e+008 N/m$^2$). Maximum translational displacement of 562 mm was noted down at this frequency. Of the infinite modes of vibration that exist on the frame structure, only the lowest frequencies are of interest [8]. The lower modes of vibration maximize the kinetic energy and maximize the strain energy, while the high modes act in an opposite manner [8]. This means that the soft and stiff parts of chassis will be apparent in the low and high modes of vibrations respectively. Therefore it is worth to note that at lower frequencies (below 156.86 Hz), no considerable stress was found and chassis was assumed to be safe with considerable factor of safety. The most affected region was the main hoop and main hoop bracings with maximum stress and displacement.

Table 5: Von Misses stresses and Factor of Safety

| S. No. | Test | Von Misses Stress (N/m$^2$) | FOS |
|---|---|---|---|
| 1 | Static Shear | 1.17*108 | 3.18 |
| 2 | Static overall bending | 2.27*107 | 16.41 |
| 3 | Lateral Bending | 1.48*107 | 25.25 |
| 4 | Static torsional loading | 1*108 | 3.73 |
| 5 | Acceleration test | 2.67x107 | 13.97 |
| 6 | Frequency analysis | $5.53*10^{10}$ (156.86 Hz) | 00 |

By dividing yield stress (3.73e+008 N/m$^2$) of chassis material with maximum Von Misses stresses induced in frame, factor of safety was calculated as shown in table 5. Considerable factor of safety was observed in all static (shear, bending and torsional test) boundary conditions. Chassis was found to exhibit high factor of safety during dynamic viz. acceleration test. Frame behavior was analyzed in frequency range of 69.12 to 204.793 Hz and observed maximum deformation (stress and deflection), more than the yield stress of frame material at a frequency of 156.86 Hz. Below 156.86 Hz, chassis was observed to be safe and experienced stress very small than the strength of material. So, 156.86 Hz can be considered as threshold value for proposed chassis. Chassis was found to have highest factor of safety in lateral bending (25.25) followed by static bending (16.41) and dynamic acceleration test (13.97) respectively.

Factor of safety was noted to be 3.73 in torsional loading mode which represents the stiff nature of frame. An ideal chassis is one that has high stiffness; with low weight and cost. If there is considerable twisting, the chassis vibrates, complicating the system of the vehicle and sacrificing the handling performance [8]. The chassis that flexes is more susceptible to fatigue and subsequent failure, and “suspension compliance may be increased or decreased by bending or twisting of the chassis [9]. Also if a chassis is well designed to handle torsional loads, bending should not be an issue [9]. The torsional rigidity can be calculated by finding the torque applied to the frame and dividing by the angular deflection. The actual calculation is done as follows, with the figure 20 showing a view looking from the front of the suspension bay.

$$K=R/\Theta \tag{1}$$

$$K= (F*L)/\tan^{-1}[(\Delta y_1+\Delta y_2)/2L] \tag{2}$$

where K = Torsional Stiffness
T = Torque
$\Theta$ = Angular deformation
F = Shear Force
$y_1$, $y_2$= Translational displacement

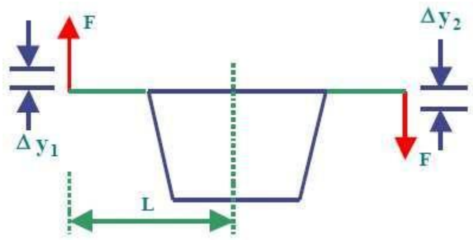


Figure 20: Front suspension bay testing loads

Force applied (F) = 1264 Nm
$y_1 = y_2 = 2.24$ mm = 0.00224m
L= 0.250 m
$K = (1264*0.250)/\tan^{-1}[(0.00224+0.00224)/2*0.250] = 615.98$ Nm/deg

Torsional rigidity of previous design was calculated to be 250 Nm/deg [11] which was an indicator of moderate stiffness and hence needed to be improved following changes in design and material. The torsional stiffness of present design was calculated to be 615.98 Nm/deg as calculated above which shows significant increase in stiffness in present model by 2.46 times than the older design. Deakin et al concluded that a Formula SAE racer, which has a total suspension roll stiffness of 500–1500 Nm/degree, requires chassis stiffness between 300 and 1000 Nm/degree to enable the handling to be tuned [10]. Torsional rigidity of University of Southern Queensland (USQ) 2004 SAE car experienced torsional rigidity of 214 Nm/degree and appeared to drive reasonably well, apart from the under-steer and other minor construction matters [10]. Increase in chassis stiffness in present work owes to good structural design having more triangulation and of course higher yield strength of IS 3074.

## 8.0 CONCLUSIONS

The dominant characteristic of structural behavior viz. torsional rigidity increased by 2.46 times with an average value of 615.98 Nm/deg without compromising on weight. Weight of chassis was observed to decrease by 11.11% with approximate value of 32 kg as compared to weight of older chassis frame. Stress distributions were found to be even and less than the yield strength of material (3.73e+008 N/m$^2$). Chassis was found to be safe significantly in static (bending) and dynamic (acceleration) modes with stress values noticeably less than the yield strength. Critical value of stress was found to be 5.53x10$^{10}$ N/m$^2$ at a frequency of 156.86 Hz. Although below this frequency, chassis was found to be safe and exhibited almost no stress. Dynamometer testing of previous car at approximate 9000 rpm with speed of 90 km/h, the maximum vibration frequency was noted to be not more than 100 Hz. So, it was expected that vibration range under same conditions will either remain same or decrease for the present design model. Hence chassis was expected to perform well in motion also.

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