ADAMA SCIENCE AND TECHNOLOGY UNIVERSITY

SCHOOL OF MECHANICAL, CHEMICAL AND MATERIALS ENGINEERING

Structural Analysis of FAW Dump Truck Chassis Frame Made of Composite Materials

A thesis submitted in partial fulfillment of the requirements for the award of the degree of Master of Science in Automotive Engineering

By

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CANDIDATE'S DECLARATION

I here by declare that the thesis which is being presented entitled "**structural analysis of FAW dump truck chassis frame made of composite materials"**, in artial fulfilmen of the requirements for the award of the degree of master of science in automotive engineering is an authe tic record of my own work carried from February 2014 to October 2014 under the supervision of N. Rmesh Babu (Associate professor) and K. Sirinivasa Reddy (Associate professor) department of Mechanical and vehicle engineering, Adama science and technology university, Ethiopia.

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NOMENCLATURE

F	Force matrix
K	Stiffness matrix
U	Displacement matrix
I_{XX}-	Moment of inertia about X axis
b	Base of the section
h	Height of the section
Z_{XX}	section modulus
σ_y	yield strength
RBM	resisting bending moment
σ_{ult}	Ultimate strength
E	Young's modulus
P	Density
T_g	Glass transition temperature
T	Torsion
P_L	Left wheel loads
P_R	right wheel loads
B	Track length
H	Bump height
P_{axle}	axle loads
ω_e	engine excitation frequency

Ke	element stiffness matrix
K	global stiffness matrix
Me	element mass matrix
M	global mass matrix
F^{nd}	nodal force vector
Q_i	amplitude vector
T	time
F^r	reaction vector
q	nodal displacement vector

LIST OF ABREVIATION AND ACRONYMS

HDVs	heavy duty vehicles
FEM	finite element methods
CAD	computer aided design
CAE	computer aided engineering
BIW	body in white
SUVs	sport utility vehicles
Max	maximum
Min	minimum
CFRP	Carbon fiber reinforced plastics
3D	three dimensional
FEA	finite element analysis
VARTM	vacuum-assisted resin transfer molding
RTM	resin transfer molded
GVM	gross vehicle mass
UDL	uniformly distributed load
CFRP	carbon fiber reinforced plastic
OEMs	original equipment manufacturer
GHG	green house gas emission
GVWR	Gross Vehicle Weight rating
LCC	Life-Cycle-Costs

ASTM	America Society for Testing and Material
MMC	Martin Marietta Composites Inc.
SMC	sheet molding compound
USCAR	U.S. Council for Automotive Research
Kg	kilogram
mm	Millimeter
kN	Kilo Newton
kPa	Kilo Pascal
MPa	Mega Pascal
GPa	Giga Pascal
m	Mass
V	Volume

ABSTRACT

The chassis frame forms the backbone of a heavy vehicle. Its principal function is to safely carry the maximum load for all designed operating conditions. This paper describes structural analysis of heavy duty truck chassis. Structural analysis involves static and dynamics analysis to determine the key characteristics of a truck chassis. The static analysis is done when the chassis frame is subjected to three different load cases such as bending, torsion and combined bending and torsion load cases acting on the horizontal C- Chanel. The maximum stress is obtained due to torsion load case. The dynamic analysis of truck chassis such as the natural frequency and mode shape were determined by using finite element (FE) method. Weight reduction is now the main issue in automotive industries. Traditionally, the most common material for manufacturing vehicle chassis has been steel, in various forms. Over time, other materials have come into use, the majority of which have been is Steel, Aluminum and composite materials. In this paper the dimensions of an existing heavy vehicle chassis of a FAW dump truck vehicle is taken for modeling and analysis of a heavy vehicle chassis, and traditional materials are replaced with composite materials [Carbon Epoxy and E- glass epoxy] with the same geometry subjected to the same pressure as that of a steel chassis. From the static analysis it is obtained that carbon epoxy has relatively higher factor of safety than steel and e glass epoxy. By using composite material for chassis frame it is possible to reduce the weight up to 79.48% for carbon epoxy and 66.67% for e glass epoxy. From the modal analysis taking the first six natural frequencies, carbon epoxy has higher natural frequencies ranges from 56.8Hz to 134.8Hz. The first two natural frequencies are in engine excitation range but all the first six natural frequencies of e glass epoxy (27.22Hz to 64.59Hz) and steel (31.97Hz to 75.88Hz) lies in the engine excitation frequency range of 13.33Hz to 83.33Hz. The results show that the road excitation is the main disturbance to the truck chassis as the chassis natural frequencies lie within road excitation frequency. Carbon epoxy is the best material that has higher strength to weigh ratio and factor of safety with less vibration and weight. Software's used in this work is CATIA V5 R19 for modeling and ANSYS

14.5 for the finite element analysis. The design constraints were stresses, deflections, natural frequency and mode shapes. Generally using composite materials for chassis frame is safe.

Key words: chassis, composite materials, stress, deformation, mode shape, natural frequency

CHAPTER 1

INTRODUCTION

1.1 Introduction about chassis

The French term Chassis was initially used to denote the frame parts or basic structure of the vehicle. It is the back bone of the vehicle. Automotive chassis usually refers to the lower body of the vehicle including the tires, engine, frame, driveline and suspension. Also the frame should be rigid enough to withstand shock, twist, vibrations and other stresses. The components of the vehicle like engine, transmission system, axles, wheels and tires, suspension, controlling systems and also electrical system parts are mounted on the chassis frame. The loads acting on a chassis are weight of vehicle and passengers, load due to road camber, side wind, cornering, and engine torque, braking torque, load due to road obstacles and finally the load due to collisions [9]

The ladder chassis frame consists of side members attached with a series of cross members. The critical point at which the maximum stress occurs can be found out by stress analysis of the frame using finite element method. The critical point is the main factor that causes the fatigue failure of the frame. The magnitude of the stress can be used to decide the life of the truck frame. The accuracy of prediction life of truck chassis is depending on the result of its stress analysis. The various cross sections used in frame construction include channel, box, hat, double channel and I section [11].

Chassis is one of the important parts that used in automotive industry. This structure was the bigger component in any automobile vehicle that the vehicle shape is dependent to this chassis. The main function of the chassis is not only support the components and payload mounted upon it including engine, body, passengers and luggage, but also to maintain the desired relationship between the suspension and steering mechanism mounting points. The greater the energy absorbed by the chassis on impact the lower the energy levels transmitted to a vehicles occupants

and surroundings, lowering the chances of injury. The chassis is subjected to stress, bending moment and vibrations due to road roughness, weather and components that mounted on it, when the truck travels along the road, Stress that is acting on chassis varies with the displacement and each part on the car chassis [1].

The chassis is the framework that is everything attached to it in a vehicle. In a modern vehicle, it is expected to fulfill the following functions:

i) Provide mounting points for the suspensions, the steering mechanism, the engine and gearbox, the final drive, the fuel tank and the seating for the occupants;
ii) Provide rigidity for accurate handling;
iii) Protect the occupants against external impact.

While fulfilling these functions, the chassis should be light enough to reduce inertia and offer satisfactory performance. It should also be tough enough to resist fatigue loads that are produced due to interaction between the driver, engine, power transmission and road conditions.

The dump truck is used for transporting materials in mining and construction. A typical dump truck is equipped with a hydraulically operated suspension with closed box structure chassis, a body which is located at the top of the chassis. The chassis supported at the front and rear side suspension of the truck. The purpose of the body is lifted rear side of the vehicle, which is to deposit the materials on the ground. Chassis is an important part of the load carrying member of a rear dump truck over which the entire equipment is structured. Chassis acting as a structural support for power train elements and also enables the dump body carrying full payload. The chassis is neither a serviceable nor a replaceable component as far as the life span of the vehicle is concerned. The life span of a chassis must be equal to or more than 30,000 running hours under normal operating condition of the vehicle [5].

The chassis development processes in the automotive and transport equipment industries are subjected to ever higher demands in terms of improved reliability, safety and performance, as well as reduced weight, production cost and development lead time. Due to the demand of huge truck for building and testing prototypes and manufacturing, there is increased emphasis on analytical durability assessment methods. In the case of special purpose vehicles, for which the

low production volumes may exclude the possibility of extensive prototype testing, such emphasis would be even more pronounced.

The first step of any durability assessment process involves the definition of the loading conditions. The automotive industry is expending constant efforts in correlating ground loading conditions with the customer profiles. Alternatively, employs standardized load time histories derived from extensive field measurements. In the heavy vehicle industry, particularly in the case of special vehicles, such information is not always readily available. Hence, the need for economical methods to describe loading conditions for design purposes is apparent. The second step in the assessment process involves calculating the stress response to the input loading. In case of automotive industry, this is commonly achieved using the dynamic finite element analysis method. Due to the cost of dynamic finite element analyses, there is an incentive to circumvent the need for analysis. Since calculations could not stipulate the use of dynamic finite element analysis method. Since heavy vehicle structures are usually welded, less complex stress-life method is commonly employed [23].

In this thesis static structural analysis and modal analysis on the ladder chassis is done. The static structural analysis includes identification of maximum stressed area and also the deformation for three different load cases. Modal analysis includes determination of natural frequency and mode shape.

1.2 Back ground of the study

The chassis is inherently important, as it is the frame, the internal structure that supports all the Components and occupants, preferably in a well balanced and effective manner. Rigidity in bending and torsion, efficient load absorption and low weight are keys to strong chassis performance [18].

The history of the ladder frame chassis dates back to the times of the horse drawn carriage. It was used for the construction of ‘body on chassis’ vehicles, which meant a separately constructed body was mounted on a rolling chassis. The chassis consisted of two parallel beams mounted down each side of the car where the front and rear axles were leaf sprung beam axles.

The beams were mainly channeled sections with lateral cross members, hence the name. The main factor influencing the design was resistance to bending [7].

A ladder frame acts as a grillage structure with the beams resisting the shear forces and bending loads. To increase the torsion stiffness of the ladder chassis, cruciform bracing was added in the 1930's. The torque in the chassis was restrained by placing the cruciform members in bending, although the connections between the beams and the cruciform are rigid. Ladder frames were used in car construction until the 1950's but in racing only until the mid 1930's [18].

The ladder frame chassis became obsolete in the mid 1930's with the advent of all-round independent suspension, pioneered by Mercedes Benz and Auto Union. The suspension was unable to operate effectively due to the lack of torsion stiffness. The ladder frame was modified to overcome these failings by making the side rails deeper and boxing them. A closed section has approximately one thousand times the torsion stiffness of an open section. Mercedes initially rectangular section and later switching to oval section, which has high torsion stiffness and high bending stiffness due to increased section depth, while Auto union. Mercedes design was further improved by mounting the cross members through the side rails and welding on both sides [7].

In the early days of vehicle design and manufacture, the chassis was the frame above which the body was mounted; the axles were mounted below by means of their springs, together with any associated steering and braking systems. During the 1920s, pioneers began studying the implications of eliminating the chassis frame and attaching the wheels whether mounted on axles or independently - directly to the body. It quickly emerged that to achieve similar orders of stiffness in bending and especially in torsion, a load carrying body should be made lighter and also more compact than an unstressed body mounted on a chassis frame. By the 1950s the stress-carrying "unitary" body had become the industry standard. The terms "chassis" had not disappeared, but had rather been transferred to those systems between the body and the road surface - the suspension linkages, springs and dampers, and the wheels themselves - together with the closely associated steering and braking systems, essential for controlling the vehicle some observers regard chassis systems and suspension systems as synonymous. However, this disregards the interdependence of suspension, steering and braking systems and the fact that they are integrated to an increasing degree, especially at the electronic level [18]. It may also be

argued that even in the early days of motoring when complete chassis were delivered to coachbuilders for body installation, the steering and the brakes (such as they were) were already installed. In present-day terms, therefore, the terms "chassis engineering" or "chassis systems" embrace a hierarchy of technologies and features that may be outlined as follows:

Suspension: This include the choice of basic geometry for optimum wheel location, the mounting of suspension members on the body (including the use of sub-frames), the springing medium and the provision of damping of vertical wheel movement.

Steering: Includes the optimization of front suspension geometry for steering, the choice of steering system, the provision of power assistance, the satisfaction of safety requirements, and the provision of "augmented stability" through interaction with the braking system.

Braking: Refers to the choice of friction system, the design of the operating linkage, the provision of servo assistance, the satisfaction of safety requirements, the provision of anti-lock braking and other enhancements such as emergency brake assist.

Wheels and tyre: Include choice of wheel and tyre size, choice of wheel material and tyre configuration, choice of spare wheel configuration.

1.3 Classification of truck

Generally, truck vehicles are designed for carrying or pulling loads. It is an automotive vehicle suitable for hauling. Definitions vary depending upon the type of truck and its application.
Truck definition and classification

Generally, truck is any of various heavy motor vehicles designed for carrying or pulling loads. Other definition of the truck is an automotive vehicle suitable for hauling. Some other definition are varied depending on the type of truck, such as Dump Truck is a truck whose contents can be emptied without handling; the front end of the platform can be pneumatically raised so that the load is discharged by gravity.

Trucks are classified based on the weight classes, as defined by the US government, ranging from Class 1 to Class 8 as listed in Table below. The second is classified into a broader category [17].

• Light Duty Truck

• Medium Duty Truck

• Heavy Duty Truck

Table 1.2.1 Truck Classification by the U.S. Department of Transportation's GVWR and Vehicle Inventory Use Service (VIUS) Categories [17].

Category	Class	GVWR	Representative vehicles
Light duty	1	0-27KN (0-6000 lbs)	Pickup trucks, ambulances, parcel delivery
	2	27-45KN (6000-10,000 lbs)	
	3	45-62KN (10,000-14,000lbs)	
Medium duty	4	62-71 KN (14,000-16,000 lbs)	City cargo van, beverage delivery truck, wrecker, school bus
	5	71-87 KN (16,000-19500 lbs)	
	6	87-116 KN (19,500-26,000 lbs)	
	7	116-147KN (26,000-33,000lbs)	
Heavy duty	8	147kN and over (33,000lbs. and over)	Truck tractor, concrete mixer, dump truck, fire truck, city transit bus

Notes: Gross Vehicle Weight Rating (GVWR): weight specified by manufacturer as the maximum loaded weight (truck plus cargo) of a single vehicle.

This thesis is mainly concentrated on the analysis of FAW dump truck chassis frame which is categorized under heavy duty truck from the above tables of truck classification. So there are different types of dump trucks listed as follow.

1.2.1 Standard Dump Truck

The standard dump truck is a full truck chassis with a dump body mounted onto the frame. The dump body is raised by a hydraulic ram lift that is mounted forward of the front bulkhead, normally between the truck cab and the dump body. The standard dump truck also has a front

axle, and one or more rear axles, which normally have dual wheels on each side. The common configurations for standard dump trucks are six and ten wheelers [17].

1.2.2 Transfer Dump Truck

The transfer dump truck is easily recognizable through the noise made during transfer. A standard dump truck pulls a separate trailer which can be loaded with sand, asphalt, gravel, dirt, etc. The box or aggregate container on the trailer is powered by an electric motor, which rides on wheels and rolls off of the trailer and into the main dump box. The biggest advantage of this configuration is that payload capacity maximized manufacturability of the short and nimble dump truck standards [17].

1.2.3 Semi Trailer Dump Truck

The semi end dump truck is a tractor trailer combination where the trailer itself contains the hydraulic hoist. The average semi end dump truck has a 3 axle tractor that pulls a 2 axle semi trailer. The rapid unloading is the main advantages of semi trailer dump truck [17].

1.2.4 Semi Trailer Bottom Dump Truck

A bottom dump truck is a 3 axle tractor that pulls a 2 axle trailer with a clam shell type dump gate in the belly of the trailer. The biggest advantage of a semi bottom dump truck is its ability to lay material in a wind row unlike the double and triple trailer configurations. This type of truck is also maneuverable in reverse as well [17].

1.2.5 Double and Triple Bottom Dump Truck

The double and triple bottom dump trucks consist of a 2 axle tractor pulling a semi axle semi trailer and an additional trailer. These types of dump trucks allow the driver to lay material in wind rows without the material having to leave the cab or stop the truck. However biggest disadvantage is the difficulty in going in reverse [17].

1.2.6 Side Dump Truck

Side dump trucks consist of a 3 axle trailer pulling a 2 axle semi trailer. It offers hydraulic rams that tilt the dump body onto the side, which spills the material to the left or right side of the

trailer. The biggest advantages of these types of dump trucks are that they allow rapid unloading and increased weight carrying capacities when compared to other dump trucks. In addition to this, side dump trucks are almost impossible to tip over while dumping, unlike the semi end dump trucks which are very prone to being upset or tipped over. The lengths of these trucks impede maneuverability and also the limit versatility of the chassis [17].

1.2.7 Off-Road Dump Truck

Off-road trucks resemble heavy construction equipment more than highway dump trucks. They are used strictly for off road mining and heavy dirt-hauling jobs, such as excavation. They are very big in size, and are perfect for having massive amounts of dirt to another location whole digging roads [17].

1.3 Statements of the problem

The truck industry has experienced a high demand in market especially in Ethiopia where by the economic growth is very significantly changed from time to time. There are many industrial sectors using this truck for their transportations such as the logistics, agriculture, factories and other industries. However, the development and production of truck industries in Ethiopia are currently much relying on foreign technology and sometime not fulfill the market demand in term of costs, driving performances and transportations efficiency.

When a vehicle is running, it is subjected to dynamic loads. Under such various dynamic excitations, the truck chassis tends to vibrate. If any of the excitation frequencies coincides with the natural frequencies of the truck chassis, then resonance phenomenon occurs. The chassis will undergo dangerously large oscillations, which may lead to excessive deflection and failure. The vibration of the chassis will also cause high stress concentrations at certain locations, fatigue of the structure, loosening of mechanical joints, and creation of noise and vehicle discomfort. To solve these problems, study on the truck chassis static and dynamic characteristics is thus essential [1].

Weight reduction is now the main issue in automotive industries. Because if the weight of the vehicle increases, the fuel consumption increases. At the same time as the weight of the vehicle increases the cost also increases which becomes a major issue while purchasing the vehicle. For

example if we take frame of cargo truck heavy vehicle frame. It is manufactured with mild Steel. Steel structures exposed to air and water, such as bridges are susceptible to corrosion. In conditions of repeated stress and more temperatures it can suffer fatigue and cracks. These are the main problems of steel and these are compensated by inducing composite materials [17].

The major challenge in today's ground vehicle industry is to overcome the increasing demands for higher performance, lower weight in order to satisfy fuel economy requirements, and longer life of components, all this at a reasonable cost and in a short period of time also give new safety requirements. In order to achieve and satisfy the above demands, designing the chassis frame made from composite material is necessary.

According to the fact that car industries should focus on developing high performance cars at lower cost, developing light weight vehicles using new materials are significantly considered in R&D section. In fact light weight cars will reduce the fuel consumption remarkably so that the running cost will be decreased. Furthermore lower fuel consumption will reduce the emission of exhaust gases to the air [16].

Table 1.3.1 Total life cycle cost comparison of BIW [16]

	Min for steel BIW	Average for steel BIW	Max for steel BIW	Min for carbon fiber BIW	Average for carbon fiber BIW	Max for carbon fiber BIW
Pre manufacturing cost ($)	284.9	381.1	481	1928	2442	3256
Manufacturing cost ($)	1995	1995	1995	994	994	994
Use cost ($)	13144.14	16398.42	19716.16	10110.66	12637.63	15166.08
Post use ($)	4.63	4.63	4.63	19.17	19.17	19.17
Total life cycle cost ($)	15428.7	18779.2	22196.8	12752.8	16092.8	19435.3

The total life cost of steel BIW is 18779.2$ while this amount is 16092.8$ for carbon BIW. This has also means that life cycle cost of carbon fiber BIW is only 16% lower than the steel [16].

1.4 Significance of the study

The thesis will have a significant impact in improving the technical capability of the country in design and analysis of the vehicle chassis frame by its own man power. Because automotive technology in Ethiopia is relying on foreign industries, so it will increase the design and analyzing capability of engineers in automotive industrial areas.

It also has an implication for establishing linkage between the university and automotive industries. More over the study focuses on static and dynamic analysis of the existing chassis frame and modifying it with composite material such as carbon/epoxy and E glass/epoxy; to reduce the cost of fuel consumption by reducing the weight of chassis frame, identify the critical stress area.

This thesis is very important for structural engineers in providing a knowledge and data about composite materials. As we know when heavy weight materials are substituted with light weight materials such as composites, aluminum, others for the same part, the thicknesses will not be the same for both light and heavy weight materials. For a better strength and support the thickness of the part made from light weight material is greater than heavy weight materials. So in this thesis the performance of composite materials are examined with the same thickness to the steel when they are subjected to the same pressure or load.

1.5 Objective of this thesis

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1.5.1 General objective

The general objective of this thesis is to analyze the structure of dump truck chassis frame made of composite materials.

1.5.2 Specific objectives

i) To observe the existing chassis and to note its features.
ii) To develop 3D CAD model of the chassis frame.

iii) To analyze statically and dynamically the model of existing chassis frame and also chassis made of composite material under the same working conditions.
iv) To know the performance of composite materials and the weight difference.
v) To compare the results obtained

1.6 Scope of the study

The scope of this work includes:

i) Literature review associated with truck chassis development, design and analysis of the commercial truck chassis frame.
ii) This study will concentrate on dump truck chassis frame.
iii) Simulation work on the existing truck chassis with ANSYS work bench.
iv) Improvement of truck chassis characteristic by changing the material.
v) Development of new truck chassis made of composite materials such as Carbon/Epoxy and E-glass/Epoxy.
vi) Modeling of the frame will be done by CATIA V5 software.
vii) Experimental analysis test is not included. The analysis is done with finite element method commercial software ANSYS 14.5.

1.8 Structure of the thesis

The scope of the research is confined to develop a 3D finite element analysis of static bending, torsion, and modal strength of a chassis and investigating the responses of chassis frame for mild steel and composite materials with the same thickness subjected to the same pressure. The thesis is structured accordingly and it is detailed below.

Chapter 2 discusses review of literature in finite element simulation of chassis. It deals with the literature related to simulation of static and dynamic analysis histories involved in the chassis design and analysis of bending, torsion and modal analysis. Secondly, it covers the application and role of composite materials for heavy duty truck and trailer parts. It also covers some of the earlier works related to modeling, design and analysis of an automotive chassis.

Chapter 3 discusses on materials and methodologies of the thesis. The aver all specifications of the vehicles such as weight specifications, engine specifications and chassis specifications and

the material properties of mild steel and composite materials (carbon epoxy and e glass epoxy) are clearly described. The CAD model of the chassis frame is also developed.

Chapter 4 deals with theories and design constraint and parameters such as section modulus, moment of inertia, strength, stiffness, and resisting bending moment of chassis frame. It includes types of chassis, chassis load cases such as bending, torsion and combined bending and torsion, load paths, selection of materials, composite manufacturing or fabrication processes

Chapter 5 deals with the static dynamic analysis of FAW dump truck chassis frame the design and development of 3-D CAD model of FAW dump truck chassis frame for FE simulation, meshing and analysis of chassis. The FE simulation for static analysis of bending, torsional and combined bending and torsion load cases to finding out the stress distribution and deformation of chassis frame for three materials such as mild steel, carbon epoxy and e glass epoxy and compare the result. It also includes modal analysis to determine the natural frequencies and mode shapes of mild steel and polymeric composite chassis frame. These two parameters are very important to know the vibration behavior of the structure. If one of the natural frequencies coincides with excitation frequency resulted from engine, transmission and road roughness the phenomena which is called resonance will occur.

Chapter 6 deals with the conclusion and recommendation depending on the result obtained from the analysis.

CHAPTER 2

REVIEW OF LITERATURE

2.1 Introduction

Basically chassis is considered as a framework to support the body, engine and other parts which make up the vehicle. Chassis lends the whole vehicle support and rigidity. Chassis usually includes a pair of longitudinally extending channels and multiple transverse cross members that intersect the channels. The transverse members have a reduced cross section in order to allow for a longitudinally extending storage space.

Many of the truck structures found in the research are subjected to internal and external loads, which affected the ride quality of vehicle. These loads and behavior can be determined through a series of processes namely, Modal Analysis, Finite Element bending and Torsion Analysis. From the global torsion analysis, it has been found that the torsion load is more severe than bending load. In order to overcome this problem, a cross bar and material selection are important to be considered during design stage. Furthermore, the overall achievement is mainly to reduce the vibration level, so that the life of the chassis structure and performance can be maximized.

The primary objective of the chassis is to provide a structure that connects the front and rear suspension without excessive deflection. When considering an automotive chassis, a frame that is easily twisted will result insignificant stability problems related to stability. Suspension setup is based upon the assumption that all four corners of the vehicle are connected by an infinitely rigid body. If the chassis is not sufficiently stiff, the structure is merely another variable in the system as it acts as a torsional spring and damper. This variability within the chassis structure would very difficult to dial in a suspension setup that will generate the necessary levels of lateral grip to be competitive. Generally speaking, a chassis which is able to resist torsional loads resulting from inertial accelerations of components experienced during cornering or from applied

loads acting on one or opposite corners (i.e. Torsion) of the vehicle will almost always be sufficiently strong.

2.2 Finite Element Analysis of Truck Chassis

The Finite Element Analysis (FEA) is becoming increasingly popular among design engineers using it as one of many product design tools. It has been used for the solution of many types of problems. FEA has become an integral part of design process in automotive, aviation, civil construction and various consumer and industrial goods industries, cut throat competition in the market puts tremendous pressure on the corporations to launch reasonably priced products in short time, making them to rely more on virtual tools (CAD/CAE) accelerate the design and development of products. FEA is used to predict multiple types of static and dynamic structural responses. For example, companies in the automotive industry use it to predict, stress, strain, deformations, and failure of many different types of components. FEA reduces the need for costly experiments and allows engineers to optimize parts before they are built and implemented. The wide and most common commercial finite element packages are (ANSYS, ABAQUS, COMSOL, ALGOR, HYPERVIEW, LS DYNA, NASTRAN etc.) [19].

The ANSYS program is a powerful, multi-purpose analysis tool that can be used in a wide variety of engineering disciplines. Using ANSYS software can avoid expensive and time-consuming development loops, so the design period is shortened. The stress analysis of a truck chassis using ANSYS software carried out by many researchers to determine and reduce the magnitude of stress of the chassis frame [19]. For example;

Dr.R.Rajappan and M.Vivekanandhan (2013) have investigated the dynamic and static load characteristics of truck chassis frame using FE model. They have identified the location of high stress area, analysed the vibration, natural frequency and mode shapes. The maximum stress occurred at the mounting brackets of the suspension system while the maximum translation occurred at the location where the symmetry and asymmetry load is acting [1].

M.Ravi Chandra (2012) and his cooperators have done a research on a structural analysis of heavy vehicle chassis frame of TATA2515 EX vehicle. The modeling and analysis was done with three different composite materials, namely carbon /epoxy, e-glass/epoxy, s-glass/epoxy, and with three different cross sections namely C, I and Box type cross sections subjected to the same pressure as that of steel chassis with the design constraints of stress and deflection. They have used software of PRO.E 5.0 for modeling and ANSYS 12.0 for analysis. From their study by employing polymeric composite heavy vehicle chassis frame for the same load carrying capacity. There is a reduction in weight of 73%-80%, natural frequency of polymeric composite vehicle chassis frame 32%-54% higher than steel chassis and 66%-78% stiffer than the steel. There steel shows that carbon/epoxy polymeric composite heavy vehicle chassis frame of I-section has superior strength and stiffness and lesser weight as compared to steel. So from this result it is observed that the polymeric composite heavy vehicle chassis frame is lighter and more economical than the conventional steel chassis with similar design specification [9].

Hemant patil et al. (2013) presented stress analysis of a ladder type low loader truck chassis structure consisting of C-beams design using CATIA V5R10 and ANSYS. Computed results were compared to analytical calculation, where it was found that the maximum deflection agrees well with theoretical approximation but varies on the magnitude aspect [12].

Patel Vijay Kumar and Patel (2012) investigated automotive chassis is an important part of an automobile. The chassis serves as a frame work for supporting the body and different parts of the automobile. Also, it should be rigid enough to withstand the shock, twist, vibration and other stresses. Along with strength, an important consideration in chassis design is to have adequate bending stiffness for better handling characteristics. So, strength and stiffness are two important criteria for the design of the chassis. This report is the work performed towards the static structural analysis of the truck chassis. Structural systems like the chassis can be easily analyzed using the finite element techniques. So a proper finite element model of the chassis is to be developed. The chassis is modeled in PRO-E. FEA is done on the modeled chassis using the ANSYS Workbench [50].

Tanaka et al (1981) studied about the stress analysis of a truck chassis with riveted joints was performed by using FEM. The commercial finite element package ANSYS version 5.3 was used

for the solution of the problem. Determination of the stresses of a truck chassis before manufacturing is important due to the design improvement. In order to reduce the magnitude of stress near the riveted joint of the chassis frame, side member thickness, connection plate thickness and connection plate length were varied. Numerical results showed that stresses on the side member can be reduced by increasing the side member thickness locally. If the thickness change is not possible, increasing the connection plate length could be a good alternative [51].

Ibrahim et al (1994) conducted a study on the effect of frame flexibility on the ride vibration of trucks. The aim of the study was to analyze the vehicle dynamic responses to external factors. The spectral analysis technique was used. From the author's point of view, the excessive levels of vibration in commercial vehicles were due to excitation from the road irregularities, which led to ride and comfort problems. In order to study the frame flexibility, the author had came out with the truck frame modeled using (FEM) and its modal properties have been calculated. The results there were found to be good agreement with the experimental analysis and the modeling technique proved to be a very powerful one [40].

Zehsaz et al (1996) explained that automotive chassis is an important part of an automobile. The chassis serves as a frame work for supporting the body and different parts of the automobile. Also, it should be rigid enough to withstand the shock, twist, vibration and other stresses. Along with strength, an important consideration in chassis design is to have adequate bending stiffness for better handling characteristics. Hence, strength and stiffness are two important criteria for the design of the chassis. Structural systems like the chassis can be easily analyzed using the finite element techniques [41].

Murali M.R. Krishna (1998) explained on the Chassis Cross- Member Design Using Shape Optimization. The problem with the original chassis was that the 17 fundamental frequencies was only marginally higher than the maximum operating frequency of the transmission and drive shaft, which were mounted on these cross members. The aim of this testing was to raise the cross-member frequency as high as possible (up to 190- 200 Hz) so that there was no resonance and resulting fatigue damaged. Firstly, a sizing optimization was attempted which indicated that the mass was a predominant factor. Four additional holes were added to the sides of the cross-member to reduce its mass. Another tests also have been conducted which the holes on the sides

had to be expanded, the bottom holes to be reduced in size, the thickness of the attachment bracket to be increased etc. Based on those testing, the fundamental frequency of the cross-member was raised by about 4 Hz, resulting in a better design [42].

Lampert and Jon (1998) studied the torsional stiffness of a vehicle's chassis significantly affects its handling characteristics and is therefore an important parameter to measure. In this work a new twist fixture apparatus designed to measure the torsional stiffness of a Winston Cup series race car chassis is described. The twist fixture is relatively light weight, adjustable, and easily transportable by one person for quick set-up on different chassis. Measured values of torsional stiffness are reported for several different chassis. The fixture applies vertical displacements (using linear, jack-screw actuators) at the front spring perches of the chassis while holding the rear perches fixed. Conventional race car scales located under the front assembly measure the resulting reaction forces due to the displacements. Dial indicators are placed at selected locations along the chassis to measure deflections. Using the dial indicator readings, the measured reaction forces and the chassis geometry, the torsional stiffness of the chassis can be calculated. Ball-joint connections between the twist fixture and chassis have been carefully designed to minimize unwanted rotational restraints [43].

Karaoglu and Kuralay (2002) explained mechanical behavior of a semi-heavy truck chassis. The ANSYS Code and the stress distributions in the chassis have been obtained and examined. For this purpose, the FE model of the truck vehicle was using 3D shell elements. To validate the FE model of the chassis; firstly, experimental modal analysis has been used. Secondly, the modes of chassis-vibration, natural frequencies and modal shapes have been obtained from the FE analysis and were compared with the results of experimental modal analysis. The dynamic forces due to the unevenness of the road have been calculated using a simple 3D dynamic model of the truck body. Then, stress analysis for the truck chassis have been carried out under loads. Different types of joints and their thickness in the chassis of truck vehicles are one of the important parameters which have significant effect on their strength. To study the effect of the connecting plates on the strength of the chassis, the strength of the welded and also the combined welded-riveted joints have been analyzed with three different plate thicknesses: 5, 8, 12 mm. The results show that the amount of stresses in chassis and connection plates decrease significantly with an

increase in the thickness of connection plates. Also, it has been shown that the use of combined welded-riveted joints reduces the stress level of the chassis. The results prove the precision of the FE modeling and also show that the numerical modeling is accurate Therefore, the stresses which have been obtained are reliable and can be used to design the chassis [33].

Romulo Rossi Pinto Filho (2003) explained the chassis by using steel with closed rectangular profile longitudinal rails and tubular section cross-member. The geometry of the chassis was measured directly in the reference vehicle real structure. Then, a modal analysis procedure was accomplished on the real chassis and a finite element model structure was formed in order to establish the real structure to the chassis structures. The natural frequencies were extracted. For the frame optimization, the author tried to use groups of numerical and programming techniques to search for the optimum value of mathematical functions. In other words, the purpose of the optimization is to facilitate in finding results that best fills out the needs. Based on the result, the analysis and experimental procedure applied had significantly improved the overall structural stiffness by 75% by maintaining the center of gravity and the total weight was increased by 6% [44].

Masahiro et al (2004) discussed dump truck in mining. Receive various dynamic loads during travel. The strength of main frames of dump truck has been predicted based on static stress analysis where dynamic loads obtained by measurement with actual machines are replaced with static loads. However, when a developed model employs a new structure, it is difficult to precisely determine the conditions that are critical for main frames on development stage. For the articulated dump truck that was developed recently, it was required to precisely predict, on design stage, the stresses that act on the frames during travel. Therefore, we introduced elastic characteristic of main frame into kinematical analysis models by using kinematical analysis software ADAMS and finite element method software NASTRAN and developed the method to calculate frame stress that occurs during travel, which was applied to the rear frame of the articulated dump truck. As a result, it was confirmed that frame stress during travel can be calculated correctly with this method [45].

Izzudin B. Zaman (2006) conducted a study on the application of dynamic correlation and model updating techniques. These techniques were used to develop a better refinement model of existing truck chassis with approximately 1 ton and also for verification of the FEA models of truck chassis. The dynamic characteristics of truck chassis such as natural frequency and mode shape were determined using finite element method [26].

Chiewanichakorn et al (2007) explained lighter fiber reinforced polymer (FRP) decks are gaining popularity among bridge owners as an alternative to replace old deteriorated heavy concrete bridge decks to increase the live load capacity of old steel superstructures requiring minimal repairs. This is more attractive in case of old truss bridges, which are relatively in good condition, but are not designed to current live loads and cannot be easily rehabilitated to improve their capacities. Replacing the heavy concrete decks on these bridges will extend their service life by reducing the dead load and thus increasing the live load capacity. When an FRP deck is used in such cases, a system level approach should be used to evaluate the bridge condition for all possible failure mechanisms [46].

Roslan Abd Rahman et al (2008) discussed stress analysis of heavy duty truck chassis. The stress analysis is important in fatigue study and life prediction of components to determine the critical point which has the highest stress. The analysis was done for a truck model by utilizing a commercial finite element packaged ABAQUS. The model has a length of 12.35 m and width of 2.45 m. The material of chassis is ASTM Low Alloy Steel A 710 C (Class 3) with 552 MPa of yield strength and 620 MPa of tensile strength. The result showed that the critical point of stress occurred at the opening of chassis which is in contact with the bolt. The stress magnitude of critical point is 386.9 MPa. This critical point is an initial to probable failure since fatigue failure started from the highest stress point [32].

Okurdi et al (2008) discussed about the one of the most important steps in development of a new truck chassis is the prediction of fatigue life span and durability loading of the chassis frame. The age of many truck chassis in Malaysia are of more than 20 years and there is always a question arising whether the chassis is still safe to use. Thus, fatigue study and life prediction on the chassis is necessary in order to verify the safety of this chassis during its operation. Stress analysis using Finite Element Method (FEM) can be used to locate the critical point which has

the highest stress. This critical point is one of the factors that may cause the fatigue failure. The magnitude of the stress can used to predict the life span of the truck chassis. In this study, the stress analysis is accomplished by the commercial finite element packaged ABAQUS [47].

Patel Vijay Kumar and Patel (2012) investigated that automotive chassis is an important part of an automobile. The chassis serves as a frame work for supporting the body and different parts of the automobile. Also, it should be rigid enough to withstand the shock, twist, vibration and other stresses. Along with strength, an important consideration in chassis design is to have adequate bending stiffness for better handling characteristics. So, strength and stiffness are two important criteria for the design of the chassis. This report is the work performed towards the static structural analysis of the truck chassis. Structural systems like the chassis can be easily analyzed using the finite element techniques. So a proper finite element model of the chassis is to be developed. The chassis is modeled in PRO-E. FEA is done on the modeled chassis using the ANSYS Workbench [50].

Haval Kamal Asker et al (2012) discussed the frame of the standard dump truck supports all types of complicated loads coming from the road and freight being loaded. So, the intensity and the strength of the frame play a big role in the truck's design. A frame of 6 wheels, standard dump truck has been studied and analyzed using ANSYS package software. The static intensity of the frame has been analyzed when exposed to pure bending and torsion stress, within two cases. First case is when the rear wheels zigzag gets over block (only one side of the chassis steps the block), and the second case is when both wheels gets over the block. Finite element model of a stress analysis of the vehicle chassis has been built using three dimension hyper elastic elements for the modeling. The results show important differences between the two case studies, especially in the torsion and deformations results obtained from the chassis model. Also, vibration modes have been analyzed during the loading conditions. The more damping ratio used, the more stabilizing of the stresses with respect to time [5].

2.2 The application of composite materials for freight truck trailer

Hauling freight by truck is a huge international industry. Retailers, manufacturers and the farm/food industry all depend on over-the-road truck shipments. According to the American

Trucking Association, 2.6 million big rigs logged more than 200 billion miles in the U.S. alone, in 2000. Yet, despite these volumes, haulers face a very competitive market, with expensive vehicles and trailers and the necessary costs of ongoing maintenance and repair. In many areas, total vehicle weight restrictions often mean that truckers must limit load size, which reduces the revenue for each trip [13].

Truck manufacturer Stevens Carrosserie N.V. (Lokeren, Belgium), which was established in the 1870s to make horse-drawn wagons, spun off its Composite trailer N.V. group, headed by Jan Verhaeghe, in 1993 to develop an all-composite big-rig trailer for more cost-effective freight hauling. The group reasoned that a lighter trailer would reduce overall vehicle weight, which would mean more cargo could be hauled per trip, reducing the cost of the transported goods. Their concept was a trailer made entirely of pultruded profiles, filament-wound structures and resin transfer molded (RTM) elements. In the U.S., Martin Marietta Composites Inc. (MMC, Raleigh, N.C., U.S.A.) had a similar idea, approached Composite trailer and eventually worked out a licensing agreement, enabling MMC to manufacture trailers for the North American market while Composite trailer supplies the European market [16].

"The only non-composite parts on the trailer are the axles, the wheels and tires, wiring harness, the drive system for the moving floor, the king pin [the connection point between the trailer and the tractor] and the landing gear legs," says MMC's vice president Grant Godwin. "We've taken the European technology, reanalyzed and optimized the design for a U.S. trailer, which is 6 ft longer and has two axles instead of three." Godwin says the composite trailer is targeted to haulers of heavy bulk goods, like agricultural products (e.g., potatoes), wood chips, or waste, which tend to reach mandated maximum loads before the trailer is full [16].

The design process started in Belgium with analysis of a conventional steel trailer and typical cargo loads, usually between 20 and 22 metric tonnes (44,000 to 48,400 lb). NE/Nastran finite element analysis (FEA) software from Noran Engineering (Los Alamitos, Calif., U.S.A.) was used to help define the composite materials and fiber architecture to meet the anticipated loads [13].

Composite trailer wanted to fabricate the components cost-effectively, so pultrusion and filament winding were the processes selected for the chassis elements. Side panels were designed for vacuum-assisted resin transfer molding (VARTM). The modeling confirmed that the trailer's chassis and superstructure could be constructed with enough structural rigidity, strength and toughness using fiberglass and vinyl ester resin, without the need for carbon fiber reinforcement [13].

At first glance, the composite design resembles a conventional steel or aluminum trailer. Two main pultruded chassis frame rails run the length of the trailer (13.6m/44 ft in the European version, 15.4m/50 ft in the U.S. version), connected at intervals by filament-wound tubular cross members with thickened, trumpet-shaped ends. The thickened ends provide a greater surface area for bonding the cross members to the rails with epoxy or isopolyester adhesive and also give additional strength for the stainless steel or galvanized bolts that are inserted to supplement the bonded joints. The filament-wound design, with both helical and hoop fibers, allows the transfer of bending and torsional loads encountered when, for example, the driver goes over a curb. The number of cross members needed is dependent on the type of trailer produced and the anticipated loads. Cross members are filament wound with E-glass roving and Dow Chemical Co. (Midland, Mich., U.S.A.) [13].

The sidewalls and doors that comprise the trailer superstructure are sandwich panels approximately 2.7m/9 ft high and fabricated to length in one piece in a VARTM process. Panels consist of inner and outer skins made up of two layers of 450 g/m2 (1.5 oz/ft2) continuous strand mat and a single ply of 580 g/m2 (17 oz/ft2) woven E-glass over a 25-mm/1-inch-thick core of structural closed-cell foam. Total finished panel thickness is 30 mm/1.18 inches, with a total fiber volume of about 38 percent by weight. The sandwich layup was designed to withstand a load of 1,200 N/270 lb force at the center of each panel, applied by a 2-inch square steel pad, without deflecting more than 4 mm/0.16 inch. The stiff yet thin panels result in increased interior trailer volume about 5 percent greater than a conventional trailer. While VARTM has been successful, the panels will probably be pultruded in the future. Composite trailer and MMC are developing pultrusion processes for producing the thin panels to length. Godwin says pultrusion

will enable the use of integral color with pigmented resins, eliminating the need to subsequently paint the panels [13].

Empty weight of an assembled trailer is less than 5,900 kg/13,000 lb, which is 25 percent less than a comparable steel or aluminum version, translating to an overall 15 percent load capacity increase. The trailer's "live floor" is made from standard pultruded plank profiles, which are connected to a drive system that moves them back and forth to discharge the cargo. The one-piece rear cargo doors are hinged with carbon fiber hinge pins [13].

The original composite trailer and its components fabricated in Belgium have undergone a lengthy series of tests, starting with laboratory testing in a load simulator. The simulator placed actuator-driven load cells at various locations on the trailer to approximate the loads that the vehicle will see in use. Strain gauges were strategically placed throughout the trailer structure and monitored for deformation or damage due to the loading scenarios. FEA was then executed with the Nastran program [13].

"We simulated cargo loads during normal driving and turns, under acceleration, and under panic stops where the load could shift," explains Godwin. The results indicated that the trailer would handle loads with less stress then conventional trailers [13].

Accelerated static and fatigue durability tests also were performed, using an industry-standard 15-year lifetime, equivalent to more than 2 million miles on the road. The chassis was clamped in a fixture and a 12,000 kg/26,400 lb, 1 Hz downward load was applied to one frame rail while a second upward load of 3,000 kg/6,600 lb at 3 Hz was applied simultaneously to the other frame rail, at the forward axle location. The frame, including the bonded cross members, successfully survived 6 million load cycles with no observed loss of strength or creep at bond lines and fastener locations [14].

These tests were followed by actual field tests of the prototype trailer on a NATO military vehicle test course in Belgium. The trailer has a lot of flex due to its composite design, which works to its advantage, says Godwin. In a slalom avoidance test, for example, a heavily loaded Composite trailer actually deforms slightly under extreme torsion, as compared to a stiffer

conventional steel trailer, which experiences much more jarring and load shifting. The damping properties of the composite materials give the trailer superior "road grip," which means it absorbs the vibrations from bumps and obstacles and distributes them through the structure much more efficiently than metallic trailers, holding the road better with less bounce, a real safety benefit. Better damping means less wear and tear on tires and suspension components, and causes less damage to sensitive loads [13].

Every day for the past two years, the European prototype has completed a 435-mile round trip between Zwolle, The Netherlands and Deinze in Belgium, hauling a test load of 25 metric tonnes (55,000 lb). So far, more than 200,000 miles have been logged, with no failures of any components, proving the robustness of the composite design and piquing the interest of the trucking industry. In addition to the obvious weight difference and increased payload capacity, other benefits include corrosion resistance, non-conductive properties, the ability to repair damage and an important selling point for European customer's recyclability [13].

2.3 The role of composite material in weight reduction

According to the Transportation Energy Data Book (2013), there were 10,770,000 heavy trucks registered in the U.S. in 2010. These trucks account for about 22 percent of U.S. transportation petroleum use and nearly 15.2 percent of the U.S. petroleum use in 2010 and 2011 (19 million barrel per day). Reports show that trucking is energy-intensive and accounted for 69 percent of freight energy use [16].

The domestic petroleum production is just 7.51 million barrel per day in 2010. The remaining petroleum is obtained from foreign sources. Therefore, improving fuel efficiency for heavy-duty trucks, particularly for Class 8 trucks, is necessary to achieve energy sustainability and support future economic development. Therefore, new technologies are needed in order to increase energy security in the transportation sector at a critical time for global petroleum supply, demand, and pricing. However, the oil dependence continues to increase unabated to the present and the oil price run up of July 2008 ($147 per barrel of crude) illustrated the rapidity with which these discontinuities can occur [30].

Additionally, there is an environmental responsibility concomitant with oil dependence to reduce greenhouse gas emissions, primarily carbon emissions from ground transport vehicles. Transportation sources emitted approximately 40 percent of all GHG emissions in the United States. Medium- and heavy-duty vehicles (above 8,500 lb gross vehicle weight rating) represented about 22 percent of the transportation emissions, up from 15 percent in 1990. In 2008, trucks accounted for 69 percent of freight energy use, consuming 2.35 million barrels of oil per day in 2008 and generating 363 million metric tons of carbon dioxide. Consequently, trucks are an important place to look for energy savings and climate change mitigation in the transportation sector. Neither petroleum reduction goals nor carbon emissions reduction goals can be achieved without new and more efficient vehicle technologies. While achieving petroleum and GHG reduction goals will be extremely challenging, such challenges represent a unique opportunity for the U.S. to establish a sustainable energy infrastructure.

The increasing price of oil across much of the world has brought the issue of fuel efficiency to the front of many car buyers' minds. The automotive industry has responded with a variety of new approaches to minimize the weight and improve performance of their vehicles. Light weighting of vehicles is one of the most realistic ways in meeting these requirements due to the inherent relationship between mass and fuel consumption. In addition, light weighting may benefit other advanced fuel-saving but load constrained technologies, such as battery-powered vehicles [31].

Studies suggest that a 10-percent decrease in vehicle weight results in a 6.6% increase in fuel efficiency. In fact, according to an many report, there is no single cost-effective technology that can accomplish the targeted fuel efficiency without significant weight reduction [13].

One of the easiest ways to reduce mass is to look at alternatives to traditional steel. Materials such as Composites, Aluminum, Magnesium, Zinc and High Strength Steels each offer unique value propositions for OEMs to consider when considering various options for vehicle mass reduction. However, the ability to introduce new lightweight materials into vehicles is not a trivial matter. Many see a new concept, or limited production, vehicle introduced to the market with lightweight materials and feel that adoption by mass produced vehicles is a simple matter of “remove and replace.” However, this is not the case, factors such as existing infrastructure, material cost, and high volume capacity become of great importance for mass production vehicles. In addition, many of the low production vehicles incorporate these lightweight materials as a method for gaining experience on

their performance. Without significant data to support durability, the risk-averse automotive culture will not adopt new materials. Therefore it often takes many years to implement lightweight technology in mainstream production vehicles [16].

Bandivadekar et al. (*2008*) showed that reduction in vehicle weight and size could significantly reduce fuel consumption and greenhouse gas emissions. They estimated that direct weight reductions through the substitution of lighter materials as well as basic vehicle design changes which, for example, maximize the interior volume for a given vehicle length and width enable secondary weight reductions as other vehicle components are appropriately downsized. A shift in vehicle size distribution away from larger vehicles also reduces average weight and initially can be accomplished by changes in production volumes. They estimated a maximum potential vehicle weight reduction of 35% at plausible cost, which allow for the additional weight of future safety requirements and convenience features. Vehicle weight reductions of this magnitude could alone result in some 12–20% reduction in vehicle fuel consumption [21].

Reduction in vehicle body mass was considered to offer the greatest opportunity for achieving near-term, cost-effective reductions in fuel consumption. This led the US Department of Energy and U.S. Council for Automotive Research (USCAR) to continue their research and development efforts for ways to reduce body mass by substituting new materials such as high-strength steel, advanced high-strength steel, aluminum, magnesium, and composites for current materials. The materials industries also conducted research to advance new materials (for example, through the Auto-Steel Partnership, the Aluminum Association, and the American Chemistry Council). Increased costs for lighter and stronger materials result from higher material costs and higher costs of component fabrication and joining. It is estimated that every 10 percent reduction in vehicle weight will result an approximate 7 percent saving in fuel. Estimates of the body-mass reduction that can be achieved in the near term vary from 10% (with mostly conventional and high-strength steels) to 50% (with a mostly aluminum structure). Adopting materials such as carbon fibers and polymer matrix composites could produce weight reductions of 25-70%, while improved manufacturing and use of high strength steel can reduce vehicle weight by 15-25% [16].

2.4 Finite element method

Finite Element Analysis (FEA), also known as the finite element method (FEM) is based on the concept that a structure can be simulated by the mechanical behavior of a spring, in which the

applied force is proportional to the displacement of the spring and the relationship F = ku is satisfied. In FEA, structures are modeled by a CAD program and represented by nodes and elements. The mechanical behavior of each of these elements is similar to a mechanical spring, obeying the equation, F = ku. Generally, a structure is divided into several hundred elements, generating a very large number of equations which can only be solved with the help of a computer. The term 'finite element' stems from the procedure in which a structure is divided into small but finite size elements (as opposed to an infinite size, generally used in mathematical integration). The endpoints or corner points of the element are called nodes. Each element possesses its own geometric and elastic properties. Spring, Truss, and Beams elements, called line elements, are usually divided into small sections with nodes at each end. The cross-section shape doesn't affect the behavior of a line element; only the cross-sectional constants are relevant and used in calculations. Thus, a square or a circular cross-section of a truss member will yield exactly the same results as long as the cross sectional area is the same. Plane and solid elements require more than two nodes and can have over 8 nodes for a 3 dimensional element. A line element has an exact theoretical solution, e.g., truss and beam elements are governed by their respective theories of deflection and the equations of deflection can be found in an engineering text or handbook. However, engineering structures that have stress concentration points e.g., structures with holes and other discontinuities do not have a theoretical solution, and the exact stress distribution can only be found by an experimental method. However, the finite element method can provide a more acceptable efficient solution more efficiently. Problems of this type call for use of elements other than the line elements mentioned earlier, and the real power of the finite element is manifested. In order to develop an understanding of the FEA procedure, we will first deal with the spring element. In this chapter explained how spring structures are used as building blocks for developing an understanding of the finite element analysis procedure. Both spring and truss elements give an easier modeling overview of the finite element analysis procedure; this is because of that each spring and truss element, regardless of length, is an ideally sized element and does not need any further division [3]

Finite element analysis, also called the finite element method, is a method for numerical solution of field problems. A field problem requires the determination of spatial distribution of one or more dependent variables [3]

Mathematically, a field problem is described by differential equations or by an integral expression. Either description may be used to formulate finite elements which can be visualized as small pieces of a structure. The elements are connected at points called 'Nodes'. The assemblage of elements is called finite element model or structure. The particular arrangement of elements is called a mesh. Numerically, a finite element mesh is represented by a system of algebraic equations to be solved for unknowns at nodes. Finite element modeling is the process of preparing a FE model. It decides about the significant features of the actual problem that can be incorporated in the model. Simulation is the prediction of the intended output results of the computational FE model [3]

2.4.1 Basic Concept of FEM

The finite element method (FEM) is a computational technique used to obtain approximate solutions of boundary value problems in engineering. Simply stated, a boundary value problem is a mathematical problem in which one or more dependent variables must satisfy a differential equation everywhere within a known domain of independent variables and specific conditions on the boundary of the domain [3]

An unsophisticated description of the FE method is that it involves cutting a structure into several elements (pieces of structure) and then reconnecting elements at nodes as if nodes were pins or drops of glue that hold elements together. This process results in a set of simultaneous algebraic equations. In stress analysis these equations are considered to be equilibrium equations of the nodes. There may be several hundred or several thousand such equations, which mean implementation of computer is mandatory [3]

2.4.2 FEM Steps

There are certain common steps in formulating a finite element analysis of a physical problem, whether structural, fluid flow, heat transfer and some others problem. These steps are usually embodied in commercial finite element software packages. There are three main steps, namely: preprocessing, solution and post processing. The preprocessing (model definition) step is critical. A perfectly computed finite element solution is of absolutely no value if it corresponds to the

wrong problem. This step includes: define the geometric domain of the problem, the element type(s) to be used, the material properties of the elements, the geometric properties of the elements (length, area, and the like), the element connectivity (mesh the model), the physical constraints (boundary conditions) and the loadings [2, 3].

The next step is solution, in this step the governing algebraic equations in matrix form and computes the unknown values of the primary field variable(s) are assembled. The computed results are then used by back substitution to determine additional, derived variables, such as reaction forces, element stresses and heat flow. Actually the features in this step such as matrix manipulation, numerical integration and equation solving are carried out automatically through commercial software [2, 3].

The final step is post processing during which the analysis and evaluation of the result is done. Examples of operations that can be accomplished include sort element stresses in order of magnitude, check equilibrium, calculate factors of safety, plot deformed structural shape, animate dynamic model behavior and produce color-coded temperature plots. The large software has a preprocessor and postprocessor to accompany the analysis portion and the both processor can communicate with the other large programs. Specific procedures of pre and post are depends upon the program [2, 3].

CHAPTER 3

MATERIALS AND METHODOLOGIES

The methodology intended for this thesis work is mainly modeling the frame with CATIA V5 and stress analysis is done with finite element analysis method commercial software called ANSYS. The specifications of the chassis frame and other data is taken from FAW dump truck assembled in **Bishoftu Automotive Industry under METEC**.

The truck chassis used for the study is with a gross weight of 32,000 kg. It consists of 2 C-channel side rails and have 5 cross members along the 2 side rails. There are some additional members like flat and gusset brackets or connection plates located at the joint between side rails and cross members to strengthen the joints. Towards the middle is a two C-channel cross members together forms I section to provide a better support and rigidity. The final 2 cross members at the front and rear side are single C-channel cross member. The present material of the truck chassis is mild steel. The data of FAW dump truck collected from bishoftu automotive industry is listed as below.

3.1 Over all specification

The vehicle selected for the purpose of this study is FAW dump truck assembled in Bishoftu automotive industry. Bishoftu automotive industry is one of a huge government owned industry under METEC that participate in assembly of different car models by importing parts from

China. This industry plays a vital role for the developments of the country in reducing the cost of importing of the assembled vehicles. They planned to manufacture some of the parts in the country by its own man power rather than importing. If it continues like this through time after a few years Ethiopia will have indigenous vehicles or its own brand like other countries passed through this process such as South Korea, India, China, Malaysia and others. These countries started by assembling then they have made indigenous vehicles of different models. So Bishoftu automotive industry has to be appreciated. All the specifications of the vehicle are obtained from Bishoftu automotive industry. Some of the specifications are taken from the manual such as over all dimensions, wheel base, tracks, engine model, and weight specifications and the rests are by measurement using measuring instruments such as caliper and meter. Some of the measured components are the thickness, height, width and length of each cross members, reinforcing members and connection plates and the distance between cross members on the side rails.

3.1.1 Specification of the vehicle

Table 3.1.1 The specification of the vehicle

Over all dimension	Length (mm)	7393/8393
	Width (mm)	2500
	Height (mm) (empty load)	2890/3280
Wheel base (mm)	3825+1350	
Track(mm)	Front	2020
	Rear	1840

Table 3.1.2 Engine specification

Engine model	WP10.340E32
Displacement(L)	9.726
Rated power kW/(r/min)	250/2200
Max torque Nm/(r/min)	1350/1200-1600

Engine speed (r/min)	800 to 5000

3.1.2 Weight specification

Max front axle load (kg) = 6500

Max rear axle load (kg) = 25500

Curb weight of vehicle (kg) = 12080

GVM (kg) = 32000

3.1.3 Chassis frame specification

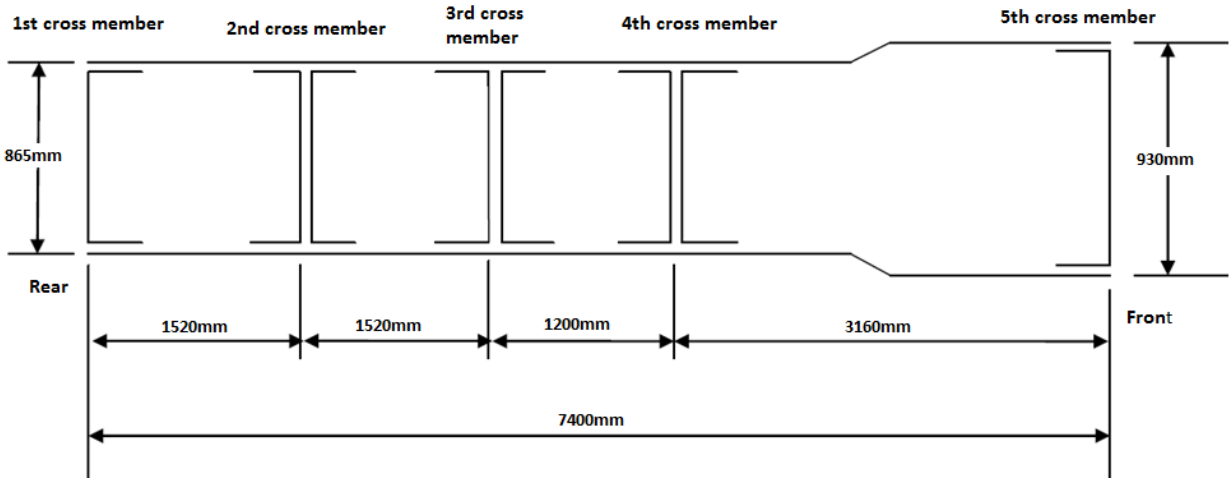


Fig.3.1.1 The dimension of FAW chassis frame

Section of main longitudinal beam (mm) = 320 x 90 x (8+8)

Frame length (mm) = 7400

Front frame width (mm) = 930

Rear frame width (mm) = 865

Front over hang (mm) = 1375

Rear over hang (mm) = 1575

3.2. CAD Model of Chassis

The chassis frame is modeled with CATIA V5 depending on the basic dimensions obtained from the manual for both steel and composite (carbon/epoxy and E glass/epoxy).

3.2.1. Solid Modeling

A solid model of an object is a more complete representation than its surface (wireframe) model. It provides more about the geometrical information which helps to represent the solid unambiguously. There are different software that is used for generating these solid models like Solid works, Pro.E and Catia V5. In this work the frame is modeled by using Catia V5 R19. All the parts that are required for constructing the frame are modeled in part module by using different commands like extrude, rotate, loft, fillet, extrude cut etc..,

The first, second up to fifth is starting from the rear towards the front. Rear end cross member is the first and front end cross member is the fifth. The following pats are modeled in Catia V5 R19 by taking all dimensions from FAW dump truck assembled in Bishoftu Automotive industry.

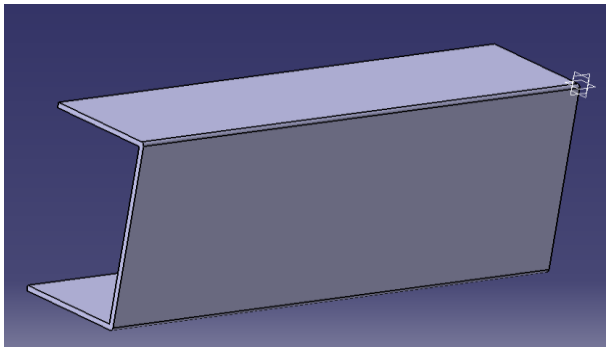


Fig 3.2.1 First cross member

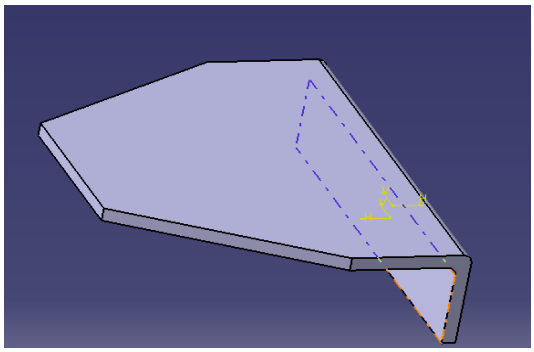


Fig 3.2.2 First connection plate

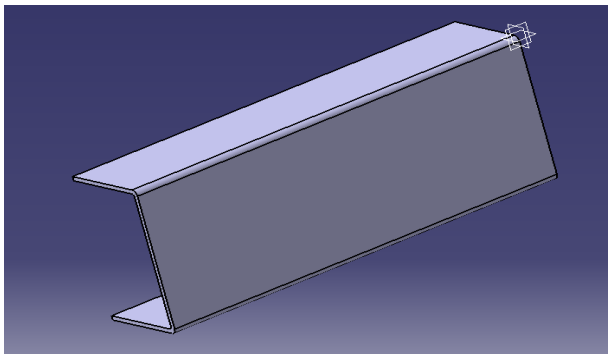


Fig 3.2.3 Second cross member

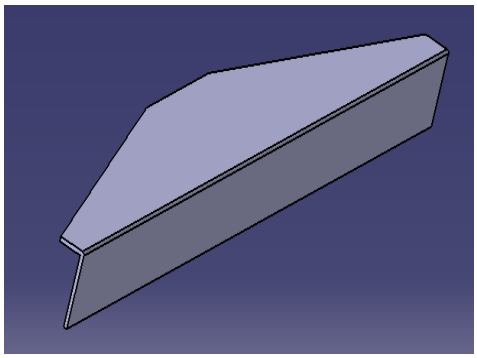


Fig 3.2.4 Second lower connection plate

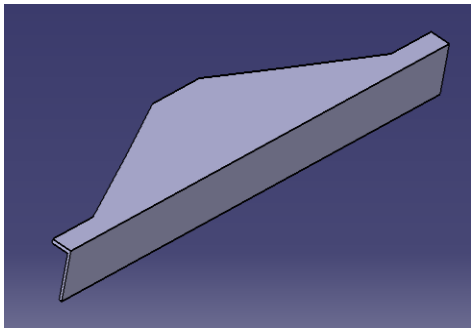


Fig 3.2.5 Second upper connection plate

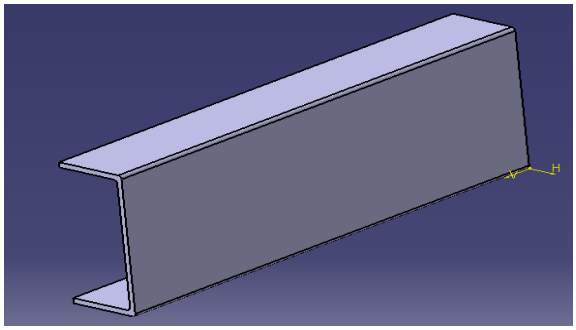


Fig 3.2.6 Secord cross member

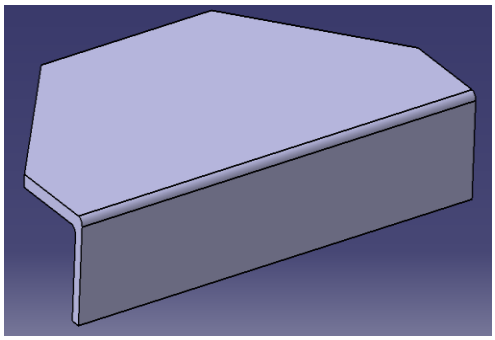


Fig 3.2.7 Third connection plate for upper and lower

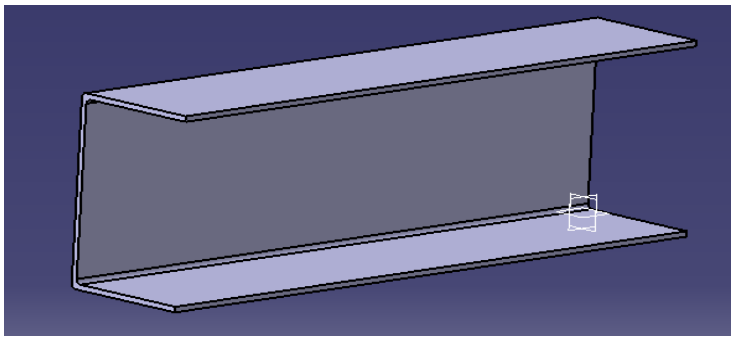
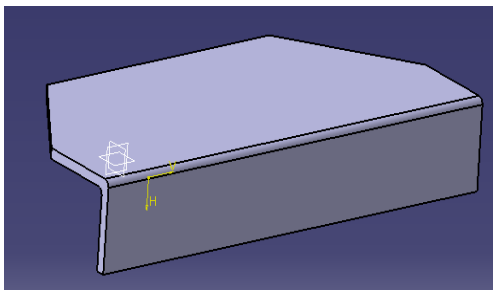


Fig 3.2.8 Forth cross member



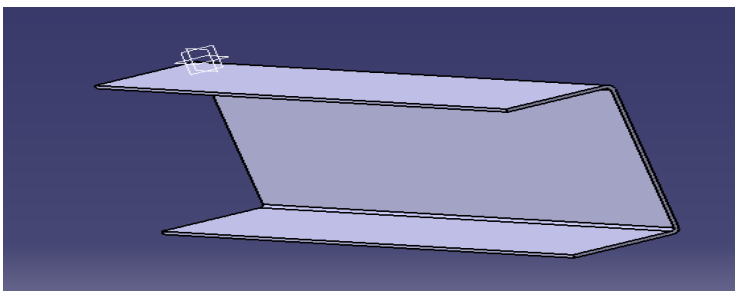


Fig 3.2.9 Forth connection plate

Fig 3.2.10 Fifth cross members

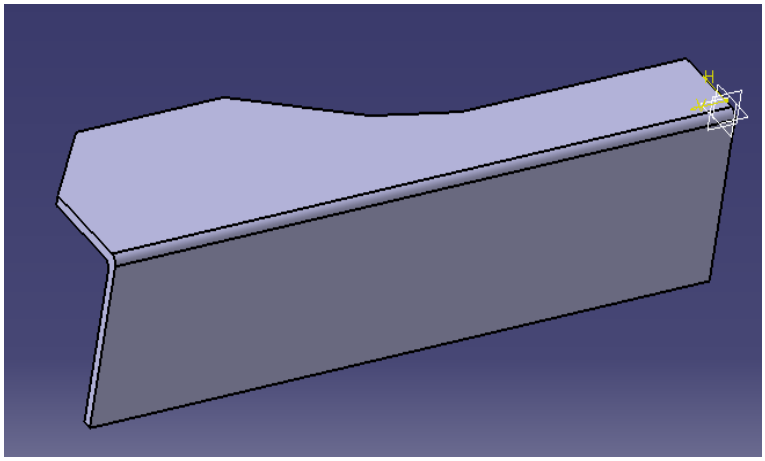


Fig 3.3.11 Fifth connection plate

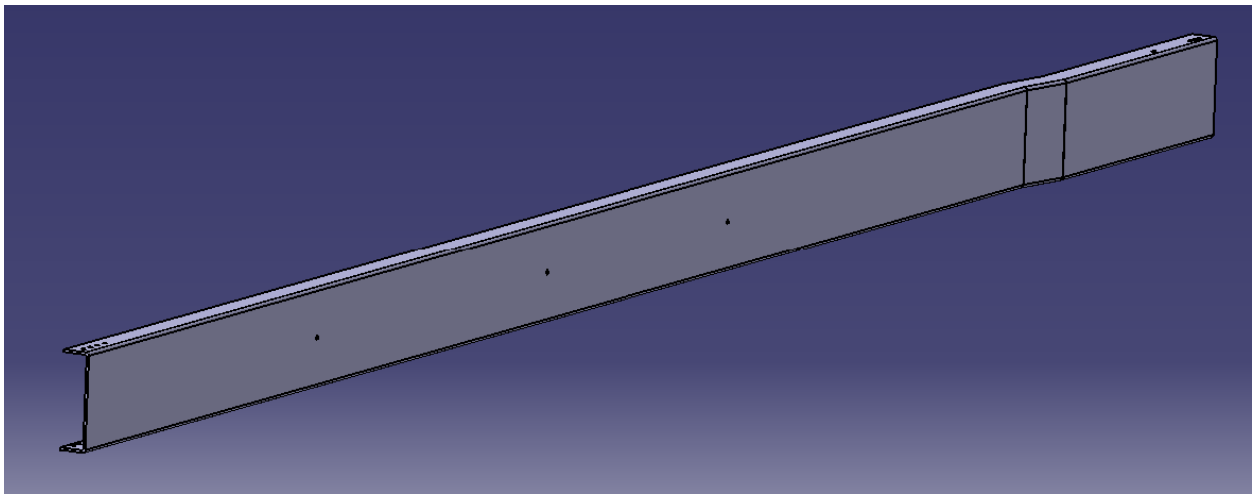


Fig 3.2.12 Main side rail

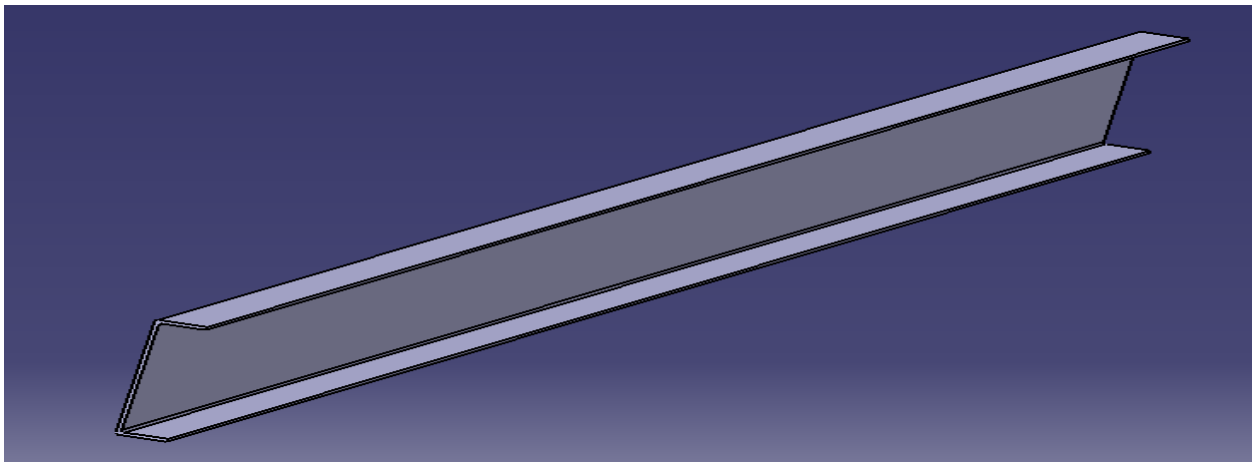


Fig 3.2.13 Supporting or reinforcing member for side rail

3.2.2. Assembly

The components that are generated in part module are imported to assembly design module and by using 'insert components' command and all these components are mated together to form the required assembly. The different views of assembly and the drawing generated in Catia V5 are as shown in the figure 3.2.14.

A 3D model of main frame structure is modeled using catia v5 software. It consists of two side rails and five cross members. Side rails have a supporting member used as a reinforcing member of C in C channel section. Both sides of longitudinal rails at the front and rear are connected to each other similarly with a single cross member. The rest of three cross members double, two C channels together makes I section and connected to the side rails using connection plates. All cross members are connected to the side rails through connection plates, which are necessary to increase the load carrying capacity and use as a reinforcements. Boundary conditions are applied on the CAD model.

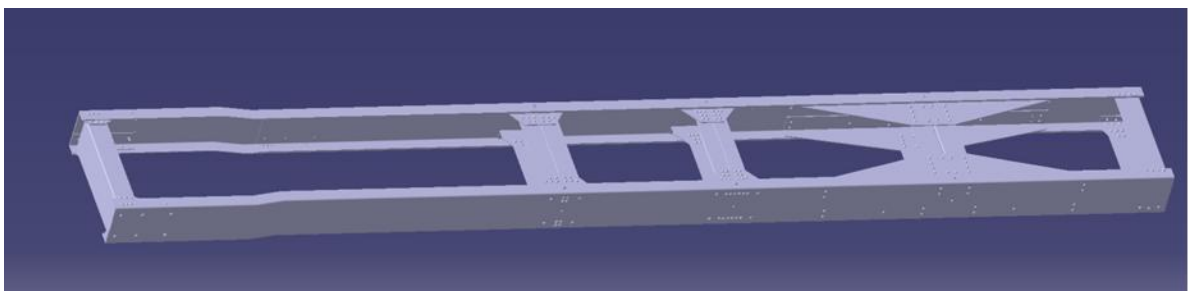


Fig 3.2.14 The assembled Catia model of chassis frame

In the chassis manufacturing technology, especially steel and its alloys are used for the material of the frame geometry. For the frame models a wide variety of materials, composite materials and different kind of alloys can be used. In the present study mild steel is compared to composite materials (carbon/epoxy and e glass/epoxy) with the same geometry and subjected to the same pressure for static and dynamic analysis.

3.3 Importing the CAD model to ANSYS workbench

The CAD model of the chassis frame is modeled and assembled in CATIA V5. This CATPART has to be changed to IGES format and it is imported to the ANSYS work bench for the analysis. Then analyzing the stress and natural frequencies developed on the chassis frame for both steel and composite (carbon/epoxy and E. glass/epoxy). And comparing the result obtained from analysis such as von misses stress, deflection, frequency, mode shape and weight of steel and composite (carbon/epoxy and E. glass/epoxy).

3.4 Material properties

The existing material of chassis is ASTM Low Alloy or mild Steel A 710 C (Class 3) with 550 MPa of yield strength and 620 MPa of tensile strength [20]. The other properties of chassis material are tabulated in table 3.4.1. The aim of the thesis is to substitute this steel with polymeric composite materials of E glass epoxy and carbon epoxy. Even if the modulus of elasticity of steel is higher, their specific modulus, specific strength and density are lower than composite materials. The material properties of carbon epoxy and e glass epoxy are tabulated in table shown below

Table 3.4.1 Material properties of the Truck Chassis [8, 20]

Material properties			
materials	Mild steel	Carbon/Epoxy	E Glass/Epoxy
Young Modulus (E_{11})	207 GPa	134 GPa	50 GPa
Poisson Ratio (v_{11})	0.3	0.3	0.3
Density	7800 kg/m^3	1600 kg/m^3	2600 kg/m^3
Yield stress (σ_{11})	550MPa	1500MPa	1062MPa

Ultimate stress (σ_{11})	620MPa	1500MPa	1062MPa

After creating the CAD model of the chassis frame, the next step is stress analysis by importing to the ANSYS workbench. Then static and dynamic response of the chassis frame is compared between mild steel and composite materials in terms of stress, deflection, natural frequency, mode shapes and deformation as well as weight.

3.5 Methodologies

The general methodologies of the thesis look the following flow chart

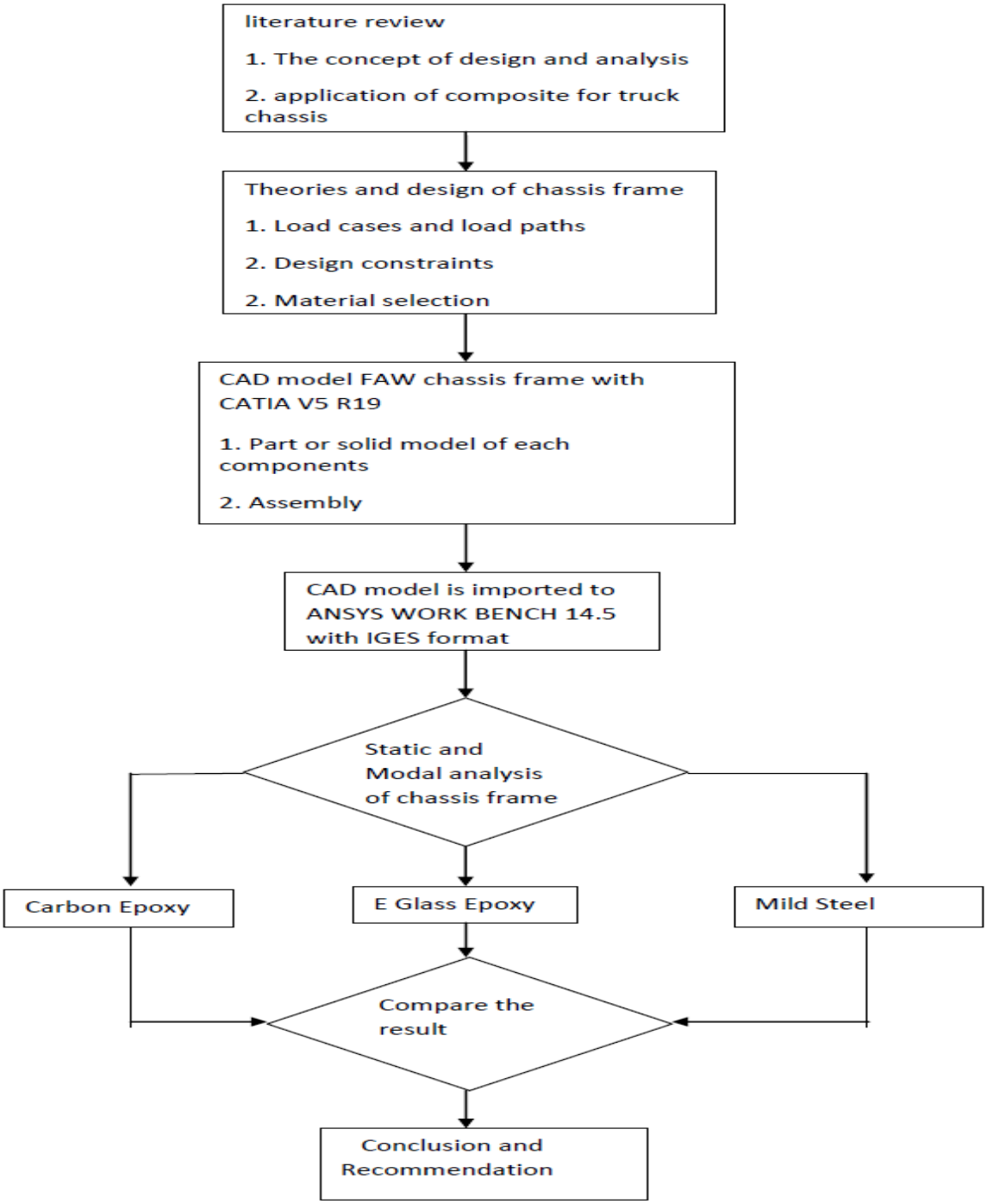


Figure 3.5.1 work flow diagram

CHAPTER 4

CHASSIS FRAME AND ITS LAYOUTS

4.1. Truck chassis for higher ton dump truck

Dump truck chassis is mainly used for coal application, and transportations of construction materials, such as stone, cements, and others. The chassis of dump truck carries the Vehicle body consists of two parts; chassis and bodywork or super structure. The conventional chassis, which is made of structured steel members, includes cross-members located at critical stress points along the side members with a channel type cross section rail structure [5].

In Ethiopia the evolutions of structural analysis in automotive industries are still far behind when compared to other countries research and development technology is not fully utilized in our country. Generally the major challenge to truck manufactures is to improve and optimize their vehicle designs to meet the market demand whole at the same time improving the vehicles durability and performance. Since the truck chassis is a major component in the vehicle system, it is often identified for refining and improving better handling and comfort.

The frame of the truck chassis is a backbone of the vehicle and integrates the main truck component systems such as the axles, suspension, power train, cab and trailer. The typical chassis is a ladder structure consisting of two C channel rails connected by cross-members. Almost all the chassis development varies in design, weight, complexity and cost. However, the Effects of changes on the frame and cross-members are not so well understood in terms of vehicle response during riding especially on bumpy and off road conditions.

After years of steady, predictable model changes, the automobile industry is in the midst of the most intense product changeover. To accomplish the need to design a moderate car, the structural engineer needs to use imaginative concepts. The demands on the automobile designer increased and changed rapidly, first to meet new safety requirements and later to reduce weight to satisfy fuel economy requirements. Hence, the finite element method and a computer dependent numerical technique have opened up a new approach to vehicle design [51].

Every vehicle has a body, which is opposed to carry both the loads along with its own weight. The body of the vehicle consists of two parts; chassis and bodywork or superstructure. The conventional chassis frame, which is made of pressed steel members, can be considered structurally as grillages. The chassis frame includes cross-members located at critical stress

points along the side members. To provide a rigid and box-like structure are secured the two main rails in a parallel position. The cross-members are usually attached to the side members by connection plates. The joint is riveted or bolted in trucks and is welded in trailers. When rivets are used, the holes in the chassis frame are drilled these holes are approximately larger than the diameter of the rivet. The rivets are then heated with an incandescent red and driven home by hydraulic or air pressure. The hot rivets conform to the shape of the hole and tighten cooling. An advantage of this connection is that it increases the chassis flexibility. Therefore, it is important to prevent high stresses in the critical area of the chassis. The side and cross-members are usually open-sectioned, because they are cheap and can easily attach with rivets [47].

4.2 Important issues in chassis design

A major challenge of the truck manufactures is to improve design and optimize the same to meet the customer requirements whole simultaneously providing better chassis durability and performance. Since the truck chassis is a major component in the vehicle system. Therefore, it is often identified for refinement and improvement to enable better handling, comfortably and life of the chassis. The chassis of the vehicle is the backbone of a vehicle and integrates the main truck component systems such as the front and rear axles, suspension, power train, cabin and body. To provide a box type structure, the cross-members secure the two main rails in a parallel position. The cross members are usually attached to the side members by connecting of rolled plates, also upon as the torque tube. The joint is welded into the rail structure of the truck chassis. When welds are used, the main focus should be on edge preparation cleanup of the welding area and provision of back-up plates. An advantage of this connection is that it increases the chassis strength and rigidity while preventing also high stresses are prevented in critical areas [23].

In all research, studies focus on improve the existing condition for betterment of riding quality, comfortably and fatigue life. There are major areas need to be established in the study to come out with proper investigation on truck chassis especially research methodology on experimental and computational analysis. The ultimate result would be improvement of vehicle quality, reliability, flexibility, efficiency, low production cost, low vibration and high fatigue life [23].

4.3 Types of vehicle chassis

The different types of vehicle chassis include:

4.3.1. Ladder Chassis:

Ladder chassis is considered to be one of the oldest forms of automotive chassis or automobile chassis that is still used by trucks and most of the SUVs till today. As its name connotes, ladder chassis resembles a shape of a ladder having two longitudinal rails inter linked by several lateral and cross braces [7]

Figure 4.3.1 ladder type chassis frame [19].

4.3.2 Backbone Chassis:

Backbone chassis has a rectangular tube like backbone, usually made up of glass fibre that is used for joining front and rear axle together. This type of automotive chassis or automobile chassis is strong and powerful enough to provide support smaller sports car. Backbone chassis is easy to make and cost effective [7].

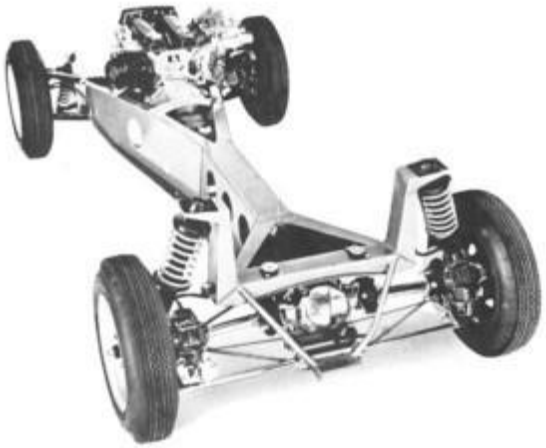


Figure 4.3.2 Backbone type chassis frame [7]

4.3.3 Monocoque Chassis

Monocoque Chassis is a one-piece structure that prescribes the overall shape of a vehicle. This type of automotive chassis is manufactured by welding floor pan and other pieces together. Since monocoque chassis is cost effective and suitable for robotised production, most of the vehicles today make use of steel plated monocoque chassis [7].

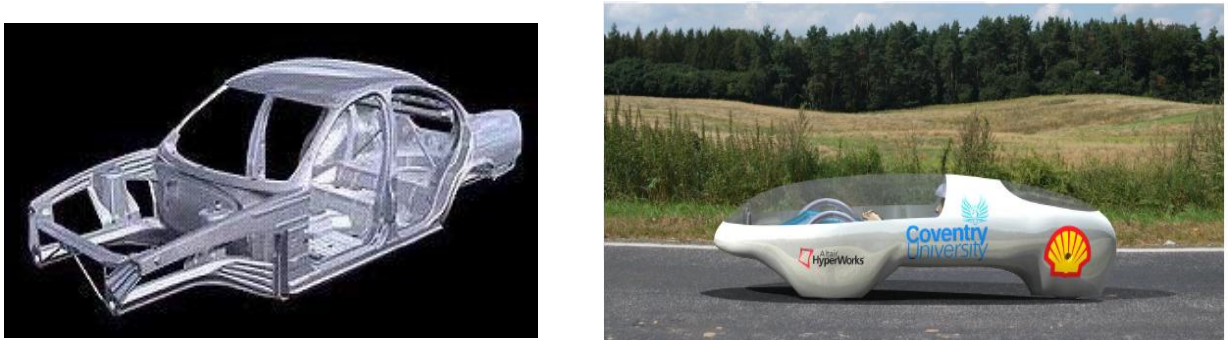


Figure 4.3.3 Monocoque type chassis frame

4.3.4 Space frame

Motor racing engineers have developed a 3-dimensional design which known as tubular space frame. Tubular space frame chassis employs dozens of circular-section tubes (some may use square- section tubes for easier connection to the body panels though circular section provides the maximum strength), position in different directions to provide mechanical strength against forces from anywhere. These tubes are welded together and form a complex structure. For higher strength required by high performance sports cars, tubular space frame chassis usually incorporate a strong structure under both doors. Tubular space frame

chassis also very strong in any direction compared with ladder chassis and monocoque chassis of the same weight [7].

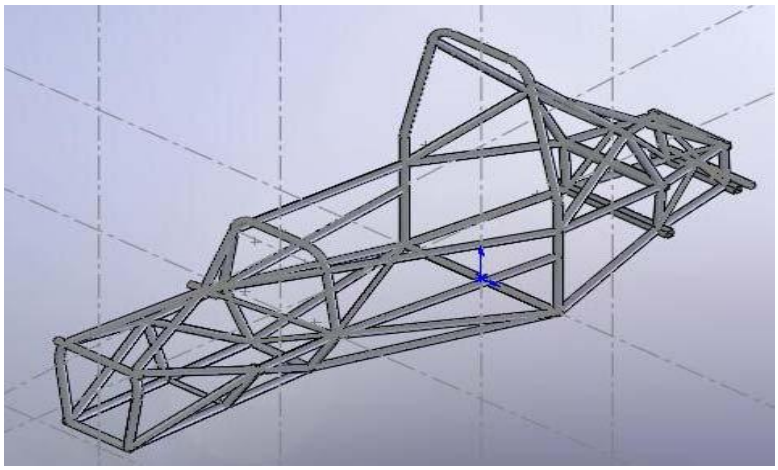


Figure 4.3.4 Spaceframe chassis Source: (Wan, 2000)

4.4 Vehicle load cases

The first step to designing a vehicle frame, or any structure, is to understand the different loads acting on the structure. Chassis structures are stressed by [7]

Internal loads

- Passengers and baggage's weight
- Reaction forces of the power train suspension

External loads

- Wheel-ground interface
- Aerodynamic loads

The main deformation modes for an automotive chassis when it is subjected to different load cases are given as:

1. Longitudinal Torsion case
2. Vertical Bending
3. Lateral Bending
4. Horizontal Lozenging

4.4.1. Longitudinal Torsion case

Torsion loads result from applied loads acting on one or two oppositely opposed corners of the car. The frame can be thought of as a torsion spring connecting the two ends where the suspension loads act. Torsional loading and the accompanying deformation of the frame and suspension parts can affect the handling and performance of the car. The resistance to torsional deformation is often quoted as stiffness in foot-pounds per degree [7].

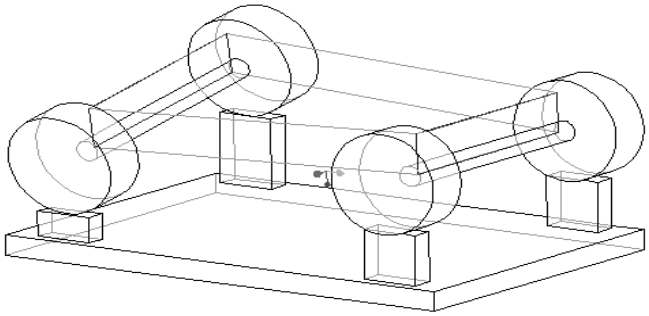


Figure 4.4.1 Longitudinal Torsion Deformation Mode

4.4.2. Vertical Bending case

The weight of the driver and components mounted to the frame, such as the engine and other parts, are carried in bending through the car frame. The reactions are taken up at the axles. Vertical accelerations can raise or lower the magnitude of these forces [7].

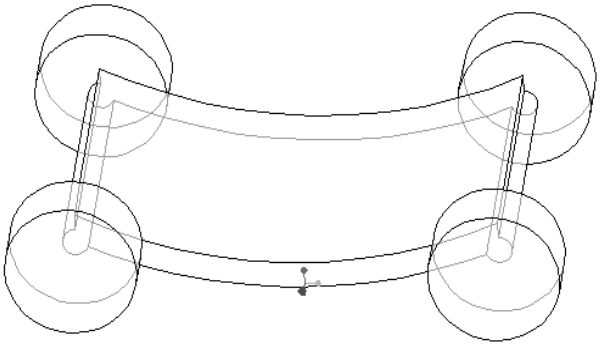


Figure 4.4.2 Vertical Bending Deformation Mode

4.4.3. Lateral Bending case

Lateral bending loads are induced in the frame for various reasons, such as road camber, side wind loads and centrifugal forces caused by cornering. The sideways forces will act along the length of the car and will be resisted at the tires. This causes a lateral load and resultant bending [7].

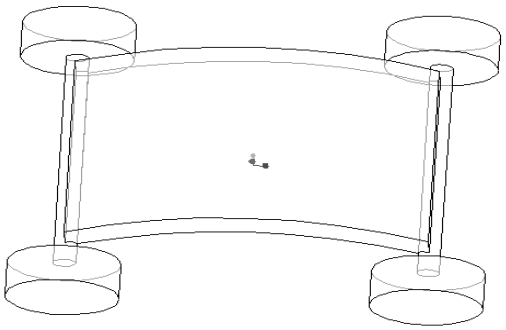


Figure 4.4.3 Lateral Bending Deformation Mode

4.4.4. Horizontal Lozenging case

Forward and backward forces applied at opposite wheels cause this deformation. 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 above [7].

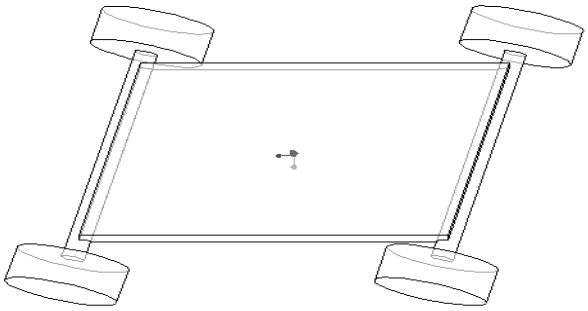


Figure 4.4.4 Horizontal Lozenging Deformation Mode

4.5. Load paths

Load paths form a complex subject that defines race car performance in terms of contact between the tires and ground, and the balancing of the car. As the engine produces force it is transferred to the tires through the power train of the car. Because of the high friction coefficient between the tires and the ground, the force from the engine results in the car moving forward. This force

however cannot be utilized to its full potential if the tires do not have a good contact path with the ground. The general rule is that you want maximum contact between the tires and the ground to increase the handling of the car and the force transferred from the engine to the ground [7].

There are three types of load transfer that will affect the overall performance of the car: longitudinal, transverse and diagonal. The load transfer axis are showed in figure 3.5.1.

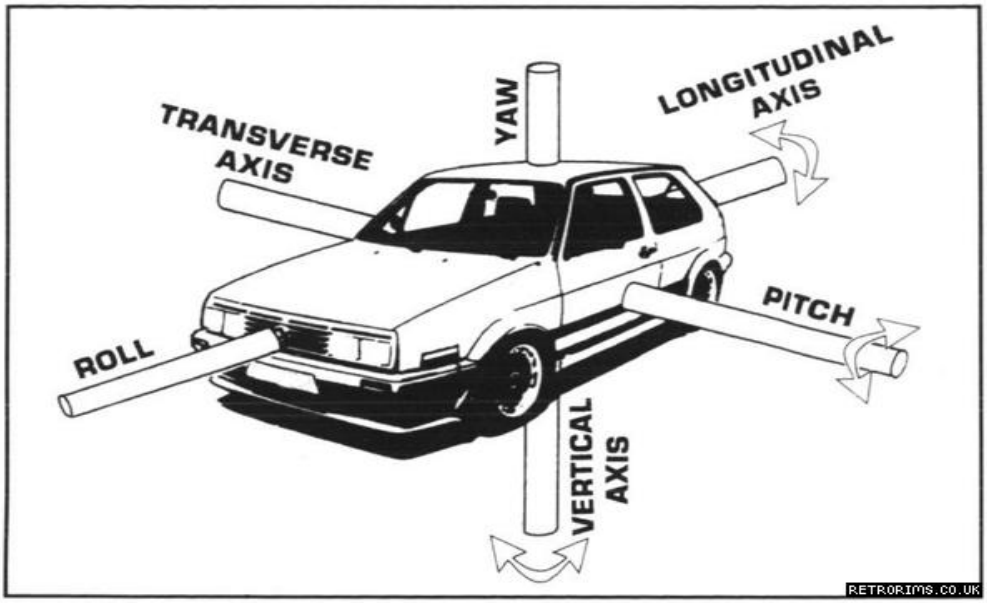


Figure 4.5.1 Relevant load transfer axes. [Retrorims 2013]

4.5.1 Longitudinal Load Transfer

This load transfer occurs in the longitudinal plane of the car as it accelerates (squats) or decelerates (dives). During acceleration the weight of the car shifts towards the rear, thus transferring more force to the rear tires and providing more grip. The opposite occurs when braking and the load transfers to the front of the car. The total weight is unchanged as it is merely transferred longitudinally. Many different design aspects, like the centre of gravity, weight and wheelbase, affect the longitudinal load path. Large amounts of longitudinal load transfer are detrimental because unloading any of the wheels can potentially make the road traction decreasing. Therefore, minimal longitudinal load transfer offers better grip for handling, making it possible to increase acceleration and decrease the time spent decelerating [7].

3.5.2 Lateral Load Transfer

Lateral force is caused by centrifugal forces when cornering and shifts weight from side to side of the car. Acting through the centre of gravity of the car, the mass wants to continue in its intended path around a corner, only resisted by the lateral forces produced by the tires [7].

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4.5.3 Diagonal Load Transfer

Diagonal load transfer is the most common and unfortunately also the most difficult transfer case to study. It occurs as the lateral and longitudinal forces are combined, for example when turning and accelerating or decelerating. The amount of load transferred in the diagonal direction depends on the torsional stiffness of the chassis as well as the spring rate, wheelbase and track width [7].

4.6 Design terminologies

4.6.1 Strength

The strength requirement implies that no part of the shape will lose its function when it is subjected to road loads. Loss of function may be caused by instantaneous overloads due to extreme load cases, or by material fatigue. Instantaneous failure may be caused by either overstressing of components beyond the elastic limit, or by buckling of items in compression or shear stress, or by failure of the joints. The strength may be alternatively defined as the maximum force which the structure can withstand. Different load cases different local component loads, but the structure must have sufficient strength for all load cases [7].

4.6.2 Stiffness

The stiffness of the structure relates the deflection produced when load is applied. It applies only to structures in the elastic range and is the slope of the load versus deflection graph.

a) Bending stiffness, which relates the symmetrical vertical deflection of a point near the centre of the wheelbase to multiples of the total static loads on the vehicle.
b) Torsion stiffness, relates the torsional deflection of the structure to an applied pure torque T about the longitudinal axis of the vehicle. The vehicle is subjected to the ‘pure torsion load case’. Twist angle is measured between the front and rear suspension mountings [7].

4.6.3 Vibration behavior

The global vibration characteristics of a vehicle are related to both its stiffness and mass distribution. The frequencies of the global bending and torsional vibration modes are commonly used as benchmarks for vehicle structural performance [7].

4.6.4 Section modulus

Basically, Section Modulus is a mathematical analysis of the cross section of a load carrying member.

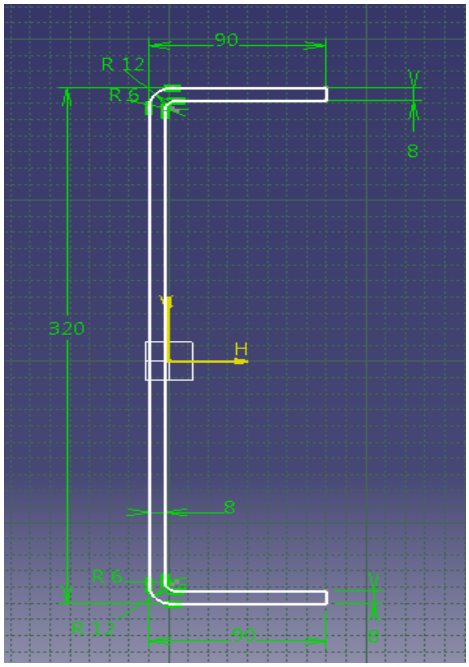

Fig 4.6.1 Section of main side rail

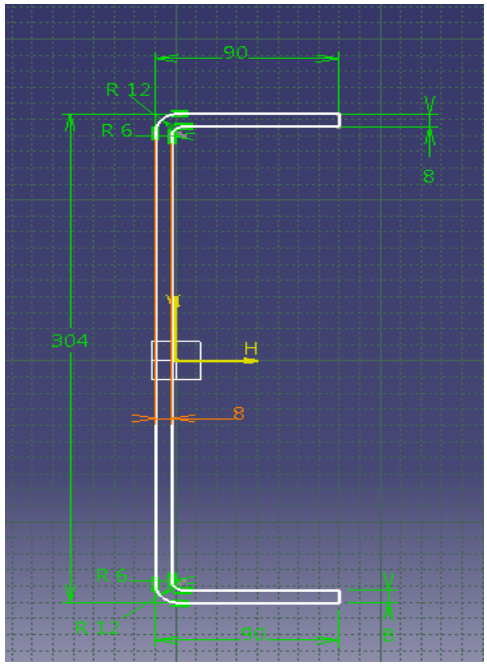

Fig 4.6.2. Supporting or reinforcing member of side rail

The moment of inertia of side rail is the combination of main side rail and supporting member. It is calculated that the moment of inertia of both C-channel section and sum up. The same is true for section modulus.

$$I_{XX} = (bh^3 - b_1h^3_1) / 12 \quad (4.1)$$

Where, b is outer width of the section

h is the outer height of the section

b1 is inner width of the section

h1 is inner height of the section

Ixx is area moment of inertia

Zxx is section modulus

I_{XX}= 98,451,456 mm^4

Section modulus is also the combination of main side rail and supporting member.

$$Z_{XX} = (bh^3-b_1h^3_1) / 6h \quad (4.2)$$

Z_{XX}= 615,321.6 mm^3

The cross section of a frame side member has several engineering "properties" used in design. Two significant mathematical properties are the Moment of Inertia and the Section Modulus. When comparing frames made of the same material, section modulus can be used to determine the frame with the larger load-carrying capacity. Frames with larger section modulus or SM can carry heavier loads. There are several formulas used in determining a channel's section modulus [18].

Section Modulus is one of several engineering properties for beam cross sections and is a factor in the flexural strength (load carrying capability) of the beam. It is defined as the Moment of Inertia divided by the distance from the neutral axis to a location of interest within the cross section. Since the Section Modulus relates to the stress in a beam cross section, the maximum stress generally occurs at the outer surface which is the greatest distance from the neutral axis [25].

In summary, the Section Modulus of a frame side member is a complex engineering property that indicates the load carrying capability. It considers the shape, height, flange width and material thickness of the side member.

4.6.5 Yield strength (σ_y)

Yield Strength is simply the maximum rated load that can be applied to a material before it permanently stretches or breaks. Since Section Modulus (Z_{xx})is a comparison of relative load carrying ability based only on the shape of a frame section, a method is needed to compare the strengths of the various materials that are used in truck frames. The number used for material strength comparison is yield strength. It represents the maximum load that can be placed on a specific size piece of the material without permanently bending or deforming the material [18].

To summarize, with Section Modulus numbers we can compare the relative load carrying ability of various frame section shapes without regard to the frame material. With yield strength information we can compare the strength per unit area of frame material without regard to the shape. What we finally need, however, is a method to compare the actual load carrying ability of frames of various shapes and materials.

The frame selected for the purpose of this study is made of milled steel with yield strength of 550MPa and tensile strength of 620MPa. Carbon epoxy and e glass epoxy has a strength up to1500MPa and 1062MPa respectively [1].

4.6.6 Resisting Bending Moment (RBM)

It is defined as the Section Modulus multiplied by the yield strength and it does allow comparison of load carrying members of any shape and material. The RBM is calculated for only for one rail [7].

For mild steel

$$RBM=Z_{XX} \times \sigma_y \quad (4.3)$$

$RBM= 615{,}321.6\ mm^3 \times 550\ N/mm^2 = 338{,}426{,}880Nmm = 338.426kNm$

For carbon epoxy

$RBM= Z_{XX} \times \sigma_y$

$RBM= 615{,}321.6mm^3 \times 1500N/mm^2= 922982400Nmm=992.98kNm$

For e glass epoxy

$RBM= Z_{XX} \times \sigma_y$

$RBM= 615{,}321.6mm^3 \times 1062N/mm^3= 653{,}471{,}539.2Nmm=635.47kNm$

From the above results composite materials have higher resisting bending moment than mild steel. That means composite material especially carbon epoxy chassis frame have a greater load carrying capacity as compared to other materials.

4.7 Vehicle weight reduction options

Reductions in vehicle weight can be achieved by a combination of

(i) material substitution;

(ii) vehicle redesign; and

(iii) Vehicle downsizing.

Material substitution involves replacing heavier iron and steel used in vehicles with weight-saving materials like aluminum, magnesium, high-strength steel, and plastics and polymer composites. Vehicles can be redesigned to optimize the size of the engine and other components as vehicle weight decreases, or to improve the vehicle's packaging and reduce exterior vehicle dimensions while maintaining the same passenger and cargo space. Finally, downsizing can provide further weight reduction by shifting sales away from larger and heavier to smaller and lighter vehicle categories [6].

The body and chassis of modern HDVs are currently made from a wide variety of materials ranging from high-strength steel to glass-reinforced composites. This diversity combined with relatively low production volumes provides an opportunity for additional weight reduction in the body and chassis while also serving as a potential entry point for validating advanced lightweight material designs. In fact, the heavy-duty industry has significant experience light weighting specific components. However, expanding these techniques to the entire vehicle requires design experience with the available new materials, joining techniques for multi-material approaches, process databases, optimized manufacturing techniques, the availability of advanced materials, established repair technologies, and lower costs materials. These barriers contribute to delays in commercial adoption of designs utilizing lighter weight materials. Significant advances are needed to bridge these gaps while maintaining or improving the performance, durability, utility, and safety of the vehicle [13].

4.8 Material selection

Mild steel, also called plain-carbon steel, is the most common form of steel because its price is relatively low while it provides material properties that are acceptable for many applications, more so than iron. Low-carbon steel contains approximately 0.05–0.3% carbon making it malleable and ductile. Mild steel has a relatively low tensile strength, but it is cheap and malleable; surface hardness can be increased through carburizing. It is often used when large quantities of steel are needed. Its drawback is higher density, less fatigue strength and susceptible to corrosion. Aluminum and magnesium are lightest metals with less modulus but their density is higher as compared to modern composite materials. Generally metals are susceptible to corrosion.

4.8.1 Selection of fibers

A. Glass fiber

Glass is the most common fiber used in polymer matrix composites. Its advantages include its high strength, low cost, high chemical resistance, and good insulating properties. The drawbacks include low elastic modulus, poor adhesion to polymers, high specific gravity, sensitivity to abrasion (reduces tensile strength), and low fatigue strength. An individual structural glass fiber is both stiff and strong in tension and compression that is, along its axis. Although it might be assumed that the fiber is weak in compression, it is actually only the long aspect ratio of the fiber which makes it seem so i.e., because a typical fiber is long and narrow, it buckles easily. On the other hand, the glass fiber is weak in shear that is, across its axis. Therefore if a collection of fibers can be arranged permanently in a preferred direction within a material, and if the fibers can be prevented from buckling in compression, then that material will become preferentially strong in that direction. Furthermore, by laying multiple layers of fiber on top of one another, with each layer oriented in various preferred directions, the stiffness and strength properties of the overall material can be controlled in an efficient manner. In the case of fiberglass, it is the plastic matrix which permanently constrains the structural glass fibers to directions chosen by the designer. With chopped strand mat, this directionality is essentially an entire two dimensional plane; with woven fabrics or unidirectional layers, directionality of stiffness and strength can be

more precisely controlled within the plane. The types of glass fibers are C-glass, S-glass and E-glass. The C-glass fiber is designed to give improved surface finish. S-glass fiber is design to give very high modular, which is used particularly in aeronautic industries. The E-glass fiber is a high quality glass, which is used as standard reinforcement fiber for all the present systems well complying with mechanical property requirements. Thus, E-glass fiber was found appropriate for this application. E-Glass / Epoxy Resin Composites are extremely strong materials used in roofing, pipes and automobiles. Composition: 54% SiO2 - 15% Al2O3 - 17% CaO - 4.5% MgO_2 - 8% B_2O - 1.5% others [8].

B. Carbon fiber

Carbon-Fiber-Reinforced Polymer, Carbon-Fiber-Reinforced Plastic or Carbon-Fiber-Reinforced Thermo Plastic (CFRP, CRP, CFRTP or often simply carbon fiber, or even carbon) is an extremely strong and light Fiber-Reinforced Polymer which contains carbon fibers. The binding polymer is often a thermoset resin such as epoxy, but other thermoset or thermoplastic polymers, such as polyester, vinyl ester or nylon, are sometimes used. The composite may contain other fibers, such as aramid, Kevlar, Twaron, Aluminium or Glass fibers as well as Carbon fiber. The properties of the final CFRP product can also be affected by the type of additives introduced to the binding matrix (the resin). The most frequent additive is silica, but other additives such as rubber and carbon nanotubes can be used [10]

CFRPs are commonly used in the transportation industry; normally in cars, boats and trains, and in sporting goods industry for manufacture of bicycles, bicycle components, golfing equipment and fishing rods. Although carbon fiber can be relatively expensive, it has many applications in aerospace and automotive fields, such as Formula One racing and wherever high strength-to-weight ratio and rigidity are required such as sailing boats and rowing shell hulls, top-end bicycles and motorcycles, As manufacturing techniques improve and costs reduce it is becoming increasingly common in small consumer goods that require strength, lightness and stiffness such as laptop bodies, tripod legs, tent poles, fishing rods, hockey sticks, bows and arrows, racquet frames, stringed instrument bodies, drum shells, golf clubs, crash helmets and billiards cues. **Composition**: Bisphenol-based epoxy 60-90%, Amine-based curing agent 1-30%, Imidazole-based curing catalyst 0.1-3% and Carbon Black 1-10% [10].

4.8.2 Resins selection

In a composite, inter laminar shear strengths is controlled by the matrix system used. Since these are reinforcement fibers in the thickness direction, fiber do not influence inter laminar shear strength. Therefore, the matrix system should have good inter laminar shear strength characteristics compatibility to the selected reinforcement fiber. Many thermo set resins such as polyester, vinyl ester, epoxy resin are being used for fiber reinforcement plastics (FRP) fabrication. Among these resin systems, epoxies show better inter laminar shear strength and good mechanical properties. Hence, epoxy is found to be the best resins that would suit this application. The main reasons why epoxy is the most used polymer matrix material are

- High strength
- Low viscosity and low flow rates, which allow good wetting of fibers and prevent misalignment of fibers during processing
- Low volatility during cure
- Low shrink rates, which reduce the tendency of gaining large shear stresses of the bond between epoxy and its reinforcement

Monolithic metals and their alloys cannot always meet the demands of today's advanced technologies. Only by combining several materials one can meet the performance requirements. In many cases, using composites is more efficient. For example, in the highly competitive automotive market, one is continuously looking for ways to lower the overall mass of the vehicle without decreasing the stiffness and strength of its components. This is possible by replacing conventional metal alloys with composite materials. Even if the composite material costs may be higher, the reduction in the number of parts in an assembly and the savings in fuel costs make them more profitable [10].

Composites offer several other advantages over conventional materials. These may include improved strength, stiffness, fatigue and impact resistance, thermal conductivity, corrosion

resistance, etc. Thus, to measure the mechanical advantage of materials, the specific modulus (E/ρ) and the specific strength (σ_{ult}/ρ) are the best parameters to be compared.

Table 4.9.1 Specific Modulus and Specific Strength of Typical Fibers, Composites, and Bulk Metals [8]

Material Units	**Density (kg/m^3)**	**Young's modulus (GPa)**	**Ultimate strength (MPa)**	**Specific modulus ($GPa\text{-}m^3/kg$)**	**Specific strength ($MPa\text{-}m^3/kg$)**
Carbon fibre	1800	230.00	2067	0.1278	1.148
Glass fibre	2500	85.00	1550	0.0340	0.65
carbon/Epoxy	1600	134.00	1500	0.08375	0.9375
e glass/Epoxy	2600	50	1062	0.01923	0.40846
Mild Steel	7800	207	620	0.02652	0.07948
Aluminium	2800	68.95	275.8	0.02457	0.1061

From the above table it is represented graphically shown below to compare material performance specific modulus and specific strength of polymeric composite materials and metal alloys. Carbon epoxy and e glass epoxy have higher specific strength with less density. The specific modulus is higher for carbon epoxy and lower e glass epoxy. Mild steel have less specific strength and good specific modulus next to carbon epoxy but its density is high.

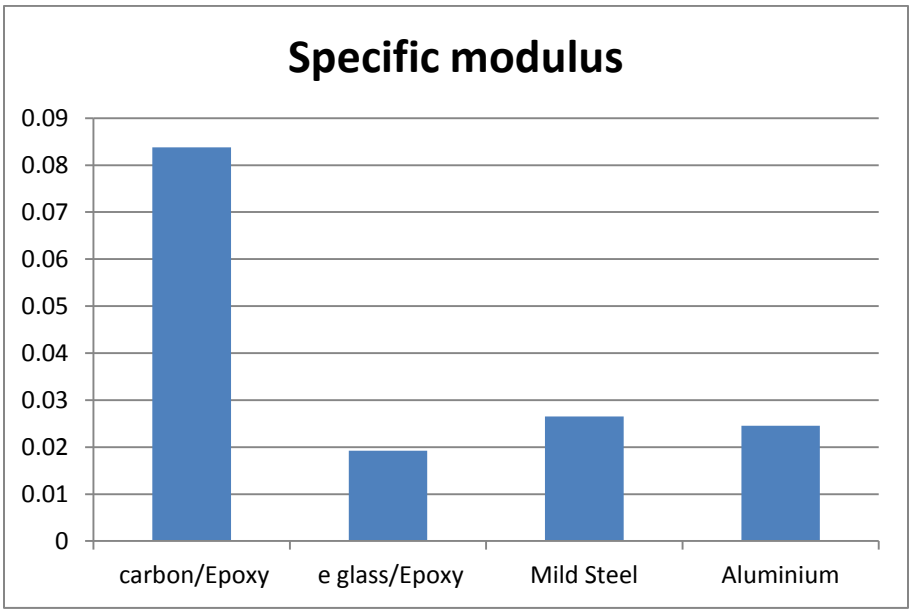

Fig 4.8.1 Graphical representation of specific modulus

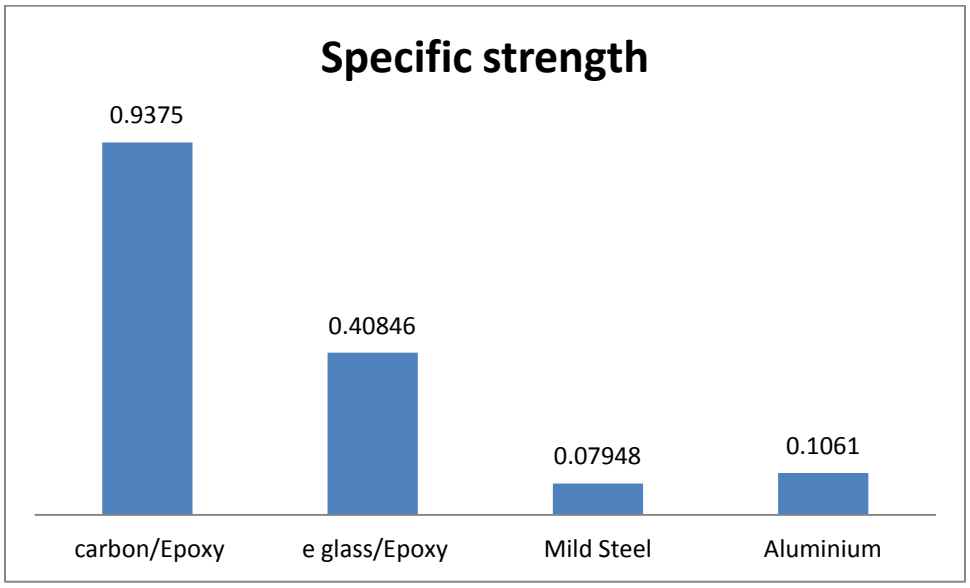

Fig 4.8.2 Graphical representation of specific strength of materials

Similarly, we can prove that, for achieving the minimum deflection in a beam under a load along its length, the lightest beam is one with the highest value of $E^{1/2}/\rho$ and $E^{1/3}/\rho$.

Table 4.8.2 Specific Modulus Parameters E/ρ, $E^{1/2}/\rho$ and $E^{1/3}/\rho$ for Typical Materials [8]

Material **Units**	**Density**(kg/ m^3)	**Young's Modulus**	E/ρ **(GPa-m3/kg)**	$E^{1/2}/\rho$ **($Pa^{1/2}$-**	$E^{1/3}/\rho$ **($Pa^{1/3}$-m^3/kg**

		(GPa)		**m3/kg)**	**)**
Carbon fibre	1800	230.00	0.1278	266.4	3.404
Glass fibre	2500	85.00	0.034	116.6	1.759
carbon/Epoxy	1600	134.00	0.08375	228.78	3.535
glass/Epoxy	2600	50.0	0.01923	86.0	1.417
Mild Steel	7800	207	0.02654	58.3	0.758
aluminium	2800	68.95	0.02462	93.79	1.4645

Form these candidate materials, the best material is that have higher specific strength, specific modulus and less density. Weight reduction is the main issue in conjunction with good strength.when the weight of the component deceases, there is a great reduction in fuel consumption so as to reduce emission which is a global issue and life cycle cost will be decreased. So to achieve these listed advantages quasi isotropic carbon epoxy and e glass epoxy are selected.

According to the requirements of truck chassis frame such as light weight, strong, stiff, resistance to fatigue load and others, carbon/epoxy and E-glass/epoxy materials are selected as the best materials that can satisfy the requirements.

4.9 Composite materials

A composite material is defined as a material composed of two or more constituents combined on a macroscopic scale by mechanical and chemical bonds. Composites are combinations of two materials in which one of the material is called the “matrix phase” is in the form of fibers, sheets, or particles and is embedded in the other material called the “reinforcing phase” [8].

Another unique characteristic of many fiber reinforced composites is their high internal damping capacity. This leads to better vibration energy absorption within the material and results in reduced transmission of noise to neighboring structures. Many composite materials offer a combination of strength and modulus that are either comparable to or better than any traditional metallic metals. Because of their low specific gravities, the strength to weight-ratio and modulus to weight-ratios of these composite materials are markedly superior to those of metallic materials. The fatigue strength weight ratios as well as fatigue damage tolerances of many

composite laminates are excellent. For these reasons, fiber composite have emerged as a major class of structural material and are either used or being considered as substitutions for metal in many weight-critical components in aerospace, automotive and other industries. High damping capacity of composite materials can be beneficial in many automotive applications in which noise, vibration, and hardness is a critical issue for passenger comfort [9].

The type and quantity of the reinforcement determine the final properties. There is a practical limit of about 70 volume percent reinforcement that can be added to form a composite. At higher percentages, there is too little matrix to support the fibers effectively. The theoretical strength of discontinuous-fiber composites can approach that of continuous-fiber composites if their aspect ratios are great enough and they are aligned, but it is difficult in practice to maintain good alignment with discontinuous fibers [8].

Discontinuous-fiber composites are normally somewhat random in alignment, which dramatically reduces their strength and modulus. However, discontinuous-fiber composites are generally much less costly than continuous-fiber composites. Therefore, continuous-fiber composites are used where higher strength and stiffness are required (but at a higher cost), and discontinuous-fiber composites are used where cost is the main driver and strength and stiffness are less important [8].

4.9.1 Laminates

When there is a single ply or a lay-up in which all of the layers or plies are stacked in the same orientation, the lay-up is called a *lamina*. When the plies are stacked at various angles, the lay-up is called a *laminate*. Continuous-fiber composites are normally laminated materials in which the individual layers, plies, or laminae are oriented in directions that will enhance the strength in the primary load direction. *Unidirectional (0°) laminae* are extremely strong and stiff in the 0° direction. However, they are very weak in the 90° direction because the load must be carried by the much weaker polymeric matrix. While a high-strength fiber can have a tensile strength of 3500 MPa or more, a typical polymeric matrix normally has a tensile strength of only 35 to 70 MPa [8].

The longitudinal tension and compression loads are carried by the fibers, while the matrix distributes the loads between the fibers in tension and stabilizes the fibers and prevents them from buckling in compression. The matrix is also the primary load carrier for inter laminar shear (i.e., shear between the layers) and transverse (90°) tension. Because the fiber orientation directly impacts mechanical properties, it seems logical to orient as many of the layers as possible in the main load-carrying direction. While this approach may work for some structures, it is usually necessary to balance the load-carrying capability in a number of different directions, such as the 0°, +45°, -45°, and 90° directions. A balanced laminate having equal numbers of plies in the 0°, +45°, –45°, and 90° degrees directions is called a *quasi-isotropic laminate*, because it carries equal loads in all four directions [8].

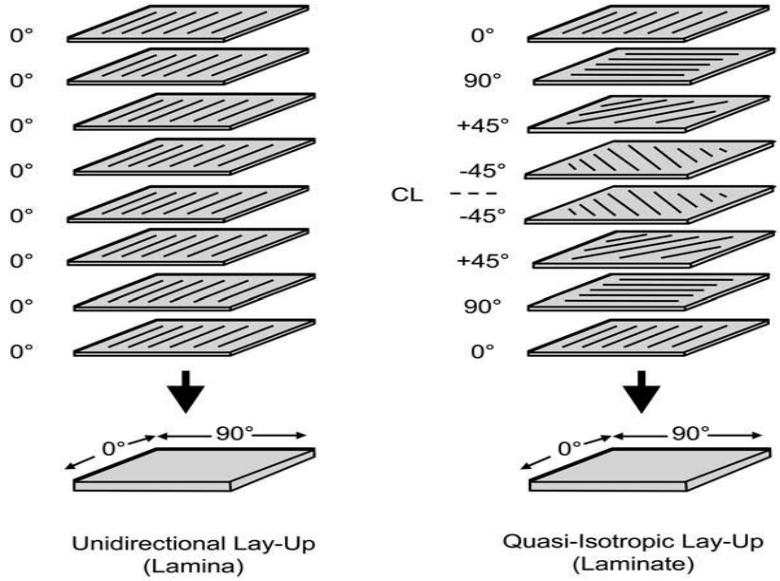

Fig 4.9.1 The difference between lamina and laminate [8]

Temperature has an effect on composite mechanical properties. Typically, matrix-dominated mechanical properties decrease with increasing temperature. Fiber-dominated properties are somewhat affected by cold temperatures, but the effects are not as severe as those of elevated temperature on the matrix-dominated properties. An important design factor in the selection of a matrix resin for elevated-temperature applications is the cured glass transition temperature. The cured glass transition temperature (*T*g) of a polymeric material is the temperature at which it changes from a rigid, glassy solid into a softer, semi flexible material. At this point, the polymer structure is still intact but the crosslink's are no longer locked in position. Therefore, the *T*g

determines the upper use temperature for a composite or an adhesive and is the temperature above which the material will exhibit significantly [8].

4.9.2 Advantages of composite materials

The main advantage of most composites materials are in the weight savings. A quick way to illustrate this advantage is in the strength to weight ratio. Different materials have different strength, which is each material can take different of amount of load for the same volume (cross sectional area) of the material. For a given design, the material used must be strong enough to withstand the load that is to be applied. If a material selected is not strong enough, the part must be enlarged to increase the load bearing capacity. But doing so increases the bulk and weight of the part. Another option is to change material to one that has high enough strength to begin with. Significant advantages over traditional materials include [13];

- ✓ Corrosion, scratch and dent resistance.
- ✓ Reduced noise vibration harshness.
- ✓ High specific energy absorption.
- ✓ Increased design flexibility.
- ✓ Reduced tooling costs.
- ✓ Dimensional stability.

4.10 Composite fabrication or manufacturing processes

Most widely used manufacturing methods for laminated fibre composites are as follows:

4.10.1 Hand Lay-up

The hand (wet) lay-up is one of the oldest and most commonly used methods for manufacture of composite parts. Hand lay-up composites are a case of continuous fibre reinforced composites. Layers of unidirectional or woven composites are combined to result in a material exhibiting desirable properties in one or more directions. Each layer is oriented to achieve the maximum utilisation of its properties. Layers of different materials (different fibres in different directions) can be combined to further enhance the overall performance of the laminated composite material. Resins are impregnated by hand into fibres, which are in the form of woven, knitted, stitched or bonded fabrics. This is usually accomplished by rollers or brushes, with an increasing use of nip-

roller type impregnators for forcing resin into the fabrics by means of rotating rollers and a bath of resin. Laminates are left to cure under standard atmospheric conditions [8]. A typical hand lay-up method is shown below

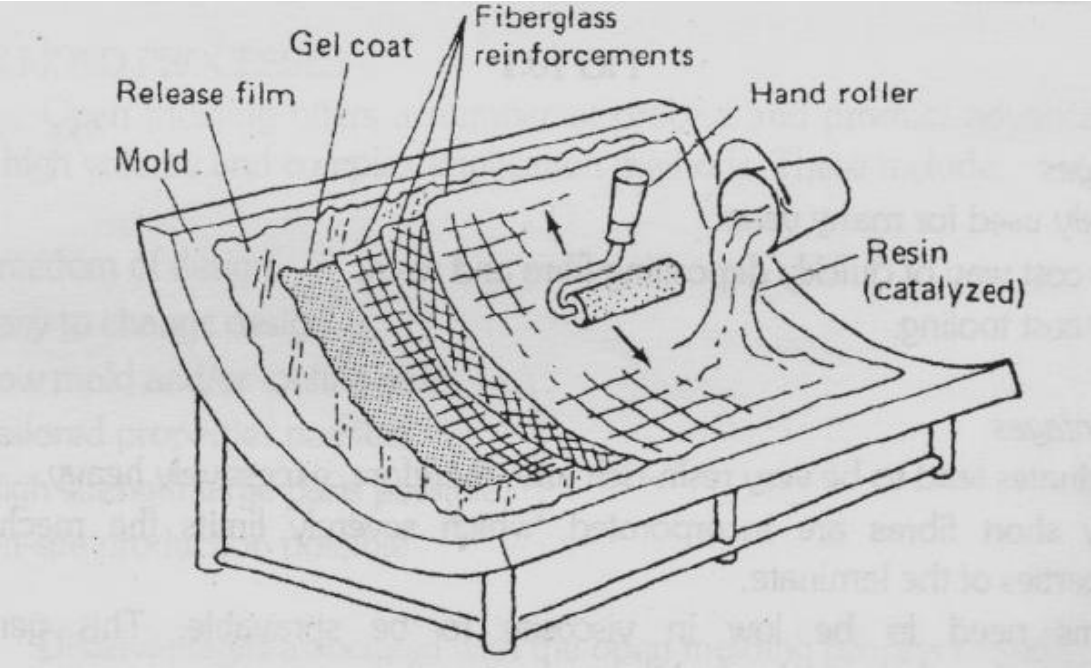

Fig 4.110.1 Hand layup process [8]

4.10.2 Spray Lay-up

In a spray lay-up method, the fiber is chopped in a hand held gun and fed into a spray of catalysed liquid resin directed at the mold (Fig 3.10.2). The sprayed, catalysed liquid resin will wet the reinforcement fibers, which are simultaneously chopped in the same spray gun. The deposited materials are left to cure under standard atmospheric conditions [8]

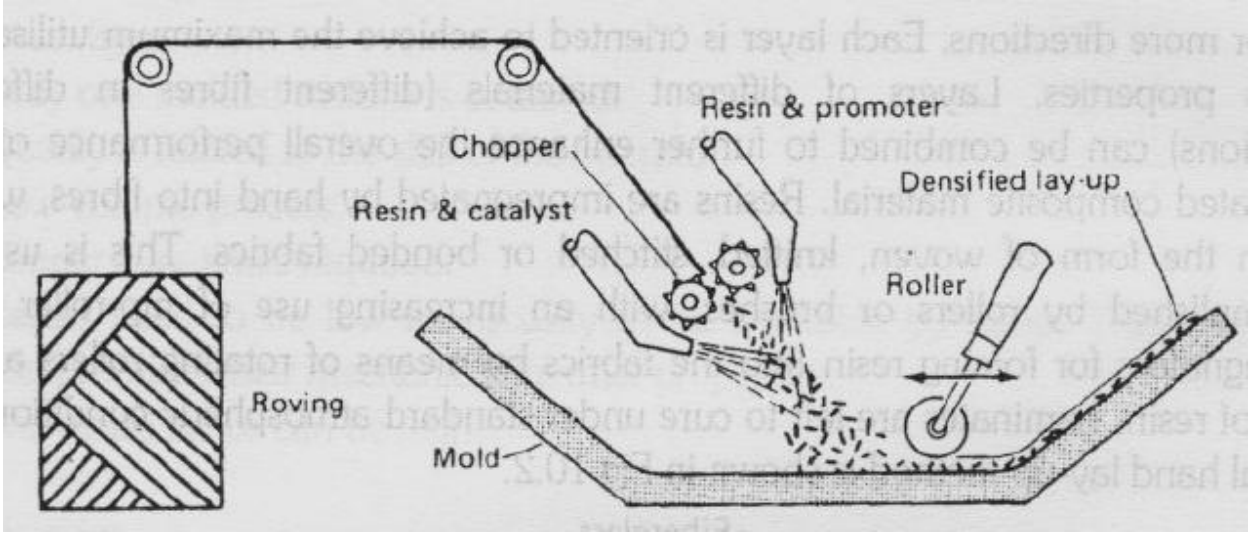

Fig 4.10.2 Spray lay up process [8]

4.10.3 Filament Winding

Filament winding is automated processes for creating parts of simple geometry wherein continuous resin impregnated fibers are wound over a rotating male tool called mandrel. Fig 10.3 depicts a filament winding process where a continuous fiber roving passes through a shuttle, which rotates and the roving is wrapped around a revolving or stationary mandrel. Two basic types of filament winding are in use - (i) the polar or planer method, and (ii) the high helical pattern winding [8].

The polar or planer method of winding utilizes a fixed mandrel and a shuttle that revolves around the longitudinal axis of the part to form longitudinal winding patterns. This type of winding is used if the longitudinal fibres are required with angle less than 25° to the mandrel axis. The polar winding schematic is shown in Fig 4.10.3.

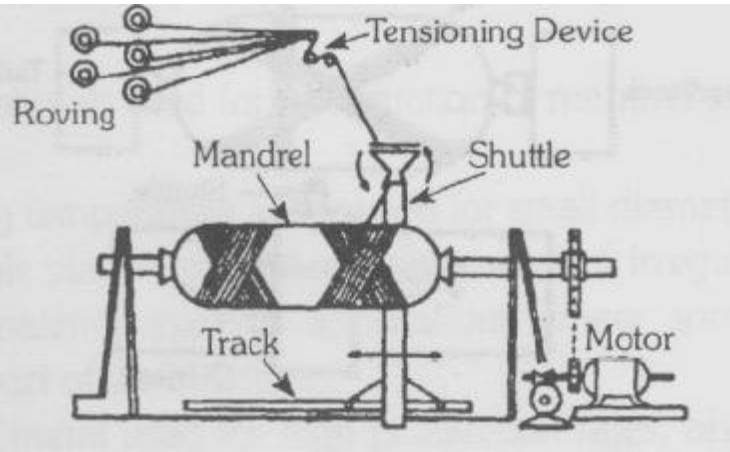

Fig 4.10.3 Polar filament winding process [8]

4.10.4 Sheet Molding Compound

Sheet molding compound (SMC) refers to both a material and a process for producing glass fiber reinforced polyester resin items. The material is typically composed of a filled, thermosetting resin and a chopped or continuous strand reinforcement of glass fiber. A SMC processing machine, (Fig 4.10.4) produces molding compound in sheet form. The glass fiber is added to a resin mixture that is carried onto a plastic carrier film. After partial cure, the carrier films are removed. The sheet molding material is cut into lengths and placed onto matched metal dies under heat and pressure [8].

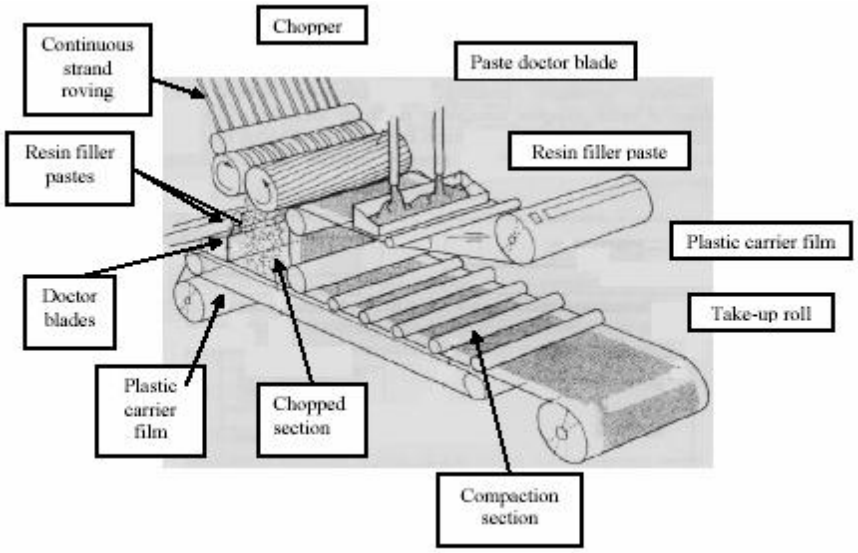

Fig 4.10.4 Sheet molding compound process [8]

4.10.5 Compression Molding

Compression molding is one of the oldest manufacturing techniques in the composites industry. The recent development of high strength, fast cure, sheet molding compounds bulk molding compounds and advancement in press technology is making the compression molding process very popular for mass production of composite parts. Fully formed parts are molded in matched metal compression molds that give the final part shape. In comparison with the injection molding process, better physical and mechanical properties can be obtained in compression molding [8].

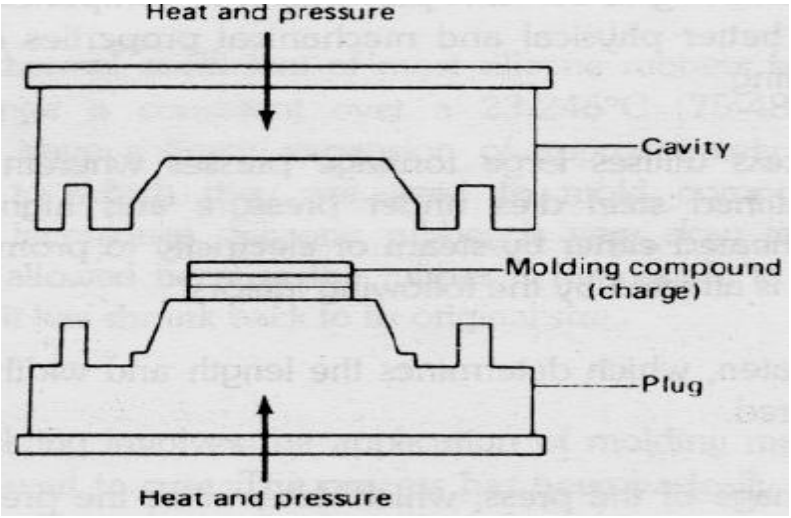

Fig 4.10.5 Compression mold process [8]

4.10.6 Pultrusion

Pultrusion is an automated process used to create shapes by pulling rovings through a shaped and heated die. The use of pultruded parts in aircraft is limited to specialized applications. Practical applications are limited to constant cross-section parts. Pultrusion is used to manufacture constant cross-section shapes, viz., I-beam, box, channels, tubings, etc.

The Pultrusion process machine consists of six different parts namely, the creel, the resin bath, the forming die, the heated curing die, the pullers and the cut-off saw. The creel is the beginning of the Pultrusion process and is the material storage system from which the fibers and mat or fabric are drawn in the correct sequence to match the design requirements of the structural shape.

Virtually all Pultrusion processes utilize a resin impregnation bath to facilitate the impregnation of the resin into the fiber structure. The use of pre-impregnated fibers eliminates the resin bath.
Two types of dies are used in Pultrusion process, namely, the forming and the heating or curing die. Forming is done immediately after the impregnation process. Forming dies are normally attached to the heating or curing die in order to provide the correct relationship between the forming and the heated curing step. The rovings go through a heated die that represents the cross-section of finished part. Curing is accomplished by heating the die. The product continuously pulled out and as it comes out of the puller mechanism, it is cu the desired length by an automatic saw. The process is continuous and can be used to manufacture extremely long sections [8]. At typical continuous Pultrusion process is shown in 3.10.6.

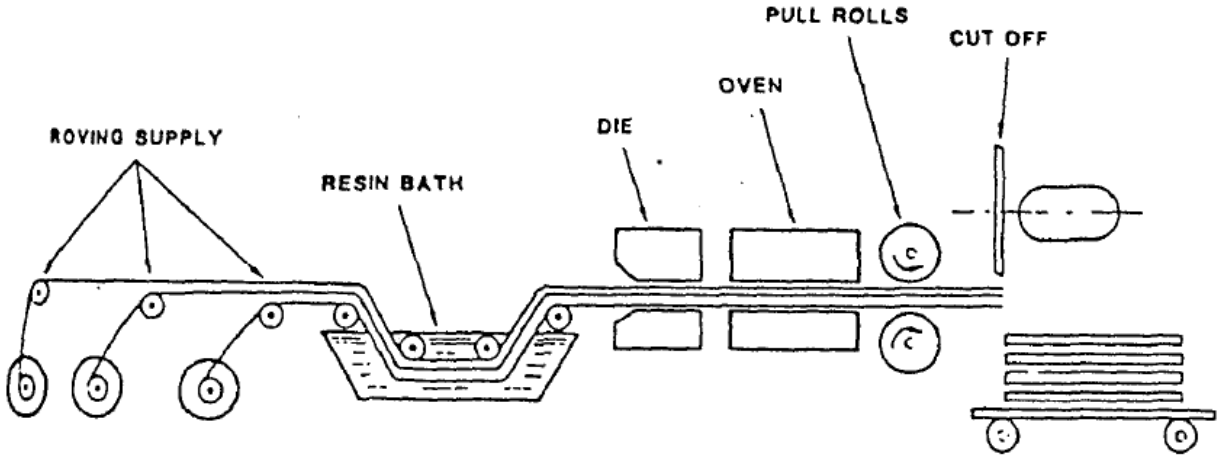

Fig 4.10.6 Pultrusion process [8]

4.11.7 Injection Molding

Injection molding refers to a process that generally involves forcing or injecting a fluid plastic material into a closed mold. It is differentiated from compression molding, in which plastic materials in a soft but not fluid condition are formed by transferring them into an open mold, which is then forcibly closed. This method is not normally used in polymeric matrix compound processes due to fiber damage in the barrel [8].

The molding compound is fed into injection chamber through the feed hopper. The molding compound is heated in the injection chamber wherein it changes into liquid form. It is forced into the injection mold by the plunger. A typical injection mold is shown in Fig 4.10.7

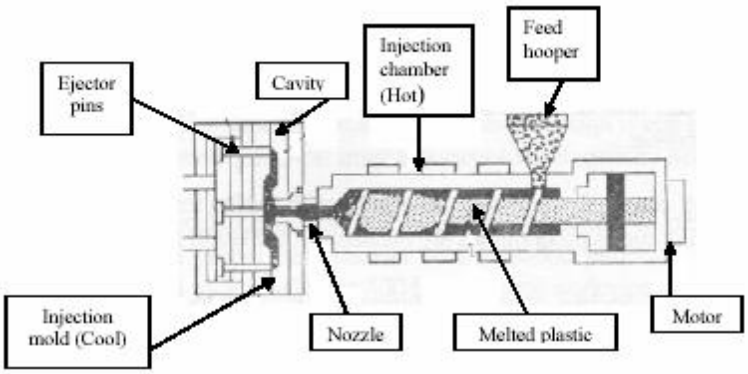

Fig 4.10.7 Injection molding process [8]

4.10.8 Resin Transfer Molding (RTM)

Resin transfer molding is a closed mold low pressure process that allows the fabrication of composites ranging in complexity from simple, low performance to complex, high performance parts and in size from small to large (Fig 4.10.8). The process is differentiated from other molding processes in that the dry reinforcement and the resin are combined within the mold to form the composite component. The fiber reinforcement, which may be pre-shaped, is placed into a tool cavity, which is then closed. A tube connects the closed tool cavity with a supply of liquid resin, which is pumped or transferred into the tool to impregnate the reinforcement for subsequent curing. Injection pressure is normally less than 690 kPa (100 psi). The displaced air is allowed to escape through vents to avoid dry spots. Cure cycle is dependent on part thickness, type of resin system and the temperature of the mold and resin system. The part cures in the mold, normally heated by controllers [8].

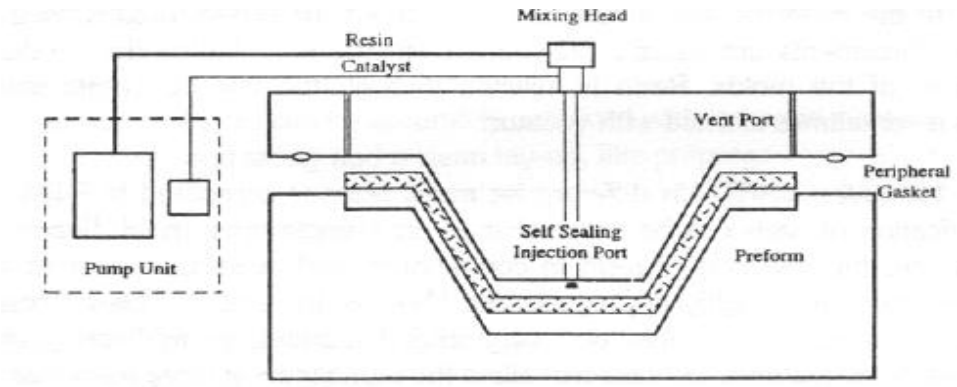

Fig 3.10.8 Resin transfer molding process [8]

CHAPTER 5

RESULTS AND DISUSSION

5.1 Introduction

The chassis forms the backbone of the dump truck and its chief function is to safely carry the maximum load, whether the vehicle is in static or dynamic condition. With the help of simulation tools like finite element method, the extent of the effects of chassis input conditions can easily be estimated with minimum experimental cost. As a test study, a 3-D finite element model was developed to predict the complete static stresses and vibration of the high strength structural chassis frame.

Main objective is to determine the static loads defined in pay load and vehicle weight acting in the form of reaction forces was adopted for static load condition. Another objective is to behavior of vibrations encountered during running condition. The load combinations are mainly three cases like bending, torsion and combined bending and torsion.

The truck chassis is the main system of the vehicle and it is integrated with the main truck component systems such as the axles, suspension, power train, cab and body. The truck chassis has been loaded by static, dynamic and also cyclic loading. Static loading comes from the weight of payload and net vehicle weights. The movement of truck affects a dynamic loading to the chassis. The vibration of engines and the roughness of roads give a cyclic loading. The existing truck chassis design is normally designed based on static analysis. The emphasis of this design is chiefly on the strength of structure to support the loading placed upon it. However, the truck chassis is loaded by complex type of loads, including static, and dynamic aspects. It is estimated that fatigue is responsible for 85 to 90% of all structural failures (or) crack propagation on the chassis. The knowledge of static and dynamic behavior of truck chassis in such environment is thus important to determine the stress and deflection as well as natural frequencies and mode shapes on the mounting point of the components like engine, suspension, transmission. Many researchers have carried out studies on truck chassis to investigate stress analysis of a truck chassis with riveted joints using FEM. Numerical results showed that stresses on the side member can be reduced locally increasing the side member's thickness. If the thickness change is not possible, then increasing the connection plate length may be a good alternative method. The objective of this research work is mainly focus to determine the vibration and static analysis on the heavy duty truck chassis.

As per the objectives of the thesis static analysis and modal analysis are performed. Static analysis is performed by assuming the vehicle is at rest. Previously it is noted that the vehicle is subjected to different load cases such as bending case, torsion case and combined bending and torsion cases as well as different load paths dynamically. So by using finite element method commercial software ANSYS the response of the ladder type chassis frame is analyzed when it is subjected to static load cases and also modal analysis is done to determine the natural frequencies and mode shapes of the frame that are necessary to know the vibration behavior .If

the excitation frequencies form the engine, transmission and road rough nesses are coincides with one of the natural frequencies resonance will happen.

5.2 Linear static analysis of chassis

This simulation is based on the condition of the truck being stationary. The ladder frame chassis was treated as a simply supported beam and loads were due to the weight of components applied to the beam. The support loads from the axles were distributed through spring hangers. The axle's reaction loads were obtained by resolving forces and taking moments from the weights and positions of the components. For practical calculations, it was recommended that the load on the chassis frame, including its own weight, is concentrated at a small number of points. These point loads were statically equivalent to the actual distributed load carried by the vehicle [20].

The stress distribution on the chassis for bending, torsion and combined bending and torsion load have been determined. According to the equation of equilibrium condition, the vertical tire reaction forces act upward direction and body weight with payload act on downward direction.

Static analysis calculates the effects of steady load conditions on a structure, while ignoring inertia and damping effects, such as those caused by time varying loads. A static analysis, however, includes steady inertia loads such as gravity and rotational velocity, and time-varying loads that can be approximated as static equivalent loads such as the static equivalent tire reaction loads and vehicle weight plus payload [1].

The chassis forms the main structure of the dump truck and its chief function is to safely carry the maximum load wherever the operation demands. Basically, it must absorb engine and axle torque and absorb the shock loads over twisting, pounding and uneven roadbeds when the vehicle moves along the road. The most important load cases are those of bending, torsion and combined bending a**n**d torsion are paramount in determining a satisfactory structure (Pawlowski, 1964). These listed load cases are simulated with FEM commercial software of ANSYS workbench 14.5 to know the response of the chassis frame in terms of stress and deformation. The basic equation governing static equilibrium condition of the system is given in equation below.

$$[K]\{q\} = \{F\} \quad (5.1)$$

Where, [K] is global stiffness matrix

{q} is nodal displacement vecor

{F} is the load vector

The global stiffness matrix is taken as the summation of element matrices, i.e.

$$[K] = \sum_1^n Ke \tag{5.2}$$

5.1.2 Mass of chassis frame

The mass of an object is a fundamental property of the object, a numerical measure of its inertia, a fundamental measure of the amount of matter in the object. Here the mass of frame is compared for three materials such as mild steel, carbon epoxy and e glass epoxy. Mathematical equation for mass is [10]

$$\text{Mass} = \text{Volume} \times \text{Density}$$

$$m = v \times \rho \tag{5.5}$$

The total volume of the frame is 0.14729m³

A. Mass of mild steel frame

We know, Density of steel = 7800kg/m³

Volume of frame = 0.14729 m³

Total mass of frame = 7800 × 0.14729 = 1148.9 kg.

B. Mass of carbon epoxy frame

We know Density of Carbon/Epoxy = 1600 kg/m³

Volume of frame = 0.14729 m³

Total mass of frame = 1600 × 0.14729 = 235.67 kg (Approx.)

C. Mass of e glass epoxy frame

We know Density of E-glass/Epoxy = 2600 kg/m^3

Volume of Frame = 0.14729 m^3

Total mass of Frame = 2600 × 0.14729 = 382.96 kg

When heavy weight materials are replaced by light weight materials it is possible to reduce the weight of vehicles and increases the payload carrying capacity. Here using carbon epoxy and e glass epoxy, it is saved 913.23kg and 765.94 kg respectively.

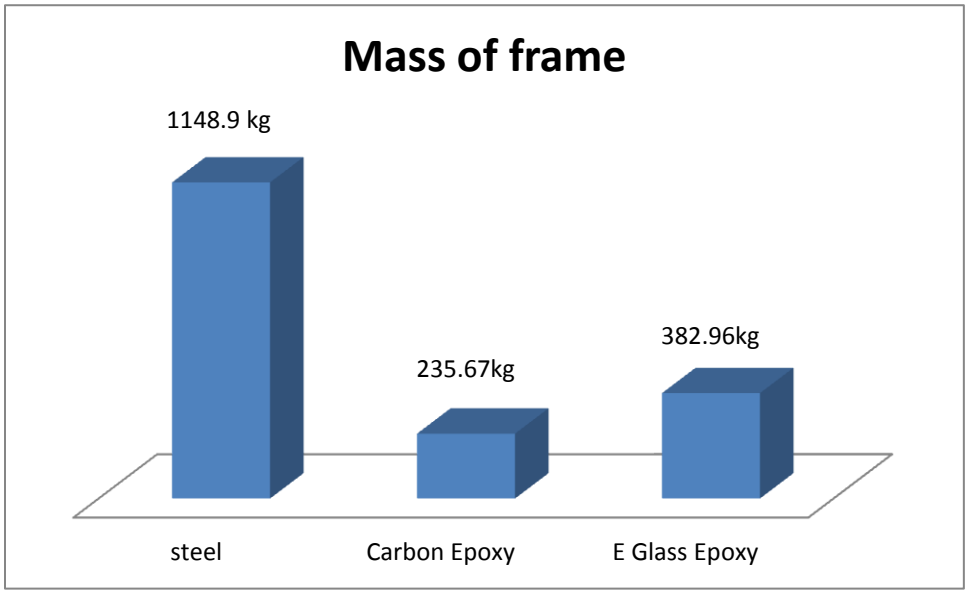

Fig 5.5.1 Graphical representation of mass of frame

5.5.3 Bending load case

It is the methodology of determining the effects of loads on physical structures and their components. Structural analysis incorporates the fields of applied mechanics, materials science and applied mathematics to compute a structure's deformations, internal forces, stresses, support reactions, accelerations, and stability. The results of the analysis are used to verify a structure's fitness for use, often saving physical tests. Structural Analysis is thus a key part of the engineering design of structures, [9]. The present frame model is converted into IGES format and it is then imported to ANSYS Workbench 14.5.There are various steps that are to be followed in

analyzing a component structurally. They are mesh generation, fixed supports, application of loads and evaluating result

A) Mesh Generation

The process for generating a mesh consists of three general steps are set the element attributes, set mesh controls (optional) and meshing the model.

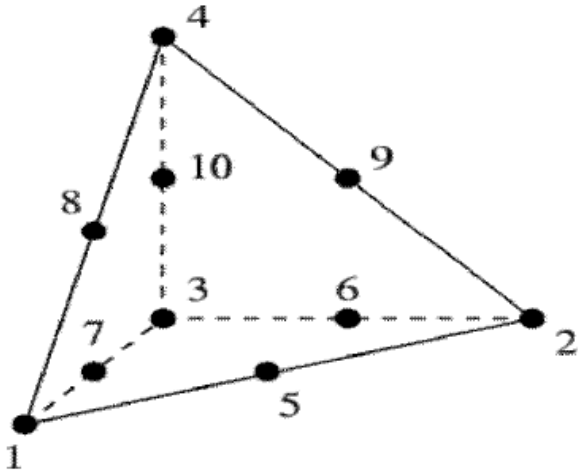

Fig 5.5.2 Ten node tetrahedral element

It is not always necessary to set mesh controls because the default mesh controls are appropriate for many models. If no controls are specified, the program will use the default settings to produce a free mesh. Alternatively, you can use the Smart Size feature to produce a better quality free mesh. Depending on the model complexity, desired mesh quality and type, and the time available to perform meshing, ANSYS Meshing provides a scalable solution. Quick automatic hex or hex-dominant mesh can be generated for optimal solution efficiency and accuracy, [10].

The meshing is done on the model with 203,977 numbers of nodes and 93,699 numbers of ten node tetrahedral elements. When meshing the sizing of mesh is done with fine relevance center and higher smoothing.

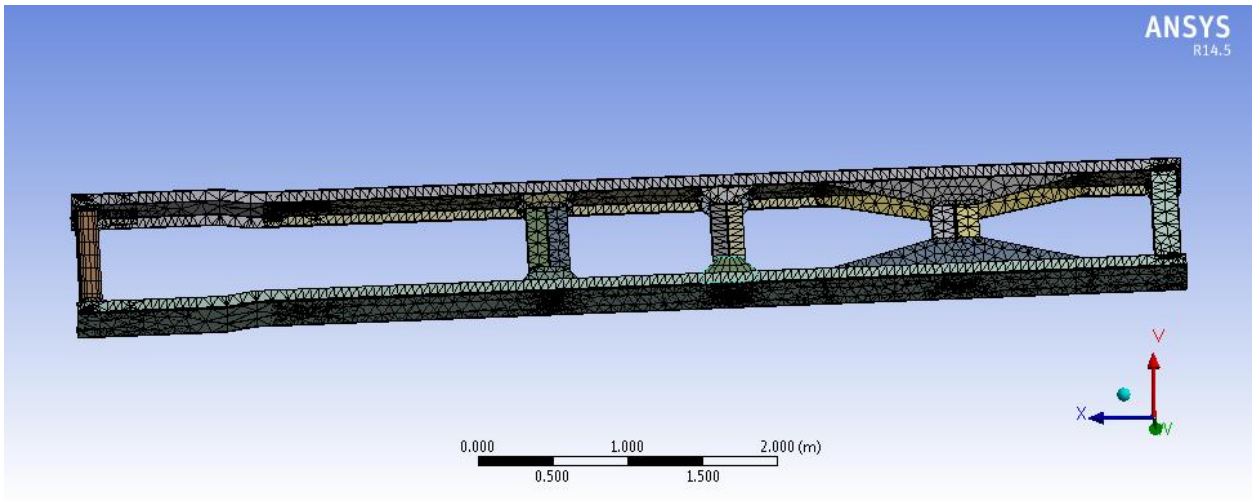

Fig 5.5.3 Meshing of the model

B) Boundary condition and application of the load

The fixed supports for the frame are placed at the wheel positions. The total number of supports is four. The first support is placed at 1375 mm from the front end at the front wheel center and the second support is placed at 1525 mm from the rear side at the position where the center of spring hunger is located. The other two supports are placed at same positions on the other side.

The load is applied on the upper flange of side rails where the pay load has contact with the frame [10].

Total GVM = 32000kg

Total weight = GVM x gravity

W= m x g (5.4)

= 32000*9.81 = 313,920N = 313.92KN

As the frame supports the body by its two side frames the load on each side members

= 313,920N/2 = 156,960N=156.96KN

Taking the load as a uniformly distributed load (UDL),

The total area on which the UDL is placed = the length of side beam * width of side beam

$$A = L \times b \quad (5.5)$$

$$= 7400mm * 90mm = 666{,}000mm^2$$

The total pressure applied = total load / total area = $156{,}960N/666{,}000mm^2 = 0.2356N/mm^2$

$$P = W/A \quad (5.6)$$

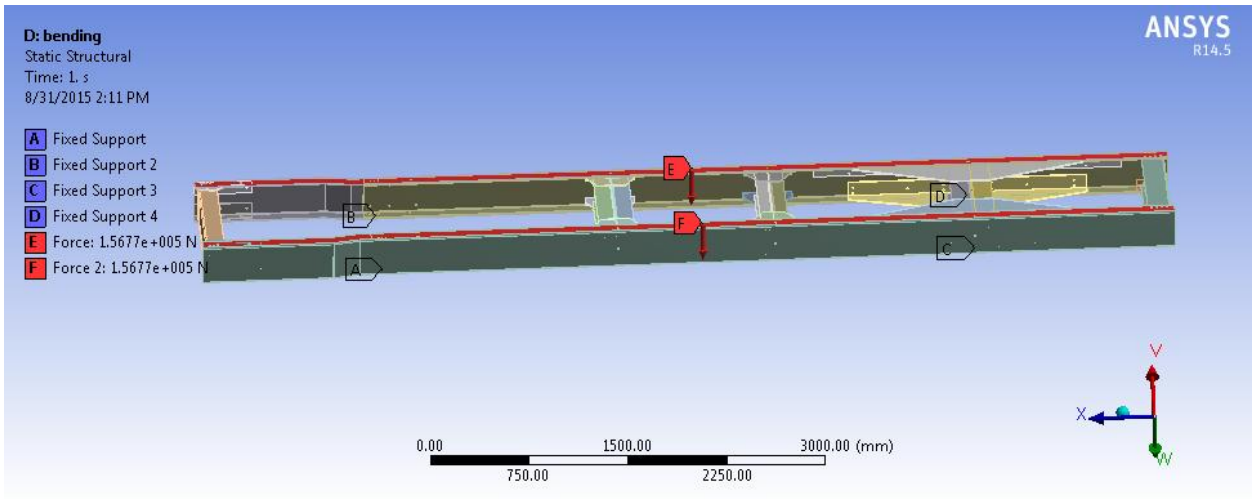

Fig 5.5.4 Fixed support and application of the load for bending case

C) Stresses distribution of mild steel for bending load case

It is a physical quantity that expresses the internal forces that neighboring particles of a continuous material exert on each other. For example, when a solid vertical bar is supporting a weight, each particle in the bar pulls on the particles immediately above and below it. These macroscopic forces are actually the average of a very large number of intermolecular forces and collisions between the particles in those molecules. There are many types of stresses developed in a component. The frame is analyzed by considering Equivalent stress [9].

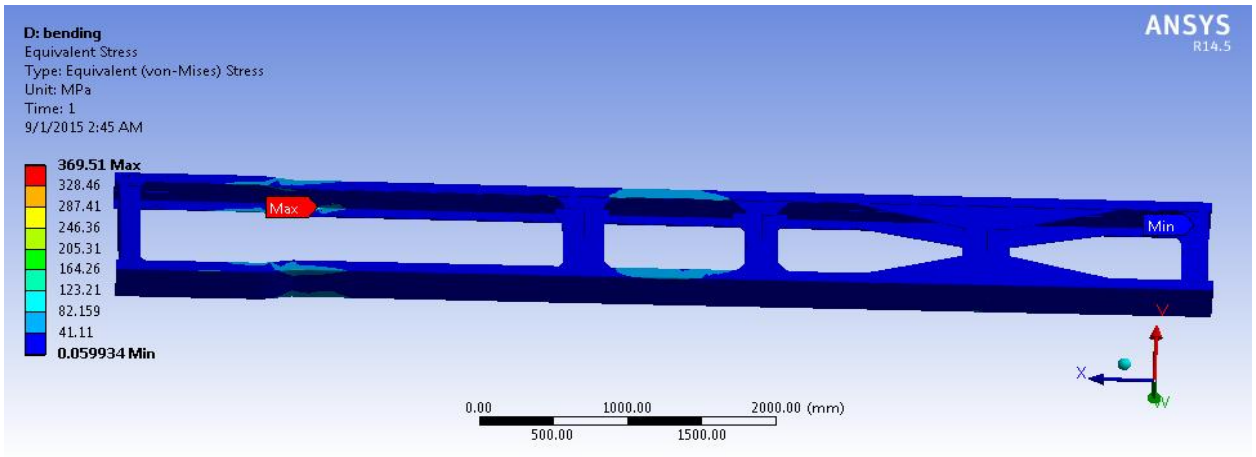

Fig 5.5.5 Equivalent von-mises stress distribution of mild steel for bending case

The maximum equivalent stress 369.51MPa is obtained around front fixed support and the minimum stress at the rear end of the frame. The yield stress of mild steel is 550MPa, so the equivalent stress is less than the yield stress means the design is safe.

The factor of safety of mild steel chassis frame due to bending load case is;

Factor of safety=yield stress/ developed stress (5.7)

FOS= 550Mpa/ 369.51MPa

FOS=1.48

D) Deformation of mild steel for bending load case

When an object is subjected to loading its shape may be changed temporarily or permanently due to applied force. This change in shape is called deformation. If the object deforms permanently it is called plastic deformation or failure. If it deforms temporarily it is called elastic deformation. While analyzing a frame it should deform elastically within the maximum loading limit so that the design is safe [19]. The values of deformation obtained in ANSYS 14.5 for mild steel are as shown in Fig. below.

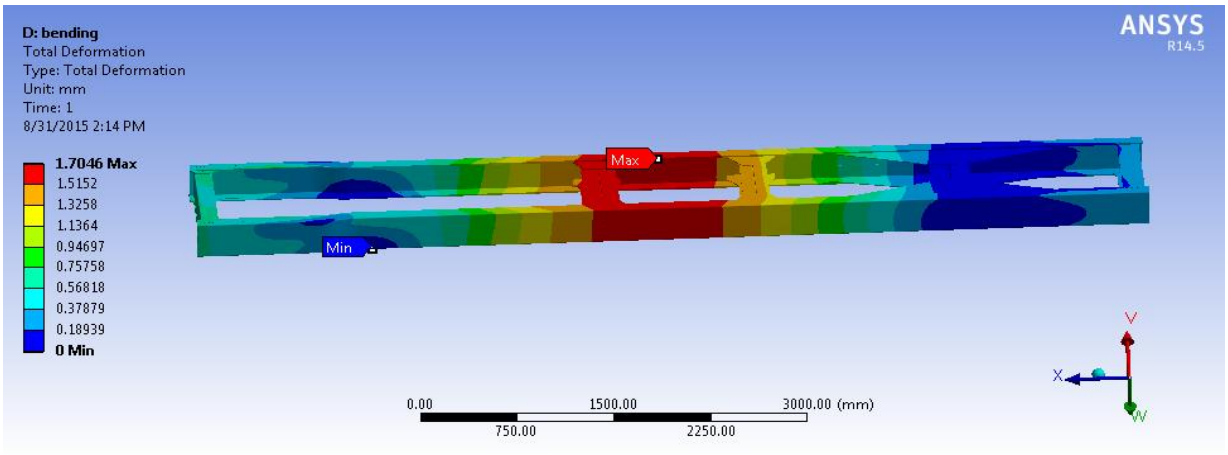

Fig 5.5.6 Total deformation of mild steel for bending case

The maximum deformation 1.7046mm and minimum deformation 0 mm is obtained at the middle of the frame and front fixed support respectively.

E) Stresses distribution of carbon epoxy for bending load case

The equivalent (von mises) stresses are considered for analyzing the frame and their respective stress distributions are as shown below

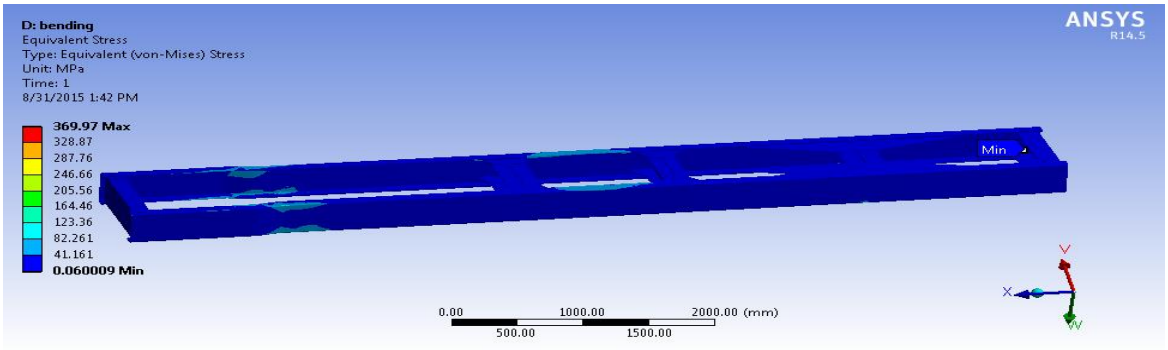

Fig 5.5.7 Equivalent vonmises stress of carbon epoxy for bending load case

The maximum equivalent stress of carbon epoxy made frame due to bending load case is 369.97MPa around the front fixed support. The ultimate strength of carbon epoxy is 1500MPa which is four times greater than the equivalent stress developed on the frame, so that the design is safe.

The factor of safety of carbo epoxy frame due to bending load case is;

FOS = 1500MPa/369.7MPa

FOS = 4.057

F) Deformation of carbon epoxy for bending case

The maximum deformation of carbon epoxy frame is 2.6364 mm at the middle of the frame.

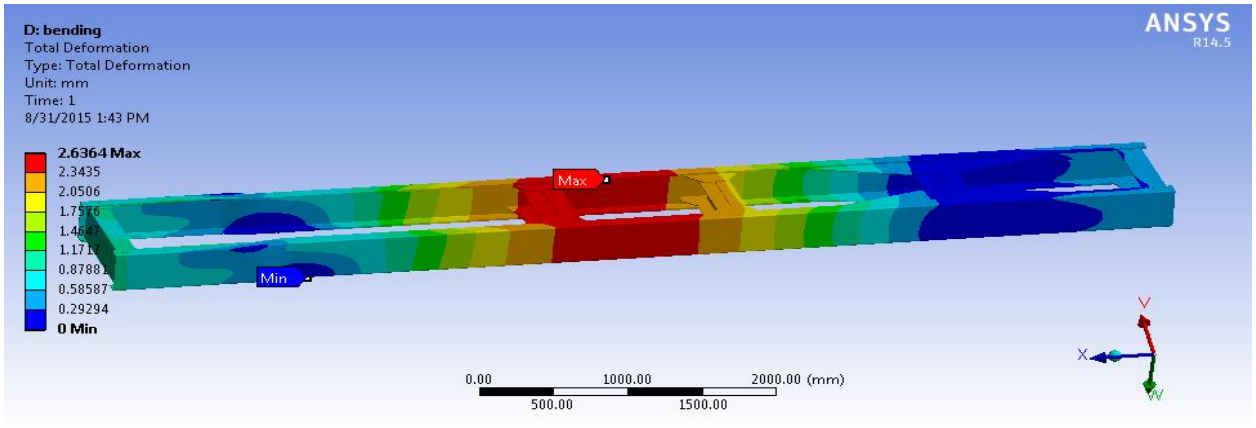

Fig 5.5.8 Total deformation of carbon epoxy for bending case

G) Stresses distribution of e glass epoxy for bending load case

The two types of stresses are considered for analyzing the frame and their respective stress distributions are as shown

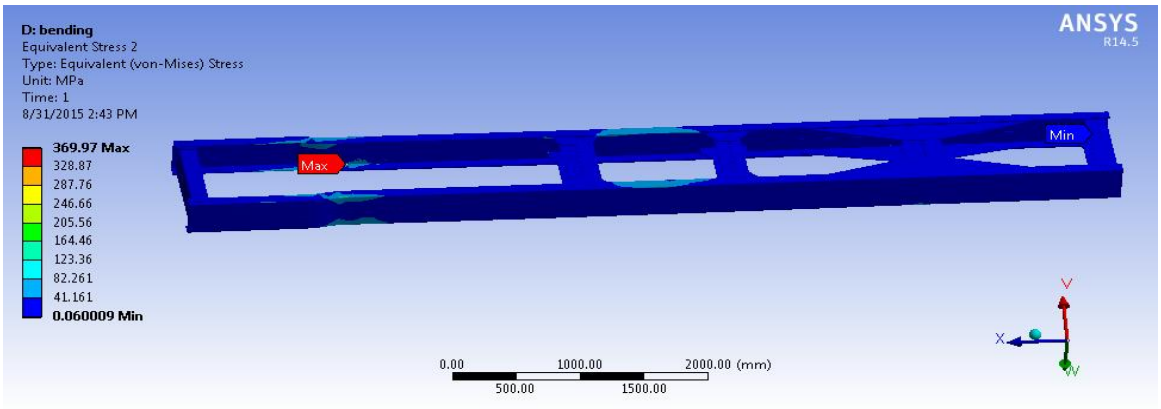

Fig 5.5.9 Equivalent stress of e glass epoxy for bending case

The equivalent stress developed on the e glass epoxy frame is maximum of 369.97 MPa and minimum of 0 MPa around front fixed support and at the rear end of the frame respectively. the ultimate strength of e glass epoxy is 1062 MPa which is three times greater than the stress developed in the frame.

The factor of safety of e glass epoxy chassis frame due to bending load case is;

FOS = 1062MPa/369.71MPa

FOS = 2.87

H) Deformation of e glass epoxy bending load case

The maximum deformation of e glass epoxy frame is 7.0656mm which is obtained at the middle and minimum at the front fixed support.

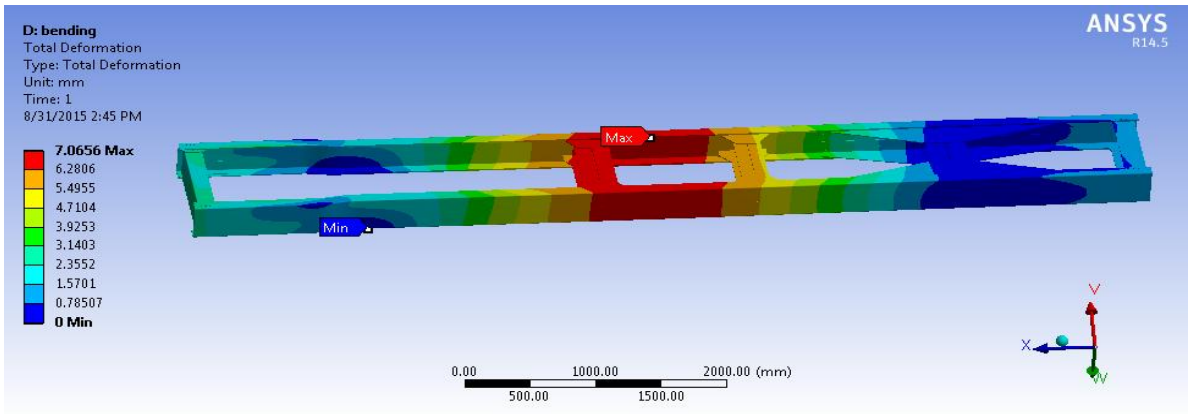

Fig 5.5.10 Total deformation of e glass epoxy for bending case

Table 5.5.1 Comparative Analysis of steel chassis and polymeric composite chassis for bending load cases.

S.No.	Parameters	steel	Carbon epoxy	E glass epoxy
1	Equivalent stress	369.97MPa	369.97MPa	369.97MPa
2	Deformation	1.7046 mm	2.6364 mm	7.0656 mm

4.5.4 Torsion load case

This simulation is based on the condition whereby one of the truck's front wheel rest on a hump, thus causing torsion to the chassis. This pure torsion load case could not occur in practice. However, it is important because it generates very different internal load in structure [7].

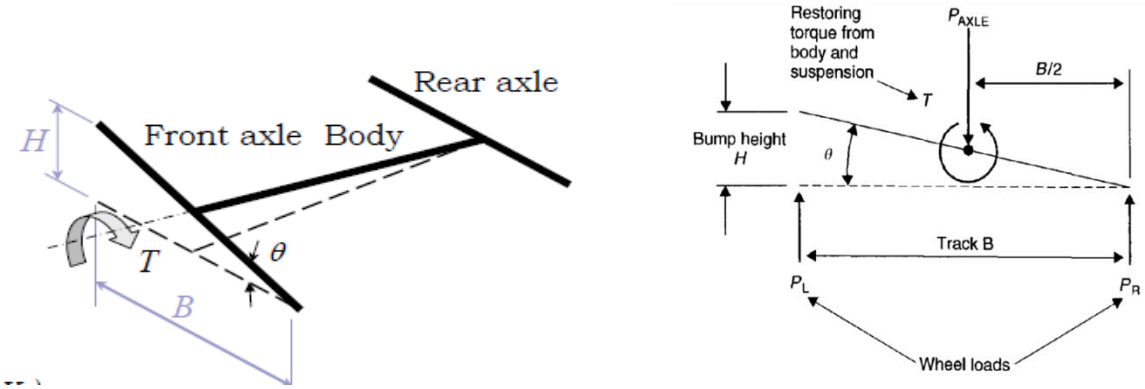

Fig 5.5.11 Torsional load and deformation pattern

Equilibrium equations, $T= (P_L-P_R) \times B/2$ (5.8)

$$P_{axle}=P_L+P_R \quad (5.9)$$

Torque T will reach a limit when right wheel lifts off, i.e., when P_R = 0.The front axle is the lightest load axle, from the manual the front axle weight is 6500 kg, then

$P_L=P_{axle}$ =6500 x 9.81= 63765 N

$$T=P_{axle} \times B/2 \quad (5.10)$$

$$\theta=H/B \quad (5.11)$$

The maximum bump height *Hmax* that cause the right wheel to lift off the ground is 200mm (pawlowski, 1969) [7].

A) Boundary condition and application of the load

Torsion load means if one of the wheels gets a bump and other wheels are at the level road [18]. As it is shown below the force is applied at the front axle of left wheel designated D and other three wheels are fixed. Fixed support means it is not allowed to move any direction, there is no translation and rotation designated as A, B and C.

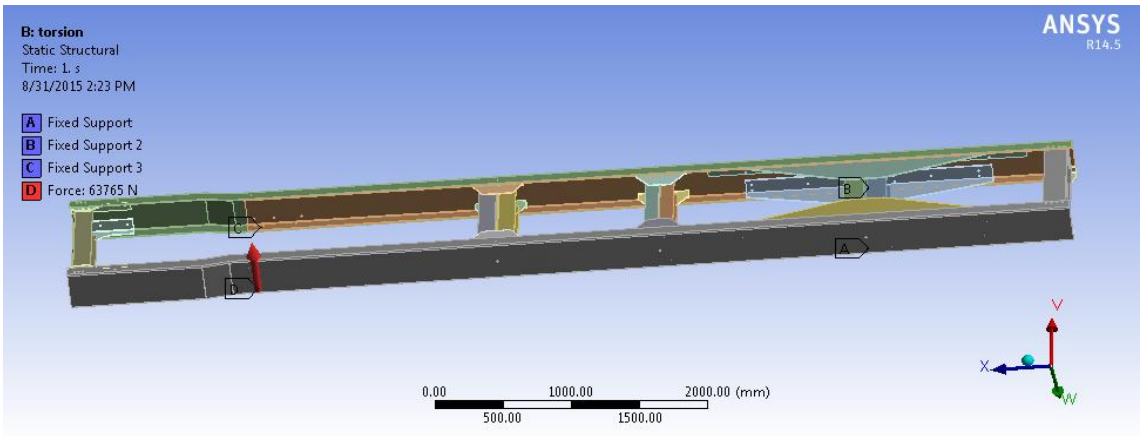

Fig 5.5.12 Application of the load and boundary condition for torsion case

B) Stress distribution of mild steel for torsion case

The maximum equivalent stress is 6076.7MPa at the front end of the frame rail imposed by the torsion load and the stress is minimum at the opposite rear end of the frame rail. This stress value is greater than the yield stress and ultimate strength of all three proposed materials. if the frame is subjected to this much amount of torsional load, even if it could not occur in practice, the frame will fail automatically.

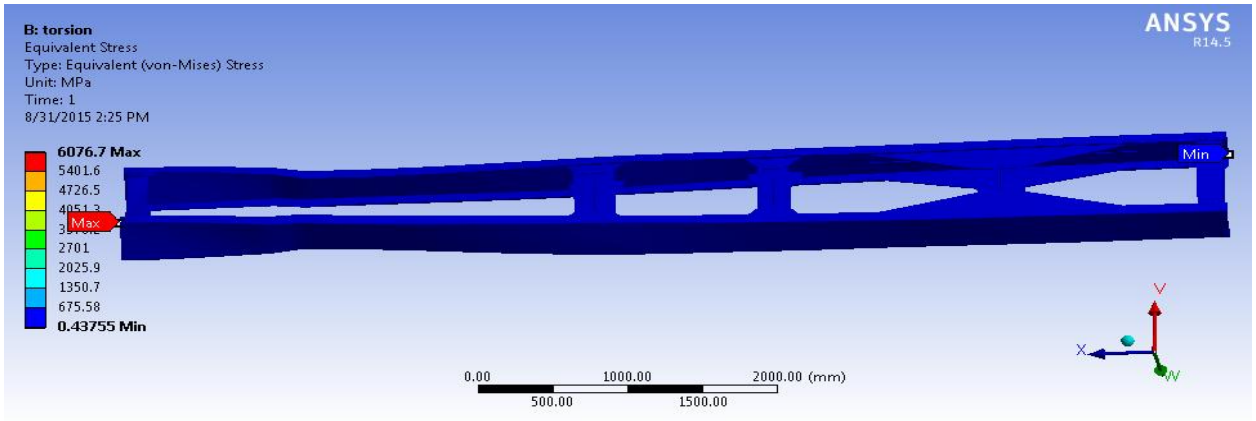

Fig 5.5.13 Equivalent von mises stress of mild steel for torsion case

C) Deformation of mild steel for torsion case

The maximum deformation is 52.559 mm obtained at the front end where the torsional equivalent stress is maximum.

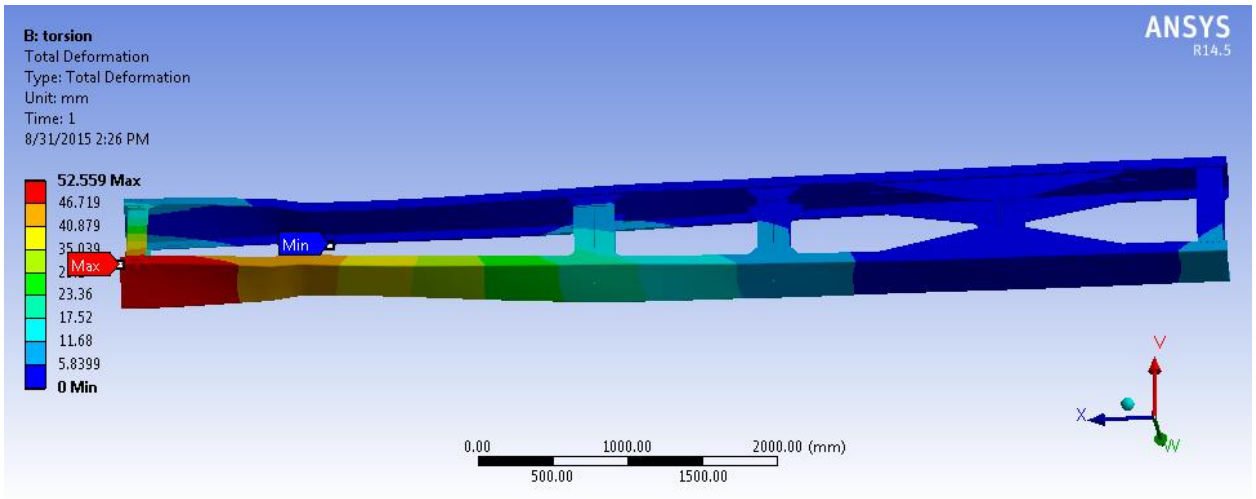

Fig 5.5.14 Total deformation of mild steel for torsion case

D) Stress distribution of carbon epoxy for torsion case

The maximum equivalent stress is 6076.7MPa at the front end and minimum at the rear end of the frame.

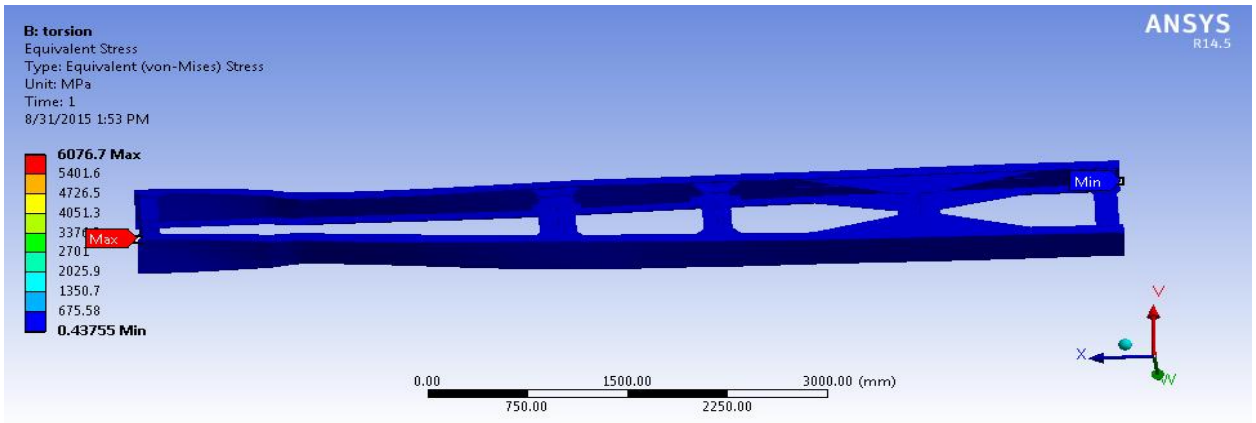

Fig 5.5.15 Equivalent von mises stress of carbon epoxy for torsion case

E) Deformation of carbon epoxy for torsion case

The maximum deformation is 81.192 mm at the front end of the frame on the side rail subjected to torsion load

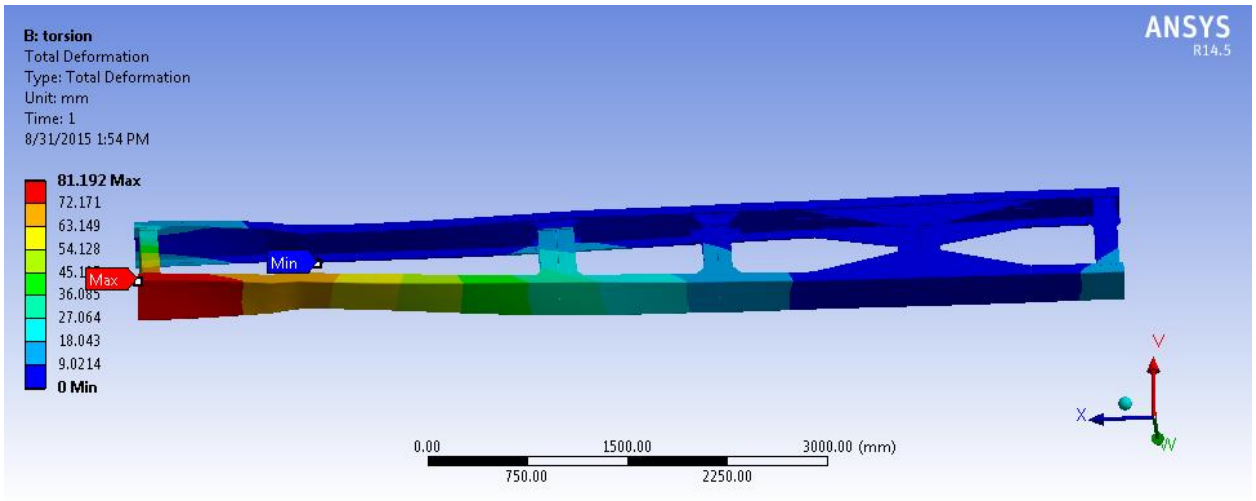

Fig 5.5.16 Total deformation of carbon epoxy for torsion case

. F) Stress distribution of e glass epoxy

The stress is maximum at the front end of the frame 6076.7MPa.

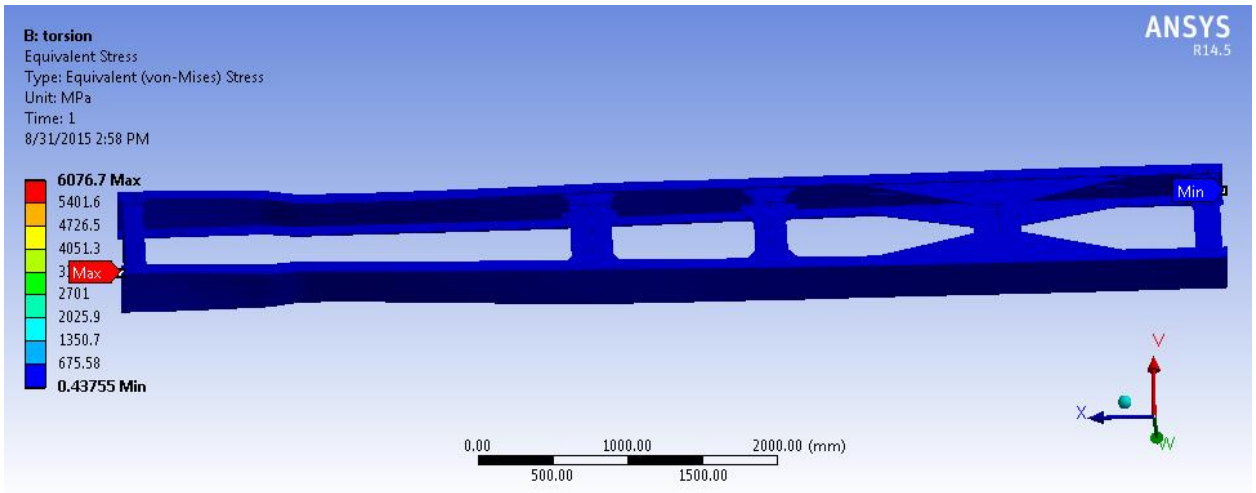

Fig 5.5.17 Equivalent von mises stress of e glass epoxy for torsion case

G) Deformation of e glass epoxy for torsion case

The deformation is maximum 217.6 mm at the front end of the frame at place where stress is maximum.

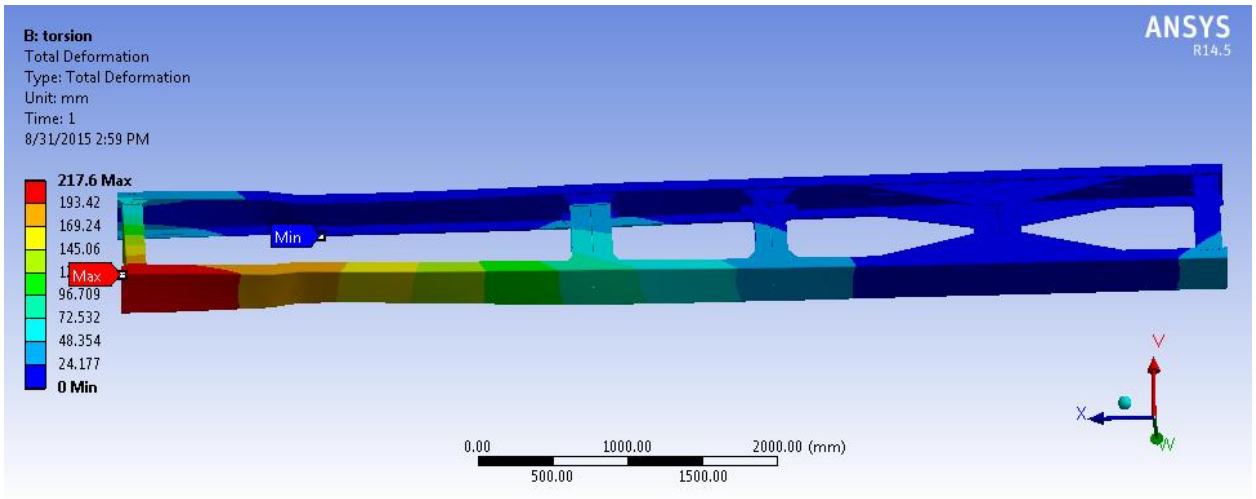

Fig 5.5.18 Total deformation of e glass epoxy for torsion case

Table 5.5.2 Comparative Analysis of steel heavy vehicle chassis and polymeric composite heavy vehicle chassis for torsion case

S.No.	Parameters	steel	Carbon epoxy	E glass epoxy
1	Equivalent stress	6076.7 MPa	6076.7 MPa	6076.7 MPa
2	Deformation	52.559 mm	81.129mm	217.6 mm

5.5.5 Combined bending and torsion load case

If the static loading cases of bending and torsion are combined the loading condition shown in Figure 4.5.8 will be achieved. This represents the situation arising if one wheel of the lighter loaded axle is raised on a bump of sufficient height to cause the other wheel on that axle to leave the ground. Pawlowski (1964) recommended that a maximum bump height of 200 mm should be considered as most cars have a suspension bump to rebound travel of 200 mm or less. The present writer considers the 200 mm bump height will lift the other wheel of the same axle off the ground. In this condition all the load of the lighter axle is applied to one wheel [7].

A) Boundary condition and application of the load for combined load cases

As it is shown from the figure below A, B and C are fixed supports at the wheel found at level ground. Torsion load is applied on the front axle left wheel 63,765 N at D where the wheel gets a hump. Bending load due to the weight of the components and payload is also applied at each side rails 156,960 kN at E and F

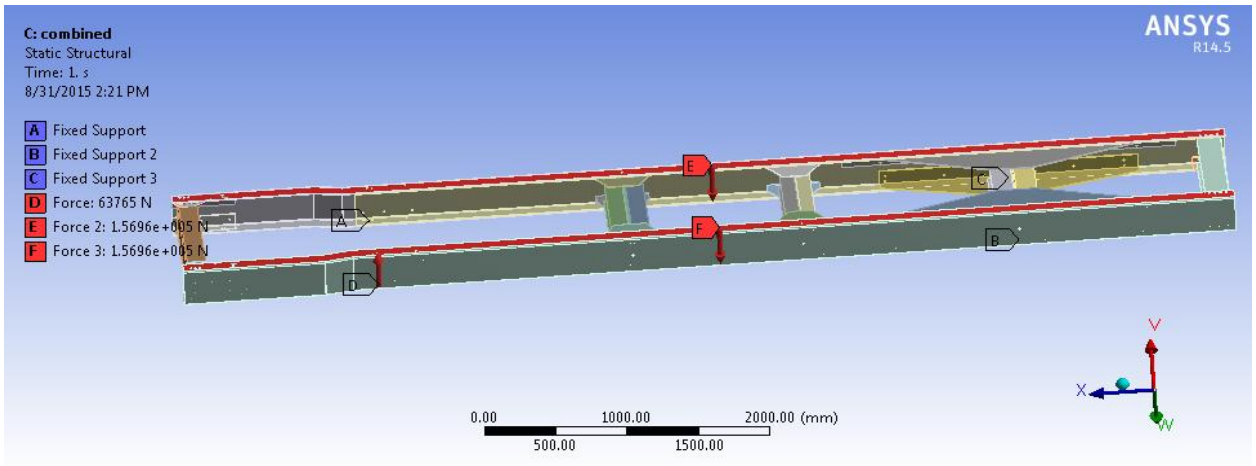

Fig 5.5.19 Application of the load and boundary condition for combined bending and torsion case

B) Stress distribution of mild steel for combined load case of mild steel

The maximum equivalent stress developed in the frame of mild steel is 1257.7MPa at the front end of the frame rail at the side torsion load is subjected. This magnitude of stress is greater than or beyond the yield strength and ultimate strength of the steel of 550 MPa and 620MPa respectively. So the frame will fail if it is subjected to this magnitude of torsion load, the design is no safe. For the design to be safe the tress developed on the frame has to be less than the yield strength of the material that the frame is made [7].

FOS = 550Mpa/1257.7MPa

FOS = 0.43 the design is not safe

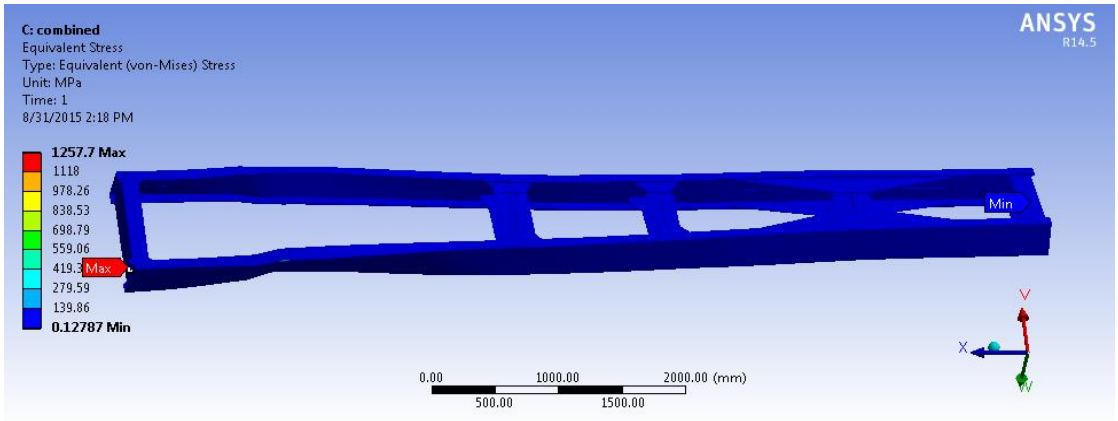

Fig 5.5.20 Equivalent von mises stress of mild steel for combined bending and torsion load case

C) Deformation of mild steel for combined load cases

The maximum deformation is 12.221mm at the front end of the frame where the stress is maximum.

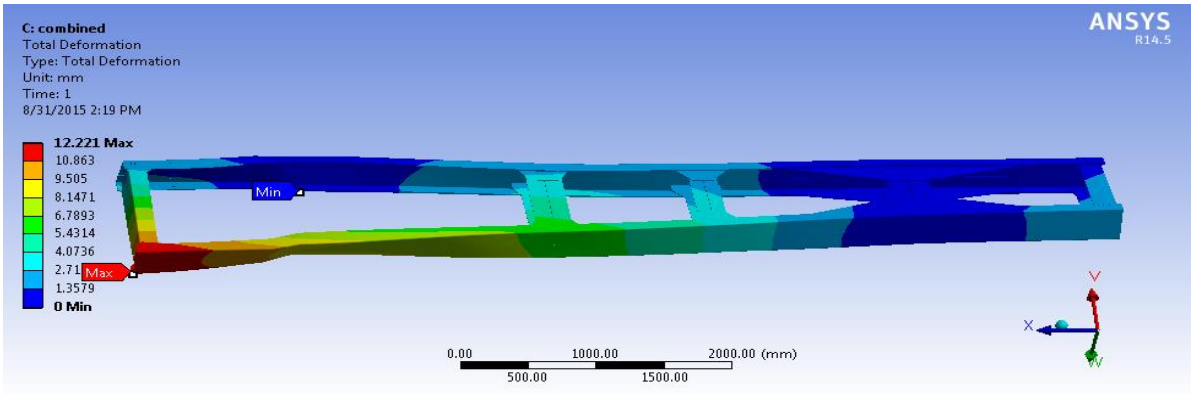

Fig 5.5.21 Total deformation of mild steel for combined bending and torsion load case

D) Stress distribution of carbon epoxy for combined load cases

The stress is maximum 1257.7Mpa at the front end and minimum at the rear end. Here the frame is made from carbon epoxy. The stress developed on the frame is less than the ultimate strength of carbon epoxy material which is 1500 MPa. So we can say the design is safe.

Factor of safety, FOS = 1500MPa/1257.7MPa

FOS = 1.19

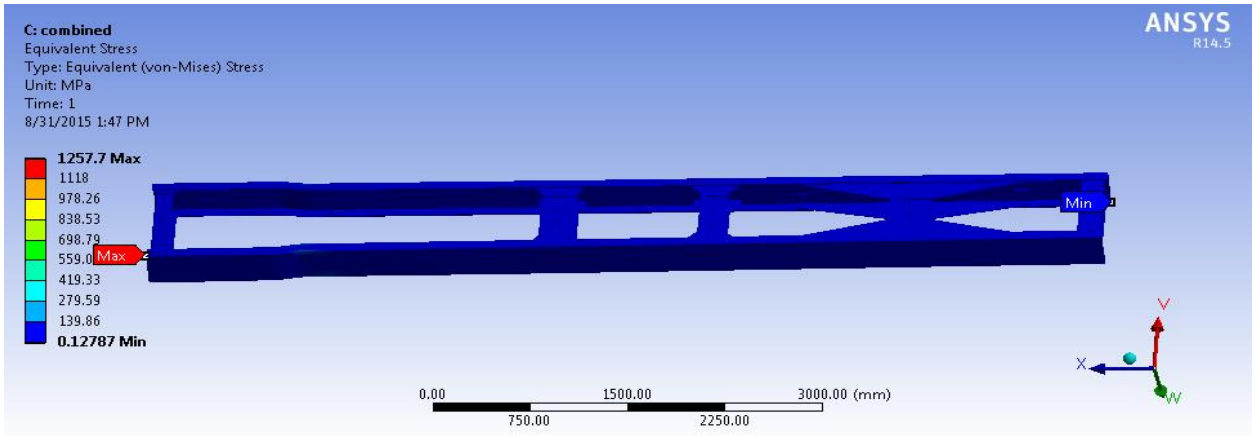

Fig 5.5.22 Equivalent von mises stress of carbon epoxy for combined bending ad torsion load case

E) Deformation of carbon epoxy for combined load cases

The maximum deformation resulted in a carbon epoxy frame is 18.878 mm at the front end of the frame where the torsional stress is maximum.

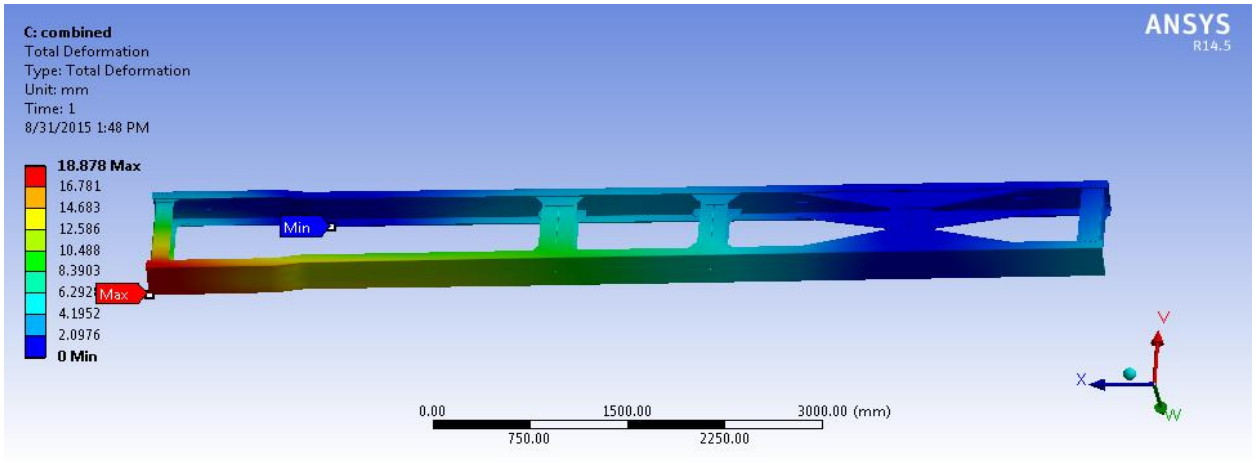

Fig 5.5.23 Total deformation of carbon epoxy for combined torsion and bending load case

F) Stress distribution of e glass epoxy for combined load cases

The stress developed on e glass epoxy frame is 1257.7MPa which is greater than the ultimate strength 1067 MPa of the e glass epoxy. If the torsion load reaches this magnitude the frame will fail automatically. So the design is unsafe.

FOS = 1067MPa/1257.7MPa

FOS = 0.85

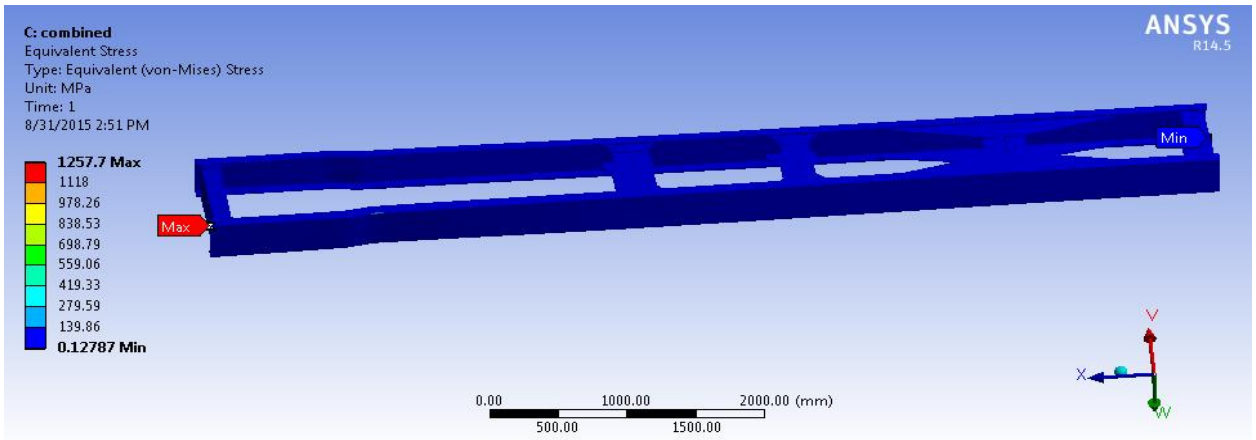

Fig 5.5.24 Equivalent vonmises stress of e glass epoxy for combined bending and torsion load case

G) Deformation of e glass epoxy for combined load cases

The maximum deformation obtained on the chassis frame due to combined load case is at the front end of side rail at which the torsion load is applied.

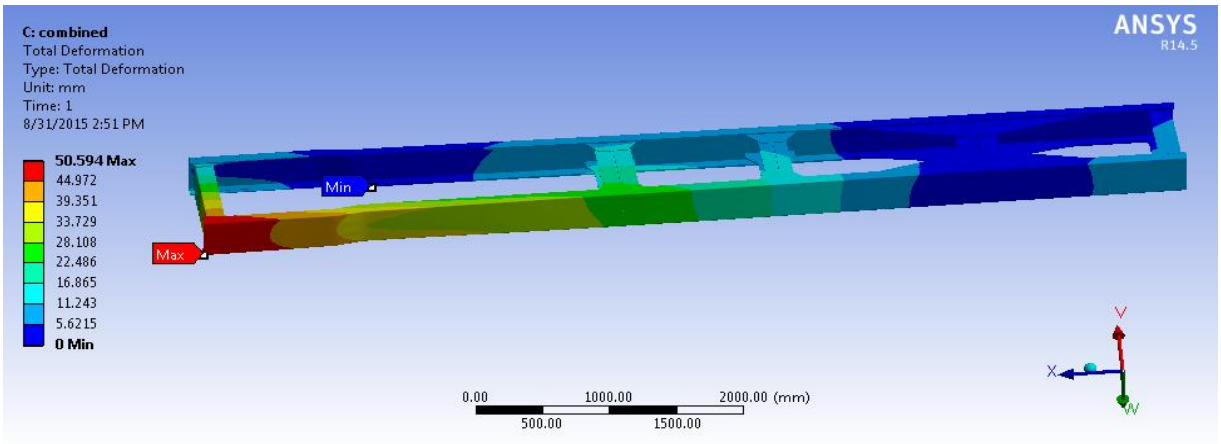

Fig 5.5.25 Total deformation of e glass epoxy for combined bending and torsion load case

From the linear static analysis of FAW truck chassis frame, for all three load cases it is obtained that the maximum stress area is around the front end, because the distance between the front two cross members (i.e. fifth and fourth cross members) is relatively high. There should be other cross member in between these two cross members to minimize the stress.

Table 5.5.3 Comparative Analysis of steel heavy vehicle chassis and polymeric composite heavy vehicle chassis for combined bending and torsion case

S.No.	Parameters	steel	Carbon epoxy	E glass epoxy
1	Equivalent stress	1257.7 MPa	1257.7 MPa	1257.7 MPa
2	Deformation	12.221 mm	18.878 mm	50.594 mm

4.6 Modal analysis

Truck chassis forms the structural backbone of a commercial vehicle. When the truck travels along the road, the truck chassis is excited by dynamic forces caused by the road roughness, engine, transmission and more. Modal analysis using Finite Element Method (FEM) can be used to determine natural frequencies and mode shapes. In this study, the modal analysis has been accomplished by the commercial finite element packaged ANSYS. Are natural frequencies of the modified chassis in suitable range? After constructing finite element model of chassis and appropriate meshing with tetrahedral elements, model has been analyzed and first 6 frequencies that play important role in dynamic behavior of chassis, have been expanded. In addition, the relationship between natural frequencies and engine operating speed has been explained. The results show that the road excitation is the main disturbance to the truck chassis as the chassis natural frequencies lie within the road excitation frequency range. For the modal analysis of the frame the boundary conditions are given as fixed support only. It is not required to give the other boundary conditions like loading and acceleration due to gravity [1].

The dynamic response of simple structures, such as uniform beams, plates and cylindrical shells, may be obtained by solving their equations of motion. However, in many practical situations either the geometrical or material properties vary, or the shape of the boundaries cannot be described in terms of known mathematical functions. Also, practical structures consist of an assemblage of components of different types, namely beams, plates, shells and solids. In these situations it is impossible to obtain analytical solutions to the equations of motion. This difficulty is overcome by seeking some form of numerical solutions and finite element methods. Automotive industry is one of the biggest users of the technology of modal analysis [15].

The modal behavior of car chassis is a part of most necessary information for the inspection into car's dynamic behavior. In this thesis the modal analysis of truck chassis has been studied. As a

truck travels along the road, the truck chassis is excited by dynamic forces induced by the road roughness, engine, transmission and more. Under such various dynamic excitations, the truck chassis tends to vibrate [1].

Whenever the natural frequency of vibration of a machine or structure coincides with the frequency of the external excitation, there occur a phenomenon known as resonance, which leads to excessive deflections and failure. The literature is full of accounts of system failures brought about by resonance and excessive vibration of components and systems. The global vibrational characteristic of a vehicle is related to both its stiffness and mass distribution. The frequencies of the global bending and torsional vibration modes are commonly used as benchmarks for vehicle structural performance. Bending and torsion stiffness influence the vibrational behavior of the structure, particularly its first natural frequency. The mode shapes of the truck chassis at certain natural frequencies are very important to determine the mounting point of the components like engine, suspension, transmission and more [1].

In general the natural frequency can be calculated using the equation (1)

$$\omega = \sqrt{k/m} \tag{5.12}$$

Where k and m stand for stiffness and mass respectively Diesel engine is known to have the operating speed varying from 800 to 5000 revolutions per minute (rpm), this translates in to excitation frequencies varying from 13.33 to 83.33 rpm. In low speed idling condition, the speed range is about 800 to 1000 rpm. This translates into excitation frequencies varying from 13.33 to 16.567 Hz. The main excitations are at high speeds, when the car is in the last gear. The natural frequency of the car 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 car chassis when the car travels along the road. In practice, the road excitation has typical values varying from 0 to 100 Hz, [1]. At high speed cruising, the excitation is about 3000 rpm or 50 Hz, [1]. Mounting of vibration components of the car on the nodal point of the chassis is one of the vibration attenuation methods to reduce the transmission of vibration to the car chassis [24].

Regarding the previous discussions about the diesel engine speed, we can say that natural frequencies are in critical range. Hence with decreasing the chassis length which increases the chassis stiffness, we increase the natural frequencies to place them in the appropriate range. In addition, with changing the gasoline tank situation and performing similar changes, we can prevent coinciding the simulation force frequencies and natural frequencies. Otherwise resonance phenomenon occurs and if these two frequencies coincide, this phenomenon destroys the chassis. As a reminder it is mentionable that validity of the results has been verified by comparing the results from a similar model with the model proposed by Mr Foui and his cooperator [24].

Therefore it is important to include the dynamic effect in designing the chassis. Many researchers carried out study on truck chassis. In their method in addition to simulating truck with finite element packaged ANSYS and being sure that structure vibration modes are in appropriate range, they vibrationally analyzed it. This paper deals with 32,000kg truck chassis that includes natural frequencies and mode shapes.

For three materials mild steel, carbon epoxy and e glass epoxy modal analysis of chassis frame is performed to determine the natural frequency and mode shapes. The finite element analysis result of modal analysis for each material is as follow.

The finite element method (FEM) governing equation for modal analysis is shown below;

$$[M]\{\ddot{q}\} + [K]\{q\} = \{0\} \tag{5.13}$$

For a linear system, the responses (displacements) will be of harmonic form. Thus,

$$\{q\} = \{Q_i\}\sin(\omega_i t) \tag{5.14}$$

Where, $\mathbf{Q_i}$ is amplitude of mode shape of i^{th} natural frequency, ω_i

Substituting in the equation above

$$(-\omega^2[M] + [K])\{Q_i\} = \{0\} \tag{5.15}$$

This gives

$$|-\omega^2[M] + [K]| = 0 \tag{5.16}$$

5.6.1 Natural frequencies of mild steel frame

Table 5.6.1 The natural frequencies of mild steel frame

Mode	Frequency [Hz]	Displacement
1.	31.977	Bending about y-axis
2.	33.786	Lateral bending about z-axis
3.	50.435	Bending about y-axis
4.	64.816	Twisting about x-axis and bending about y-axis
5.	65.151	Twisting about x-axis
6.	75.885	Twisting about x-axis and bending about y-axis

Heavy duty diesel engine works in range of 800 to 5000 rpm engine speed. The equation used to change engine speed to frequency is

$$\omega_e = N/60 \quad (5.17)$$

The engine excitation frequency range is in between 13.33Hz - 83.33Hz. Due to the engine the main excitation is when the engine runs 2000rpm and above. At low speed there is no risk of vibration due to engine excitation. Road excitation is in range of 0 - 100Hz. Excitation from the road is the main disturbance [1].

5.6.2 Deformation and mode shapes of mild steel frame

a) First mode shape of mild steel

The maximum translation is 1.9662mm obtained at the front end of the frame.

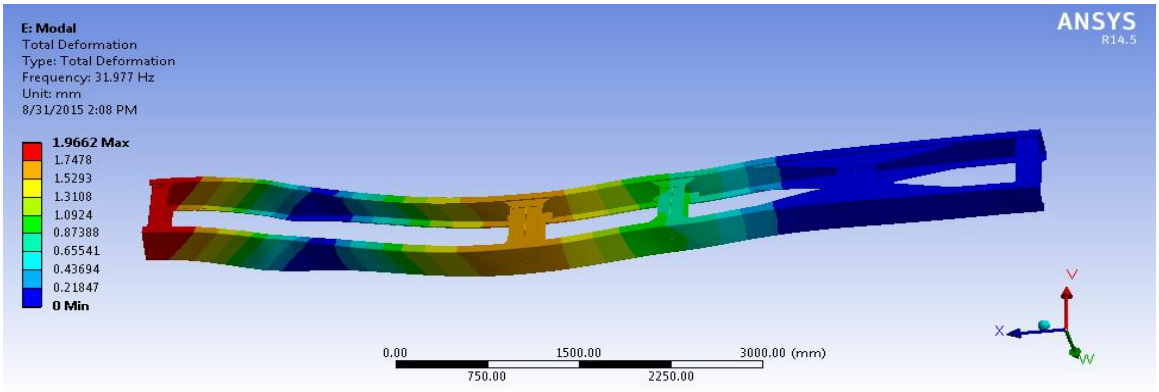

Fig 5.6.1 First mode shape of mild steel

b) Second mode shape of mild steel

The maximum translation is 3.1806mm obtained at the front end of the frame.

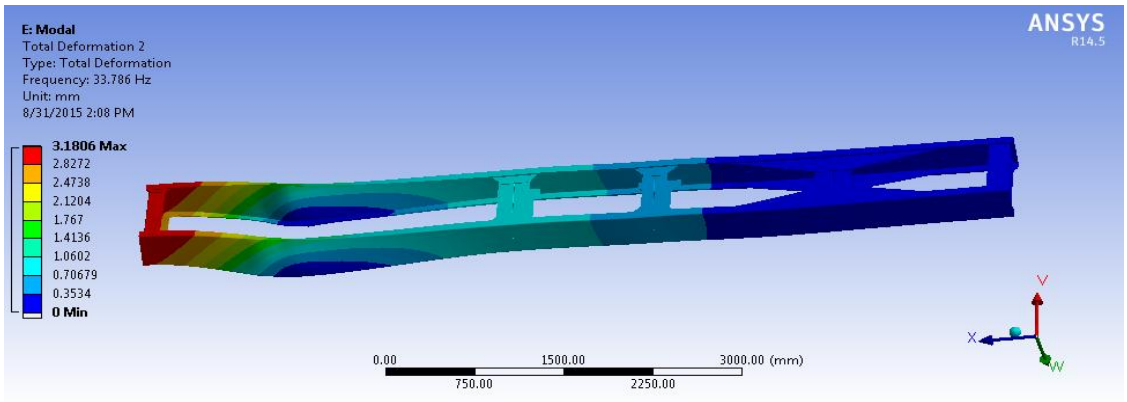

Fig 5.6.2 Second mode shape of mild steel

c) Third mode shape of mild steel

The maximum translation is 1.6627mm obtained at the rear end of the frame.

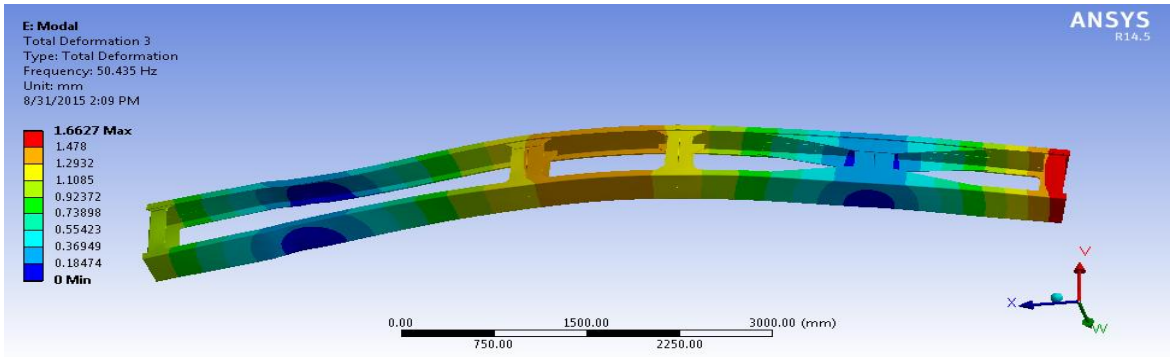

Fig 5.6.3 Third mode shape of mild steel

d) Forth mode shape of mild steel

The maximum translation is 3.3048mm obtained at the front end of the frame.

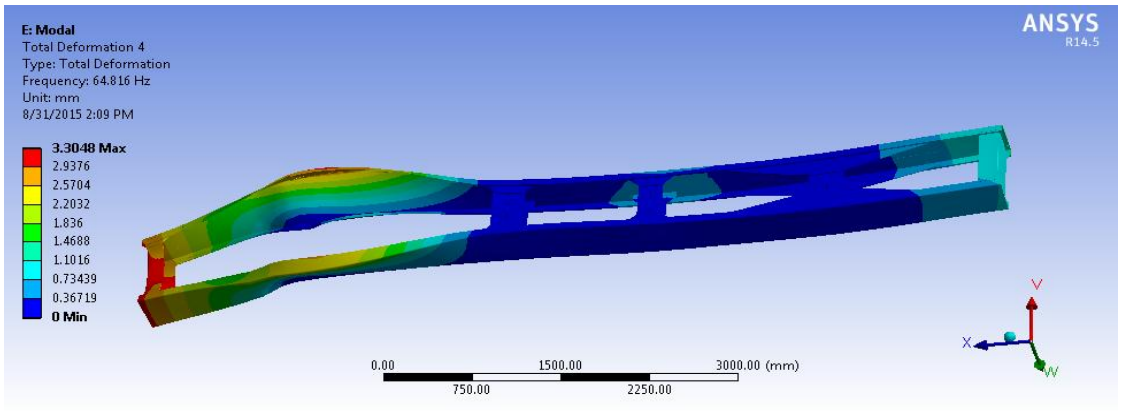

Fig 5.6.4 Forth mode shape of mild steel

e) Fifth mode shape of mild steel

The maximum translation is 2.3534mm obtained at the rear end of the frame.

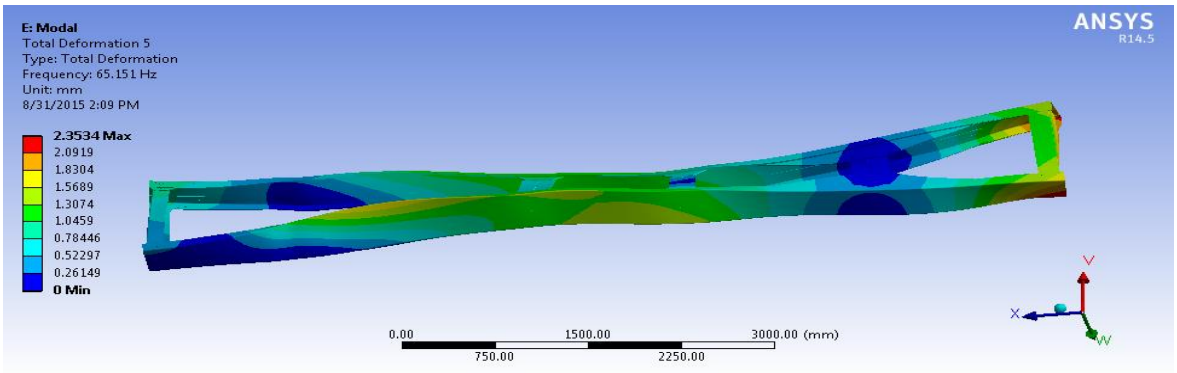

Fig 5.6.5 Fifth mode shape of mild steel

f) Sixth mode shape of mild steel

The maximum translation is 3.2579mm obtained around the front end of the frame.

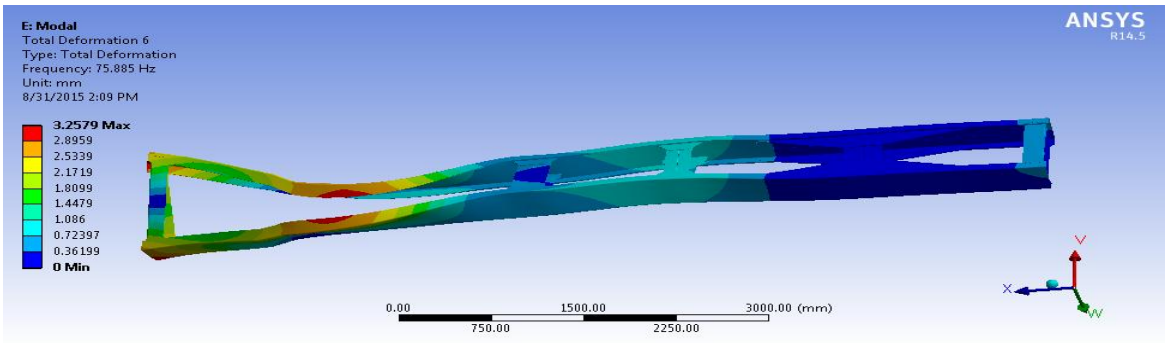

Fig 5.6.6 Sixth mode shape of mild steel

5.6.3 Natural frequencies of carbon epoxy frame

Table 5.6.2 The natural frequencies of carbon epoxy frame

Mode	Frequency [Hz]	Displacement
1.	56.806	Bending about y-axis
2.	60.02	Lateral bending about z-axis
3.	89.595	Bending about y-axis
4.	115.14	Twisting about x-axis and bending about y-axis
5.	115.74	Twisting about x-axis
6.	134.81	Twisting about x-axis and bending about y-axis

The natural frequency of carbon epoxy is greater than mild steel. The first two natural frequencies 56.8 and 60.03 are critical, means they have a probability to coincide with engine excitation. Here also road excitation is the main disturbance. So the main excitation is disturbance due to engine is at high speed.

5.6.4 Deformation and Mode shapes of carbon epoxy frame

a. First mode of carbon epoxy

The maximum translation is 4.3413mm obtained around the front end of the frame.

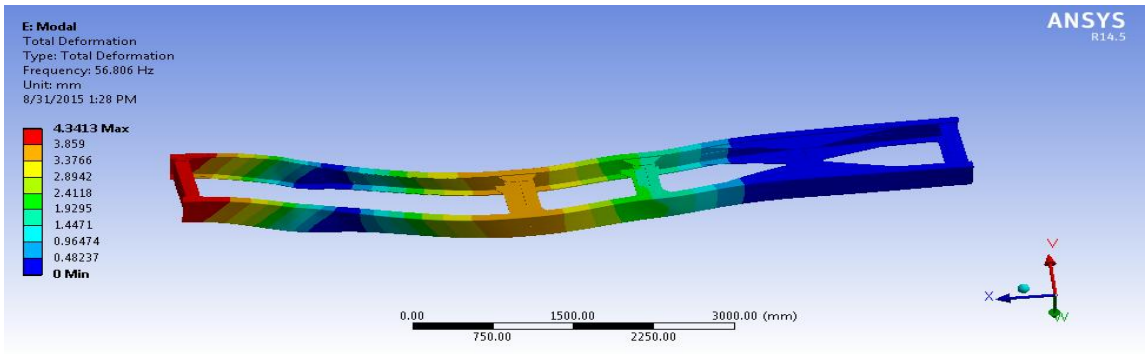

Fig 5.6.7 First mode of carbon epoxy

b. Second mode shape of carbon epoxy

The maximum translation is 7.0225mm obtained around the front end of the frame.

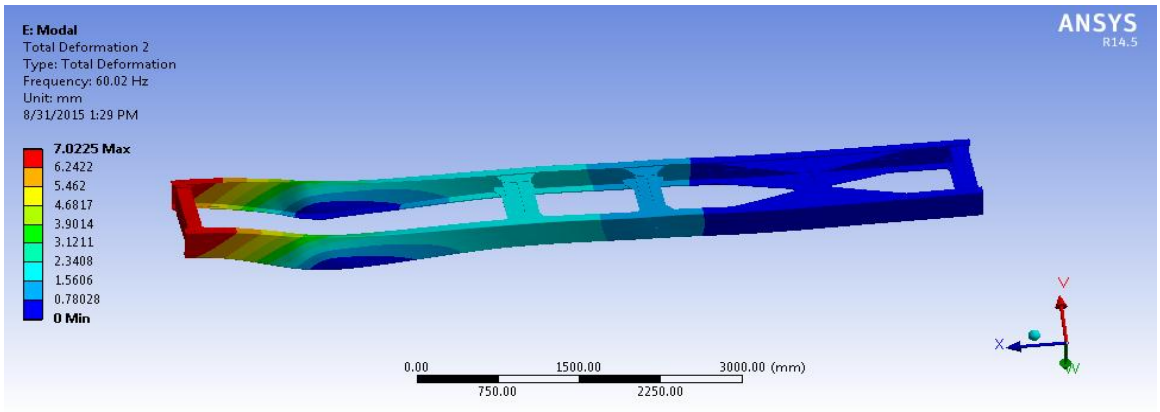

Fig 5.6.8 Second mode shape of carbon epoxy

c. Third mode shape of carbon epoxy

The maximum translation is 3.6711mm obtained around the rear end of the frame.

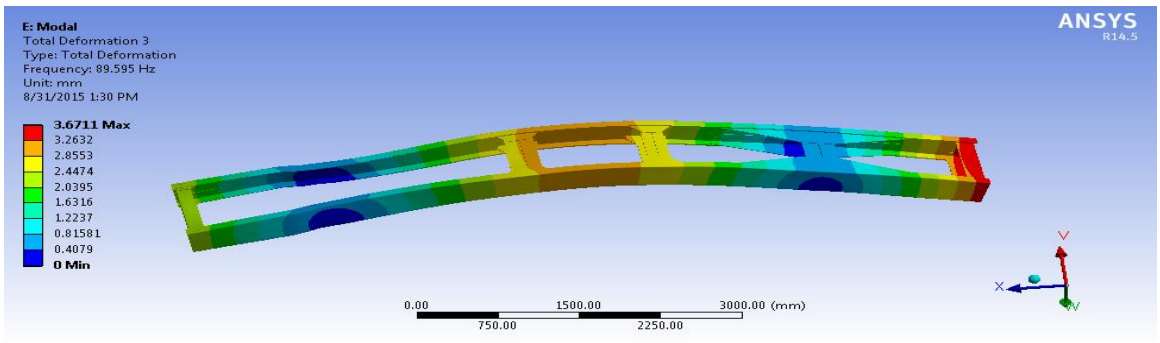

Fig 4.6.9 Third mode shape of carbon epoxy

d. Fourth mode shape of carbon epoxy

The maximum translation is 7.2967mm obtained around the front end of the frame

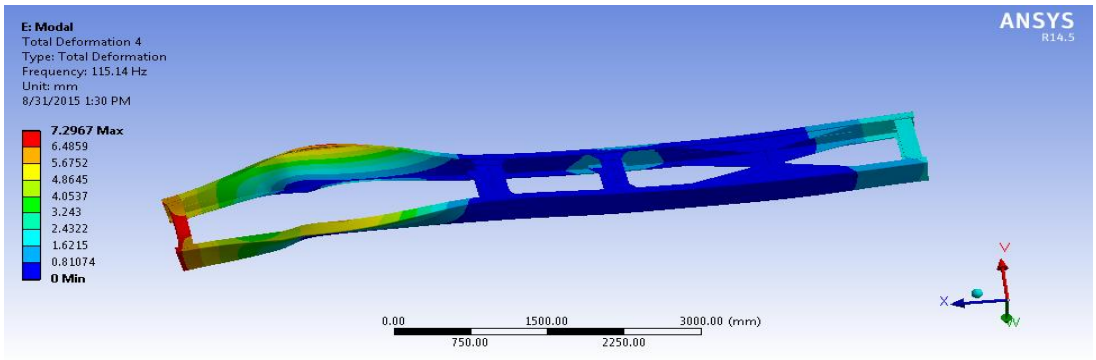

Fig 5.6.10 Fourth mode shape of carbon epoxy

e. Fifth mode shape of carbon epoxy

The maximum translation 5.1961mm obtained around the front end of the frame.

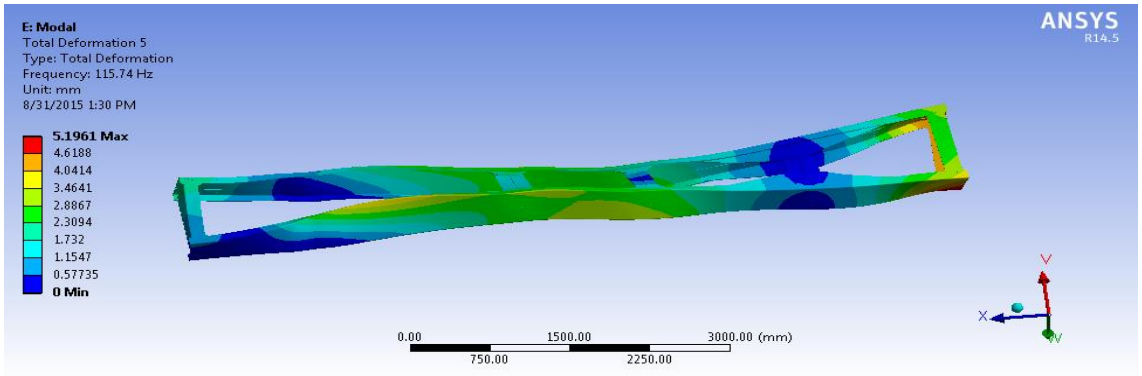

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Fig 5.6.11 Fifth mode shape of carbon epoxy

f. Sixth mode shape of carbon epoxy

The maximum translation is 7.1932mm obtained between fourth and fifth cross members.

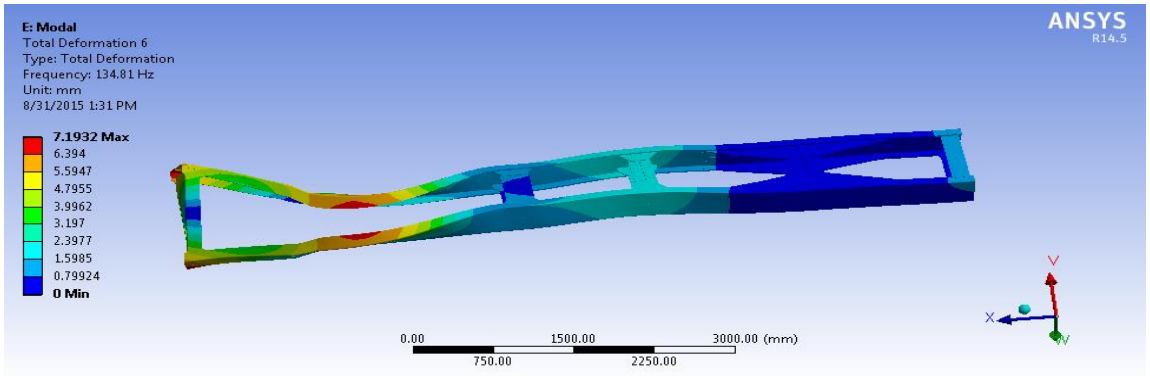

Fig 5.6.12 Sixth mode shape of carbon epoxy

5.6.5 Natural frequencies of e glass epoxy frame

Table 5.6.4 The natural frequencies of carbon epoxy frame

Mode	Frequency [Hz]	Displacement
1.	27.221	Bending about y-axis
2.	28.761	Lateral bending about z-axis
3.	42.933	Bending about y-axis
4.	55.175	Twisting about x-axis and bending about y-axis
5.	55.46	Twisting about x-axis
6.	64.597	Twisting about x-axis and bending about y-axis

At low speed the vibration is less duet engine excitation but at high speed it is high. because engine excitation frequency is in between 13.33 to 83.33 Hz, taking the engine speed range is 800 to 5000rpm. Here also road excitation is the main disturbance.

5.6.6 Deformation and mode shapes e glass epoxy frame

a) First mode shape of e glass epoxy

The maximum translation is 3.4056mm obtained at the front end of the frame.

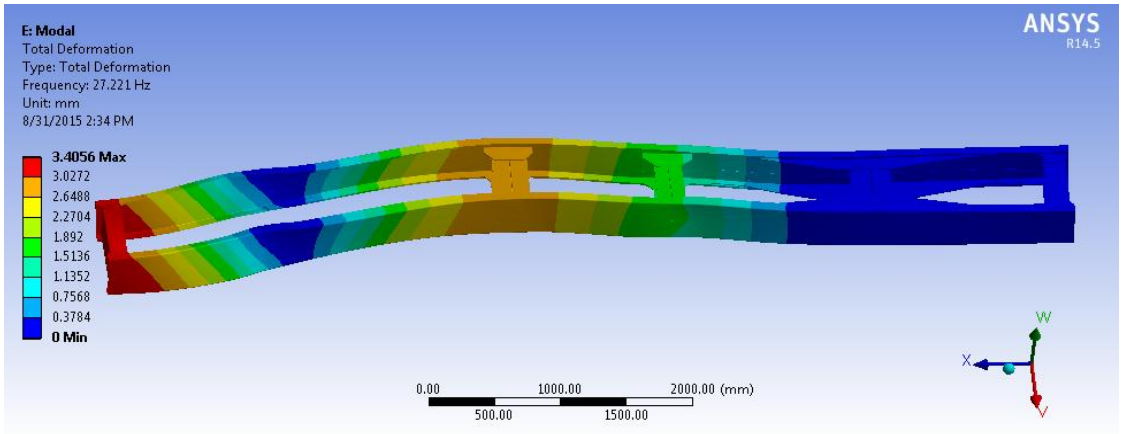

Fig 5.6.13 First mode shape of e glass epoxy

b) Second shape of e glass epoxy

The maximum translation is 5.5089mm obtained at the front end of the frame.

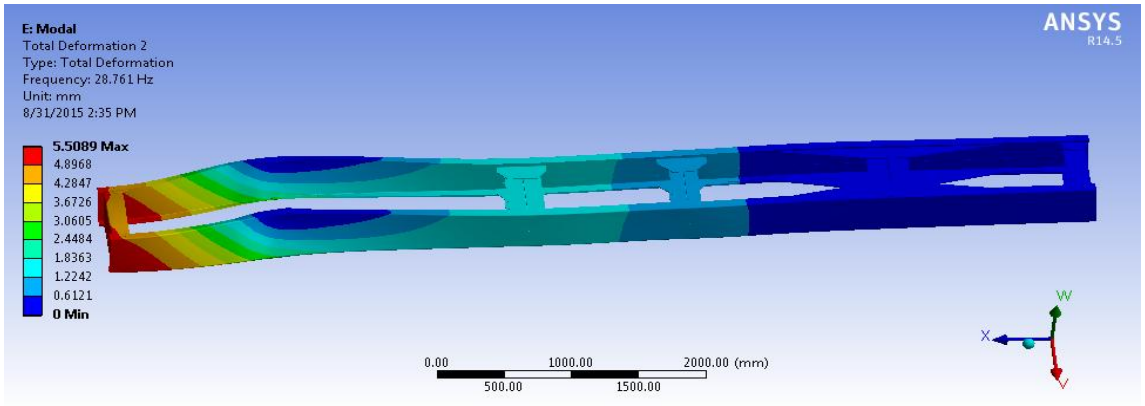

Fig 5.6.14 Second mode shape of e glass epoxy

c) Third mode shape of e glass epoxy

The maximum translation is 2.8799mm obtained at the rear end of the frame.

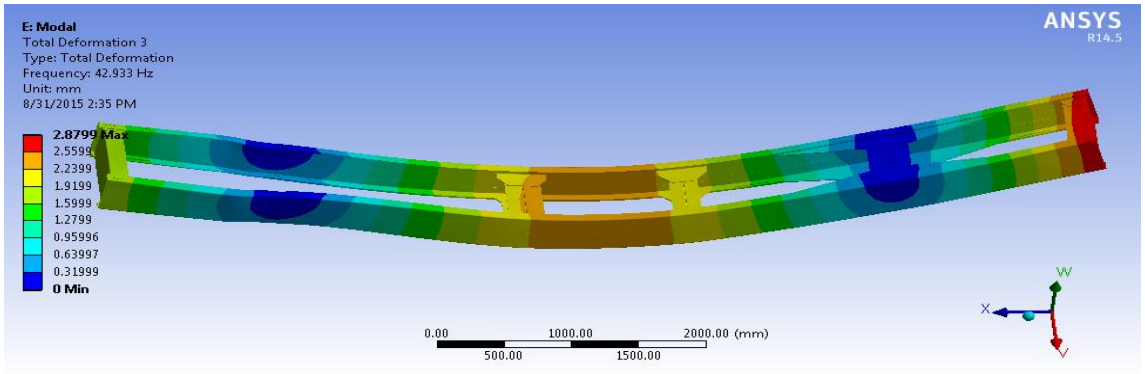

Fig 5.6.15 Third mode shape of e glass epoxy

d) Fourth mode shape of e glass epoxy

The maximum translation is 5.724mm obtained at the front end of the frame.

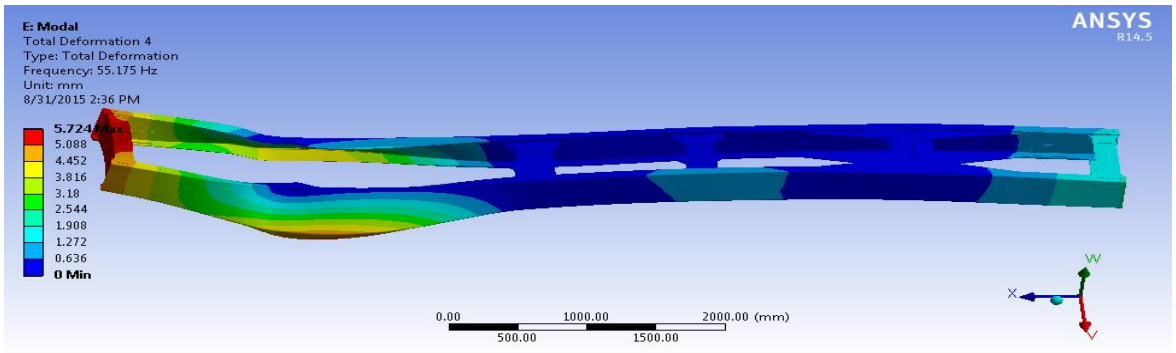

Fig 5.6.16 Fourth mode shape of e glass epoxy

e) Fifth mode shape of e glass epoxy

The maximum translation is 4.8762mm obtained at the front end of the frame.

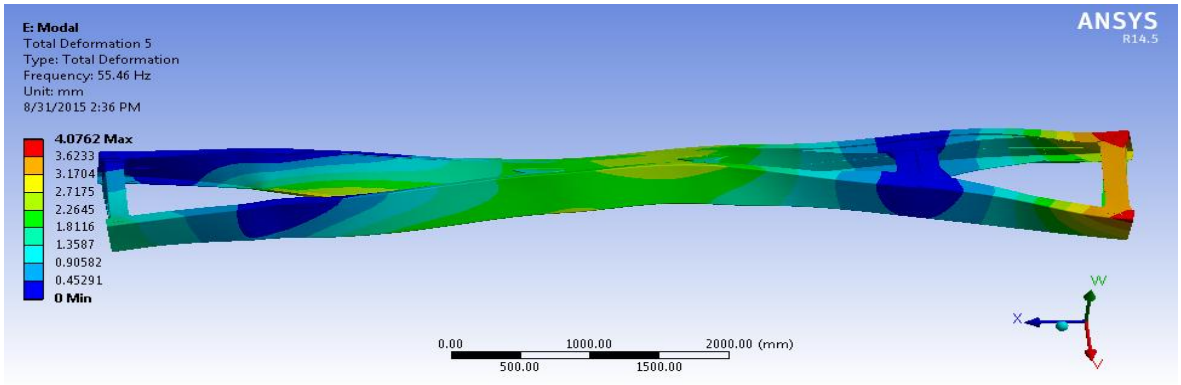

Fig 5.6.17 Fifth mode shape of e glass epoxy

f) Sixth mode shape of e glass epoxy

The maximum translation is 5.6428mm obtained around the front end of the frame.

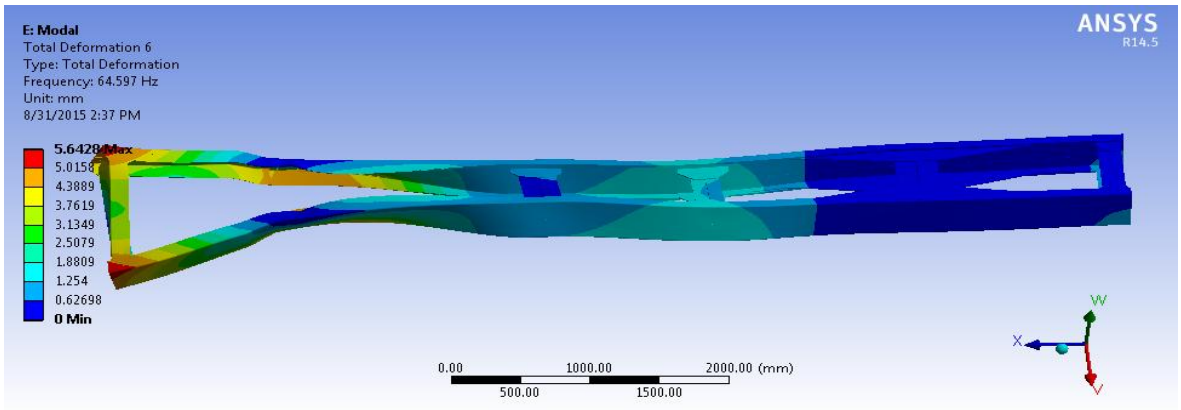

Fig 5.6.18 Sixth mode shape of e glass epoxy

From the modal analysis of frame made of three materials mild steel, e glass epoxy and carbon epoxy individually, most of the maximum displacements are obtained at the front. From the analysis result carbon epoxy frame has less vibration as compared to e glass epoxy and mild steel frame.

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusion

In this work, a heavy vehicle chassis frame is modeled using CATIA V5 R19. Static structural and modal analysis is done on the chassis frame using ANSYS work bench 14.5. The main intention of the thesis is static and modal analysis of FAW dump truck chassis frame using three materials steel, carbon epoxy and e-glass epoxy. Present used material for chassis is steel. The static analysis has been done for three load cases namely bending, torsion and combined torsion and bending case. The finite element analysis result shows that the equivalent stress for steel, carbon epoxy, and e glass epoxy due to bending load case, torsion load case and the combination of bending and torsion load case is 367.97MPa, 6076.7MPa and 1257.7MPa respectively.

Table 6.1.1 Equivalent stress and deformation of chassis frame due to static load cases

Materials	Equivalent stress (MPa)			Total deformation (mm)		
	Bending load case	Pure torsion load case	Combined bending and torsion load case	Bending load case	Pure torsion load case	Combined bending and torsion load case
Mild steel	367.97	6076.7	1257.7	1.7046	52.559	12.221
Carbon epoxy	367.97	6076.7	1257.7	2.6364	81.192	18.878
E glass epoxy	367.97	6076.7	1257.7	7.0656	217.6	50.594

Although the magnitude of stress developed on the frame of each material for a particular load case is the same, the total deformation value is different. For all load cases with the same stress the maximum deformation is obtained on e glass epoxy frame and less deformation on mild steel frame. The deformation of carbon epoxy is less than e glass epoxy and greater than steel, because composite materials have less stiffness than steel but strength and stiffness to weight ratio is

higher. A material that has more stiffness exhibits less deformation. Carbon epoxy and e glass epoxy has a greater factor of safety than steel, that means the maximum stress developed in thus two polymeric composite materials is less and less as compared to their yield ultimate strength.

For the linear static analysis, the stress distribution and deformation profile of the truck chassis subjected to three loading conditions have been determined. Due to bending load case the maximum stress of the truck chassis is occurred at the front where fixed support is applied, while the maximum translation is at the middle between third and forth cross members. These values are acceptable as compared to the yield strength of the three chassis materials and the tolerance allowed for the chassis.

Due to torsion load cases the maximum stress and translation or deformation of truck chassis occurred at the front end of side rail where the torsion load is applied. These values are not acceptable as compared to the yield strength of each three chassis material.

Due to combined bending and torsion load case the maximum stress or deformation of truck chassis occurred at the front end of side rail where the torsion load is applied. These values are not acceptable for steel and e glass epoxy material of the chassis as compared to their yield strength. For carbon epoxy the maximum stress is acceptable as compared to its yield strength. Carbon epoxy is the best material that performs well when it is subjected to bending and combined bending and torsion cases except pure torsion load case.

I have considered composites Carbon Epoxy and E – Glass Epoxy for chassis material. By observing structural analysis results, the stress values for Carbon Epoxy and E Glass Epoxy are less than their respective allowable stress values or ultimate strength. So using composites for chassis is safe. By using composites instead of steel, the weight of the chassis reduces 4.8 times for carbon epoxy and 3 times glass epoxy because density of steel is more than the composites.

By employing a polymeric composite heavy vehicle chassis for the same load carrying capacity, there is a reduction in weight of 79.48% for carbon epoxy and 66.67% for e glass epoxy**.** From the results, it is observed that the polymeric composite heavy vehicle chassis is lighter and more economical than the conventional steel chassis with similar design specifications. Material

substitution is the best way to reduce weight of the vehicle so as to reduce fuel consumption and emission which is the global issue.

Fuel consumption is an important factor for the customer. Although the fuel consumption will benefit the customer, it will help the manufacturer to sell the car. In addition, higher fuel consumption will result higher CO2 emission. For fuel consumption, it has been assumed that 10% weight reduction results to 5% fuel efficiency.

Table 6.1.2 weight comparison of steel and composite materials

Materials	Weight (kg)	Saved weight(kg)	Weight reduction in percent (%)
Mild steel	1148.9	-	-
Carbon epoxy	235.67	913.23	79.48
E glass epoxy	382.96	766.24	66.67

From the modal analysis result all the first six natural frequencies of steel and e glass epoxy chassis frame are in a range of engine excitation frequencies at high speed when the engine is working above 2000rpm.The natural frequency of steel varies in between 31.97Hz and 75.88Hz and for e glass epoxy it varies in between 27.22Hz and 64.59Hz. So there is a probability to resonate. For carbon epoxy frame only the first two natural frequencies 56.8Hz and 60.02Hz respectively are in engine excitation frequency range of 13.33 to 83.33Hz. The natural frequencies of carbon epoxy in ranges of 56.8Hz and 134.8Hz are greater than steel and e glass epoxy. The frame that has a natural frequency greater than 100Hz is a vibration free. because all excitation frequencies from engine, transmission and road roughness are in between 0 and 100Hz. The main disturbance is road excitations. Carbon epoxy is better than other materials because it has less risk of vibration and good load carrying capacity with light weight as compare to steel and e glass epoxy.

From both static and modal analysis result, it is obtained that the maximum deformation and stresses are obtained at around the front part of the chassis frame. This is because the distance between the front two cross members is 3160 mm which is a large distance. So in order to reduce

and minimize the stress and deformation from around the front end, there should be reinforcement on the side rail or additional cross member between fourth and fifth cross members especially around front fixed supports.

6.2 Recommendations

Due to of time constraint complete analysis could not be done in this thesis. I would like to recommend the future researchers should continue to work in this direction and try to reduce the problems of automotive industries in Ethiopia.

Here in this thesis only static and modal analyses are performed. The research can be extended in the future in the following areas.

1. Fatigue analysis to determine the fatigue life of chassis and factor of safety for all static loading conditions such as bending, torsion and combined bending and torsion. Because fatigue is more critical as far as structural failure is concerned.

2. Dynamic analysis that includes transient, harmonic and random vibration analysis.

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APPENDIX

A. Actual model of chassis

Figure A1. FAW dump tuck displayed for sale in Bishoftu Automotive Industry

Figure A2. Semi- finished FAW Dump Truck in assembly room in Bishoftu Automotive Industry

Figure A3 FAW dump truck frame on the assembly stand the cabin is mounted on the frame in Bishoftu Automotive Industry.

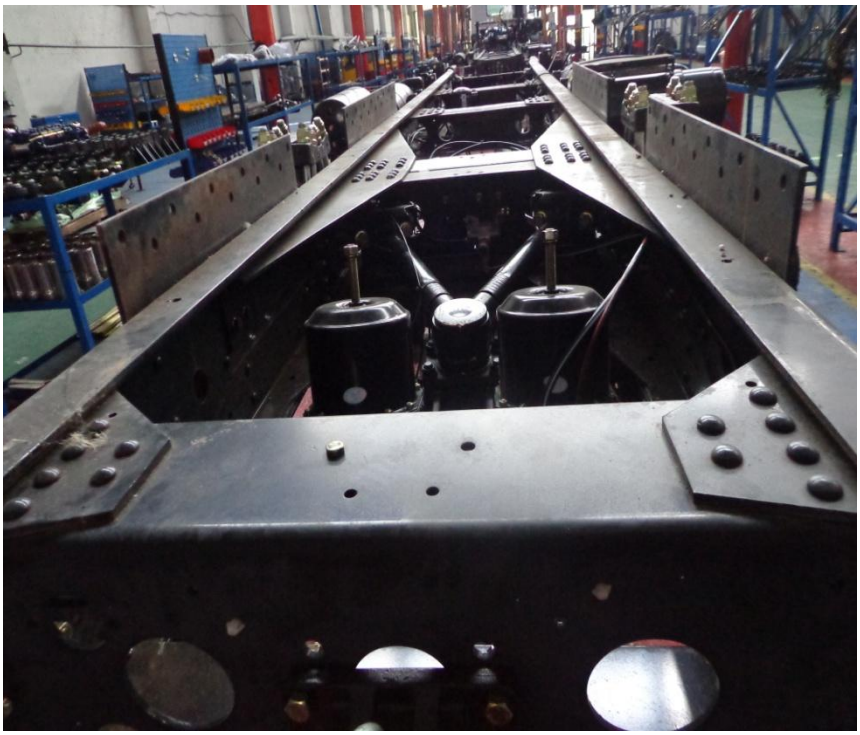

Figure A4 the frame on the stand in bishoftu automotive industry

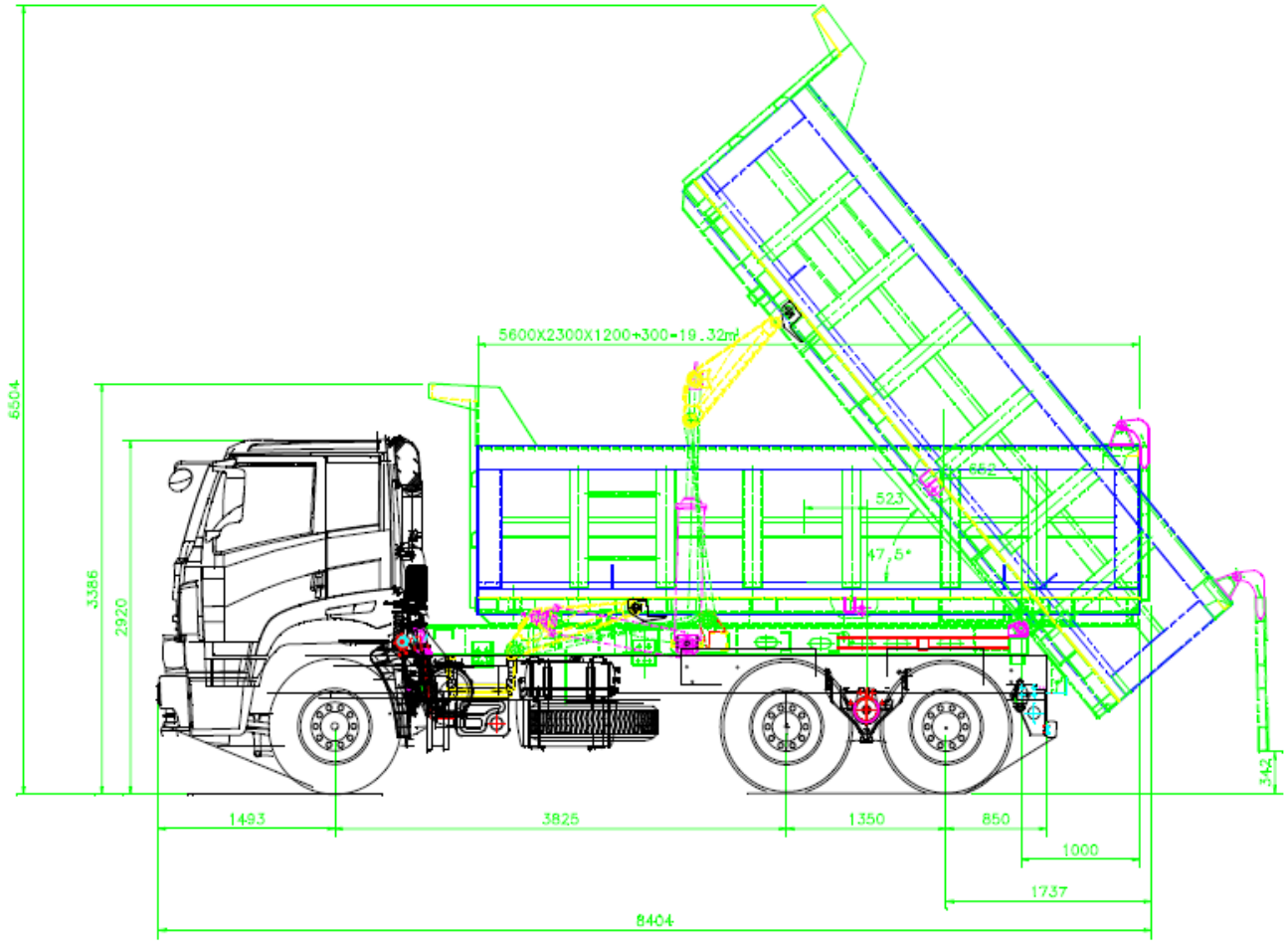

Figure A5. The overall dimension of FAW dump truck taken from the manual in Bishoftu automotive industry