PRE-STRESSED DYNAMIC SIMULATION OF TRUCK CHASSIS

Thesis submitted in partial fulfilment of the requirements for the degree of

Master of Engineering in Automobile Engineering

By

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CERTIFICATE OF RECOMMENDATION

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This foregoing thesis entitled "**Pre-Stressed Dynamic Simulation of Truck Chassis**" is hereby approved as a credible study of an engineering subject carried out and presented in a manner satisfactory to warrant its acceptance as a prerequisite to the degree for which it has been submitted. It is understood that by this approval the undersigned does not endorse or approve any statement made, opinion expressed, or conclusion drawn there in but approves the thesis only for the purpose for which it has been submitted.

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NOMENCLATURE

$[\boldsymbol{M}]$	=	Mass matrix
$[\boldsymbol{K}]$	=	Stiffness matrix
$[\boldsymbol{C}]$	=	Damping matrix
$[\boldsymbol{S}]$	=	Change in Stiffness matrix
$\{\boldsymbol{u}\}$	=	Displacement vector
$\{\boldsymbol{F}\}$	=	Time varying load vector
$\{\boldsymbol{F}^{nd}\}$	=	Time varying nodal load vector
$\{\boldsymbol{F}^{s}\}$	=	Modal analysis load vector
$\boldsymbol{\sigma}_{var}$	=	Variable stress
$\boldsymbol{\sigma}_{mean}$	=	Mean stress
$\boldsymbol{\sigma}_{e}$	=	Endurance stress
$\boldsymbol{\sigma}_{ut}$	=	Ultimate stress
$\boldsymbol{\omega}$	=	Angular velocity
$\boldsymbol{\omega}_{n}$	=	Natural frequency
$\boldsymbol{\phi}$	=	Phase angle
$\boldsymbol{\zeta}$	=	Damping ratio
$\boldsymbol{r}$	=	Transmissibility ratio
$\boldsymbol{n}$	=	Ratio of input frequency vs natural frequency
$\boldsymbol{N}$	=	Number of cycles
$\boldsymbol{a}$	=	Fatigue constant
$\boldsymbol{b}$	=	Fatigue constant

ABSTRACT

In this thesis work, a theoretical and comparative dynamic analysis of a ladder frame truck chassis has been done. After thorough literature study, it was found that comparatively less emphasis was given to the prestressed analysis while assessing the dynamic performance of a vehicle. On Indian roads, as seen, trucks and many heavy-duty vehicles tend to operate higher than their allowable payload limit. In this paper, static stress analysis and pre-stressed modal analysis were performed on TATA 1612 chassis model to see if significant changes occur to the natural frequencies. The chassis model was simulated using ANSYS 16.0 and three materials, namely. Structural steel, Aluminum alloy, and Carbon fibre were considered for the analysis. It was found that the effect of pre-stressing is more significant as the density of the material reduces and is most effective for aluminum alloy-made chassis. Also, it was observed that the change is more prevailing in the lower order mode frequencies for all the materials. The resultant change in frequencies as obtained in our analysis is significant, up to 39% in the case of structural steel and 50% in the case of aluminum. The load increment analysis results showed that the values increase or decrease with the increment of pre-stressing and few mode frequencies were tending to align themselves and approach the road profile frequency or engine operating frequency range. The harmonic analysis has been performed to locate the point of maximum stress concentration and deformation for both load free and pre-stressed chassis, also calculating the critical stress value which affects chassis' dynamic characteristics. Based on the overall maximum stress value on the chassis in base payload as well as overloading, fatigue analysis has been performed to show that although the chassis design can endure an infinite life, the fatigue life is reduced by a factor of 10 when loading is increased by 40%.

1. INTRODUCTION

1.1 OVERVIEW OF CHASSIS

1.1.1 WHAT IS CHASSIS?

Chassis is the main structural element of automobile which supports the loading from various sources. In a vehicle, the chassis acts as the main supporting structure, similar to the skeleton of a living organism. It bears all the stresses on the vehicle in both static and dynamic conditions. Every vehicle has a chassis frame, irrespective of their structure and purpose [1].

The Chassis has the following major functions:

- It endures the weight of the vehicle body and other vehicle components mounted on it.
- Supports the load of driver, passengers as well as luggage.
- Acts as a barrier for the stresses arising due to bad road conditions to the vehicle body.
- Withstands stresses during braking and acceleration of the vehicle.

1.1.2 TYPES OF CHASSIS

In general, the chassis can be classified into four categories.

Ladder frame chassis: It is one of the most primitive form of chassis currently in use [2]. The ladder frame chassis has got its name from its structure, which looks like a ladder. During earlier times when technology was not in its full boom, the simple structure of these chassis made it successful. Generally, it has two longitudinal beams which are supported by short/cross beams. Among the several advantages of this type of chassis, it is easily manufactured. This simpler structure also helps in mounting the components much easily. These types of chassis are heavy and generally used in vehicles like trucks, trailers, etc.

Advantages

- Easier to assemble as parts can be easily put in.
- Easier to fix as parts are not permanently attached.
- Construction method makes it durable and tough.

Disadvantages

- The ladder chassis does not have a strong torsional rigidity which makes it bad for cornering.
- It is heavy in weight and not ideal for passenger cars.

Monocoque Chassis: Monocoque is French word for shell. This type of chassis has an unibody construction where a shell-like structure is constructed around the vehicle and is connected to the base plate. This type of chassis is mainly in demand for passenger vehicles nowadays.

Advantages

- It has superior torsional rigidity.
- It has a cage-like construction which makes it safer that other chassis frames.

Disadvantages

- Both chassis and frame are joined as unibody structure which makes it heavy.
- Cannot be mass produced due to difficulty in manufacturing.

Backbone Chassis: Generally, this type of chassis has two wheelbases connected by a long rectangular cylindrical structure, which looks like a backbone. This cylindrical structure also acts as the drive shaft of the vehicle, which can also be a disadvantage[3].

Advantages

- A good torsional rigidity allows it to withstand more twists than ladder chassis.
- The driveshaft is covered by the chassis.
- Good for off-roading.

Disadvantages

- As the driveshaft is covered, it becomes hard for repairing and maintenance purpose.
- The manufacturing of backbone chassis is expensive

Tubular chassis: also known as space frame chassis, tabular chassis provides unparallel safety to the passenger or the driver. Due to its three-dimensional construction, these were an upgraded form of ladder frame chassis. Tubular chassis are rarely used on passenger cars.

Advantages

- Better rigid body dynamics than other chassis considering same weight.
- Best choice for race cars due to lightweight and better rigidity than other chassis.
- Offers the best weight/rigidity ratio out of all four types.

Disadvantages

- Tubular chassis are complex structures and need to be made manually.
- They are hard to manufacture and cannot be mass-produced.

1.2 FINITE ELEMENT ANALYSIS – OVERVIEW

1.2.1 WHAT IS FEA?

FEA is a general numerical method where we can solve complex geometrical model by dividing them into simpler, easy to comprehend parts that are called finite elements [4]. This process of dividing and breaking up the system is done by space discretization, generating the mesh of the object. This develops finite number of points or nodes which acts as numerical domain for the analysis. These nodes are assigned with their particular algebraic equations which are later grouped in to a large system of equations and solved for the entire model.

1.2.2 ADVANTAGES OF FEA

The process of subdividing the entire system in smaller domains and finite elements has many advantages.

- Complex geometry which is difficult to solve analytically can be broken down into simple parts accurately.
- Any major point of interest, such as stress points can be further analysed separately.
- Nonlinear effects of material can be accurately analysed.

- Real life testing of a system can be minimized, making the manufacturing process cost effective.

1.3 DYNAMIC ANALYSIS

Finite element analyses play an important role in assessing the performance of a structural component (for e.g., chassis) in real life. We ascertain the loading conditions and run simulation on the FE model and at the end, outputs like displacement, stress, plastic strain, life cycles etc are obtained.

The loading conditions are applied in static as well as in dynamic conditions of the body. Types of analysis performed during dynamic conditions of the body are Modal Analysis, Harmonic Analysis, Transient response analysis, Random Vibration analysis etc.

1.3.1 MODAL ANALYSIS

Modal analysis is the process of studying and analysing the natural frequencies of a component to understand its dynamic characteristics. Mode shapes are obtained through modal model equations to study various effects such as deformation, stress etc in case of a vibration input is given to the system. The matrix form of modal analysis [5] is given in Equation 1.1.

$$[\boldsymbol{M}]\{\ddot{\boldsymbol{u}}\} + [\boldsymbol{K}]\{\boldsymbol{u}\} = \{\boldsymbol{0}\} \qquad \textbf{\textit{1.1}}$$

1.3.2 HARMONIC ANALYSIS

Harmonic analysis deals with dynamic response of a system under periodic loading, considering the input in frequency domain. Whenever a system is excited with sinusoidal input, the output is also sinusoidal with different phase and amplitude. The matrix form of harmonic analysis [5] is given in Equation 1.2.

$$[\boldsymbol{M}]\{\ddot{\boldsymbol{u}}\} + [\boldsymbol{C}]\{\dot{\boldsymbol{u}}\} + [\boldsymbol{K}]\{\boldsymbol{u}\} = \{\boldsymbol{F}\} \qquad \textbf{\textit{1.2}}$$

{F} is a time varying load vector given by Equation 1.3,

$$\{F\} = \{F^{nd}\} + s\{F^{s}\} \quad \textit{1.3}$$

Where,

$\{F^{nd}\}$ = time varying nodal force

s = load vector scale factor

$\{F^{s}\}$ = load vector from the modal analysis

1.3.3 TRANSIENT RESPONSE ANALYSIS

Transient analysis deals with the response of a system in time domain. Transient means something that faces with time. Both loading and response is in time domain instead of frequency domain.

1.3.4 RANDOM VIBRATION ANALYSIS

Random vibration analysis deals with any inputs which is random in nature, for e.g., a speed breaker or pavement in case of an automobile. In this analysis, both input and output are statistical in nature.

1.4 FATIGUE ANALYSIS

Most of the failure in engineering components occurs due to fatigue. Fatigue occurs when a body experiences cyclic loading which is below its yield stress, for a period of time. Depending on the value of stress and the number of cycles the components the body can withstand before failure, fatigue can be categorized into three levels [6].

- Low cycle fatigue (when number of cycles is between 0 to 10^3)
- High Cycle Fatigue (when number of cycles is between 10^3 to 10^6)
- Infinite Fatigue Life (when number of cycles is greater than 10^6)

The maximum value of loading for which the component can withstand infinite fatigue life, is called endurance stress. A sample S-N curve has been given in Figure 1.1 for better understanding.

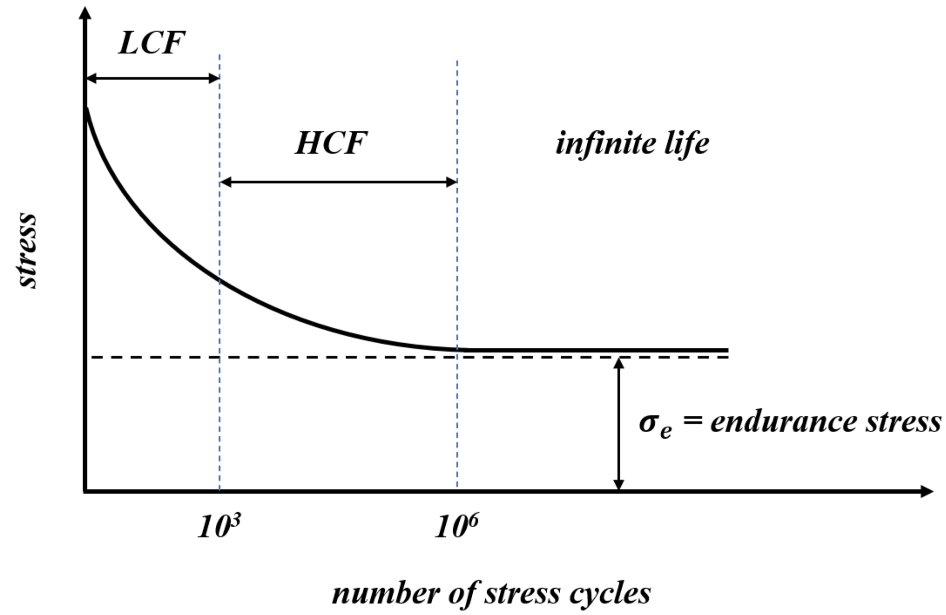


Figure 1.1 S-N curve for fatigue failure

As per the Goodman's criteria (Equation **1**.**4**),

$$\frac{\boldsymbol{\sigma_{var}}}{\boldsymbol{\sigma_e}} + \frac{\boldsymbol{\sigma_{mean}}}{\boldsymbol{\sigma_{ut}}} = \frac{\mathbf{1}}{\boldsymbol{F.O.S}} \qquad 1.4$$

Where,

$\boldsymbol{\sigma_{var}}$ = Variable stress

$\boldsymbol{\sigma_e}$= Endurance stress

$\boldsymbol{\sigma_{mean}}$ = Mean stress

$\boldsymbol{\sigma_{ut}}$ = Ultimate stress

$\boldsymbol{F.O.S}$ = factor of safety

Here, variable stress means the amount of stress that is completely reversed in the cycle, and mean stress is the average of maximum and minimum stress value.

The endurance stress for structural steel is considered to be,

$\boldsymbol{\sigma_e} = \mathbf{230}$ **MPa** ($\sigma_e = 0.5\sigma_{ut}$; for material whose ultimate tensile stress is < 1400 MPa [3])

The parameters have been further explained through Figure 1.2.

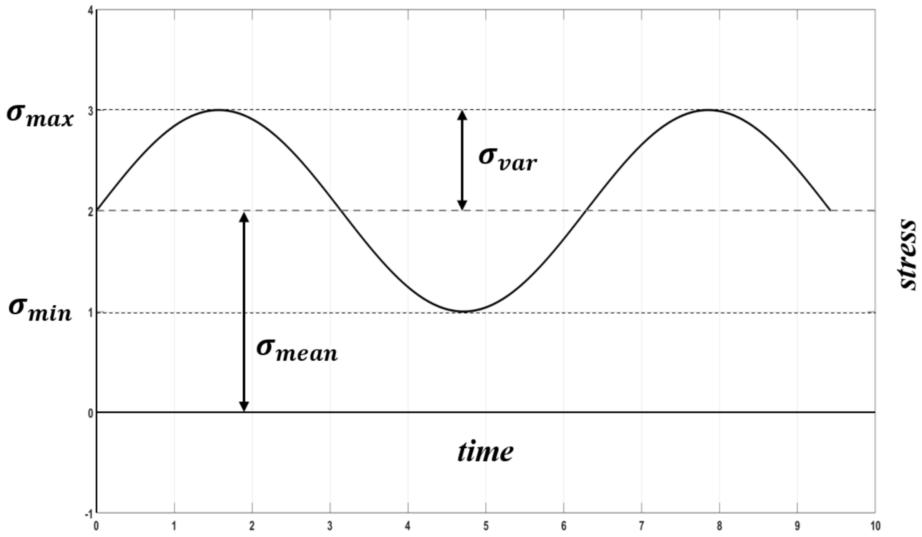


Figure 1.2 Cyclic loading showing different stress parameter for fatigue

According to the S-N curve approach, if completely reversed variable stress is given to the system, then the life of the system can be calculated by Equation 1.5,

$$\boldsymbol{N} = \left(\frac{\boldsymbol{\sigma_{rev}}}{\boldsymbol{a}}\right)^{\frac{1}{b}} \qquad \textbf{\textit{1.5}}$$

$$b = \left(-\frac{1}{3}\right)\log\left(\frac{\sigma_{ut}}{\sigma_e}\right) \qquad \textbf{\textit{1.6}}$$

$$\log(a) = \log(\sigma_{ut}) - 3b \qquad \textbf{\textit{1.7}}$$

Where**,**

N = Number of cycles

a, b = Fatigue constants

2. LITERATURE REVIEW

In the last two decades many experimental and numerical studies (material selection [7-17], structural analysis [19-27], [47-51], dynamic analysis [18-46]) have been performed by several researchers and scholars. These works mainly revolves around finding the natural frequencies of an automobile chassis and then using the selected natural frequencies which falls under the road vibration frequencies, to find out the mode shapes and thereafter the harmonic analysis is done using the results.

These studies can be classified into four major categories, namely the chassis materials selection, static structural analysis, modal analysis and the harmonic response analysis using finite element method.

2.1 CHASSIS MATERIALS

When it comes to designing of an automobile chassis, material plays a key role. Different materials have been seen in the manufacturing process of an automobile, depending on the use case scenarios of the particular vehicle model. Many forms and alloys of steel [4][7], aluminum [6][11] and composites such as carbon fibre [2][5][8], etc have been in demand for their respective properties. It has been noticed in previous studies related to material selection that structural steel, aluminum, carbon epoxy, and e-glass epoxy composites were chosen as the main alternatives for chassis material.

The main objective for material selection must be to reduce the weight of the chassis as much as possible without compromising on the strength, rigidity and other requirements of the chassis. Minimisation of weight can make the performance and efficiency of the automobile better [7]. It is also equally important to enhance the vehicle performance at minimum manufacturing cost. In general, the composite materials can replace the conventional materials due to its effective weight reduction property. For each 10% reduction in weight fuel efficiency can be increased up to 8% in the case of passenger vehicles, W. J. Joost et al [2]. Srinidhi Rao et al [9] identified structural steel as the bestsuited material, whereas Dasarath Palagiri et al [10] concluded carbon epoxy to be the most suitable material. High specific strength steel (HSSS), an aluminum infused steel has shown impressive weight reduction ratio Saxena et al [11] without much increment in stress value. In one experiment by Bhowmik et al [12] another form of steel, HSLA was found out to be the best material for designing the chassis.

Many experiments and research were also conducted on carbon fibre composites. In the manufacturing of bumpers, the carbon fibre reinforced composites are found to be used due to its appreciable reduction in weight and better strength, John et al [13]. Reduction in weight, durability, toughness, part integration and reduction, crashworthiness, and aesthetic appealing, etc are the main advantages of carbon fibre, Ahmad et al [14]. One study concluded that composite material strength yield is 40% more than aluminum and 11% less than steel strength yield (T. Pravilonis et al [15]).

Steel showed least amount of deformation in the mode shapes and static loading [4], [7] and carbon epoxy gave least deformation when considering weight ratio [5], [8], [10]. Structural steel or other alloys of steel have by far been the most used material for chassis. Aluminum and carbon fibre uses, though having some major advantages, are somewhat limited due to their cost ineffectiveness in mass production vehicles. Although, aluminum has found itself to be primary materials for automotive body panels, Y. Ota et al [16] and lightweight cars [17].

By far, structural steel is the most common material for chassis due to its cost effectiveness, strength etc. when compared to aluminum, which enhances its overall productivity. Aluminum on the other hand, has less weight yet comparable strength to steel, and due to this property, a vehicle’s body and chassis are also being manufactured apart from engine blocks, panels etc. Apart from less fuel consumption, making a chassis lighter also has various benefits for driving dynamics.

These also says that other lightweight alternatives such as carbon-fibre are also seeing an enhancement in usage for premium and exotic vehicles where costs are less of a concern.

2.2 QUASI-STATIC ANALYSIS OF CHASSIS STRUCTURE

Different authors have performed FE analysis on chassis structure under quasi-static loading, limiting the stress value within the optimum capacity, and redesigning the chassis to improve performance and reduce weight, without exceeding the safety factor for stress, strain, displacement, etc.

Srinidhi Rao et al [9] have performed a complete dynamic analysis of a ladder frame chassis with Ansys and used three different materials namely structural steel, aluminum alloy and carbon epoxy composite. Structural modifications to avoid resonance and enhancing the performance were made. Similar kind of work was done by Pravin Renuke [18], to find the mode shape of the frame and modify it so that the natural frequencies fall out of the natural road range but with increasing the chassis stiffness.

Kiran Ghodvinde et al [19] have done the stress distribution and displacement under various loading conditions in in the Kit car chassis and Chevy truck chassis. In the same year Denish S. Mevawala et al [20] have done a static stress analysis for an all-Terrain vehicle of roll cage skeleton model. Various tests like front impact side impact and rear impact were conducted all using. Hirak Patel at al [21] have done the static load analysis for stress in a truck chassis frame and its design optimization for weight reduction where 17% of kerb weight was reduced keeping the stress and defamation under the limit. Ashutosh Dubey et al [22] have done static load analysis and power spectrum analysis using FFT analyzer, for chassis frame. Worst case loading conditions and over loading conditions were considered.

Majority of the studies that were conducted, focused on the weight reduction of the automobile chassis by structural optimization (Marco Cavazzuti et al. [23]). Various techniques were mentioned for optimization of chassis. Emphasis was given in reducing the number of physical tests conducted for structural safety and using finite element analysis for this purpose by P. K. Sharma et al. [24], he performed a stress analysis on TATA Turbo Truck SE 1613. In the overloading conditions the maximum values of stress and deflection with the help of analytical calculations.

A study by Vijayan et al [25] showed that I-section beams, when tested against C-section and box beams, performed very well and induced very low stress and deformation. This paper also compares steel with composite material e-glass epoxy using different cross section and with the aim of safely designing heavy vehicle chassis.

Many studies have emphasized on the fact that carbon fibre or epoxy have shown much better results against steel when conducting the structural analysis and better weight to stress ratios were observed V. Chintada et al. [26], J.S. Nagaraju and U.H. Babu [27]. But at the same time this accounts for much higher cost and less production affordability.

In general, the major construction parameters for static structural analysis showed that C and I-section beams have been used mostly for longitudinal and cross frames of a ladder chassis respectively. The major loading could be distributed depending on the location of the payload and various vehicle components mounting.

2.3 DYNAMIC ANALYSIS ON CHASSIS STRUCTURE

2.3.1 MODAL ANALYSIS

Modal analysis of the chassis has been done by several researches in past few decades. F. A. Rodrigues et al [28] performed all kinds of dynamic analysis including modal, harmonic and transient. Results showed even if the structure matched the static characteristics, dynamic analysis such as modal and harmonic allows identification of weaker spots, resulting to the rectification of the model.

Vehicle dynamics concerns with different effects in the vehicle that can improve of deteriorate the overall performance of the vehicle in different driving conditions, Vivaz Lopez et al [29]. Understanding about vehicle structural dynamics is required to find the relationship between the various frequencies and their respective mode shapes to find their effects in an automotive chassis. Past works have shown the importance of studying the effect of irregular road surfaces on natural vibrations of an automobile as they impose a great effect on the automobile chassis as well as passengers and the loads carried by the vehicle, Prashant et al [30].

While performing modal analysis, several authors have considered unconstrained vibration of the chassis. If the structure is unconstrained, the first six modes of the structure become zero. Few researchers have taken the boundary conditions into account, restricting the degrees of freedom of the body.

In designing and fabricating a space frame/tubular chassis is important to avoid resonance frequency which could cause excessive displacement and failure, Mohammad Al Bukhari

Marzuki et al [31]. Han fui et al [32] showed as most of the chassis' natural frequencies lie within the road frequency range it acts as the major disturbing element. Also, the suitable mounting locations of components like engine and suspension system were chosen by mode shape results. Mohammad Reza Forouzan et al [33] showed as the natural frequencies fall out of the natural range due to increment of mass due to mounted components, this can be compensated with increasing the chassis stiffness, such as by decreasing it's length. Any chance of resonance can be prevented by this method and the frequencies can be kept within desired range. Components like the fuel tank, fuel filter, electronic air processing unit (EAPU) and their mounting brackets, etc. are mounted on the chassis frame, Shuvam Bhise et al [34]. The results of modal analysis should be kept in consideration to ensure that these components do not produce any vibrations when put into operation.

Goutham Solasa et al [35] performed a free free modal analysis, ignoring the boundary conditions and loads, noting the effect of any contact connections should be calculated based on their status at the beginning of the static analysis.

The natural frequency and the mode shape depend on the mass and stiffness matrix of the system. The terms in the both mass (M) and stiffness (K) matrices depend on the geometry, material properties and boundary condition. Therefore, the form of the M-matrix and the K-matrix can be changed by changing the geometry of the structure which in turns avoids the resonating frequencies, YU Baojun et al [36]. Madhu Ps et al [37] did a modification of design to reduce deflection of structure and to find out the most critical mode. Obed Lungmuana Darlong et al [38] also did various design modifications, a total of 11 modifications were carried out. The modified chassis was presented as the more ideal chassis with a reduced weight, reduced induced Von-Mises's stress and from the modal analysis results presented favorable values. Cost is a significant factor in the decision-making process in industry. A modified and reduced weight of a chassis make sure that the production, manufacturing, and material cost are in limit. Higher weight reduction is always desired and major factor in any optimization scheme, Abhishek Agarwal et al [39].

Vinodbabu Chintada et al [26] showed the deformation in CNT reinforced CFRP chassis frame is less in compared with traditional materials such as steel and magnesium alloys which are used in truck chassis. CFRP, as said earlier, also reduced the weight of the vehicle to substantial amount.

2.3.2 HARMONIC ANALYSIS

The harmonic analysis, uses frequency response or phase response curves over time domain response to determine which vibration modes affects significantly to the dynamic response of the chassis. In general, three harmonic response analysis methods are used, full, reduced, and mode superposition. Another quite expensive method is to do a transient dynamic analysis with the harmonic loads considered as time-history loading functions, 2018, K Ashok Reddy et al [40]. The main aim of harmonic analysis is to reduce the experimental testing of chassis by conducting the same in simulation.

Muhammad Al Bukhari Marzuki et al [41] have performed harmonic analysis of space frame race car chassis. The results found that the natural frequencies produced five mode shapes and all within 100hz which is safe and comfortable for human usage. In the same year Obed Lungmuana Darlong [38] performed harmonic analysis in a truck chassis and showed the most critical frequencies. Typically, the road profile frequency is 45-55 Hz. And the enginge operating frequency at 3000 rpm comes out to be about 25-30 Hz [32].

Static and dynamic loads such as weight and road roughness vibrations were analysed and applied to chassis model, Mahendra Verma et al [42]. ISO 8606 was used to calculate road roughness and the dynamic forces acting to the chassis frame were extracted using of a quarter-vehicle model simulation during a constant velocity motion.

As we can see by our observation, though many studies considering various real-life problems have been performed in both modal and harmonic analysis, there are few areas that are yet to be analysed. This thesis aims at solving one of those less approached area of pre-stressed analysis of chassis structure.

In papers like [9], [43], the loading for the harmonic analysis were applied by considering the vibrating part of the chassis, such as engine to be the main causing agent. This, however does not take into account the payload in case of heavy vehicles such as trucks, etc. where the loaded materials are not in constant contact with the chassis and may cause in or out of phase vibration with different amplitudes.

For this part of the thesis work, two approaches were taken into consideration, analytical and experimental data outsourcing. For any two body in non-rigid contact with each other experiencing forced vibration, the external load will be transmitted to the second body in accordance to their transmissibility ratio and the lag in phase angle ($\emptyset$). This ratio generally depends on the damping ratio (ζ) and natural frequencies of the body, as given in Equation 2.1 and Equation 2.2.

$$\textit{Transmissibility ratio (r)} = \sqrt{\frac{\mathbf{1 + (2\zeta n)^2}}{\mathbf{(1 - n^2)^2 + (2\zeta n)^2}}} \qquad \textbf{\textit{2.1}}$$

$$\tan \emptyset = \frac{\mathbf{(2\zeta n)^2}}{\mathbf{1 - n^2 + (2\zeta n)^2}} \qquad \textbf{\textit{2.2}}$$

This approach seemed mostly accurate, however to calculate $\boldsymbol{\zeta}$ the experimental setup of the structure was required which was not possible considering resource limitations. A second approach was taken to outsource reliable data related to acceleration endured by truck cabin and chassis, from past works. Abdelkareem et al [44] had done analysis on the designing of damper for truck. His results showed the RMS cabin acceleration to vary from 0.50 m/s^2 to 0.98 m/s^2. The average road unevenness was measured at 20mm. Zhao et al [45] had done a similar analysis with 3-DOF model and the results showed the cabin acceleration to be in the range of 0.47m/s^2 to 0.64 m/s^2, when the truck was going from 65 kmph to 85 kmph. Considering these two results, a value of acceleration = 0.57 m/s^2 was taken for our analysis. The resultant harmonic force was calculated based on this data and the payload mass was considered.

2.4 PRE-STRESSED ANALYSIS OF STRUCTURAL COMPONENTS

For prestressed modal analysis, the governing equation takes the form (Equation 2.3),

$$([\boldsymbol{K}+\boldsymbol{S}]-\boldsymbol{\omega_i}^2[\boldsymbol{M}])\{\boldsymbol{\phi_i}\}=\{\boldsymbol{0}\} \qquad \textbf{\textit{2.3}}$$

Where [K] is the stiffness matrix, [M] is the mass matrix, and [S] is the change in stiffness matrix due to prestressing [46].

What we have clearly seen from these papers is that more attention has been given to performing modal analysis when the chassis is in free condition and without stress. The pre-stress analysis is rare and comparatively less emphasis was given, as found in the literature survey [39-41][45].

Bedri et al [47] performed prestressing effects in a shell structure and found out that compressive are tensile loading can affect the natural frequencies of a structure differently. The effect of pre-stressing was also taken into consideration by Saran et al [48], while analysing the analysis of dynamic performance of an aircraft wing.

Naina Mohamed et al [49] performed the modal analysis with pre-stressed chassis. The result shows that though there are minimal effects of pre-stressing in comparison with unloaded structure for higher modes, the first few mode shapes recorded the highest change in mode frequency, (in the order of 0.1-0.2 %) with respect to the increasing load factor. But the change of frequencies when the material was changed was not addressed. Also, the chassis type taken into account was space frame chassis, generally used for lighter vehicles. Such a vehicle will not perform in high loading conditions.

2.5 SCOPE OF WORK

When looking at the past papers, many researches and experiments on the static structural analysis can be seen. Leaf spring/supports, reinforcement brackets which supports subassemblies to the chassis, washer and weld locations etc were found out to be the most critical location [50]. The stress and deformation are also compared for different types of loading and chassis structure with different cross sections [19], [25], [51].

Modal analysis is used to determine the mode shapes and the natural frequencies of any component. This is used to analyse the behaviour of that particular component when it is subjected to a load of similar frequency and to avoid failure due to resonance. The result of modal analysis is further used to study various other types of analysis such as harmonic or transient [52], [53].

In the case of vehicles such as trucks or any vehicle that is designed to carry a load much higher than normal, the components may not behave as pre-assumed in the free condition or even in the elastic domain. Stress stiffening can change the natural frequencies of a system which impacts dynamic responses of the system. Given the limitation in the current body of knowledge and the importance of understanding the pre-stress characteristic of the chassis frame, there is a scope for further analysis of the effect of pre-stressing on the chassis structure.

This work has emphasized the effect of prestressing on modal and harmonic analysis, and for different materials that are currently in use for chassis manufacturing.

3. OBJECTIVES AND METHODOLOGY

3.1 OBJECTIVES

This study approaches with the following objectives.

1. To design a ladder frame chassis model that can withstand rated payload as well as overloading.
2. Select three materials generally used for chassis' manufacturing and to analyse their respective properties and change in the results due to them.
3. To perform a pre-stressed modal analysis on the chassis made from three different materials, and compare the results obtained with that of the load free chassis side by side, to show if any changes occur.
4. Perform Harmonic analysis with mode superposition principle on both load free and pre-stressed chassis, and to find out the maximum stress developed due to harmonic loading.
5. Perform a manual fatigue analysis to check if the maximum stress endured by the chassis in base payload as well as in overloading condition results an infinite life, and to find out the reduction in fatigue life with increasing stress.

3.2 METHODOLOGY

The methodology of the thesis has overall been categorized into five parts. Namely, the chassis structure, meshing, materials used, boundary conditions and loading for static and modal analysis, and loading for harmonic analysis with pre-stressing effect from the quasi-static analysis to see the effect of loading on the dynamic behaviour of the chassis.

3.2.1 MODELLING THE CHASSIS STRUCTURE

As we have stated earlier, the ladder frame chassis is one of the simplest and oldest constructional form of chassis. There are two longitudinal members which form the backbone of the chassis joined by several cross members attached to them for additional support. In our finite element analysis, static structural, modal, and harmonic analysis were performed on the TATA 1612 ladder-frame type chassis model. The rated loading criteria of the model were obtained from available source and the structure was created using design modular in the ANSYS workbench.

Two types of beams were designed for overall construction of the chassis. The longitudinal or the side members were constructed using a C-section beam and the cross/ short members were constructed using I-section beam. Both the C-section and the I-section beam's sectional view are shown in the Figure 3.1 for a detailed overview. All dimensions are in mm.

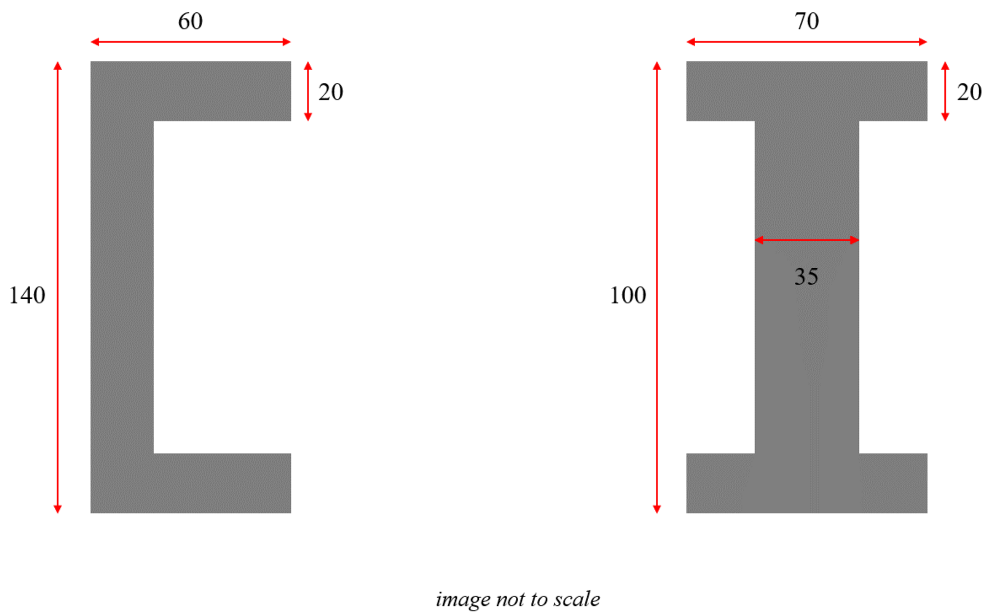


Figure 3.1 Sectional View of C-Beam and I-Beam

In the Figure 3.2 below, the top and side view of the chassis have been provided. Table 3.1 shows all the dimensional parameters of the chassis in a concise form.

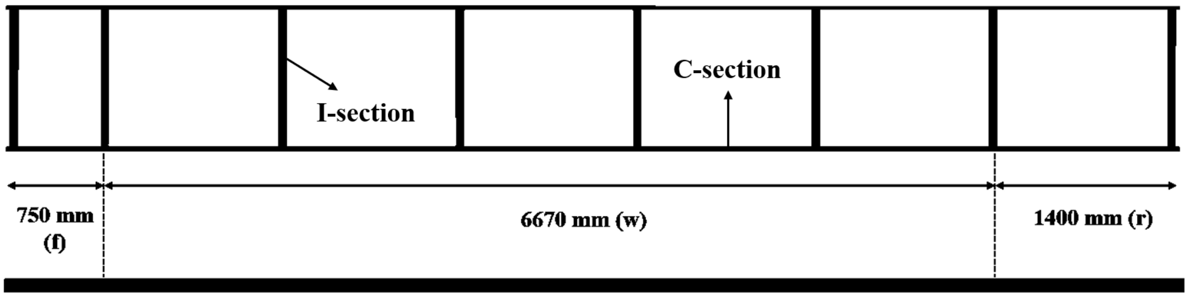


Figure 3.2 Top and Side View of The Chassis

Table 3.1 Design Specification of The Chassis

Components	Dimensions
C-section beam (outer frame)	140mm x 60mm x 20mm (length = 8810mm)
I-section channel (inner/cross frame)	100mm x 70mm x 35mm (length = 884mm)
Rear overhang(r)	1400mm
Front overhang(f)	750mm
Wheelbase(w)	6670mm

3.2.2 MESHING

Basic theme of FEA is to interpolate the results for entire domain by making calculations at limited number of points. The aim of meshing is to divide a continuous structure that has infinite degrees of freedom, into smaller parts creating finite degrees of freedom using discretization. Considering the application, different types of meshing can be used. Namely, beam, shell and solid elements (1D, 2D and 3D). Elements can be identified and selected based on the dominant geometrical dimensions.

For our analysis a line body construction of the chassis was created. The Beam 189 type element was used in the meshing purpose, as shown in Figure 3.3.

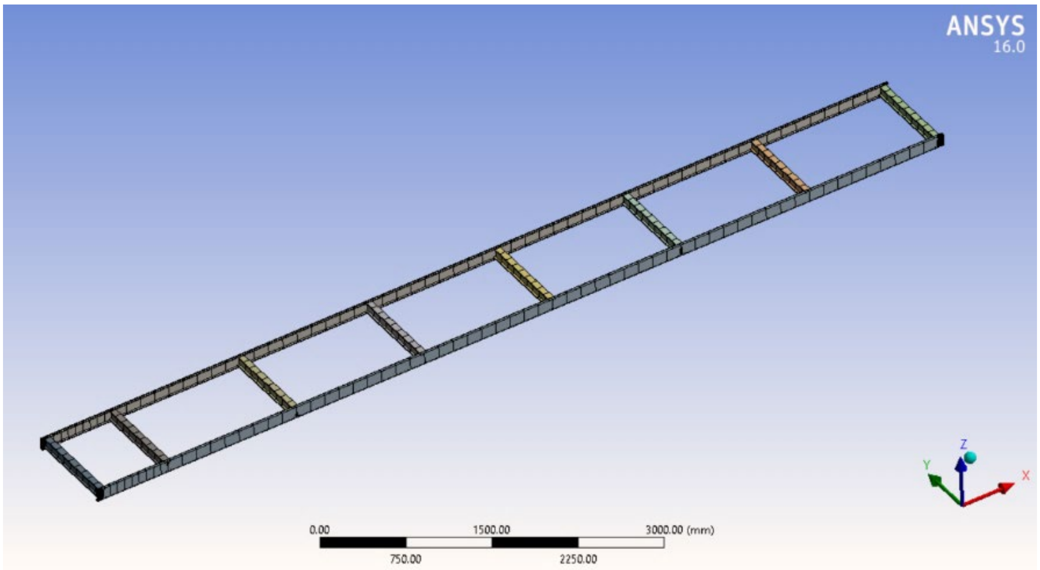


Figure 3.3 Model Meshing

The structure of the chassis is uniform, symmetric, and the cross section does not vary with the length. Beam elements are typically used in such cases. The most obvious reason for using beam elements would be speed of numerical simulation. Beam elements are far more computationally efficient than 2D or 3D elements since they have less DOFs.

In finite element analysis (FEA) as the geometry, or spatial domain, is meshed more finely convergence occurs, which means arriving nearest to the true solution. Mesh convergence involves choosing an optimal mesh by running multiple simulations with different levels of fine/coarse mesh, reducing the element size and calculating the result of this process on the accuracy of the solution.

Typically, as the mesh size is reduced, the solution becomes more accurate as the performance of the critical point of the product is finely analysed across its spatial domain. But to attain a more accurate result, longer runtime is required and complexity of the process is also increased.

Keeping theses in mind, a particular size of meshing was chosen for the analysis. As the model in use was designed by line body modelling, the edge sizing was tested for 3, 5, 7, and 9 divisions per edge length. The convergence results that were obtained by comparing the natural frequency of the chassis for the first 16 modes showed that 9 divisions per edge length can be considered as the optimum mesh size. The percentage change while going from 3 to 9 divisions have been shown in Table 3.2 below.

Table 3.2 Mesh convergence for 3 to 9 divisions in edge sizing

Percentage Change in Results		
Edge Sizing 3 to Edge Sizing 5	**Edge Sizing 5 to Edge Sizing 7**	**Edge Sizing 7 to Edge Sizing 9**
0	0	0
0	0	0
0	0	0
0.013176	0	0
0.004247	0	0
0.011764	0	0
0.005772	0	0
0.005188	0	0
0.028088	0.005107	0
0.009669	0.001934	0
0.059985	0.007499	0.001875
0.010903	0.001558	0
0.013882	0	0.001542
0.023864	0.002386	0.001193
0.024226	0.002202	0
0.271186	0.03073	0.006358

As we can see the change in percentage continues to reduce as we go from 3 to 9 and attains almost negligible value in 9 divisions, thus latter was chosen as the optimum number for mesh sizing.

Details of the optimized FE mesh are given in Table 3.3.

Table 3.3 No of Nodes and Elements

Number of total nodes	696
Number of contact elements	135
Number of solid elements	234
Number of total elements	369

3.2.3 MATERIALS CONSIDERED

As per the overall observation, three materials were chosen for our analysis, namely structural steel, aluminum alloy and carbon fibre.

Here, a comparison between structural steel, aluminum alloy, and carbon fibre has been made alongside to see their respective deformation, equivalent stress, and mode frequencies generated. The material properties of the following elements have been given in Table 3.4. and are collected from various data sources [9], [54].

Table 3.4 Material Properties of Steel, Aluminum and Carbon

	Structural Steel	**Aluminum alloy**	**Carbon Fibre**
Density(ρ)	7850 kg/m^3	2770 kg/m^3	1500 kg/m^3
Young's modulus (E)	200 GPa	71 GPa	155 GPa
Poisson's ratio (μ)	0.30	0.33	0.38

3.2.4 LOADING AND BOUNDARY CONDITIONS

The chassis structure has been divided into sections and the loads have been categorically distributed in each section as uniformly distributed load (UDL). The maximum allowable payload has been taken from standard available data and few modifications have been made as per general observation, for example, specific load (Figure 3.4) has been applied on the front section where the engine and gear assembly are supposed to be mounted [16]. Overall, the rear section has been acted upon by more loads as the whole payload is placed on the rear side of the vehicle.

As for the engine and the gearbox assembly, one specific I-beam was selected and the weight was applied. The rest of the weight, i.e., payload and the vehicle body weight, has been divided on the longitudinal C-beams. Figure 3.4 shows the distribution of the loads on the chassis.

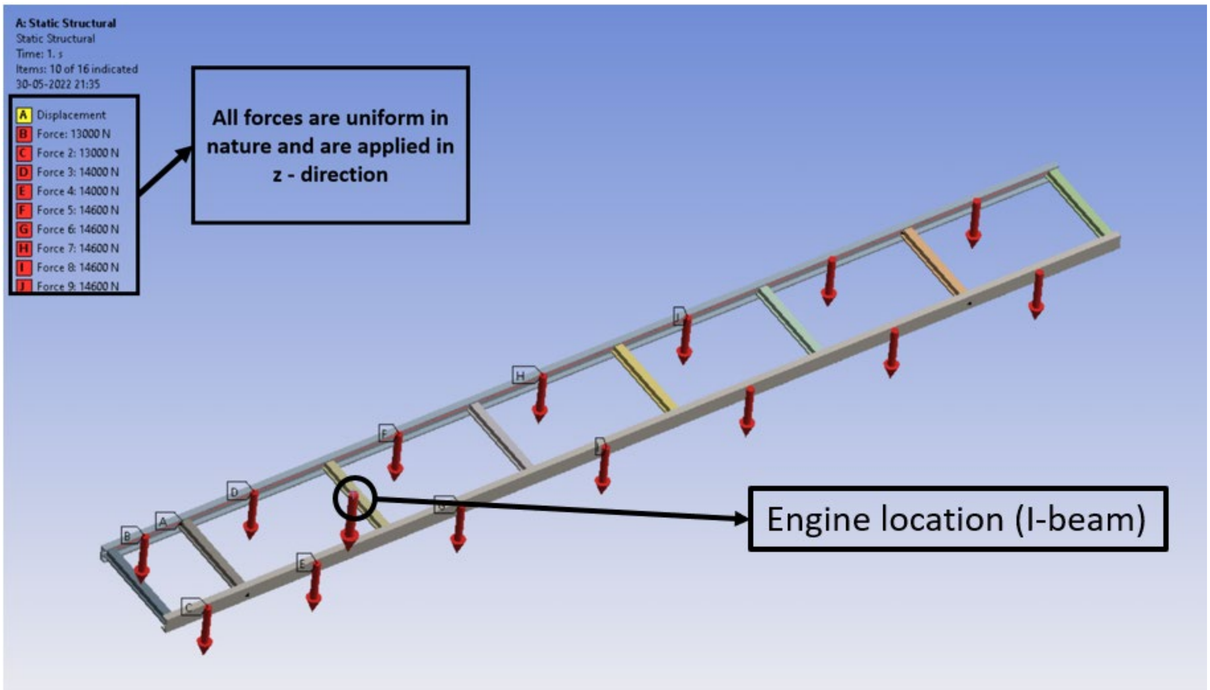


Figure 3.4 Distribution of The Loads on The Chassis

As for the payload, four load categories have been taken into account for the analysis, no load condition, base payload condition, 1.4 times of the payload condition (Table 3.5), and 1.8 times of the payload condition. These increment for the payload seemed necessary as there are frequent occurring of such scenarios where the vehicle operate and drives with a much higher payload than normal.

In this analysis, a scaling factor of 1.4 times the base payload has been considered to be the general case as we are concerned with the overloading which is a normal occurrence in the day-to-day life on Indian roads.

Table 3.5 Payload Calculation of The Chassis

Load criteria	**Loads**
Deadweight acting on the chassis	4700kg
Allowable payload	11500kg
Overloading considered	1.4 x 11500 = 16100kg
Total load	20800kg = 204 kN

The boundary conditions of the model have been shown in the Figure 3.5. Any unrestrained rigid body in three-dimensional space has six degrees of freedom (6-DOF). As per our case, the chassis structure is in fully loaded conditions and considering that, optimum supports and boundary conditions have been provided. The four wheel-base of the chassis, as shown in the image are constrained to behave as a hinged support, barring the translational motion and allowing rotational motion on all the axis.

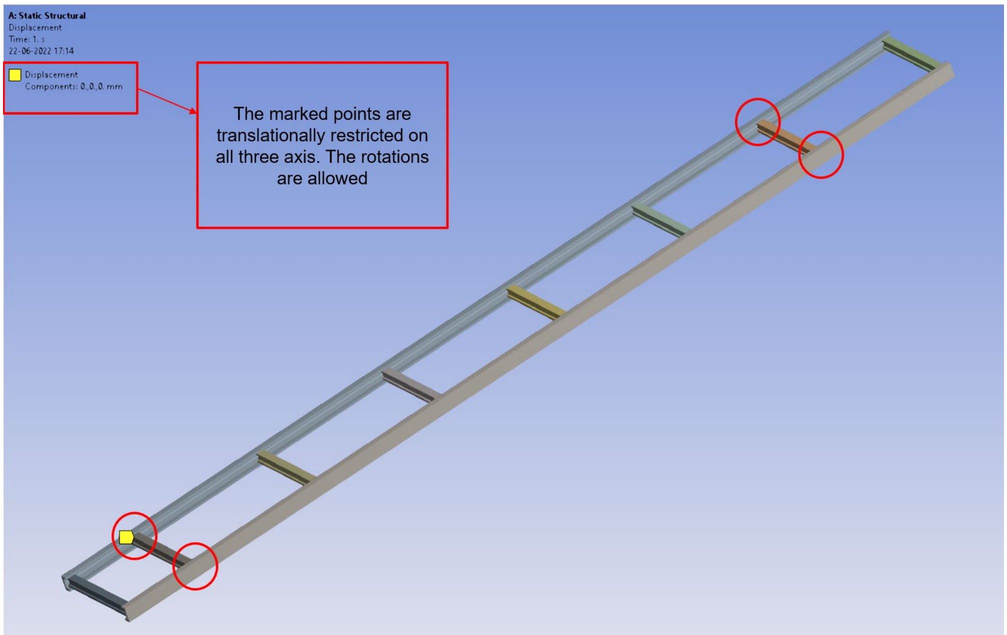


Figure 3.5 Boundary Conditions of The Model

3.2.5 HARMONIC ANALYSIS

For the harmonic analysis part, the acceleration of the periodic motion of the payload was set as 0.57 m/s^2, based on the observation [47][48] and the subsequent acting force was found out by calculation.

Total payload = =16100 × 9.81 N=157941 N

Harmonic load = 16100 × 0.57 = 9177 N

The force applied on the chassis for our analysis is shown in the Figure 3.6 below. The distribution of force has been made as per the same ratio with the base payload distribution.

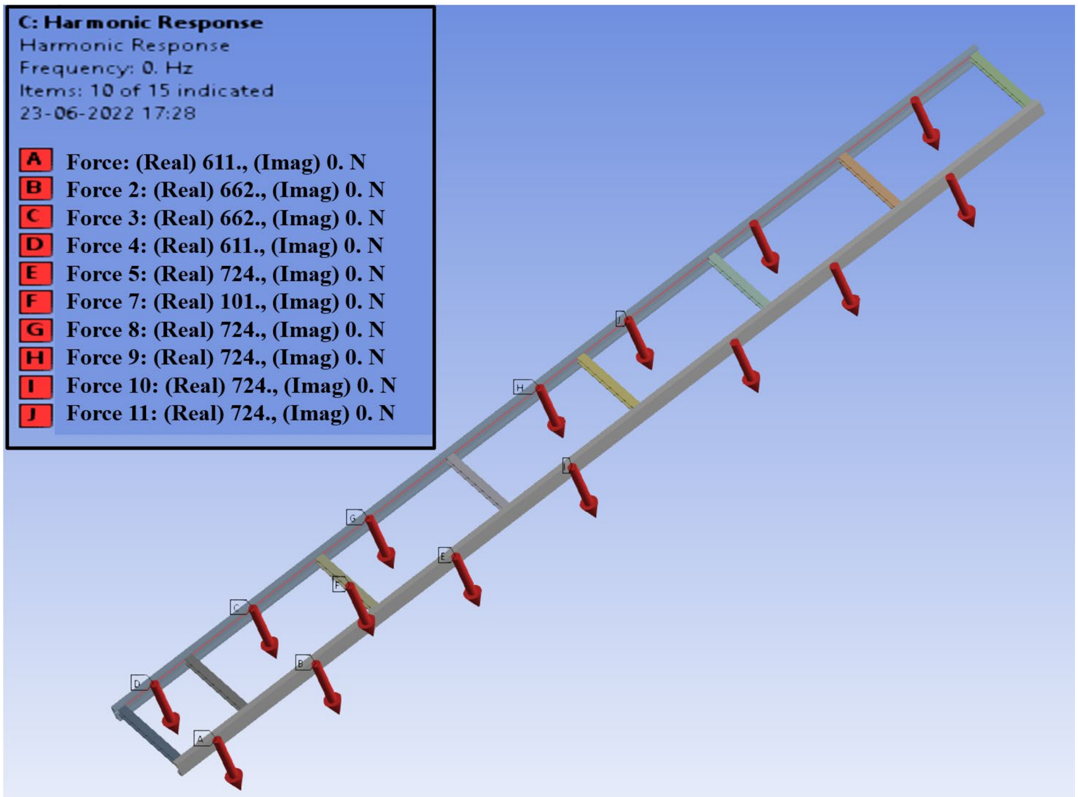


Figure 3.6 Loading for harmonic analysis

The simulations on the chassis model were run between 0-60 Hz frequency with a step division of 1 Hz. The damping co-efficient of the structure was considered to be 0.2.

4. RESULTS AND DISCUSSION

The analysis part can be categorized into four subparts, namely, STATIC STRUCTURAL ANALYSIS, MODAL ANALYSIS, HARMONIC ANALYSIS and FATIGUE ANALYSIS. The used chassis model has been simulated under various kinds of loads, static and dynamic, and also for all the three materials that have been used for this purpose.

4.1 STATIC STRUCTURAL ANALYSIS

Total deformation and maximum combined stress have been calculated for each material. The load considered for the static analysis, is 1.4 times of the actual rated payload. The result obtained can be seen in Table 4.1 below.

Table 4.1 Static structural analysis results

	Structural Steel	**Aluminum alloy**	**Carbon Fibre**
Maximum deformation	74.53 mm	125.36 mm	85.832 mm
Maximum combined stress	291.39 MPa	198.25 MPa	268.15 MPa

The deformed shape and stress of the chassis under static analysis has been shown in Figure 4.1 and Figure 4.2. The following figures have been generated considering structural steel as the material.

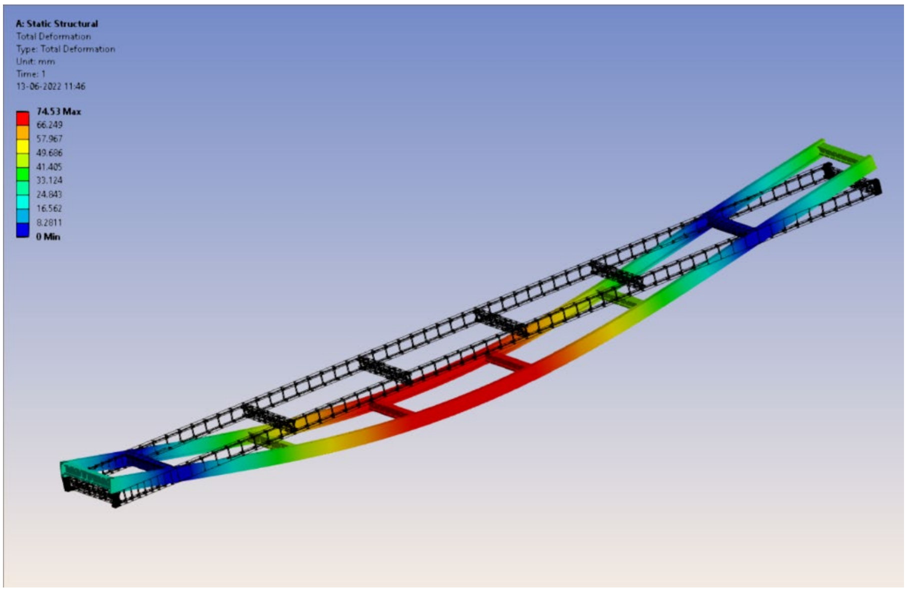


Figure 4.1 Deformation due to static loading

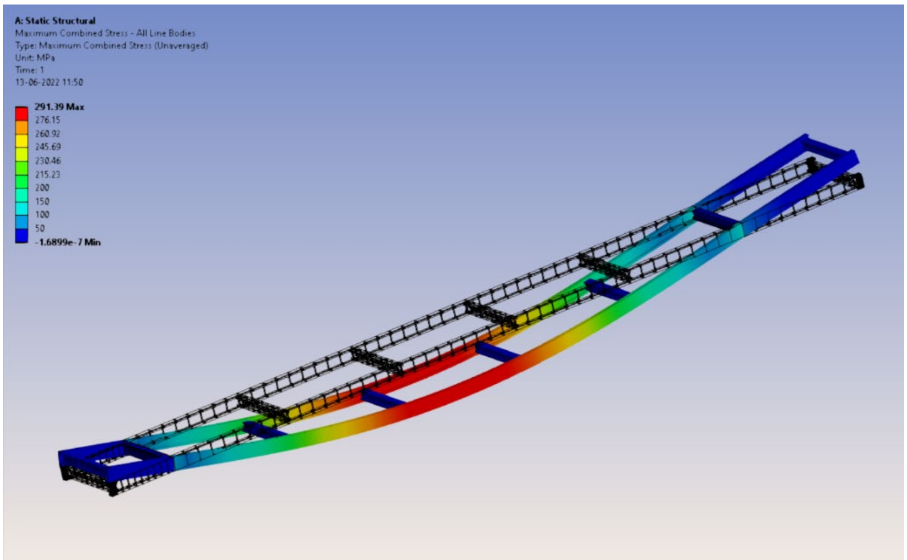


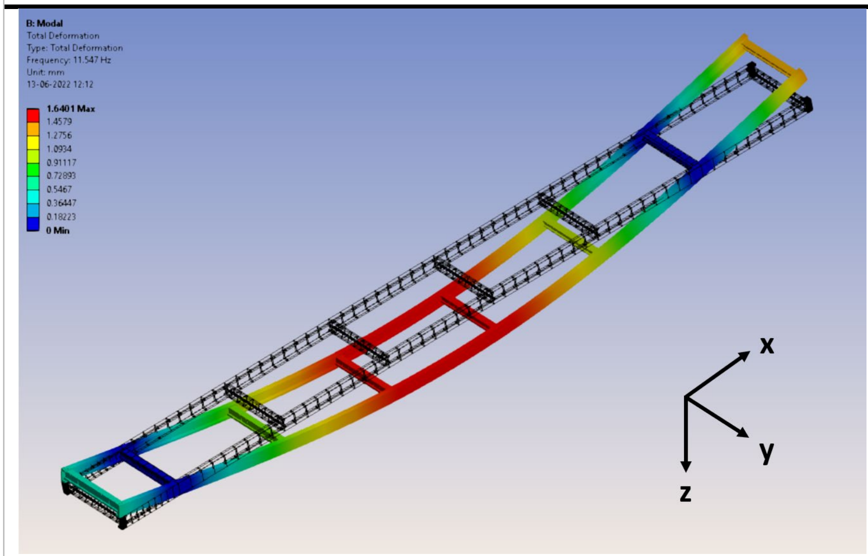
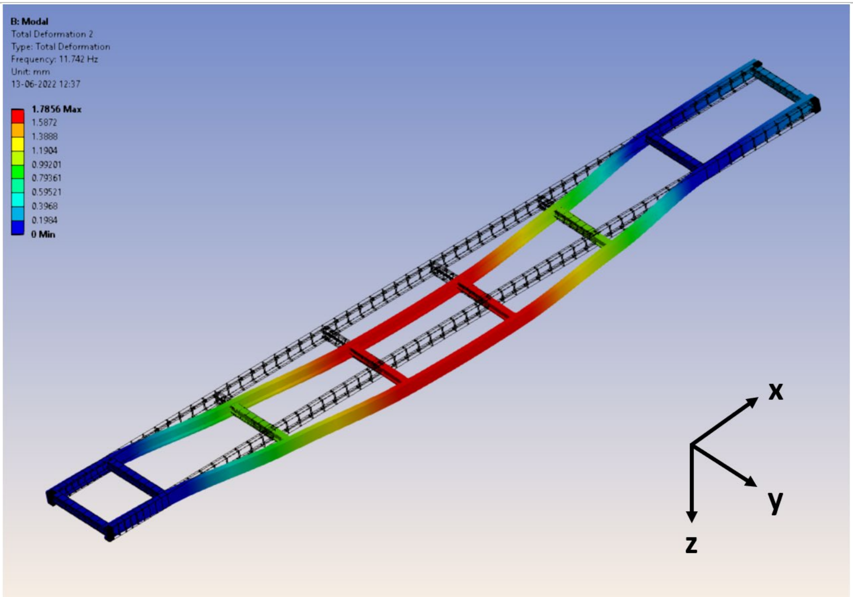
Figure 4.2 Combined stress due to static loading

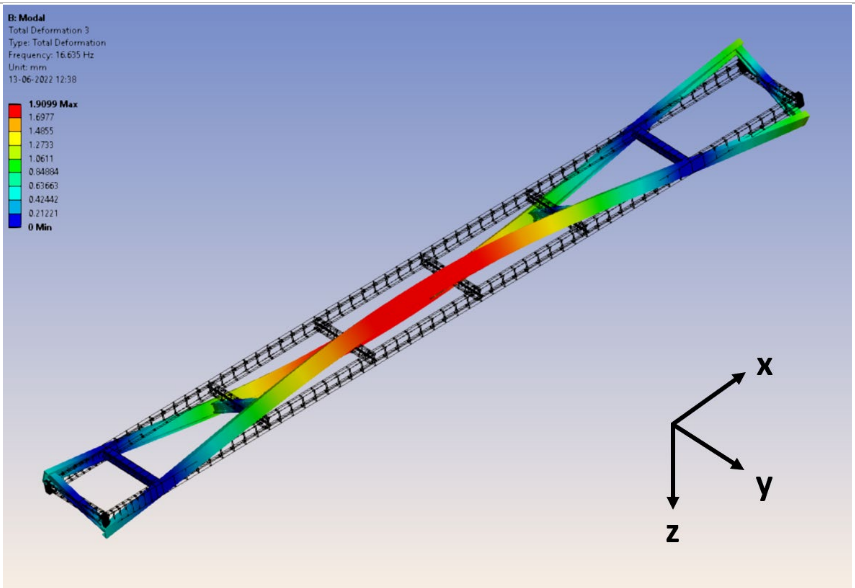
As we can see above, in Figure 4.1, the maximum deformation of 125 mm has been observed for aluminium alloy and gradually decreasing for carbon fibre and steel, with structural steel being the least deformed. The deformation of the three materials is as per expectation and in accordance to their decreasing young’s modulus. All stress values are within limit.

4.2 MODAL ANALYSIS

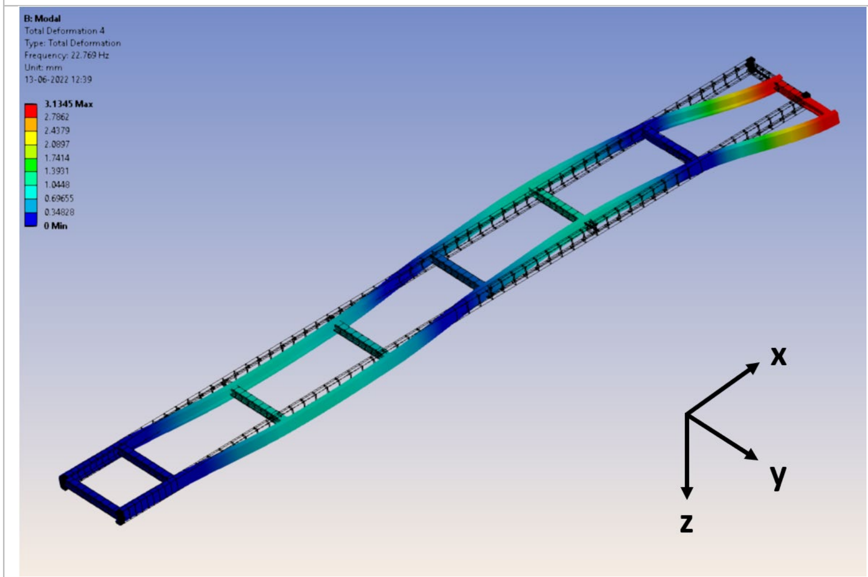
In Table 4.2, all the obtained mode shapes have been shown. The displacement nature of the modes has been discussed below.

Table 4.2 First sixteen mode shapes of the chassis

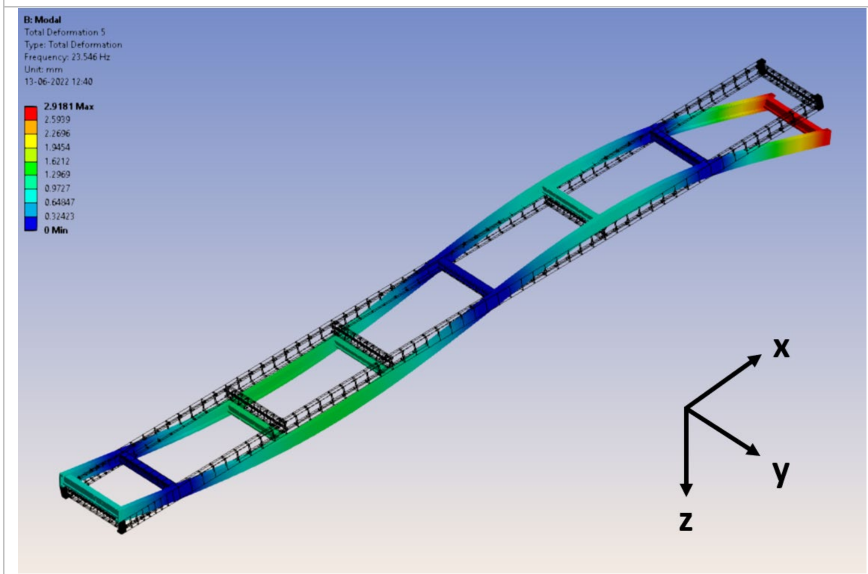
<table>
<tr><th>Mode shapes</th><th>Description</th></tr>
<tr><td>
</td><td>Mode 1: normal bending mode about Y axis.</td></tr>
<tr><td>
</td><td>Mode 2: normal bending mode about Z axis.</td></tr>
</table>



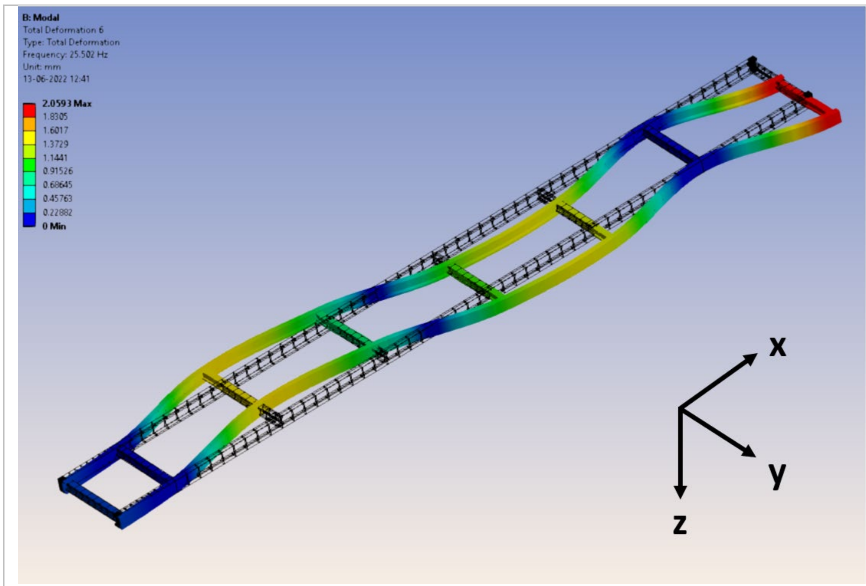
Mode 3: twisting mode about X axis.



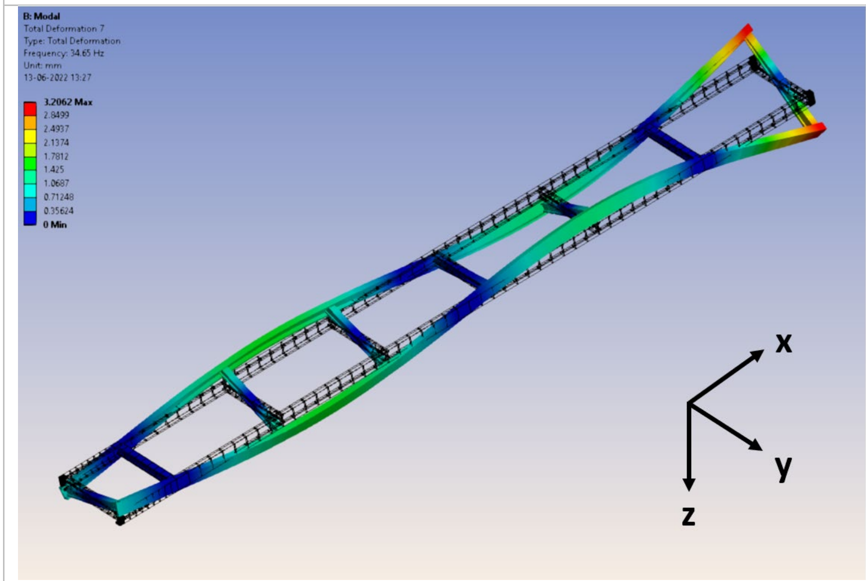
Mode 4: normal bending mode about Z axis.



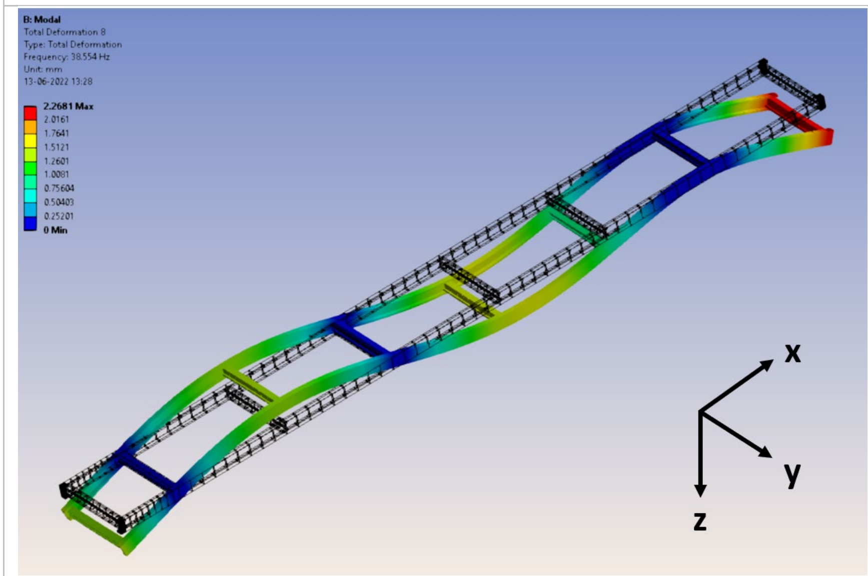
Mode 5: normal bending mode about Y axis.



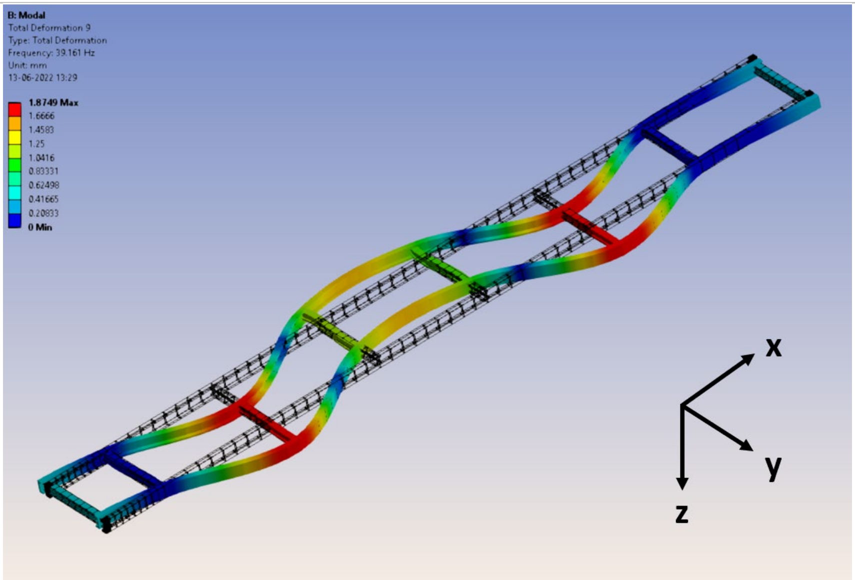
Mode 6: higher order bending mode about Z axis.



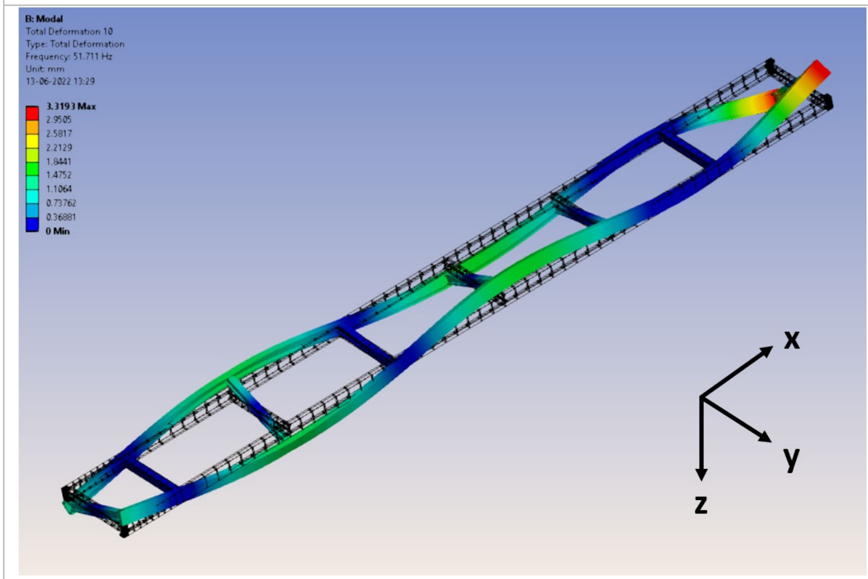
Mode 7: twisting mode about X axis.



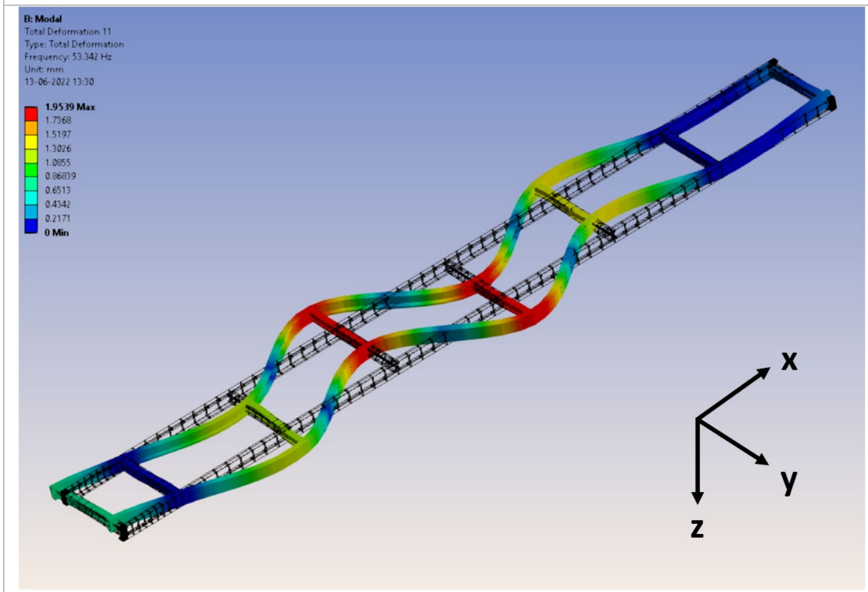
Mode 8: higher order bending mode about Y axis.



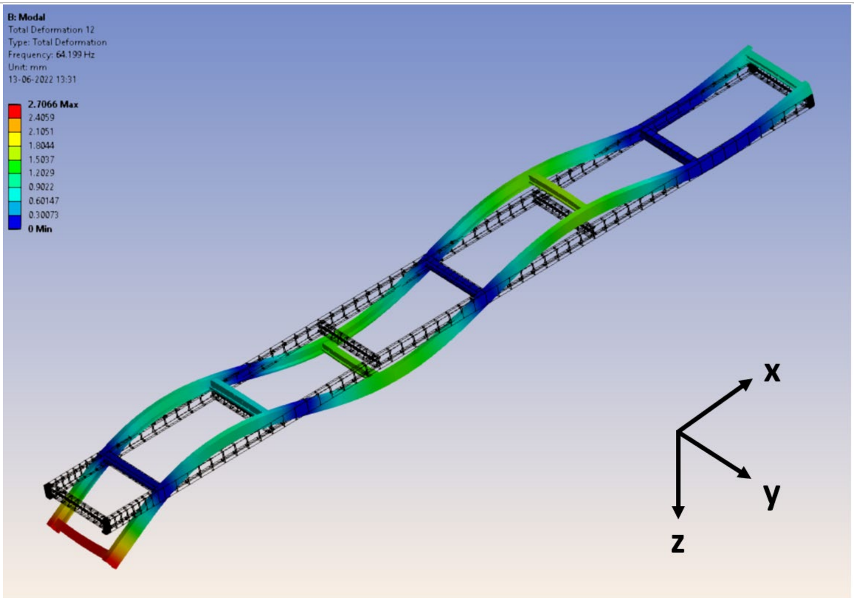
Mode 9: higher order bending mode about Z axis.



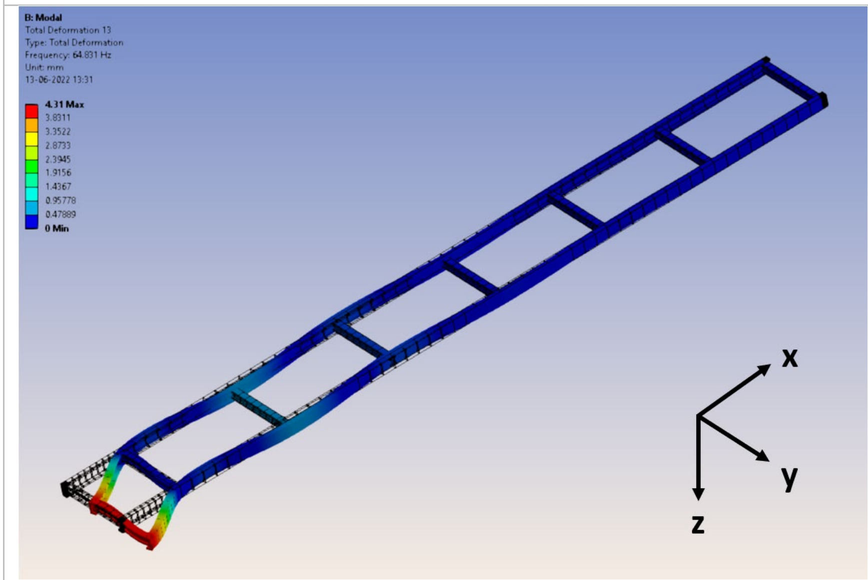
Mode 10: higher order twisting mode about X axis.



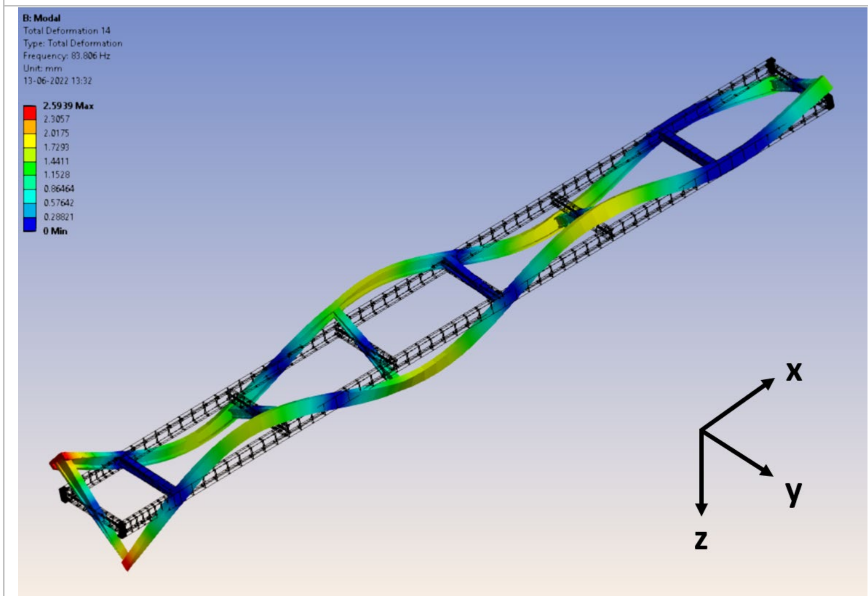
Mode 11: higher order bending about Z axis.



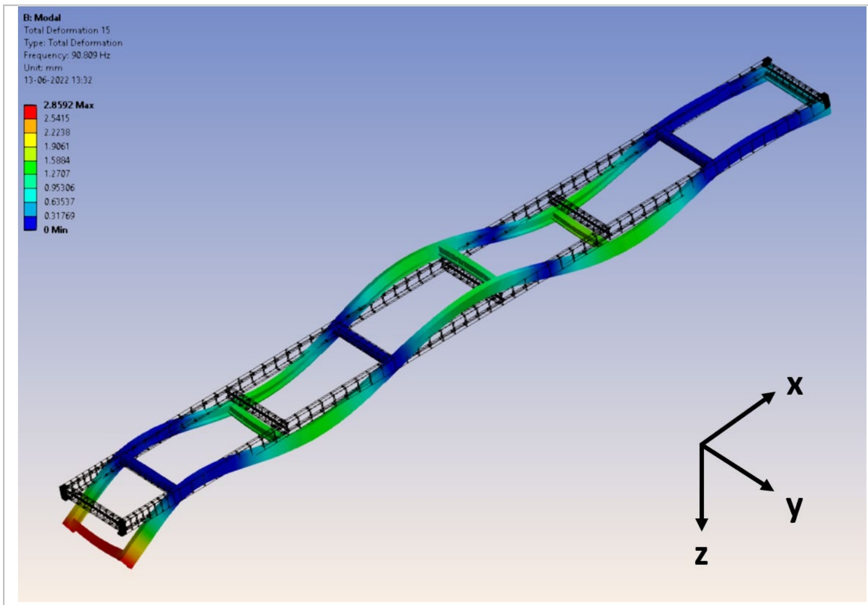
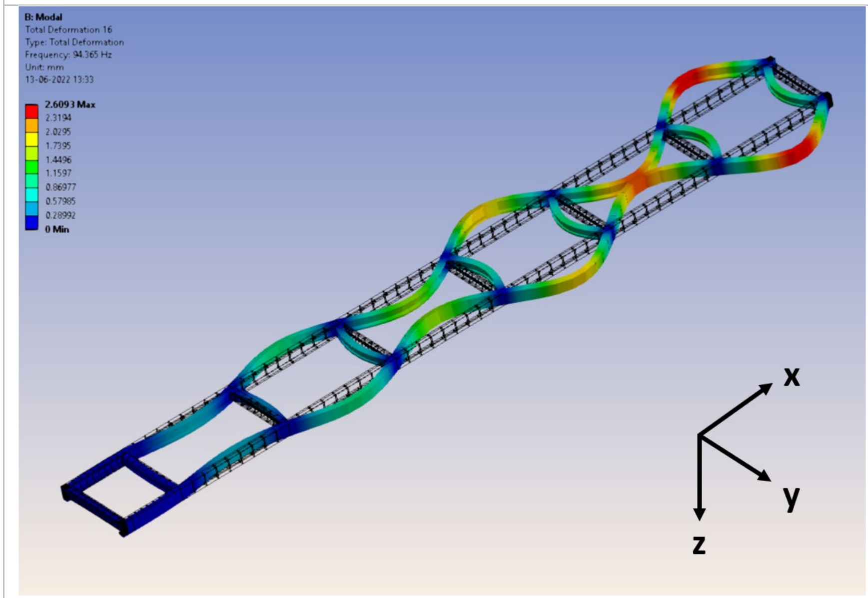
Mode 12: higher order bending about Y axis.



Mode 13: normal bending mode about Z axis.



Mode 14: higher order twisting mode about X axis

	***Mode 15: higher order bending about Y axis.***
	***Mode 16: higher order bending about Z axis.***

Mode 1, 2, 4, 5, and 13 are seen as normal bending modes, bending about y or z-axis. Mode no. 3, 7, 10, and 14 are twisting modes about X-axis. While mode no. 6, 8, 9, 11, 12, 15, 16 are higher-order bending modes. All higher order bending modes are out of plane modes except 16th mode, which is in-plane bending modes. Typically, in plane modes will have higher natural frequency given the ladder "steps" provide additional support.

4.2.1 COMPARISON OF MATERIALS

- A side-by-side comparison of mode frequencies of all the materials has been made for both prestressed chassis and load-free chassis when considering the overloading as

X=1.4, X being the factor of payload, in Table 4.3, Table 4.4, Table 4.5. There is a significant change in the mode frequencies where prestressing has been applied. The change in structural steel is less as compared to the other two materials, whereas the greatest number of changes has been observed in the case of carbon fibre chassis. (The highlighted modes in red fall in operating engine frequency, whereas green highlighted modes fall in the road profile frequency.)

Table 4.3 Effect of pre-stressing in structural steel

Mode No.	**Pre- stressed Modes**	**Load free Modes**	**Percentage change**
1	11.547	6.9958	39.41
2	11.742	10.22	12.96
3	16.635	12.968	22.04
4	22.769	21.136	7.17
5	23.546	22.285	5.36
6	25.502	24.167	5.23
7	34.65	33.345	3.77
8	38.554	35.293	8.46
9	39.161	37.386	4.53
10	51.711	48.648	5.92
11	53.342	50.697	4.96
12	64.199	62.897	2.03
13	64.831	64.467	0.56
14	83.806	82.183	1.94
15	90.809	89.721	1.20
16	94.365	90.535	4.06

Table 4.4 Effect of pre-stressing in aluminum alloy

Mode No.	Pre- stressed Modes	Load free Modes	Percentage change
1	14.133	7.017	50.35
2	16.721	10.249	38.71
3	21.717	12.926	40.48
4	23.512	21.196	9.85
5	25.729	22.350	13.13
6	29.514	24.236	17.88
7	36.785	33.324	9.41
8	40.771	35.395	13.19
9	45.473	37.490	17.56
10	53.639	48.788	9.04
11	60.678	50.740	16.38
12	66.031	63.063	4.49
13	66.61	64.640	2.96
14	86.803	82.297	5.19
15	92.997	89.932	3.30
16	99.291	90.793	8.56

Table 4.5 Effect of pre-stressing in carbon fibre

Mode No.	Pre- stressed Modes	Load free Modes	Percentage change
1	24.555	14.088	42.63
2	25.486	20.574	19.27
3	35.24	25.694	27.09
4	46.307	42.549	8.12
5	48.195	44.869	6.90
6	52.658	48.651	7.61
7	69.986	66.524	4.95

8	78.362	71.051	9.33
9	81.212	75.247	7.34
10	104.27	97.937	6.07
11	110.24	101.53	7.90
12	130	126.54	2.66
13	130.71	129.71	0.77
14	169.14	164.81	2.56
15	183.27	180.37	1.58
16	192.05	182.25	5.10

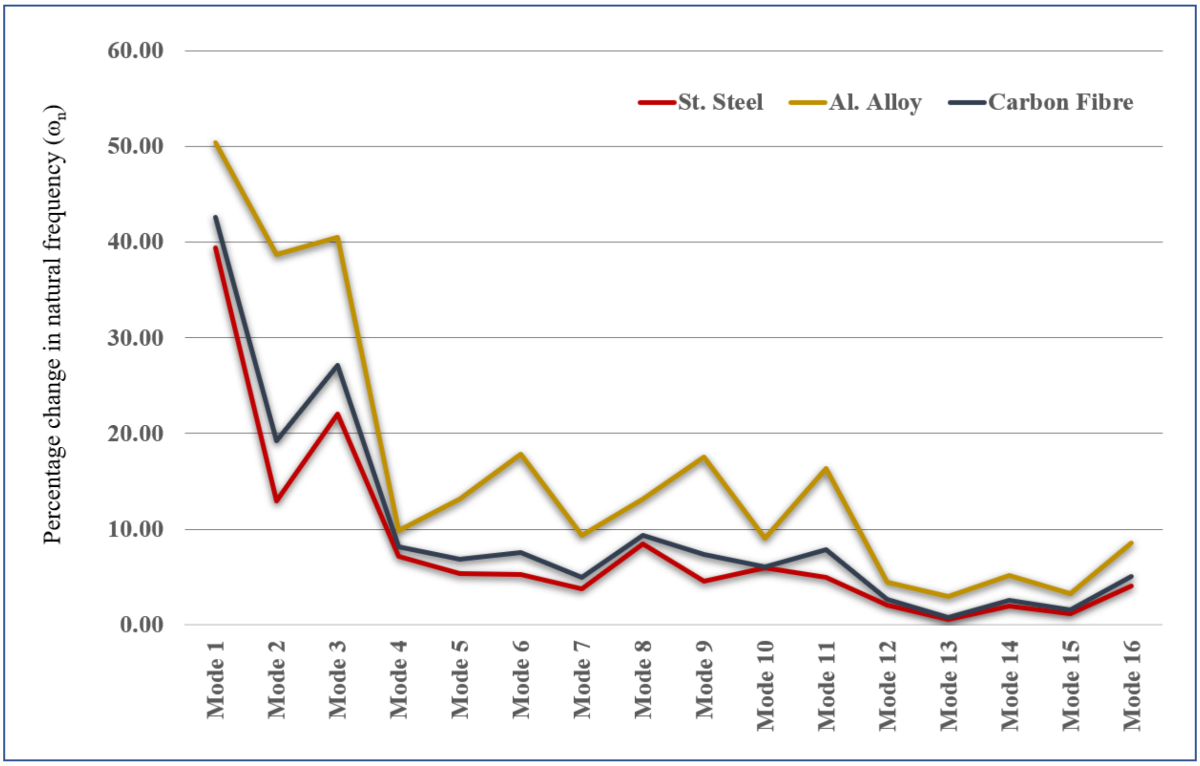


Figure 4.3 Percentage change between load pre-stressed and load free modes in all three materials

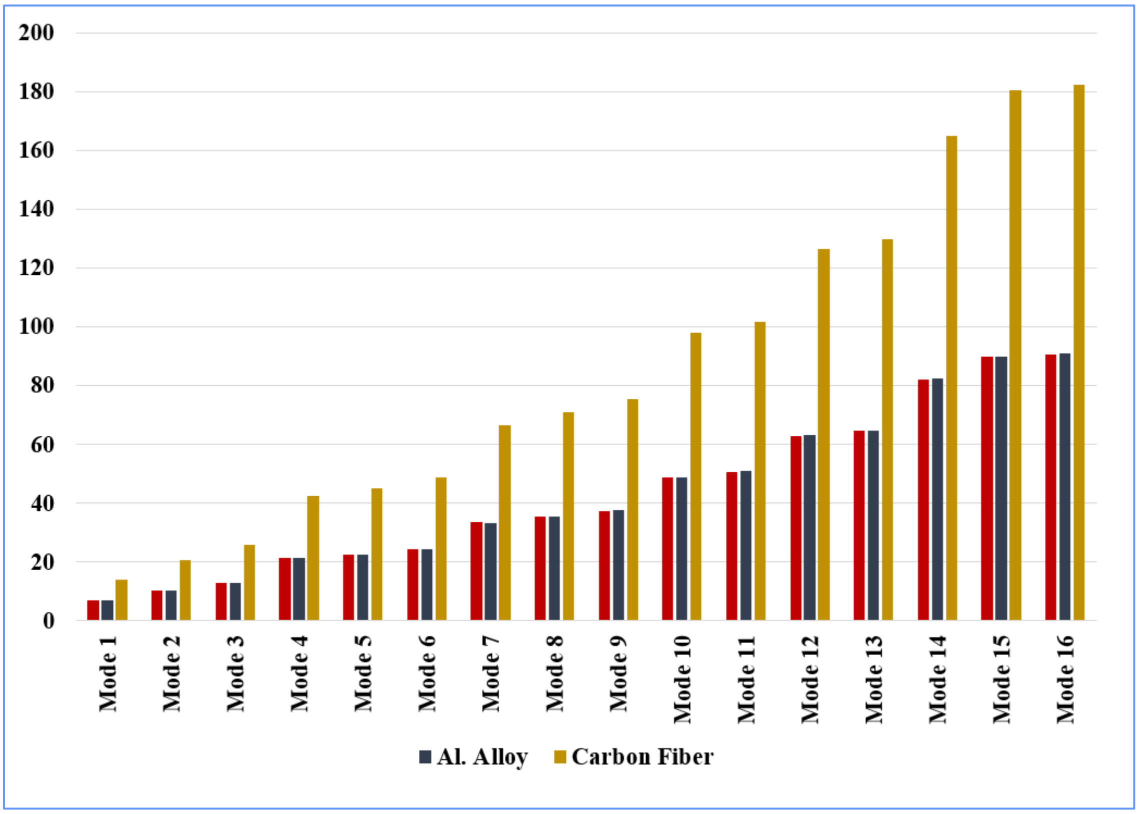


Figure 4.4 Comparison between load free modes of all three materials

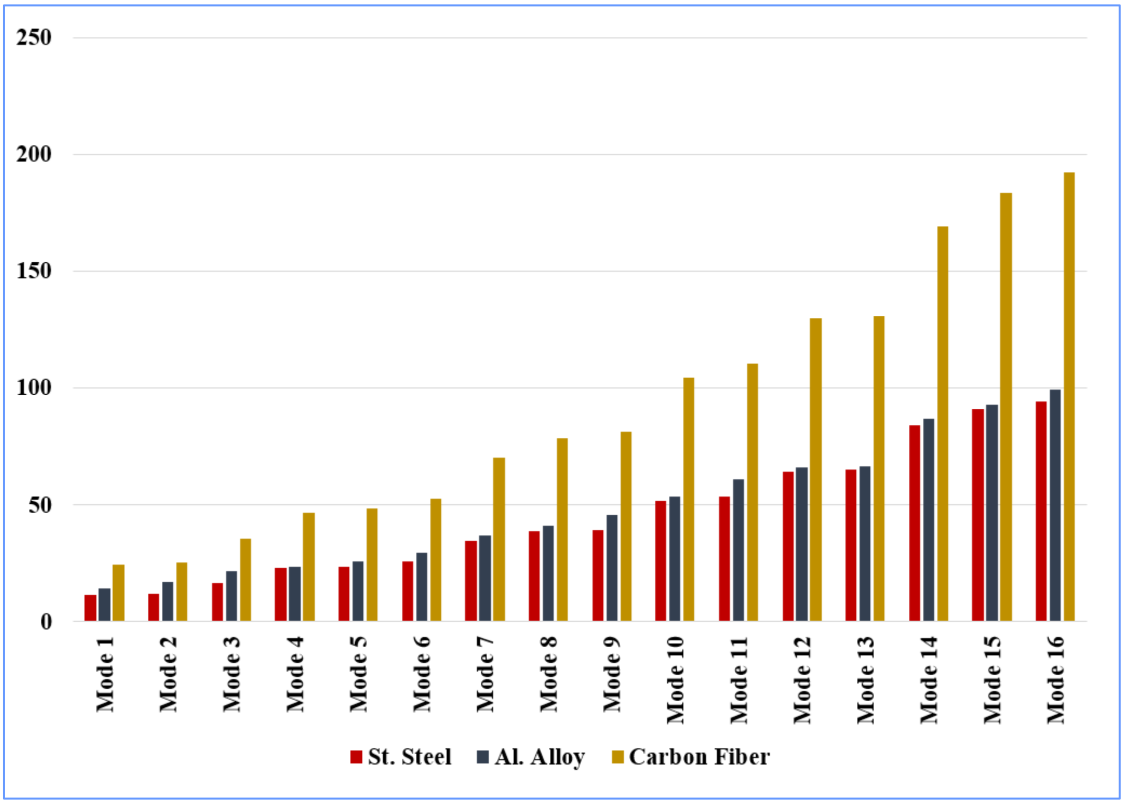


Figure 4.5 Comparison between pre-stressed modes of all three materials

The change in values of the frequencies is more prevailing in the lower order modes, for all three materials. Also, Young's modulus of carbon fibre is between that of steel and aluminum alloy, giving a pre-stressing effect to medium values, aluminum being the largest and steel being the least affected.

The following points can be deduced as seen from Figure 4.3 and the Figure 4.4, Figure 4.5. We can observe that the load-free frequency values for structural steel and aluminum alloy chassis are almost similar, but aluminum shows the maximum percentage change out of all three when pre-stressing is considered.

- Load-free modes of aluminum alloy and steel have the same natural frequencies whereas CF has significantly higher natural frequencies. The possible explanation could be that the ratio of densities and Young's modulus for Aluminum alloy and steel is approximately the same at 2.5 whereas CF has significantly lower (approximately 1/5.2) density compared to steel but it's Young's modulus is nearly 3/4 of that of steel.
- We can observe that from the 12^{th} order modes the percentage change in all the material is somewhat similar and a lot less as compared to the previous modes.
- The highlighted modes in red, for all three materials tend to align themselves and fall in operating engine frequency, whereas green highlighted modes tend to approach the road profile frequency.
- Although in past few pieces of research, the resultant change in frequencies was reported to be in the range of 0.1-0.2%, in our analysis we have found the change to be a lot higher, up to 39% even in the case of structural steel.

4.2.2 COMPARISON FOR INCREASING LOADS

A comparison has also been made for different amounts of payload acting on the chassis and the output mode frequency response for the base model, i.e., structural steel chassis in the following Table 4.6.

(X is denoted as a fraction of allowable payload).

Table 4.6 Change in natural frequencies with increment of pre-stressing

Mode No.	X = 0.6	X = 1	X = 1.4	X = 1.8
1	8.8083	10.286	11.547	12.183
2	10.755	11.263	11.742	12.624
3	14.309	15.525	16.635	17.628
4	21.863	22.406	22.769	22.996
5	22.722	23.142	23.546	23.924
6	24.483	24.93	25.502	26.136
7	33.796	34.231	34.65	35.041
8	36.638	37.929	38.554	38.922
9	37.779	38.169	39.161	40.304
10	50.275	51.364	51.711	52.036
11	51.046	51.863	53.342	54.716
12	63.34	63.775	64.199	64.598
13	64.586	64.706	64.831	64.959
14	82.732	83.275	83.806	84.311
15	90.084	90.449	90.809	91.152
16	91.924	93.204	94.365	95.379

The following deductions have been made by observing Table 4.6.

- Mode no. 10, 11 can be seen approaching the threshold value of 50-55 Hz (approximate average road profile frequency). The frequency in load-free conditions was 50.275 and 51.046 Hz. Similarly, the 5^{th} and 6^{th} mode are in the range of engine frequency.
- For natural frequencies close to the values of operational frequency, i.e., 55 Hz or 24-30 Hz, there may be a significant impact of small changes in the natural frequency value as the same could be approaching the exact road frequency.

4.3 HARMONIC ANALYSIS.

In this study, Harmonic analysis was done with the help of mode superposition principle. The inertial due to the vibration of the payload acts a harmonic load. However, to calculate the acceleration of the payload, one needs to calculate the transmissibility factor which in turn requires obtaining the damping ratio through experimental procedure. Therefore, this method of obtaining the acceleration of the payload remains outside of the scope of the present study. Another approach was taken to outsource reliable data related to acceleration and a value of acceleration = 0.57 m/s^2 was taken for our analysis. Analytically, the harmonic load occurring due to payload was calculated and applied on the structural steel made chassis. Refer to section 2.3.2 for further details.

Firstly, the deformation of the chassis in both pre-stressed and load free conditions was plotted by frequency response analysis. The result has been shown below, with the graph plotted in Figure 4.6 and Figure 4.7 respectively.

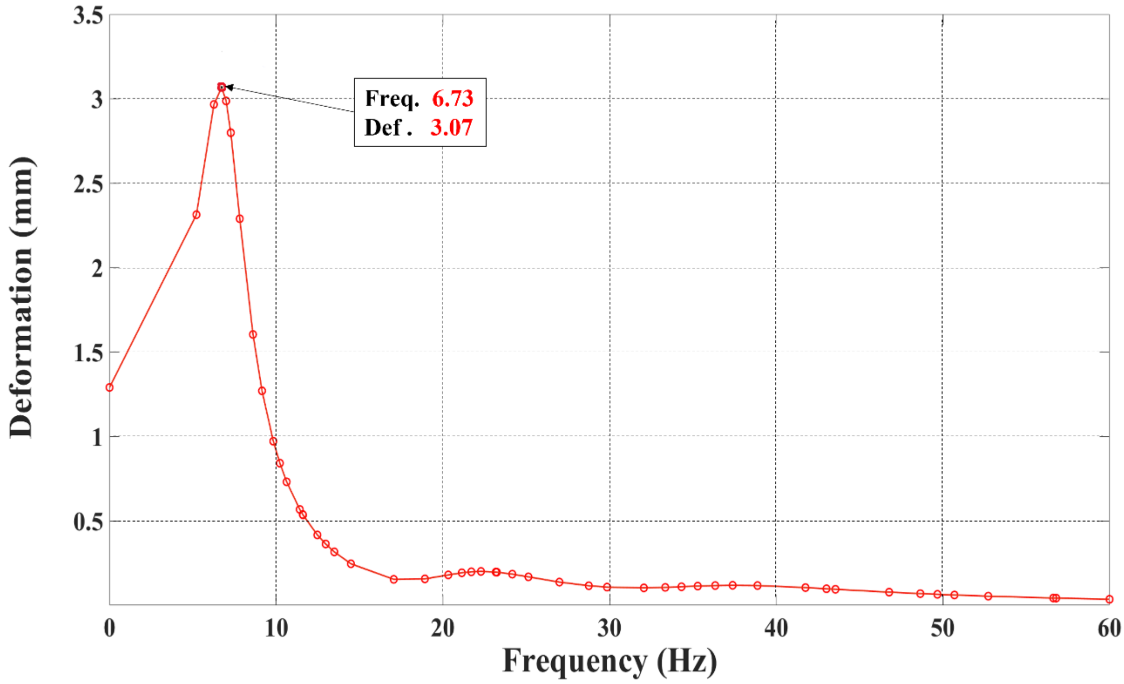


Figure 4.6 Deformation vs Frequency for load free chassis

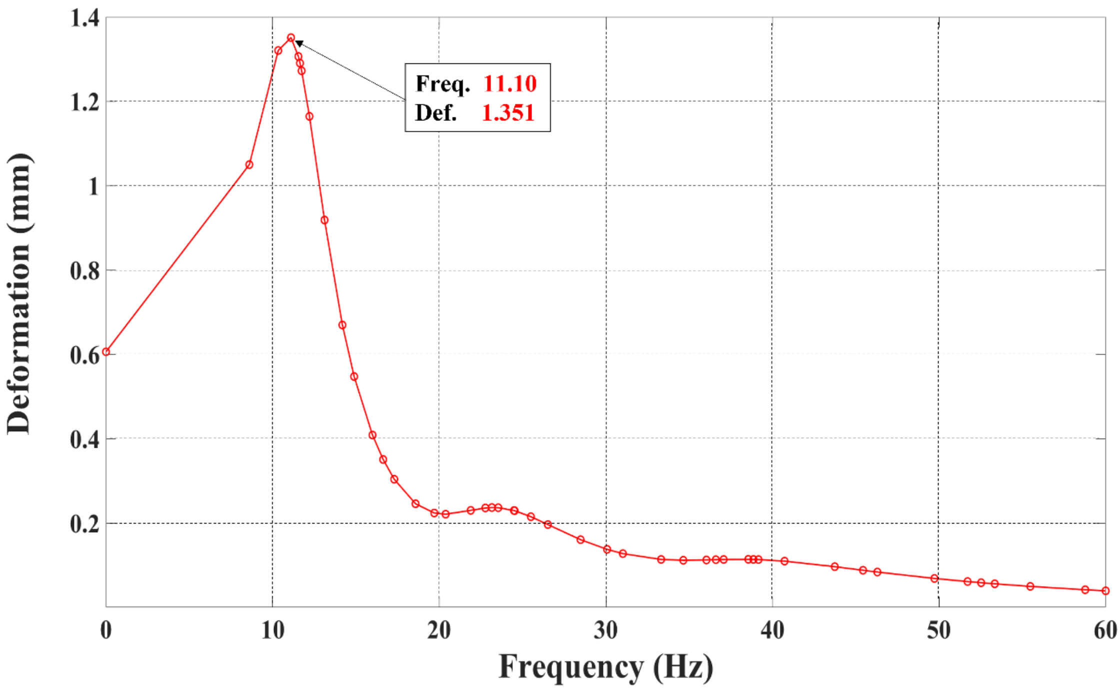


Figure 4.7 Deformation vs Frequency for pre-stressed chassis

The clustered result around the maximum deformation point has been analysed for both cases and is given in Table 4.7 below.

Table 4.7 Clustered result about the maximum deformation point

Pre-Stressed Chassis		**Load Free Chassis**	
8.60	1.050	5.21	2.313
10.34	1.320	6.26	2.966
11.10	1.351	6.73	3.067
11.55	1.306	7.00	2.986
11.65	1.290	7.28	2.798
11.74	1.272	7.81	2.289
12.21	1.164	8.61	1.604
13.12	0.918	9.15	1.270

As we can see, the maximum deformation occurs at 11.1 Hz for the pre-stressed chassis and 6.73 Hz for load free chassis. This is because of pre-stressing; the resonant frequency has shifted and pre-stressing can both increase and decrease (e.g., buckling) stiffness.

4.4 FATIGUE ANALYSIS

The bending stress due to harmonic response of the pre-stressed chassis is shown in Figure 4.8.

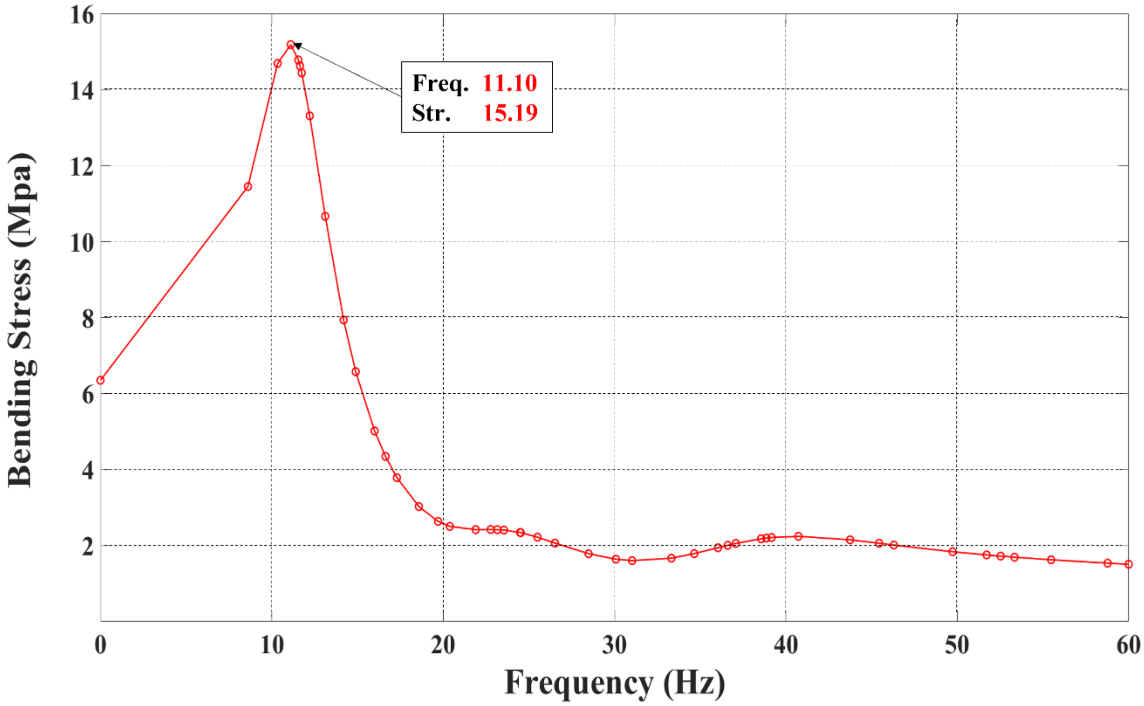


Figure 4.8 Bending stress on z-axis due to harmonic frequency

Now, by the data given in Table 4.8,

Payload Factor	X=0	X=1.4
Max Stress due to Static loading	229.29 MPa	291.39 MPa
Max Stress from Harmonic analysis	11.95 MPa	15.19 MPa
Total stress on the chassis (Maximum)	241.24 MPa	306.58 MPa

Table 4.8 Stress data for fatigue analysis

CASE – I (Base payload condition)

Calculating from the above values following the Goodman's criteria, mentioned in section 1.4,

$\sigma_{\mathrm{mean}} = 120.62\ MPa$

$\sigma_{var} = 120.62\ MPa$

$\sigma_{\mathrm{ut}} = 460\ Mpa$

$\sigma_{\mathrm{e}} = 230\ Mpa$

Thus,

$$\frac{120.62}{230} + \frac{120.62}{460} = \frac{1}{F.O.S} \qquad \textbf{4.1}$$

$$F.O.S = 1.271$$

Using the S-N approach formulae,

$$N = \left(\frac{120.62}{a}\right)^{\frac{1}{b}} \qquad \textbf{4.2}$$

Where (from equation 1.6 and equation 1.7),

$$b = \left(-\frac{1}{3}\right)\log\left(\frac{\sigma_{ut}}{\sigma_e}\right)$$

$$\log(a) = \log(\sigma_{ut}) - 3b$$

Calculating a and b, we get,

$$b = -0.1003$$

$$a = 919.72$$

Thus,

$$\boldsymbol{N = 6.3 \times 10^{8}} \textbf{ cycles}$$

The resultant value satisfies the infinite fatigue life criteria.

<u>CASE – II (1.4 times Payload)</u>

$\sigma_{\text{mean}} = 153.29\ MPa$

$\sigma_{var} = 153.29\ MPa$

Now,

$$\frac{153.29}{230} + \frac{153.29}{460} = \frac{1}{F.O.S} \tag{4.3}$$

$$F.O.S = 1.003$$

Using the S-N approach formulae,

$$N = \left(\frac{153.29}{a}\right)^{\frac{1}{b}} \tag{4.4}$$

a and b, as calculated from earlier case,

$$b = -0.1003$$

$$a = 919.72$$

Thus,

$$\boldsymbol{N = 5.7 \times 10^{7}} \textbf{ cycles}$$

Though, the resultant value satisfies the infinite life criteria, the fatigue life is reduced by a factor of 10 when increasing payload value by 40%.

5. CONCLUSION

1. The chassis structure was analysed under static loading and the values of deformation and combined stress was obtained as an output, ensuring the parameters for safe performance of the vehicle under overloading conditions.
2. Modal analysis was conducted for the truck chassis, evaluating the first 16 modes for both pre-stressed and load free conditions.
3. Structural steel has been observed to be the least affected by pre-stressing out of all three materials, though few modes were seen to undergo a change of up-to 39%, and aluminum was the most affected and the changes due to pre-stressing was up-to 50%. Whereas carbon fibre chassis' results fell between the range of 0 to 42%
4. The lower modes were seen to have a major percentage change of natural frequencies for all three materials. The change reduced as the mode no and natural frequencies increased.
5. A number of modes were observed to change values and align themselves towards the range of road profile frequencies or the engine operating frequencies, these can affect the dynamic performance of the vehicle significantly.
6. Harmonic analysis showed that the critical frequency value increases due to pre-stressing, though the increased stiffness also meant lower output response.
7. Fatigue analysis has been performed to show that the fatigue life is reduced by a factor of 10 when loading is increased by 40%

6. SCOPE OF FUTURE WORK

By observing and learning from the ample studies conducted by authors and researchers on various parameters of chassis modelling and dynamics, a humble approach with limited resource and capacity has been taken towards analysing the effect of pre-stressing on heavy vehicle chassis. This study was done in order to gain a deep and thorough understanding of their real-life performance. Many steps are yet to be taken for overcoming the drawbacks and to reach towards perfection. The same or a better approach could be taken for a complex chassis design considering the joints and stress concentration points. Also designing and testing the chassis in real life would provide a validation to the obtained results. Furthermore, to counter any sudden disturbances of an uneven road surfaces, Transient and Random vibration response of the chassis considering pre-stressing could ensure the overall dynamic performance of the vehicle.

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