ADDIS ABABA UNIVERSITY
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SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING
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Structural Design & analysis of passenger rail vehicle car body

By

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A Thesis submitted to the School of Graduate Studies of Addis Ababa University in partial fulfillment of the requirements for the Degree of Masters of Science in Mechanical Engineering

(Railway stream)

Advisor:-Dr.Ing Demis Alemu.

SCHOOL OF MECHANICAL & INDUSTRIAL ENGINEERING
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DECLARATION

I hereby declare that the work which is being presented in this thesis entitled as "Structural Design & analysis of passenger rail vehicle car body" is original work of my own. The work contained in this thesis has not been previously submitted for a degree or diploma at any other higher education institution. To the best of my knowledge and belief, the thesis contains no material previously published or written by another person except where due reference is made and all the resource materials used for this thesis had been accordingly acknowledged.

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ABSTRACT

The purpose of the study is to design and analyze a rail vehicle car body structure that solves the issue of strength & comfort. Modern rail vehicles passenger Car bodies are designed as lightweight structures with the aim to minimize mass and thus operational energy demand. At the same time the vehicle car body shall be designed as stiff as possible to withstand exceptional loads. To achieve these, both rigid and flexible body vibration frequencies shall be filtered by appropriate design of suspension system. The materials used in the design and construction of passenger rail vehicle car body structure is structural steel(st 36) that has greater resilience to different production methods and available in local market compared to other materials used for the same application. The result shows the selected sections for the different components of the rail vehicle are capable of carrying the intended purpose. However, some sections are critically loaded beyond the allowable yield stress. These sections have to be reinforced to make them stiff enough that they bear the load. In addition, the first six natural frequencies are calculated using ANSYS design and modeling software. The frequency is compared to the sensitivity range of human being in vertical ride comfort. The design and analysis is applicable to Rail Way Corporation & local manufacturer's that has plan to substitute import by local manufacturing of passenger rail vehicle car bodies

ACKNOWLEDGMENT

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My appreciation goes to Ethiopian Railways Corporation for its financial support and Addis Ababa Institute of Technology for its unreserved provision of resources for undertaking this thesis and relevant information and advice.

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LIST OF SYMBOLS

m_1	Design mass of rail vehicle car body without the bogie
m_2	Mass of a single bogie
m_1'	Dead weight of vehicle in working order
m_3	Mass of normal design pay load
m_4	Mass of exceptional payload
m_w	Mass of wheel
c_1	Damping coefficient of primary suspension
c_2	Damping coefficient of secondary suspension
k_1	Spring stiffness coefficient of primary suspension
k_2	Spring stiffness coefficient of secondary suspension
$z_c(t)$	Rail vehicle Car body displacement
$z_b(t)$	Bogie vertical displacement
$z_w(t)$	Rail vehicle wheel displacement
$z_t(t)$	Displacement of track
g	Acceleration due to gravity
σ_{all}	Allowable working stress
σ_y	Yield stress
z	Section modulus

M	Bending moment
$M_{\max}$	Maximum bending moment
R_2	Reaction force at point 2(bolster support)
R_3	Reaction force at point 3(bolster support)
L_1	Span of support between bolsters
L_2	Span of support between bolster
w	Load intensity
f	Frequency
F_{pa}	Compressive force at passage ways and access area
A_{pa}	Area of access and passage ways
F_{cv}	Superposition of compressive and vertical load
F_c	Compressive load
F_{tv}	Superposition of vertical and tensile load
P_{tv}	Pressure due to superposition of compressive & tensile load
P_{cv}	Pressure due to superposition of compressive & vertical load

CHAPTER ONE

INTRODUCTION

1.1. BACKGROUND

Rail travel is undoubtedly one of the safest modes of transportation & by far the least expensive for long and short distance. Its preference over other means of travel has grown from the time of the industrial revolution. Construction of a passenger car body structure having sufficient stiffness with respect to vertical lateral bending, cross-section as well as stiff in bending & torsion is the crux of the thesis..

The railway train running along a track is one of the most complex dynamical systems in engineering. It has many degrees of freedom, the interaction between wheel and rail involves both complex geometry of wheel tread and rail head and non conservative forces generated by relative motion in the contact area, and there are many non-linearities. The long history of railway engineering provides many practical examples of dynamical problems which have degraded performance and safety. Dynamic and static instabilities, and excessive response to track irregularities and other features of track geometry, can result in poor ride quality and high stresses and can contribute to derailment. High frequency interaction between wheel and rail can lead to damage to the contacting surfaces and corrugation of the rails, and excessive noise and vibration. This requires appropriate strength, stiffness and fatigue properties of the components to be able to stand these loads. On top of that, quality of a vehicle, as a system, which include efficient energy consumption, safety, and provision of comfort to the user are highly desired.

Although, design phase by using the FE analysis of the railway vehicle models take certain time, it can be used easily for a number of cases such as different crash scenarios with different speeds and all stages of design and improvement by using its economy and flexibility.

Static structural, vibration, crashworthiness and optimization studies of railway vehicles should be performed to provide optimum safety, comfort and energy saving. Full-length & width modeling approaches are used to determine static structural, vibration and crash behaviors of railway vehicles depending on the symmetry of vehicle. Nevertheless, half- width / full-length and half-width/half length modeling approaches can be used only for certain types of simple static structural loading such as static symmetrical compression and symmetrical tension loading conditions and cannot be validated for complex static structural loading and dynamic loading conditions. Hence, full-length simulations are important for the validation of designs and to provide the greatest possible accuracy. To obtain static structural behavior of the railway vehicles, i.e., stress and strain distribution, different static loading cases defined in standards such as EN12633-1:2010 and EN 12577:2008+A1:2010 can be used in FE analyses. The natural frequencies and elastic mode shapes of railway vehicles, which represent its vibration characteristics, can be obtained by using FE methods.

In this paper, considering the above mentioned concepts, analyses of structural stress, vibration of structures & passenger are completed. Full-length detailed FE models are used to assess static, vibration and structural behaviors of the railway vehicles. To obtain structural behavior of the railway vehicles, different static loading cases defined in standards are used. To obtain vibration behavior of vehicles normal mode (free vibration) analyses are performed. .As a result of all simulations, structural weaknesses and suggestions for improving are suggested.

All the above largely demand refined and complex design and manufacturing procedure and discipline during the production stage. This, in turn, requires good understanding of the internal systems of the vehicle and the characteristics of the different body structures in reaction to static and dynamic loads.

Vehicle dynamics, a discipline of broader significance, is an area where the basics of analyses on vehicles are dealt with. Forces/loads acting on rail vehicles can be categorized as road, aerodynamic, and gravity loads. The response of the passenger rail vehicle structure to these loads is dealt with in vehicle dynamics.

The responses of a vehicle are defined in terms of deflections, stresses, and strains, natural frequencies, random response functions, fatigue life and so on. Evaluation of the above is what puts the basis on which robustness of a rail vehicle system or design is ascertained in terms of its mechanical behavior.

Different researches have been carried out regarding the performance, the response of components to static and dynamic loads, crashworthiness, safety and others by different institutions and automotive companies. Particularly, with the growing simulation capability using computers, researches are facilitated that are aimed at achieving better quality products. The application of computer aided engineering (CAE) analysis to problems of this sort, in combination with prototype development and testing, enables to achieve structures having longer fatigue life, reduced cost, light weight and improved comfort. In light of this purpose, as stated earlier, advancements in the area are growing further.

In our country and further, the use of researches and computer aided engineering analysis is far below what it should be. However, as mentioned earlier, these are the bases for achieving structure having longer fatigue life, reduced cost and reduced weight. Thus, this prompts implementation of existing theoretical knowledge and facilities towards addressing the problem in the area.

For the analysis, FEM, commercial design software (ANSYS), CATIA V5-R16, together with motive power design and vehicle dynamics are applied.

1.2. OBJECTIVE OF THE STUDY

1.2.1. General objectives

The theme of the thesis is to design & analyze the structure of a passenger rail vehicle car for structural stiffness, and ride comfort as per the requirement of Ehiopian Railways Corporation. The purpose of conducting research on this topic in Ethiopia is to understand the dynamics of rail vehicles cars when exposed to loading conditions such as but not limited to exceptional load, quasi static loads, variable speed limits and come up with optimal rail vehicle car body structure.

1.2.2. Specific objectives

The primary objective of the design & analysis of passenger rail vehicle car structural bodies in Ethiopia is to avoid dependence on foreign skill and resource for every item that can be designed and manufactured within the country. Domestically designing rail vehicle structural car bodies for passenger comfort saves import of components and saves foreign currency and hence has a huge economic advantage.

1.3. MTHODOLOGY

To fulfill the objectives of the study the following techniques are used.

i. Literature Review:-

Survey of books, journal articles, proceedings of international conferences, auto manufacturer catalogues from reliable websites, and other relevant literature is done.

ii. Data Collection:-

Data regarding the rail vehicle body structure are collected from specifications already set by Ethiopian rail Ways Corporation.

The data include dimensions of the structure, material used to construct the structure, and construction methods. Other data used in the thesis are obtained from books, and reliable websites.

iii. Modeling and Analysis:-

The finite element model development is done by CATIA V5-16 and the corresponding static and modal analysis is performed using ANSYS work bench.

iv. Conclusions and Recommendations.

1.4. ORGANIZATION OF THE THESIS

The body of this thesis is divided into five main chapters. The first chapter discusses background and objectives of the study. In addition, the details of the model passenger rail vehicle used in the analysis are also discussed in the same chapter. The second chapter covers the review of some of the journal articles, proceedings and publications which were referred to during the course of the thesis. Also, in relation and comparison with previous works, what is done in this thesis will be stated. The mathematical and finite element modeling is discussed in the third chapter. Multi body models representing different modes of a rail vehicle structure are presented. This chapter also covers discretization of the solid model of a passenger rail vehicle used for the analysis into finite elements and the mathematical formulation of these elements. The results obtained from the static and modal analysis of the vehicle are included in the fourth chapter. Finally, the fifth chapter covers conclusions drawn based on the results of the analysis, and recommendations for future work.

1.5. SCOPE AND LIMIT OF THE THESIS

In real time application, numerous types of analysis can be carried out regarding behavior of a vehicle. In this research, static and modal analysis of a passenger rail vehicle structure is carried out. Nonlinear behaviors, which components could experience, are not dealt with and, linear and isotropic material model is used for the entire analysis. The analysis is carried out for the main load bearing substructure of the rail vehicle, i.e., under frames and cross bearers, side sills of the passenger rail vehicle car body structure.

The analysis of rail vehicle body design and analysis requires a super fast computer and the results obtained on this thesis might have been refined in the presence of these resources.

CHAPTER TWO

LITERATURE REVIEW

The dynamics of the railway vehicle represents a balance between the forces acting between the wheel and the rail, the inertia forces and the forces exerted by the suspension and articulation. Of these, the basic characteristics of the wheel-rail interface such as friction, geometry, and the elasticity in the contact area are hardly under the control of the designer.

In a complete model of the dynamics of a railway vehicle, the vehicle is considered to be assembled from wheel sets, car bodies and intermediate structures which are flexible, and which are connected by components such as springs and dampers.

The rail vehicle car body plays an important role in rail vehicle dynamics. The primary interest is structural stiffness & ride comfort. In this, the range of frequency & structural flexibility of the rail vehicle body is of major concern. Key parameters are proposed to select the rail vehicle car body Eigen modes for inclusion in a flexible model, and to quantify the interaction between passengers & rail vehicle car body [1].

Before putting a railway vehicle into service, it must be certified. Static and dynamic tests are performed according to international standards during the certification procedures. Some standards, specifications and requirements for static stress tests, vibration and crashworthiness of rail vehicle structures are developed. Numerical methods and experimental measurements are used to determine static and dynamics behavior of railway vehicles. Experimental measurements are very time consuming, expensive and they cannot be used at all stages of design. Hence, today numerical methods are important tools in static and dynamic analyses of railway vehicles. Although numerical simulations do not have aforementioned disadvantages of experimental methods, they must be verified by experiments to obtain realistic results. FE method is a powerful numerical engineering analysis tool, and widely used in static and dynamic stress analyses of railway vehicles [6].

Whole-body vibrations are transmitted to the human body of the passengers in a bus, train or when driving a car. The ISO 2631 standard provides an average, empirically verified objective quantification of the level of perceived discomfort due to vibrations for human passengers [10].

The ride quality of a modern railway vehicle is mainly determined by forces of inertia acting on the rail vehicle car body and therefore on the passenger. These accelerations result from the vibrations of rigid body modes as well as elastic or flexible modes of the car body. So as to keep these accelerations as small as possible, the rail vehicle body and the secondary suspension have to be designed carefully.

One possibility to improve the ride quality is the usage of active (controlled) secondary suspensions Another or additional possibility is the use of active vibration damping of the elastic car body structure, which is also the focus of this research[20].

2.1. PASSENGER RAIL VEHICLES

To this group belong all types of railway vehicles intended for the transport of passengers, ranging from main line vehicles, suburban and urban transit stock to tramways. Passenger vehicles are divided into five structural design categories into which all vehicles may be allocated.

According to European norm for design, there are five categories of rail vehicles, with an indication of the types of vehicle generally associated with each other:

a. Category P-I e.g. coaches;
b. Category P-II e.g. fixed units and coaches;
c. Category P-III e.g. underground, rapid transit vehicles and light railcar;
d. Category P-IV e.g. light duty metro and heavy duty tramway vehicles;
e. Category P-V e.g. tramway vehicles.

As a general requirement; rail vehicle car body should have limited weight. Trains exert forces on the infrastructure due to their weight which results in wear of the tracks. High weight also increases the wear on wheels, axles, brakes, shock absorbers, etc. The car body should also be sufficiently stiff. From a safety perspective the car body should not flex outside the track gauge during operations or show significant vertical displacement due to the passenger load.

The main parts of a railway vehicle are the body structure and the bogies. A bogie for passenger rail vehicles consists mainly of two wheel sets, the bogie frame, the bogie related traction and braking components, the primary and the secondary suspensions and also the anti-rolling device between bogie and car body. The bogies support the structure of the car body [6].

The rail vehicle car body refers to the load carrying structure, doors, windows, interior with seats etc, inner lining and so-called comfort systems for lighting, heat, ventilation and sanitation.

The technical equipment for propulsion, braking etc is by definition not included in the rail vehicle car body, even though this equipment usually is attached/hinged on this (commonly under the rail vehicle car body). Sometimes the concept rail vehicle car body is limited to only the load carrying structure of the vehicle.

The rail vehicle car body usually holds the main part of the vehicle mass. The body structure of passenger rail vehicle car is assembled from substructures, which consist of several parts and are fabricated by spot welding. These substructures are classified into two groups. One of them consists of members and pillars that are of the closed-section type, or box-type, substructures, and the other consists of panel parts, such as a roof panel, center floor panel, rear floor panel etc. The stiffness of the body structure for passenger cars depends mainly on the substructures of members

The rail vehicle car body must meet a number of requirements, including: safety requirements set up for crash scenarios, derailment, projectiles impacts, pressure waves in tunnels, etc. It must be strong enough as not to fail during typical maximum loads or during cyclic loading. A large amount of these requirements are, covered by the norm EN 12663-1:2010 [18]. Railway vehicle bodies shall withstand the maximum loads consistent with their operational requirements and achieve the required service life under normal operating conditions with an adequate probability of survival.

The capability of the railway vehicle body to sustain required loads without permanent deformation and fracture shall be demonstrated by calculation and/or testing. The strength of connections between structural members of the rail vehicle car body shall exceed the ultimate load-carrying capacity of the weakest member joined. Further increase in deflections will result in a decrease in the load capable of being carried by the structure.

The assessment is based on the following criteria:

a) Exceptional loading defining the maximum loading which shall be sustained and a full operational condition maintained;
b) Margin of safety as defined in such that the exceptional load can be considerably exceeded before catastrophic fracture or collapse will occur;
c) Service or cyclic loads being sustained for the specified life without damaging the structure.

The design should, furthermore, be reasonably easy to manufacture and maintain, it should be possible to repair damages to the rail vehicle car body.

A modern rail vehicle should look modern. Beside these requirements, the rail vehicle car body must full fill comfort requirements. For passenger rail vehicles, the body must provide the correct environment, e.g. a good ride comfort, the right lighting, space, temperature, fresh air and a low sound level. Due to the coupling of the bogies and the car body the elastic modes of the car body are excited by track irregularities (broadband excitation).Vibrations excited from the track and bogies, can lead to severely reduced comfort, it is therefore; desirable to separate the natural frequencies of the car body from those of the track and bogies. In most coach applications it is important to consider the car body structural flexibility so that car body vibrations and their negative effect on ride comfort are represented. Rail vehicle car body accelerations increase in amplitude and frequency when the structural flexibility is modeled in the vehicle – track simulations.

Frequencies about 10 Hz are usually prominent and for vertical accelerations ,this can cause significant discomfort as people are most sensitive to vertical accelerations of 8-10 Hz. Slender car bodies in vehicles running at fairly high speed on relatively poor track are most prone to this kind of acceleration amplification[1].

For the ride quality of a railway vehicle next to the rigid body movement the first elastic modes contribute the predominant part. For passenger rail vehicles, typically, four different first elastic mode shapes with distinct characteristics arise. These are the vertical and horizontal, the diagonal distortion mode & bending mode. To classify the different mode shapes the deflection and the distortion of the cross section can be used, respectively.

The finite-element detailed model of the body structure for passenger cars can be used as a tool that can predict its performance. Based on these parameters, the eigen frequencies of the rail vehicle car body will be calculated to make sure that the system is stiff enough and doesn't resonate. Rail vehicle car body vibration can be dealt with by focusing either on the stiffness of the structure or on the suspension system .Otherwise, it becomes susceptible to car vibration and passenger discomfort. The standard Static, dynamic & fatigue analysis and vertical, lateral & longitudinal acceleration will be taken as granted from EN 12663-1:2010 [6][12][19].

In general, a passenger rail vehicle car body shall have the following characteristics:- [9]

- Low weight to reduce the energy consumption and the track forces,
- good ride comfort on board even at high speed ,
- high and flexible seating capacity,
- Continuous (clear and open) passenger compartment
- Large hollow aluminum extrusion profiles or steel , large stiffness
- Lighter in weight to reduce the dead weight and operational cost of the locomotive.
- Recyclable, low cost
- Good corrosion resistance, can realize none painting design
- Large automatic welding range, high production efficiency

In order to reduce weight, the rail vehicle car body structure is made with an assembly of Aluminum extrusions up to 21m long, with machined apertures for windows and doors. The various extrusions can be welded or bolted to one another. Fig 1 shows a typical rail vehicle car body assembly [7].

With the primary aim of a further reduction in mass, in addition to metallic materials and plastics, mainly fiber-reinforced synthetic (FRS) materials are increasingly used in rail vehicle car body construction.

The outstanding, weight-specific properties of FRS, the excellent damping property, high resistance to environmental degradation and the high functional integration to various demands to be achieved makes FRS material can appear as a promising material for railway applications.

For the production of components made of fiber-reinforced plastics, there are numerous manufacturing processes. For use in rail vehicle car body construction press method and bending techniques are used in the first place. When using large FRS parts from the press usually executed in steel sheeting a differential design is replaced by large plastic components.

Components that form rail vehicle car body structure are created using various manufacturing processes such as casting, extrusion and welded plate's assembly. Various materials are considered; mainly aluminum, steel and Glass-Reinforced Plastic (GRP).They are designed and welded in accordance with AWS D15.1, AWS D1.1 for steel, and AWS D1.2 for aluminum. Such rail vehicle car bodies are assembled using bolts, rivets and welding which are assessed during FE validation. [18]

A computer aided design (CAD) or finite element (FE) analysis can give the mass, centre of gravity position, and mass moments of inertia. However, in most applications the mass of the car body structure is less than half the car body mass. For a coach the additional mass is a result of the interior layout and interior/exterior equipment. For powered vehicles, especially locomotives, the car body also carries some of the traction equipment. Often, it is quite hard to keep accurate records of all the pertinent masses and positions. Usually, the mass of this equipment is merged to the metal structure model. In many vehicles – track dynamics simulations of the payload, i.e., passengers or goods, also needs to be considered. The corresponding mass might also be combined with the car body structure model.

In dynamic simulations of railway vehicles, it is desirable to represent the car body structural flexibility by a limited number of degrees of freedom. These rigid body degrees of freedom are: longitudinal, lateral, vertical, roll, pitch, and yaw motions. The most common way of finding such a representation is to use the body's Eigen modes found from Eigen value analysis of the free body. Then, we get six rigid body modes and a number of structural modes, preferably those with the lowest Eigen frequencies [8]. The car body vibrations can be divided in two types of modes: *rigid* and *flexible*. The rigid body modes that affect vertical ride comfort are bounce, pitch and roll. These modes normally lie in a relatively low frequency range, around 1 Hz. The flexible modes are twisting and bending deformations of the car body due to constraints acting on it.

In a first step complete structure of rail vehicle bodies were developed, optimized and evaluated based on global stiffness requirements. The knowledge gained in this step was used as requirements for the strength and stiffness of panels during the continued development of the multi-functional optimization [13].

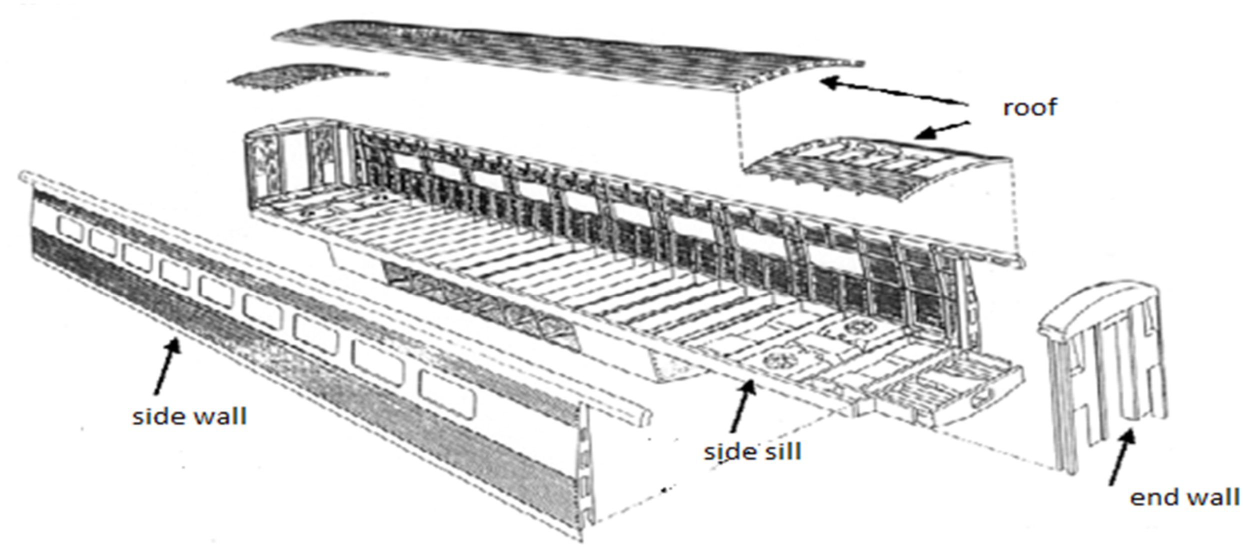


Fig.1. Main elements of a passenger rail vehicle structure [7].

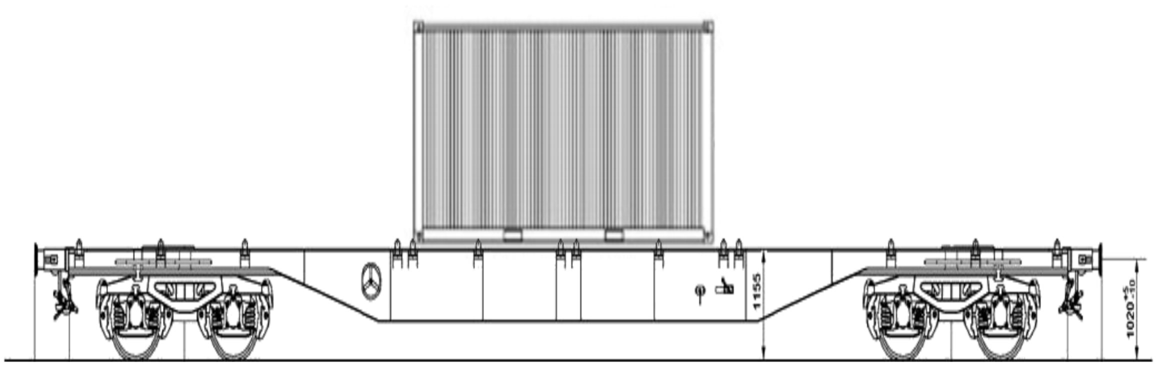


Fig. 2. Under frame of a rail vehicle loaded [18]

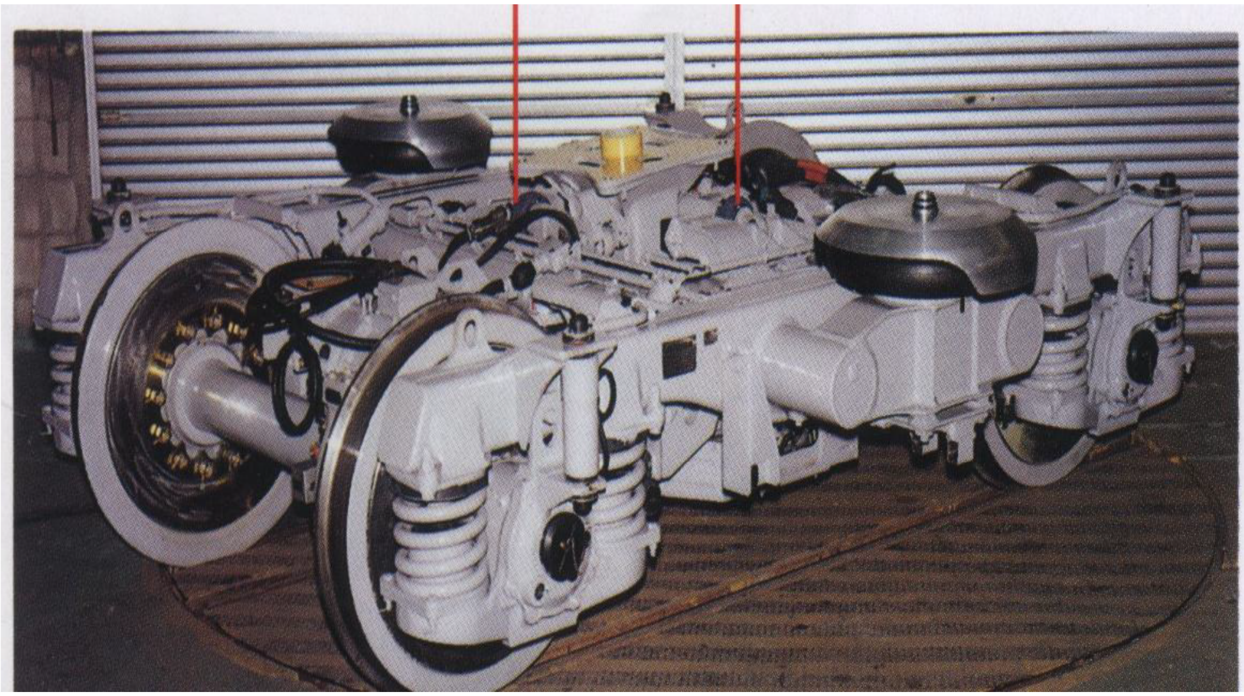


Fig 3. Bogie of a rail vehicle.

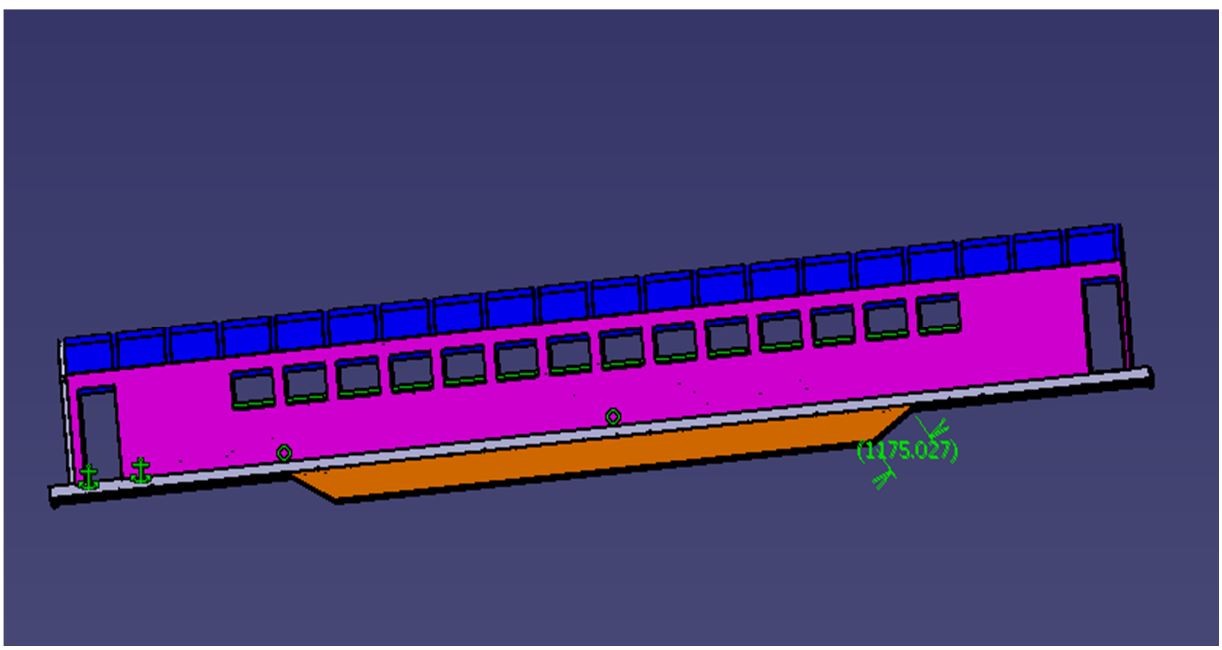


Fig 4. Typical 3D model of a passenger rail vehicle

2.2. DESIGN LOAD REQUIREMENTS

The rolling stock car body structure is facing severe problems on wear and reliability with continuously improving operational speed of trains. Most of the works are aimed for lightweight structures, improved running life time, the fatigue life prediction .With the increase of commercial speeds of railway tracks, the car body structures of these vehicles are subject to serious fatigue problems. In the past years, the general strength evaluation method of a car body was usually performed by a static load test and finite element analysis (FEA). Much of the research in modern locomotive vehicle industry is aimed to produce safe, reliable and light weight car body structure to tackle the increased speeds of railway vehicles. All loads used in this passenger rail vehicle car body structural design has incorporated any necessary allowance for uncertainties in their values derived from on-track tests or other sources of information.[15]

2.2.1. definition of the design masses

For the case under consideration, the dead weight of the vehicle (bogie +car) is 44.5 ton. And the mass of a single 209P bogie is considered to be 3 ton .According to [3] the following definitions are applied.

a. Dead weight

The dead weight of the vehicle in working order is the mass of the structure, bogies, staff and consumables together.

b. Consumables

It is the weight of sand, water, catering materials, fuel, food and beverages including water for drinking and cooking, clean water of wash basin supply reservoirs and of toilet supply, reservoirs.

c. Standing area

According to EN 15663, the standing area is calculated with tip up seats and folding tables in closed position and by taking into account half the projected area of internal stairways.

Standing area doesn't include:-

- Projected areas of normal seats (including backs and armrests) on the floors plus a 300 mm deep area for the feet of the seated passengers, which extends over the full width of the seat;
- Projected areas of fixed tables on the floors;
- Area restricted to drivers and other train crew and/or areas where standing is prohibited;
- Steps and other areas that are only used when boarding or alighting;
- Areas, except stairways, that due to their limited dimensions (width or length under 300 mm), are unsuitable for standing;
- Areas where the height is less than 1 850 mm;
- Areas which are used for toilets, washing areas or similar.

d. Luggage area

According to EN 15663, the luggage area does not include overhead racks.

Table 1. Definition of mass cases for different rail vehicle [18] categories

Definition	Symbol	Description
Design mass of the vehicle body in working order	m_1	The design mass of the vehicle body in working order according to EN 15663 without bogie masses.
Design mass of one bogie or running gear	m_2	Mass of all equipment below and including the body suspension. The mass of linking elements between vehicle body and bogie or running gear is apportioned between m_1 and m_2
Normal design payload	m_3	The mass of the normal design payload as specified in EN 15663.
Exceptional payload	m_4	The mass of the exceptional payload as specified in EN 15663.

2.2.2. design loads

This is to define load requirements to be used for the design of heavy rail transit vehicles, including static loads representing normal and exceptional conditions. The structure of the car body shall be completely assembled with the loads of all equipment included before the specified loads are applied. Each specified force shall be applied over the minimum area necessary to limit local yielding or buckling, with its center of action at the location specified.

Rail vehicles, except non-passenger carrying locomotives with non-structural equipment hoods, shall be designed to rest on their roofs so that any structural damage in occupied areas is limited to roof sheathing and framing members. Deformation to the roof sheathing and framing is allowed to the extent necessary to permit the vehicle to be uniformly supported directly on the top chords of the side frames and end frames. For this condition, the allowable stress for the structure of the occupied zones of the car body shall be one-half yield or one-half the critical buckling stress, whichever is less.

Non-passenger-carrying locomotives with non-structural equipment hoods shall be designed such that in the event of a rollover, the operator's cab will maintain a survivable volume. The manufacturer shall show by layout and calculation (classic or FEA) that the locomotive is capable of resting upside down at two or more points of contact while simultaneously maintaining a survivable volume within the operator's cab. The points of contact may be a major piece of equipment (for example, the diesel engine and transformer), one end of the platform or the other (depending on the location of the center of gravity) or structural members added to satisfy this requirement. Deformation of equipment enclosures and operator's cab roof sheathing is allowed to the extent necessary to permit the vehicle to be supported as described above. The allowable stress for structural members added to the structure specifically for this load case shall be one half yield or one half the critical buckling stress. The load applied to the structural members shall be determined from a static balance calculation while the locomotive is upside down assuming only the truck adjacent to the operator's cab is still attached to the structure

Table 2. Load cases for different rail vehicle categories [18]

Category	Exceptional pay load(m_4)	Service (Fatigue) pay load(m_3)
B-I main line passenger rolling stock [1,2,3]	• 1 passenger per seat • 4 passengers per/m^2 in passage ways, access and service areas • 300kg/m^2 in luggage areas • passenger mass =80kg	• 1 passenger per seat • Up to 2 passengers per/m^2 in passage ways, access and service areas • 300kg/m^2 in luggage areas • passenger mass =80kg
B-II Inner & outer sub urban passenger rolling stock [1,2,3]	• 1 passenger per seat • 5-10 passengers per/m^2 in passage ways, access and service areas. • 300kg/m^2 in luggage areas • passenger mass =70-72kg	• 1 passenger per seat • Up to 6 passengers per/m^2 in passage ways, access and service areas • 300kg/m^2 in luggage areas • passenger mass =70-72kg
B-III Metro & rapid transit rolling stock [3,4]	• 1 passenger per seat • 5-10 passengers per/m^2 in passage ways, access and service areas • 300kg/m^2 in luggage areas • passenger mass =70-72kg	• 1 passenger per seat • Up to 6 passengers per/m^2 in passage ways, access and service areas • 300kg/m^2 in luggage areas • passenger mass =70-72kg
B-IV Trams	• 1 passenger per seat • 6-8 passengers per/m^2 in passage ways, access and service areas • 300kg/m^2 in luggage areas	• 1 passenger per seat • up to 6 passengers per/m^2 in passage ways, access and service areas • 300kg/m^2 in luggage areas • passenger mass =70-75kg

	• passenger mass =70-75kg	
B-V - B-VI Freight rolling stock [7]	• Maximum pay load	• Maximum pay load
B-VII [8] locomotives[8]	• Zero payload, i.e is vehicle in normal working order with all supplies	• Zero payload, i.e is vehicle in normal working order with all supplies

Table 3. Compressive force at buffers and/or coupler attachment [18]

Force in killo Newtons(kN)

Locomotive	**Passenger rolling stock**					**Freight wagons**	
Category	Category P-I	Category P-II	Category P-III	Category P-IV	Category P-V	Category F-I	Category F-II
2 000	2 000	1 500	800	400	200	2 000 a	1 200 a
a Compressive force applied to the draw gear stops "c", if this draw gear stop is used (see EN 12663-2).When the compressive force is applied on side buffers, then half of the value shall be used for each buffer axis.							

Table 4. Tensile force at coupler attachment [18]

Force in killo Newtons(kN)

Locomotive	**Passenger rolling stock**					**Freight wagons**	
Category L	Category P-I	Category P-II	Category P-III	Category P-IV	Category P-V	Category F-I	Category F-II
1 000 a	1 000 a	1000	600 b	300 b	150 b	1 500 c 1 000 d	1 500 c 1 000 d
a. A higher force (e.g. 1 500 kN) may be necessary for certain types of coupling. b. These values can be adjusted but shall cover the maximum force which can be developed in							

normal operation or emergency recovery. c. Tensile force of 1 500 kN applied to the draw gear stops "a", if this draw gear stop is used (see EN 12663-2). d. Tensile force of 1 000 kN applied to the draw gear stops "b", if this draw gear stop is used and for other types of coupler attachment (see EN 12663-2).

Table 5. Lifting and jacking at one end of the vehicle at the specified positions [18]

Load in Newtons(N)

Locomotive	Passenger rolling stock					Freight wagons	
Category L	Category P-I	Category P-II	Category P-III	Category P-IV	Category P-V	Category F-I	Category F-II
$1.1xgx(m_1 + m_2)$						$1.1xgx(m_1 + m_2 + m_3)$	
NOTE: The other end of the vehicle should be supported in the normal operational condition							

Table 6. Lifting and jacking the whole vehicle at the specified positions [18]

Load in Newtons(N)

Locomotive	Passenger rolling stock					Freight wagons	
Category L	Category P-I	Category P-II	Category P-III	Category P-IV	Category P-V	Category F-I	Categor y F-II
$1.1xgx(m_1 + 2m_2)$						$1.1xg(m_1 + 2m_2 + m_3)$	
NOTE: The other end of the vehicle should be supported in the normal operational condition							

Lifting and jacking

The forces in Table 5 and Table 6 represent the lifted masses. The equations are given for a two-bogie vehicle. The same principle shall be used for railway vehicles with other suspension configurations.

The mass to be lifted is based on the vehicle mass without payload (except for freight wagons which are lifted in the laden condition). It may not include bogies or the full payload in some operational requirements. In such cases, the value m_2 and/or m_2 in the following tables shall be set to zero or reduced to the specified value.

When it is necessary to lift vehicles of class P-I to P-V with payload, this shall be part of the specification.

Where the load cases include loads that are distributed over the structure, they shall be applied in analysis and test in a manner that represents the actual loading conditions to accuracy commensurate with the application and the critical features of the structure.

Table 7. Superposition of loads on the rail vehicle body [18]

Load in Newtons

Superposition Cases	Locomotives Category L	Passenger rolling stock Category P-I, P-II, P-III,	Freight wagons Category F-I, F-II
Compressive force and vertical load		Table 2 and g x($m1$ + $m4$)	Table 2 and $gx(m_1 + m_3)$ Table 3 and $gx(m_1 + m_3)$
Tensile force and vertical load		Table 5 and g x($m1$ + $m4$)	Table 5 and $gx(m_1 + m_3)$

Maximum operating load

The maximum operating load as defined in Table 8 corresponds to the exceptional payload of the vehicle

Table 8. Maximum operating load for the vehicle [18]

Load in Newtons

Locomotive	Passenger rolling stock					Freight wagons	
Category L	Category P-I	Category P-II	Category P-III	Category P-IV	Category P-V	Category F-I	Category F-II
$1.3gx(m_1)$	$1.3gx(m_1 + m_4)$					$1.3gx(m_1 + m_3)$	-
a If the application produces a higher proof load (e.g. due to dynamic effects or loading conditions) then a higher value shall be applied and defined in the specification.							

2.3. ELEMENTS OF PASSENGER RAIL VEHICLE CAR BODY [16]

a) under frame

The under frame shall be an integral unit extending from end to end of the car and shall be composed of the center sill , collision posts, end sill, transition members, floor beams, floor pans, cross bearers, and side sills. If the Center sill is not incorporated into the design, the side sills must be made of floor beams that are continuous from side sill-to-side sill. Center and side sills shall be braced by cross members. The under frame structure shall be designed to meet all applicable FRA rules, AAR requirements, and APTA standards.

The center sill is the major load carrying plat form that comprises two longitudinal frames running along the length of the car body, side sill, bolsters, floor beams and cross bearers.

b) Center Sill

The primary component of the under frame is the center sill. As the vehicle is supported about two bogie bolsters at 18m for the case under this study, it can be simplified to a simply supported beam & hand calculation results will dictate the approximate size of the center sill section to be used. Hence it will be used as an input for finite element analysis.

The critical loading case for the rail vehicle under frame is the combined compressive and vertical load which is the biggest of all the loading scenarios.

c) Cross bearers

These are transverse sections that are integral part of the body frame. They are manufactured from 5mm thick St 36 plate bent to a U-profile and welded to the web sides of the under frame and external side frame. They are spaced to a dimension that suits chuquer plate sheeting and

These members are supposed to carry distributed load coming out from passengers. Optimization of the cross bearers is possible due to the fact that bending moments decrease as we go far away from the longitudinal center sill.

d) Side sill

The external limiting size of the rail vehicle car platform is the side sill. Its production is similar to that of the end sill and cross bearer. They are manufactured by shearing and bending a 5mm thick mild steel plate and formed to a C-profile along the length of the rail vehicle car body and are used as a base to the side wall posts. Side sills are not only structural component rather they have to be aesthetically attractive as they are the exposed surface of the rail vehicle car body.

e) Front and Rear sill

Rear and front sills are welded to the side sill along the lateral direction. They house spring loaded draw bar used for coupling two car bodies together. Name plate, reflectors, electrical light system is installed on to this frames.

It is the first to be exposed during crash and shall be reinforced by a collision post extended from itself with the under frame

f) Side wall

The side wall structural member has columns and girts designed to carry the load from roof and passenger luggage. Moreover; it should be strong enough to maintain the shape of the rail vehicle car without deformation during jacking and lifting. It also has door and window frames. Side wall should give proper ergonomics to passengers boarding the train and allow comfortable environment during different season of the year and climatic condition.

g) Roof framing and sheathing

i. The projected area in units of mm^2 of the portion of the roof supported by carlines divided by the sum of the section moduli in units of mm^3 of the carlines at any longitudinal section shall not be more than 340 mm^{-1}

ii. Flat roof sheets of mild steel with 221 MPa yield strength and without reinforcements shall be of a minimum thickness of 1.3 mm. Metals of other strengths may be used of a thickness in inverse proportion to yield strength.

iii. Metal roof sheets of a lesser thickness may be used provided the sheets are reinforced to produce at least an equivalent sectional area at right angle to roof sheets

The roof structure should be capable of resisting the rollover weight of the car body and bogies together during accident. To fulfill this requirement a Mild Steel UPN 150 bent to the profile of a dome is used as the rafter and a purlin size of UPN 100 of the vehicle car.

h) Collision posts

The APTA standard outlines the minimum structural design requirements for collision posts at the ends of occupied vehicles. The end of a vehicle that is designed to lead a train must protect occupants of that vehicle from the intrusion of objects the train has struck in a collision. As such, higher strength requirements are necessary for the lead ends of such vehicles.

The requirements of this standard are intended to result in an energy absorbing end structure above the under frame. Therefore, requirements for collision posts in the following sections include the ability of the post to absorb a significant amount of energy by undergoing severe deformation without failure of the post or its connections during an overloading condition.

i) Couplers and drawbars

A coupler does three things: connects one car with another, holds the connection, and then disconnects the two cars If used, drawbars, semi-permanent couplers, standard AAR Type F interlocking couplers, and APTA RP-M-003 Type H tight lock couplers may be considered as providing the required climb, bypass, and overturn resistance if the following additional requirements are met:

a). The structural arrangement shall include a coupler carrier designed to resist the 445KN vertical downward thrust from the coupler shank at any possible horizontal position of the coupler. The acceptance criteria for this condition shall be no permanent deformation of the coupler carrier, supporting car body structure, and intervening connections.

b). The structural arrangement shall include a buffer beam above the coupler, which shall be designed to resist the 445KN. vertical upward thrust from the coupler for any horizontal position of the coupler. The acceptance criteria for this condition shall be no permanent deformation of the buffer beam, supporting car body structure, and intervening connections.

j) horizontal Framing Members

End frame collision and corner posts may be connected by horizontal intercostals as necessary to resist the lateral components of the design loads. In addition, a structural header may be used to connect the tops of the collision and corner posts.

Cab-end framing shall include a horizontal structural member between the collision post and corner post on each side at a height equivalent to the bottom of the windshield. The structural shelf shall support a load of not less than 67 kN applied transverse to the member at any point on its span without permanent deformation of any part of the vehicle structure.

k) End frame sheathing

Cab end frame sheathing shall be:

- . Equivalent to a 13-mm steel plate with 172-MPa yield strength. Material of higher yield strength may be used to decrease the required thickness of the material provided an equivalent level of strength is maintained.
- Designed to inhibit the entry of fluids into the occupied cab area of the equipment.

- Connected to underlying framing members sufficient to develop the full strength of the sheathing.

2.4. TRUCK TO CAR BODY ATTACHMENT STRENGTH

A mechanism for attaching the completely-assembled truck, including the bolster, if used, to the car body shall be provided in accordance with the requirements of APTA.

2.4.1. Horizontal attachment

The ultimate horizontal strength of the attachment mechanism shall be sufficient to secure the entire truck to the car body in a manner which will prevent separation of the truck from the car body during derailments and collisions in which a horizontal load of minimum 1112 kN is applied in any direction at any point on the truck frame through the center of truck rotation. The required resistance to a1112 kN horizontal load shall be available at any possible position of the truck in its vertical suspension travel, including the condition of the car raised off the track with the truck hanging from the car, and shall not depend upon external vertical loading. The strength of the truck and supporting car body structure shall be sufficient to develop the ultimate horizontal strength of the attachment mechanism as specified.

2.4.2. Vertical attachment

The vertical strength of the attachment mechanism shall provide a minimum factor of safety of 2, based on the yield strength of the structural material used in the truck, car body, and the elements of the attachment mechanism, during jacking or lifting of the car body with the truck hanging from the car body.

2.5. VEHICLE AND TRACK SYSTEM MODEL

The vehicle system model used in this study consists of a full length rail vehicle car body supported on a bogie, while the side frame is supported on two wheel sets. The primary suspension, connecting the wheels and the bogie frame is modeled as a parallel combination of a linear spring and a viscous damping element. The secondary suspension connecting the bogie frame and the car body is also modeled by parallel spring and damping elements. The mass of the car body M_1 , bogie mass M_2 , wheel mass M_w are coupled through the suspension elements, as shown in Fig 5. The total vehicle system model is represented by a 2-DOF dynamic system that includes the car body vertical motion.The primary suspension stiffness and damping elements are represented by k_1 and c_1 respectively, while c_2 and k_2 represent the stiffness and viscous damping coefficient due to secondary suspension.

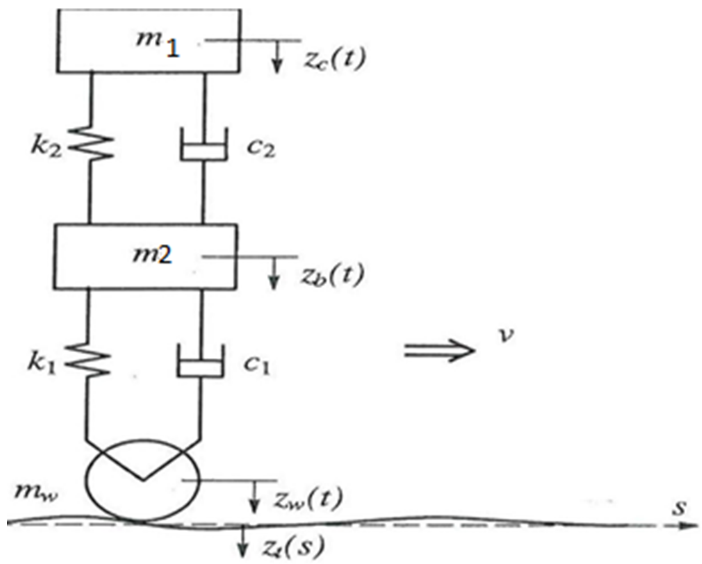
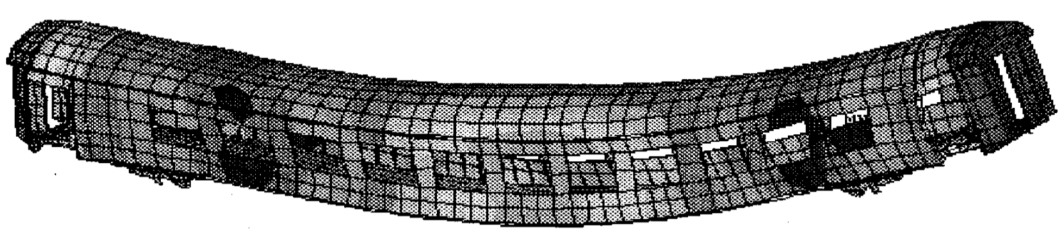


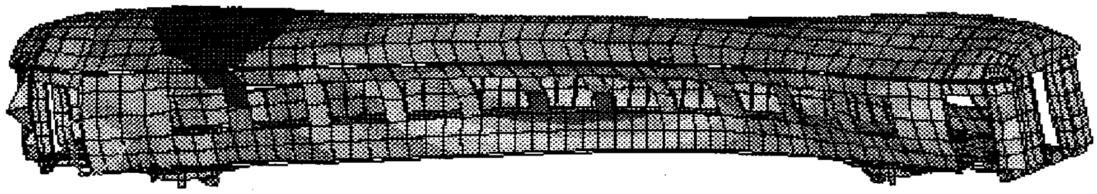
Fig.5. A quarter vertical model of a rail vehicle

The design requirement is to avoid the natural frequency of the rail vehicle not to coincide to the excitation frequency induced from the track to the car body. Otherwise, the vehicle will resonate. Fig 6 shows the typical natural modes of high speed car body. The natural frequencies of the car body give a good impression of how stiff the construction is. Especially the vertical bending mode gives a hint of how well the car body will manage the payload.

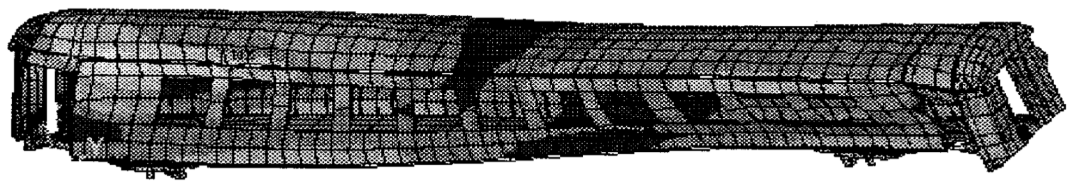
(a)



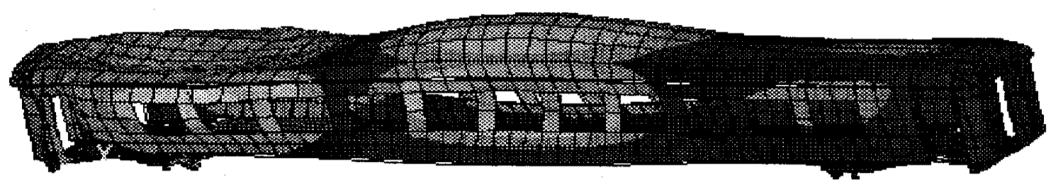
(b



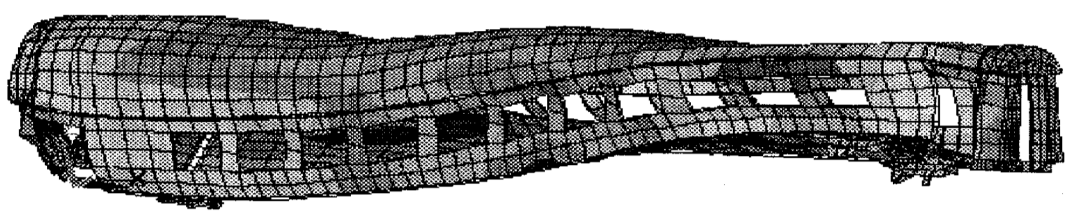
(c)



(d)



(e)



(f)

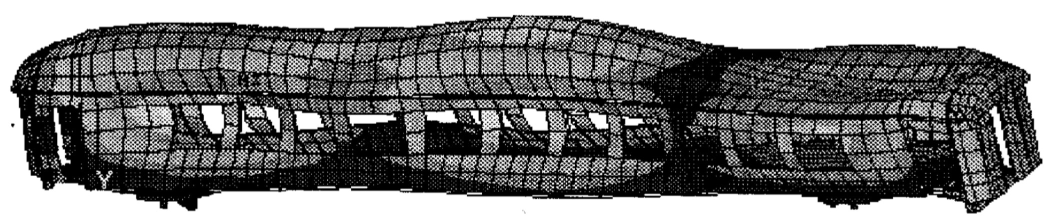


Fig 6. (a). First vertical bending,(**b).** first lateral bending(**c).** 2[nd] vertical bending torsion1 **(d).** breathing1 (**e).** Torsion 2 (**f).** Breathing 2 [1]

There are two vibration levels, essentially rigid-body vibration & structural flexibility vibration. Structural flexibility is more important than rigid body motion in the vertical direction. This is to a large extent due to comfort-weighting in the vertical direction. Structural flexibility is almost negligible in the lateral direction, also to a large extent depending on comfort-weighting.

CHAPTER THREE

PHYSICAL & MATHEMATICAL MODELING

3.1. PHYSICAL MODEL

In order to define the car body structure, a structural sketch is prepared using CATIA V5 modeling and analysis software. This model is exported to ANSYS -12 for final result analysis. The purpose of the structural sketch is to define the primary car body structure in advance of formal stress analysis and structural drawings.

The structural sketch includes a side view; a top view showing one longitudinal dimension of the roof and one longitudinal dimension of the under frame; and typical car body cross-sections.

Cross-sections of the structural members, showing the shape of each member is included in Fig 4. The members shown include, to the extent used in the particular design, typical side frame and door frame posts; end, side, draft and center sills; belt, top, and roof rails; collision and corner posts; bolsters, floor beams and cross bearers; roof carlines and purlins; roof sheathing or corrugation; side frame sheathing or wall corrugation.

A computer aided design (CAD) or finite element (FE) analysis can give the mass, centre of gravity position, and mass moments of inertia. However, in most applications the mass of the car body structure is less than half the car body mass. For a coach the additional mass is a result of the interior layout and interior/exterior equipment. For powered vehicles, especially locomotives, the car body also carries some of the traction equipment. Often, it is quite hard to keep accurate records of all the pertinent masses and positions. Usually, the mass of this equipment is merged to the metal structure model. In many vehicles – track dynamics simulations of the payload, i.e., passengers or goods, also needs to be considered. The corresponding mass might also be combined with the car body structure model.

In this thesis, category P-I(coaches) is considered in the study. The reference design and parameter is based on CNR Changchun Railway vehicles Co. ltd and rolling stock technical reference from Ethiopian Rail Ways Corporation.

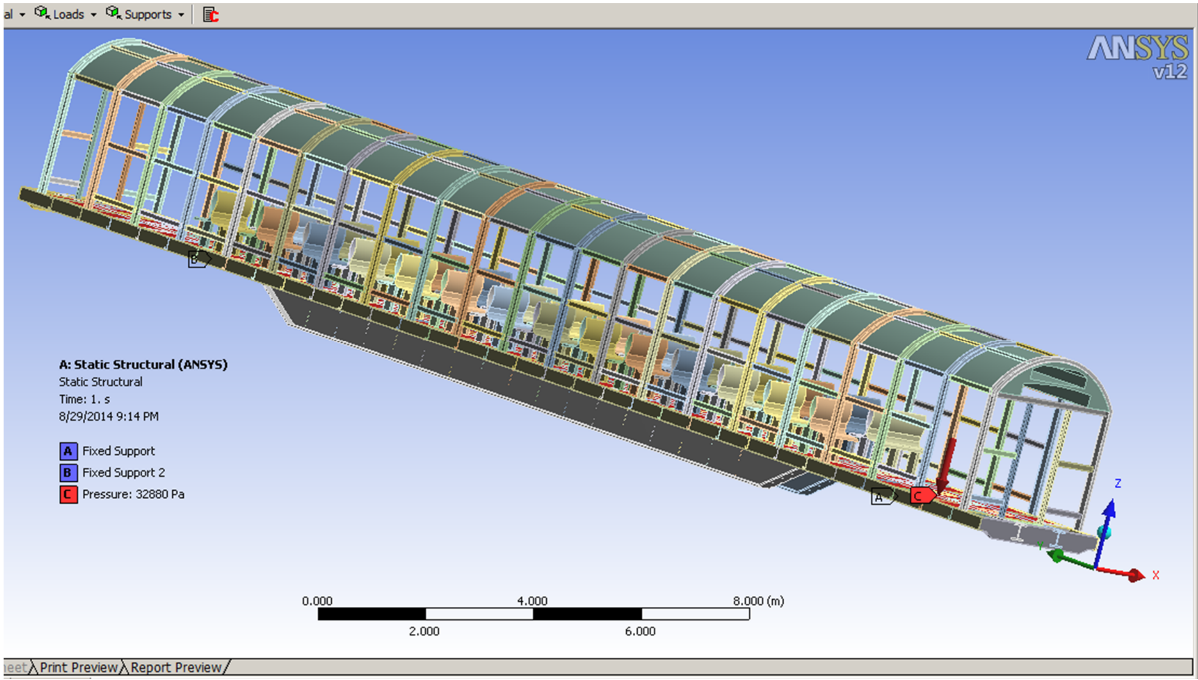


Fig 7a. Isometric view of a passenger Rail vehicle model

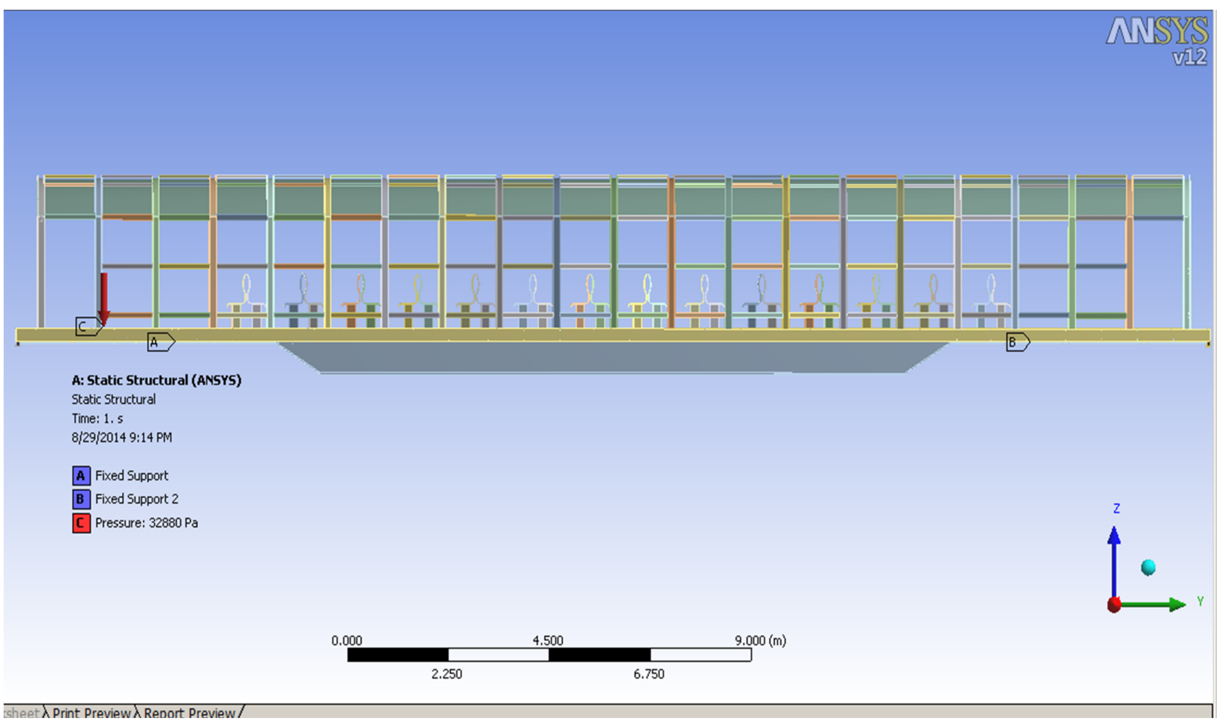


Fig 7b. Front view of a passenger Rail vehicle model

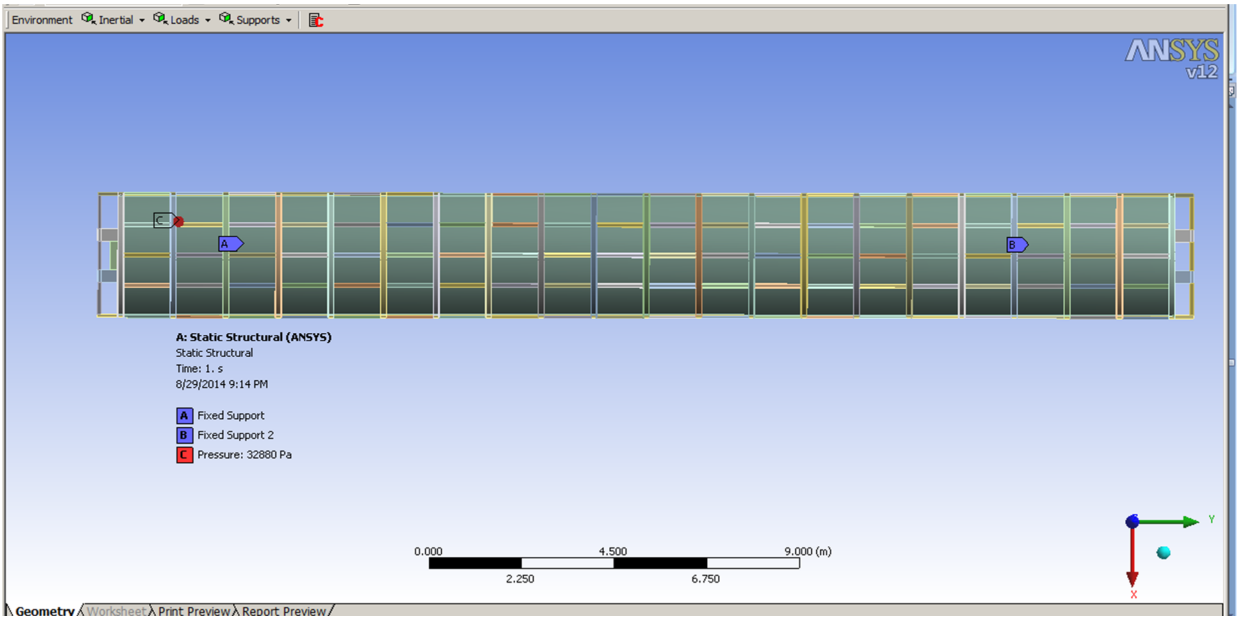


Fig 7c. Top view of a passenger Rail vehicle model

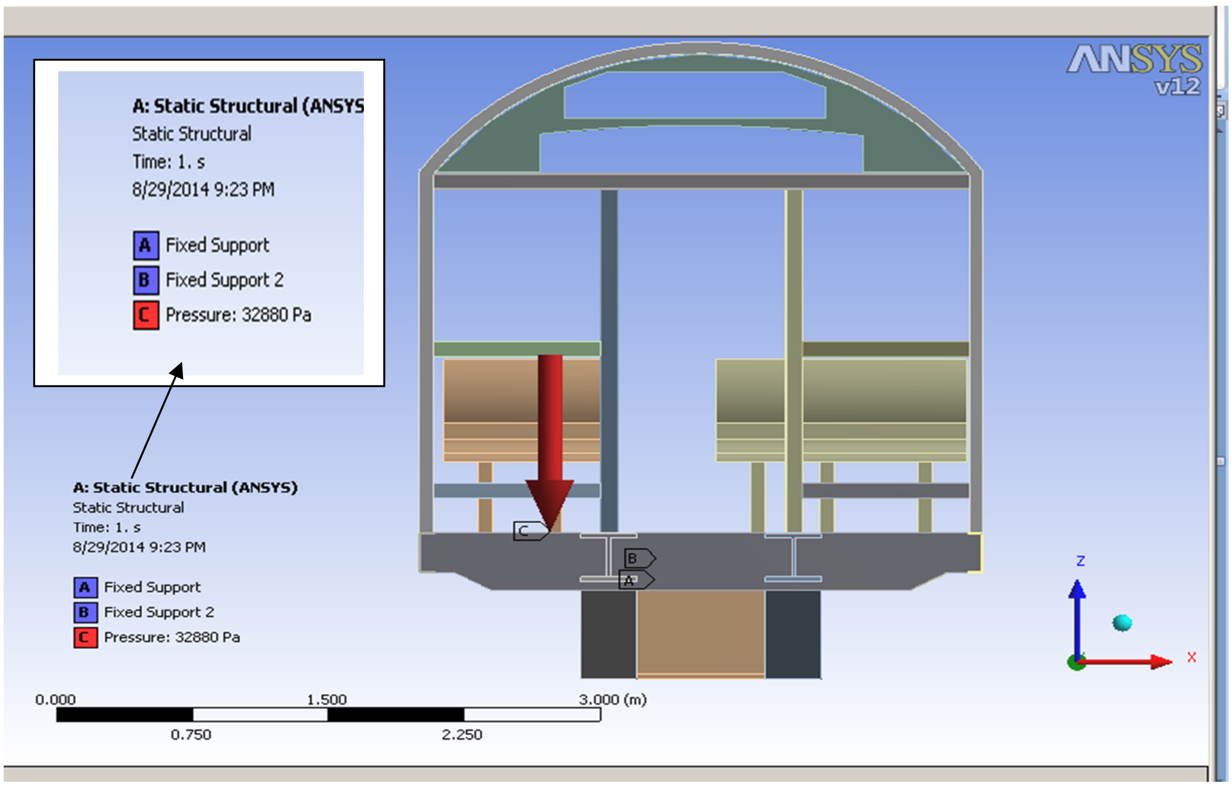


Fig 7d. Side view of a passenger Rail vehicle model

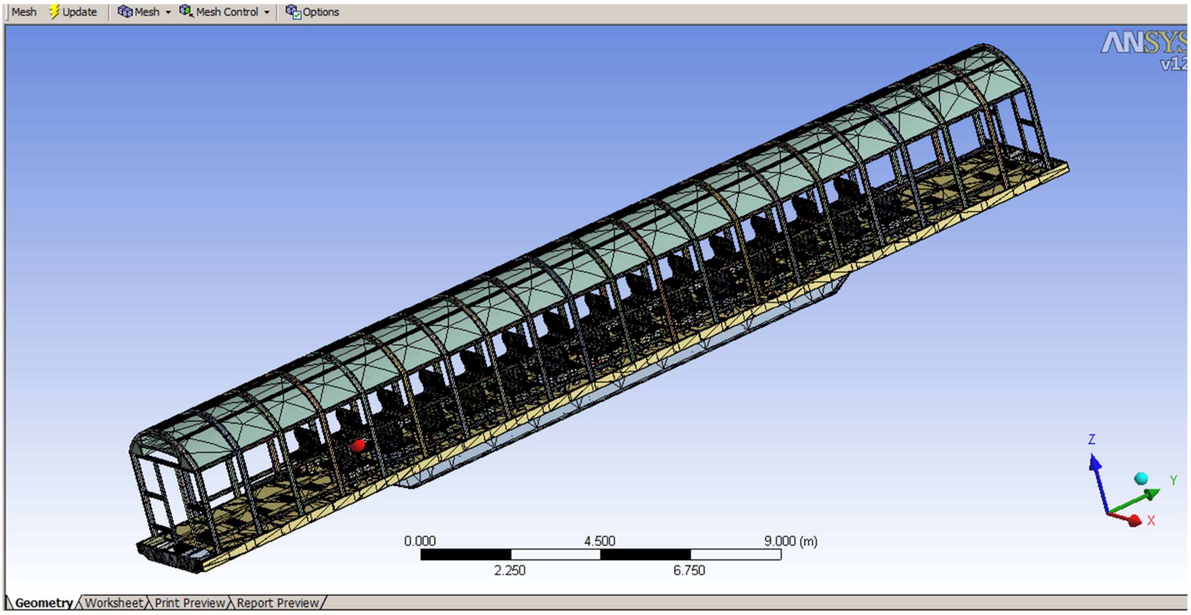


Fig 8. Meshed model of a passenger rail vehicle

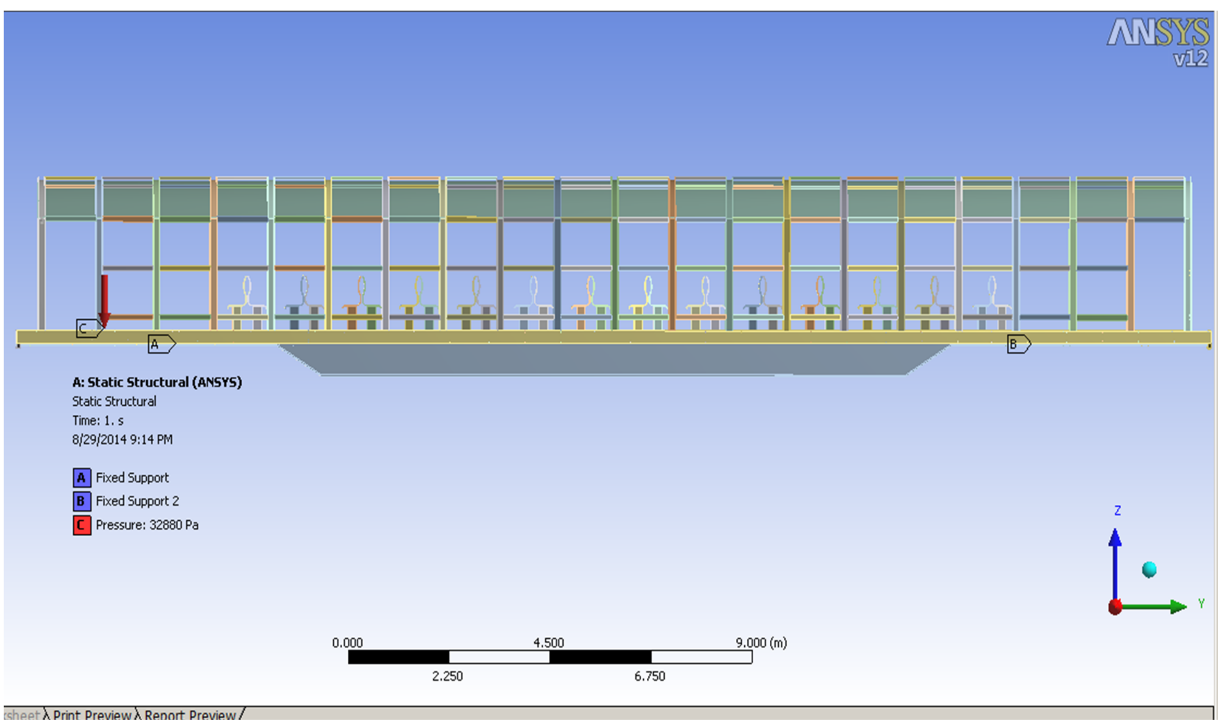


Fig 9. Fixed support points at the bogie bolster

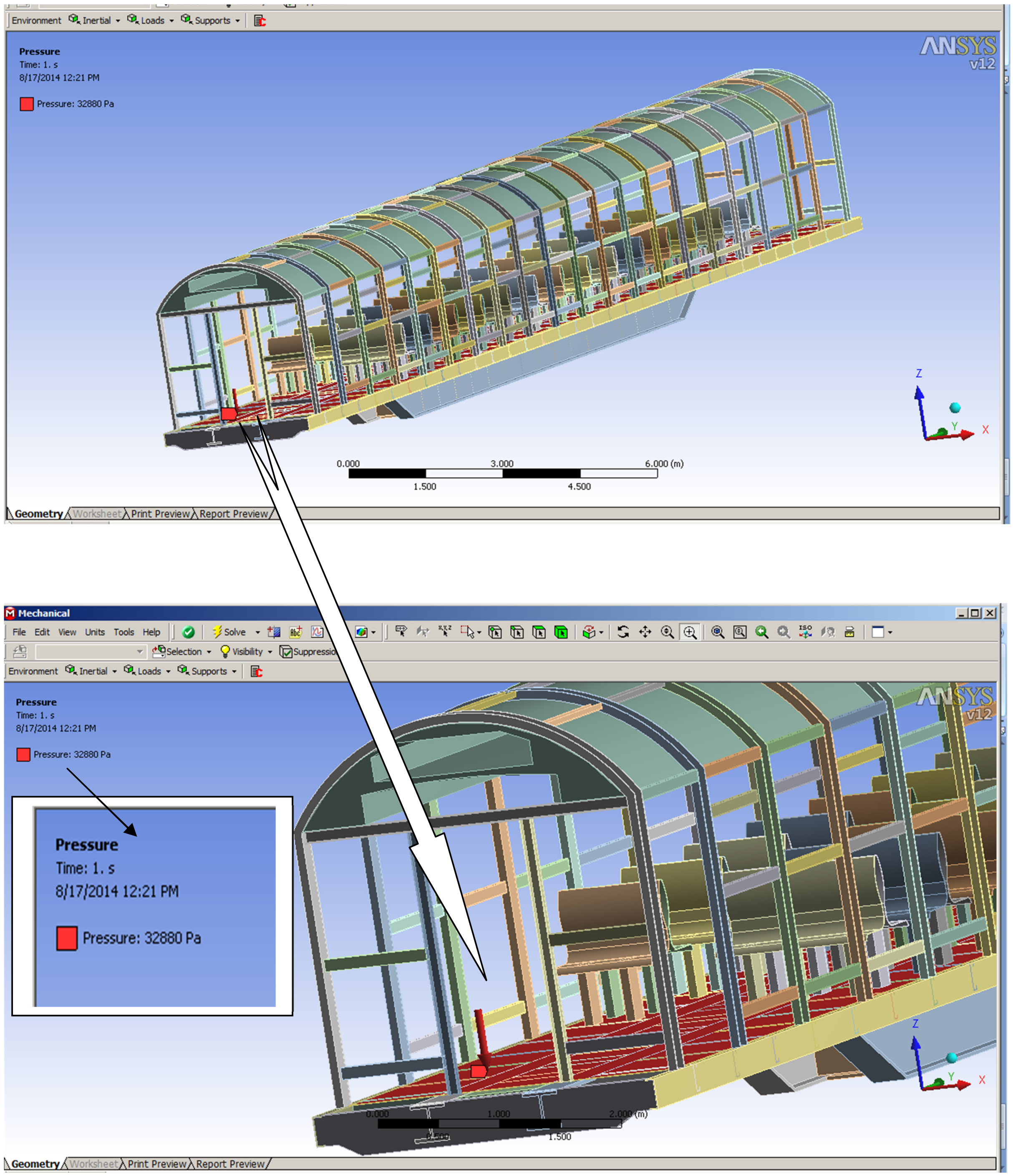


Fig 10 . Vertical &compressive load applied on the vehicle

Table 9. Main technical parameters of 25 G hard seated passenger coach

Type	Passenger coach
Design speed	120km/h
Model No	25G or YZ25B
Gauge	1435mm/ standard gauge
Car body length	25500mm
Car body width	3106mm
Min. curve radius	Coupler: 145m
Car body height	4433mm
Height from coupler center to rail roof	880mm
Bogie	209P
Height from coupler center line to top rail	880mm
Fixed distance between two cars	18000mm
Dead weight	≤46t
Axle load	≤17t
Fixed passenger capacity	118P
Bogie mass	3000kg

CHAPTER FOUR

ANALYSIS AND RESULTS

4.1 ANALYSIS & ACCEPTABLE STANDARDS

For the passenger car categorized under P-I, the load analysis is conducted in accordance to the EN12663-1:2010. All load cases are considered and the design is checked for exceptional loading (m_4) rather than fatigue or working load. This is because the Exceptional load case imposes the maximum load & requires high material strength of the passenger rail vehicle structural components. Normal working load (m_3) loads are not used in the calculation as it is not the scope of this thesis to determine the fatigue life of rail vehicle car body structure.

4.1.1 Finite Element Model Creation

The wide range of components used in the design of railway bogies and rail vehicle car bodies result in a large variety of modeling methods and element types being employed in the creation of the FE model. The analysis is a solution approach that comprises importing of 3D model from CATIA V5-R16 to ANSYS work bench that is then processed in order to clean up unnecessary details or components. The imported Model is then meshed to act like single entity. Finer mesh and close to exact solution can be achieved by controlling the mesh size. However, as the parts of rail vehicle body are enormous and complex, considering a fine mesh is time consuming.

4.1.2 Superposition of compressive loading

The total number of seated passengers is 118 and each passenger is assumed 80kg from Table 2 .The total area of passage ways, access and area by subtracting 4m from each side of the vehicle length is approximately, 17.76m long and 0.35m wide.

$A_{pa} = 0.35mx17.76m = 6.216m^2$ (Ignoring areas occupied by toilet and catenaries)

Compressive force at passage ways & access areas is

$$F_{pa} = 4\frac{person}{m^2}x6.216m^2 \text{..}(1)$$

$$m_4 = Npassenger\frac{80kg}{passenger} + F_{pa}$$

Hence the total exceptional load on the rail vehicle car body is:

$$m_4 = 118 passenger x \frac{80kg}{passenger} + 1989.12kg = 11429.12kg$$

$$m_4 = 9440kg + 1989.12kg = 11429.12kg$$

The expected total dead weight of vehicle and two bogies is 44.5ton from Table 2. The vehicle may need to be jacked to a place where it can be maintained or attached to another locomotive. In such case, a hooking point is located in such a way that it will enable the load being carried safely without deformation. This is in accordance to the center of gravity and moment of inertia of the vehicle body being jacked.

For lifting and jacking at one end of the vehicle structure, the empirical relationship is:

$$F_l = 1.1xgx(m_1 + m_2) \quad \text{....(2)}$$

$$F_l = 1.1x9.81x(38.5 + 3)$$

$$F_l = 447.8kN$$

Whereas; when the lifting and jacking made for the whole vehicle structure at specified location including both bogies, a combined force due to the vehicle self weight and external exceptional load is the maximum loading case applied to the rail vehicle structure according to Table 6.

$$F_l = 1.1xgx(m_1 + 2m_2) \quad \text{....(3)}$$

$$F_l = 1.1x9.81x(38.5 + 6)$$

$$F_l = 480.2kN$$

Superposition of Static compressive & vertical force of the vehicle structure from Table 7 is

$$F_{cv} = 2000kN + gx(m_1 + m_4) \quad \text{....(4)}$$

$$F_{cv} = 2000kN + 9.81x(38.5 + 11.429)kN = 2488.4kN$$

$$F_{cv} = 2488.4kN$$

The maximum operating load shared among the 4 axles of the rail vehicle bogies indicate the axle load.

$$F_{\max} = 1.3(\frac{m_1 + m_4}{4}) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(5)$$

$$F_{\max} = 1.3(\frac{38.5 + 11.429}{4}) = 16.2 tonperaxle$$

This result is consistent to Table 9 (i.e. axle load <17 ton)

The Superposition of Static tensile force & vertical load for the vehicle structure is

$$F_{tv} = 1500kN + gx(m_1 + m_4) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.(6)$$

$$F_{tv} = 1500kN + 9.81x(38.5 + 11.429)kN = 1989.9kN$$

$$F_{tv} = 1989.9kN$$

For the design, the maximum pressure is contributed from superposition of vertical and compressive load acting over an area of 25.5m length and 3.06m wide platform.

$$P_{cv} = \frac{F_{cv}}{A} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.(7)$$

$$P_{cv} = \frac{2488.4}{25.5mx2.96m^2} kN = 32.88kpa$$

4.1.3. Structural car body Component design

a. under frame

The under frame shall be an integral unit extending from end to end of the car and shall be composed of the center sill , collision posts, end sill, transition members, floor beams, floor pans, cross bearers, and side sills. If the Center sill is not incorporated into the design, the side sills must be made of floor beams that are continuous from side sill-to-side sill. Center and side sills shall be braced by cross members. The under frame structure shall be designed to meet all applicable FRA rules, AAR requirements, and APTA standards.

The center sill is the major load carrying plat form that comprises two longitudinal frames running along the length of the car body, side sill, bolsters, floor beams and cross bearers.

The under frame-mounted equipment has been designed on a modular concept and is totally enclosed within a welded connection of tapper I-beam, profile cut and bent sheet metal fairing. This fairing reduces aerodynamic disturbance caused by under frame-mounted equipment, particularly at higher speeds. Whilst enclosed equipment has considerable benefit to offer, it has been necessary to design the equipment modules in such a manner that maintenance is not jeopardized and so particular attention has been paid to this. Access doors and disposition of equipment are such that day-to-day and normal programmed maintenance is disposed correctly.

The under frame is exposed to cyclical load due to the loading and unloading of passengers & goods. The maximum load of the center sill is due to the self weight of the vehicle above it, quasi static load due to curve negotiation and dynamic load due to road waviness or rail corrugation, and live load of passengers on board.

b. Center Sill

The primary component of the under frame is the center sill. As the vehicle is supported about two bogie bolsters at 18m for the case under this study, it can be simplified to a simply supported beam & hand calculation results will dictate the approximate size of the center sill section to be used . Hence it will be used as an input for finite element analysis.

The critical loading case for the rail vehicle under frame is the combined compressive and vertical load which is the biggest of all the loading scenarios mentioned above.

The combined load due to Compressive force and vertical force is 2488.4kN from equation 4.

The dead weight of the vehicle in working order is

$$m_1' = m_1 + 2x(m_2) \quad \text{...(8)}$$

For calculation in this study, m_1 doesn't include the weight of two bogies

$$m_1' = 44.5ton$$

The dead load of the vehicle in working order without considering the bogie mass is

$$m_1 = m'_1 - 2x(m_2) \text{ ……………………………………………..(9)}$$

$$44.5ton - 2x3.ton = 38.5ton$$

Assumption:

Let us consider half of this superposed compressive & vertical force to be shared among two sides of the vehicle chassis running along the length and the remaining half is carried by the side cross bearers and side sill.

$$\frac{F_{cv}}{2} = 2488.44x0.5KN$$

$$\frac{F_{cv}}{2} = 1244.22kN$$

Load shared by each under frame beams is

$$\frac{F_{cv}}{4} = 1244.22x\frac{1}{2}kN$$

$$\frac{F_{cv}}{4} = 622.11kN$$

This load is considered distributed on the beam that is supported 18m along its span.

The load intensity $w = 622.11x\frac{1}{25.5}kN = 24.4\frac{kN}{m}$

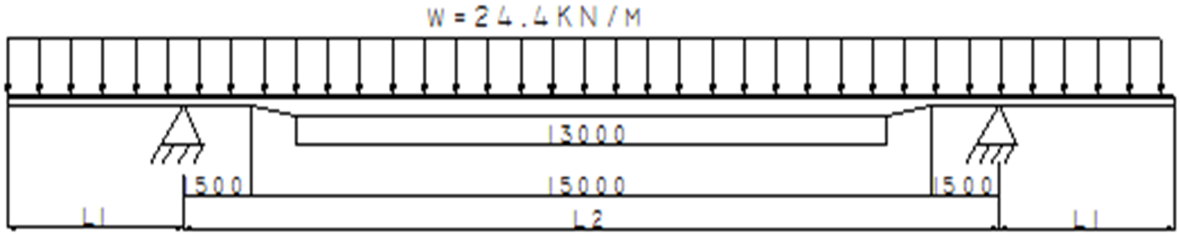


Fig 11. Modelling of Load distribution on the centre sill

The flexural stress on the chassis beam is given by the formula:

$$\sigma = \frac{Mc}{I}$$

$$Z = \frac{I}{c}$$

$$\sigma = \frac{M}{Z}$$ where Z is section modulus and c is the centroid of the center sill section.

The yield strength of the selected center sill is structural steel beam St 36 having σy=460Mpa).

Reaction force at the bogie bolsters centers.

$$R_1 + R_2 = wx(2L_1 + L_2)$$

$$R_1 = R_2$$ (as there is uniform distributed load at symmetric distance)

$$R_1 = 24.4x12.75kN = R_2$$

$$R_1 = 311.1kN = R_2$$

The Bending Moment at the two fixed bolster support ends is equal and it is the maximum among the beam section.

$$M\ @\ R_2 = R_1 x L_2 - \frac{w}{2} x(2L_1 + L_2) x \frac{L_2}{2}$$

$$M\ @\ R_1 = M\ @\ R_2 = 311.11x18 - \frac{24.4}{4}(18x25.5)$$

$$M\ @\ R_1 = 2799.9kN$$

The maximum deflection is at the center of the beam. i.e. at 12.75m from one end.

Section characteristic of center sill

$$Z_{\max} = \frac{M_{\max}}{\sigma_y}$$

$$Z_{\max} = \frac{2800kNM}{460Mpa}$$

$$Z_{\max} = 6086cm^3$$

The standard Beam element that has similar characteristics is selected from standard table for metal manufacturer's catalogue annexed . The property of the center sill is summarized in Table 10.

To decrease the weight of the car, optimization of the beam is made by reducing the section near the bolster support position. This will enable us to consider a tapper element at the low stressed area.

Considering moment at the bogie fixed ends to decrease the section beyond the bogie support on both sides of the vehicle, $M_1 \,\&\, M_4 = 0$

$$M = \frac{wxl_1^{\ 2}}{2}$$

$$M = \frac{24.4x3.75^2}{2}kN - m$$

$$M_{\max} = 171.56kN - m$$

$$Z = \frac{171.56}{460Mpa} kN - m$$

The section modulus required for this bending moment $Z = 372.95cm^3$ which is less than the selected section. This is because the flanges have to conform to the geometry of the main center sill calculated above.

The section that has this property and complements with the middle frame is 305x305 column.

Table 10. Summary of middle center sill material

Universal beam 914x305	
Radius of gyration	
Along x-x	36.78cm
Along y-y	6.42cm
Elastic modulus	
Along x-x	$10891cm^3$
Along Y-Y	$1014cm^3$
Mass per meter	289 kg/m
Flange width	307.8mm
Flange thickness	32mm
Web height	926.6mm
Web thickness	19.6mm
Crossectonal area	$369.8cm^2$

Universal beam 305x305	
Radius of gyration	
Along x-x	14.2cm
Along y-y	8.02cm
Elastic modulus	
Along x-x	$2991cm^3$
Along Y-Y	$1034cm^3$
Mass per meter	198 kg/m
Flange width	314mm
Flange thickness	31.4mm
Web height	339.9mm
Web thickness	19.2mm
Crossectonal area	$252.8cm^2$

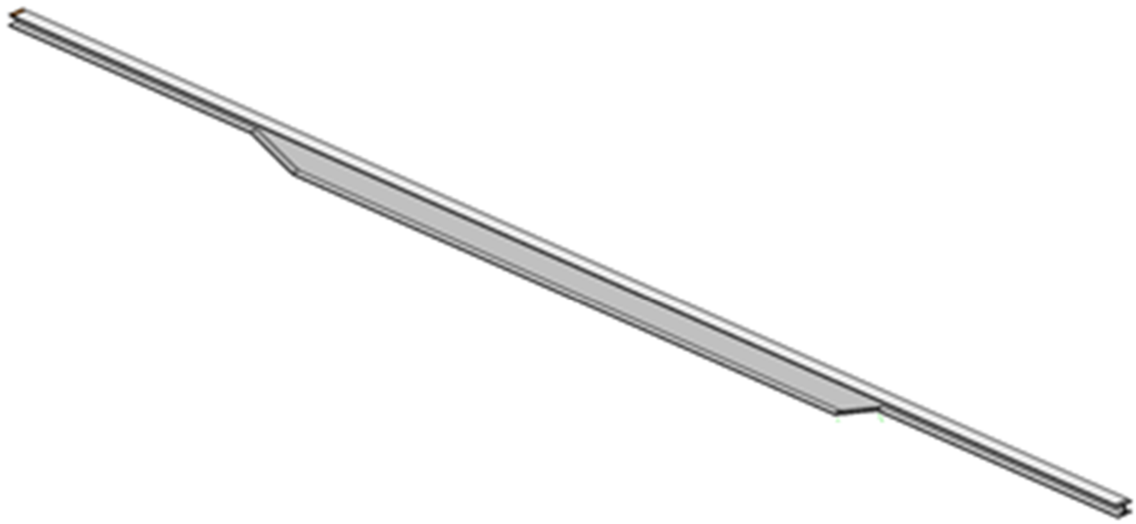


Fig 12. The designed center sill for the vehicle under the study

4.2. RESULTS

The solution method follows the loop shown in **Fig 13**

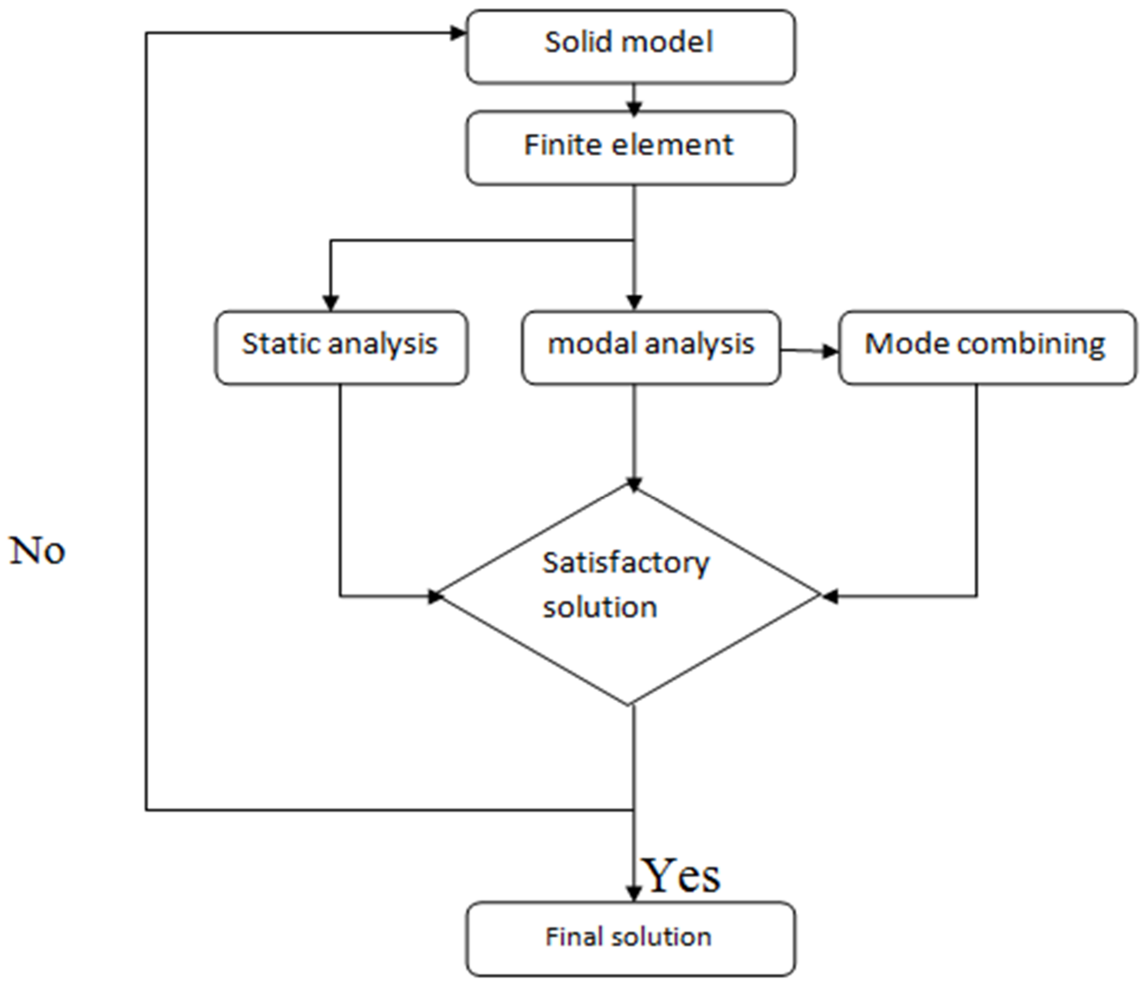


Fig13. Solution method for structural and modal analysis

4.2.1. Static structural results

The static analysis is concerned with determination of the response of the rail vehicle car body to steady state loads. The corresponding stresses and strains are obtained using strain-displacement and strain-stress (constitutive) relations. The deflections between two nodes are determined using assumed shape functions. The solution to the equation gives nodal translational and rotational displacements.

In the analysis, the situation considered is the standstill condition of the vehicle on a level road loaded to the maximum payload. The other loads considered include dead weight of the engine, and weight of the vehicle itself. By the finite element analysis using a computer 397 bodies ,576,889 nodes and 231705 elements are solved by applying a distributed pressure of 32880pa that arises from a combined vertical and compressive load on the under frame. This creates the maximum stress on the center sill. Particularly, it is maximum at transition points of change in crossection and near bolster support. The value of stress is 401.8Mpa in this spots and it is less than 460Mpa in all other sections. Under static conditions, for more portion of the segment, the safety factor is between 1 and 15.

The vertical deflection includes deflection of the spring and static deflection of the vehicle structures. The vertical deflection is observed to be maximum at the center of the rail vehicle structural body and the under frame. As seen in the figure below, of the other components, the under frame gets highly stressed. This is attributed to the fact that all the static loads including the weight of the vehicle body and the cabin, and the engine are taken by the under frames and this causes bending loads in it. The deformation has maximum intensity at the center of the frame. The value ranges up to 3.9 mm. This is illustrated in Table 14, 15 and 16.

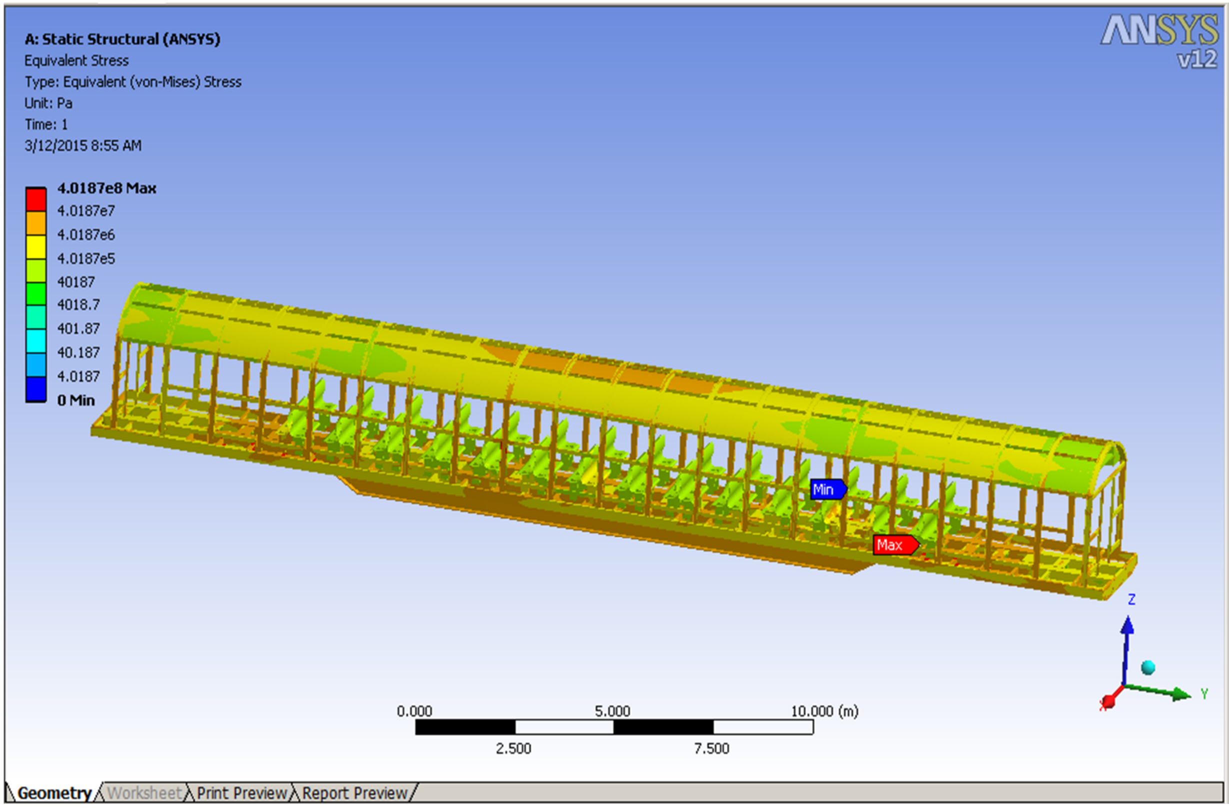


Fig 14a. Stress distribution on rail vehicle car body-Isometric view

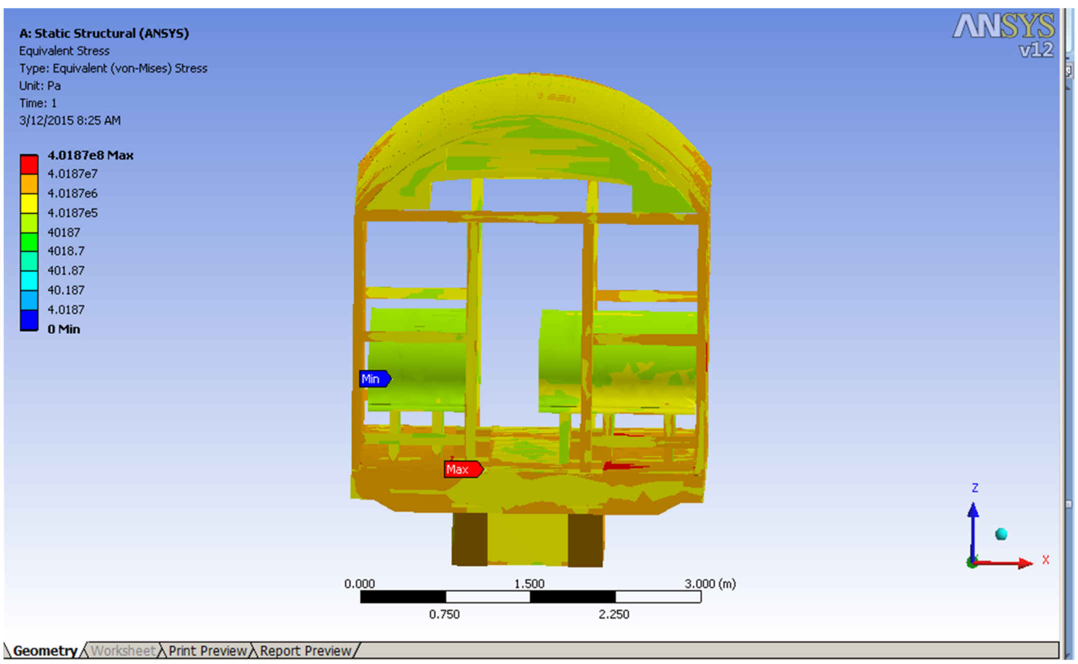


Fig 14b. Stress distribution on rail vehicle car body -end view

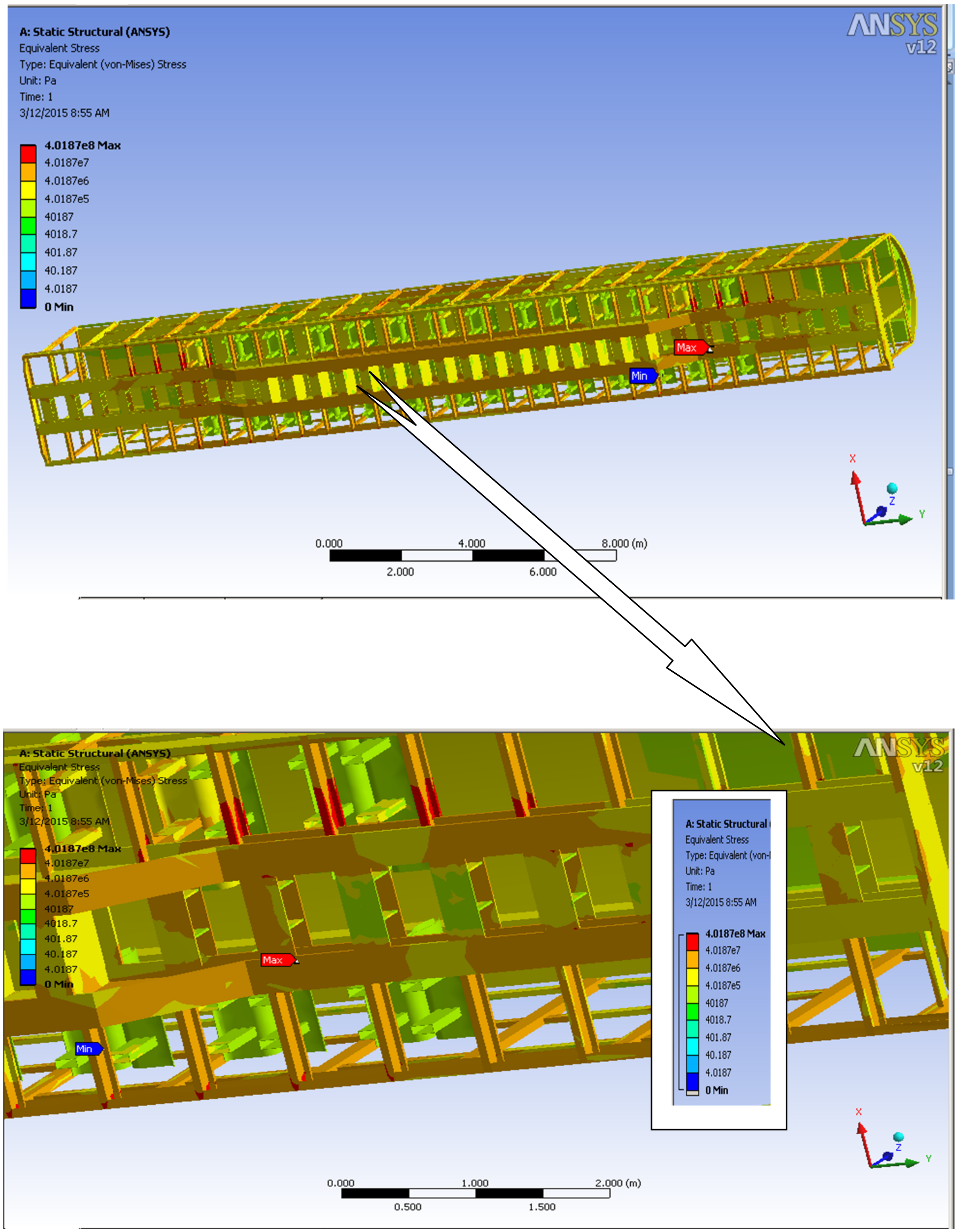


Fig14c. Stress distribution on the rail vehicle car body-bottom view

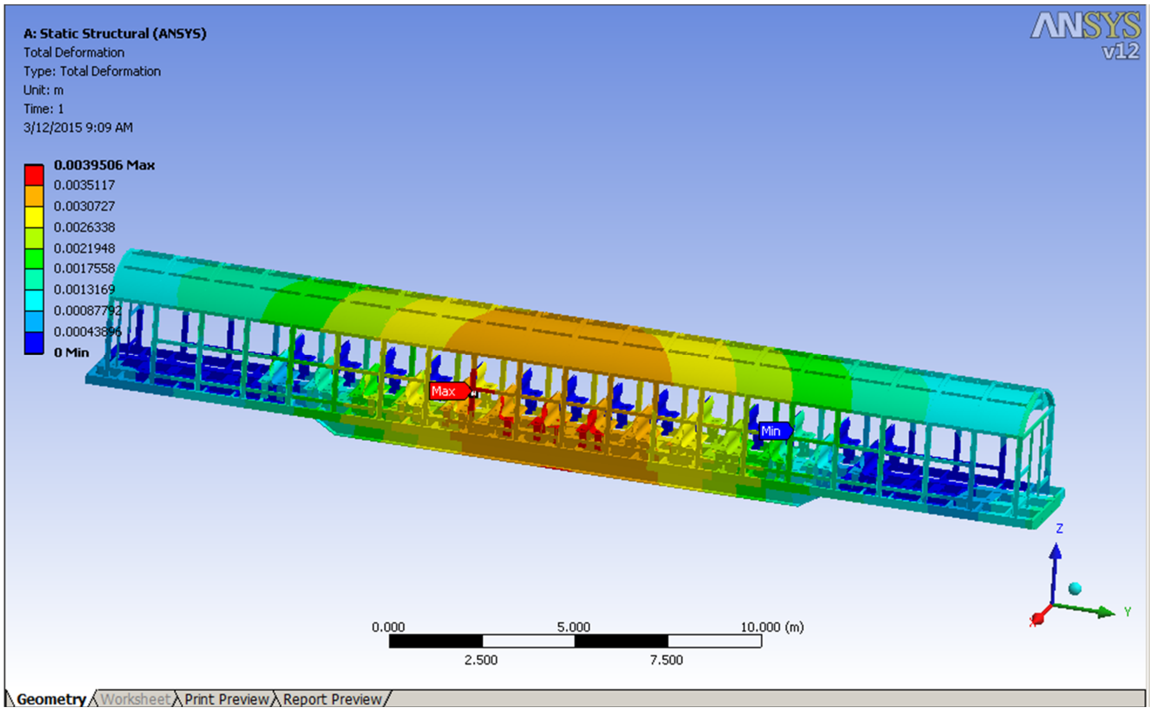


Fig 15a. Total deformation of passenger rail vehicle center sill

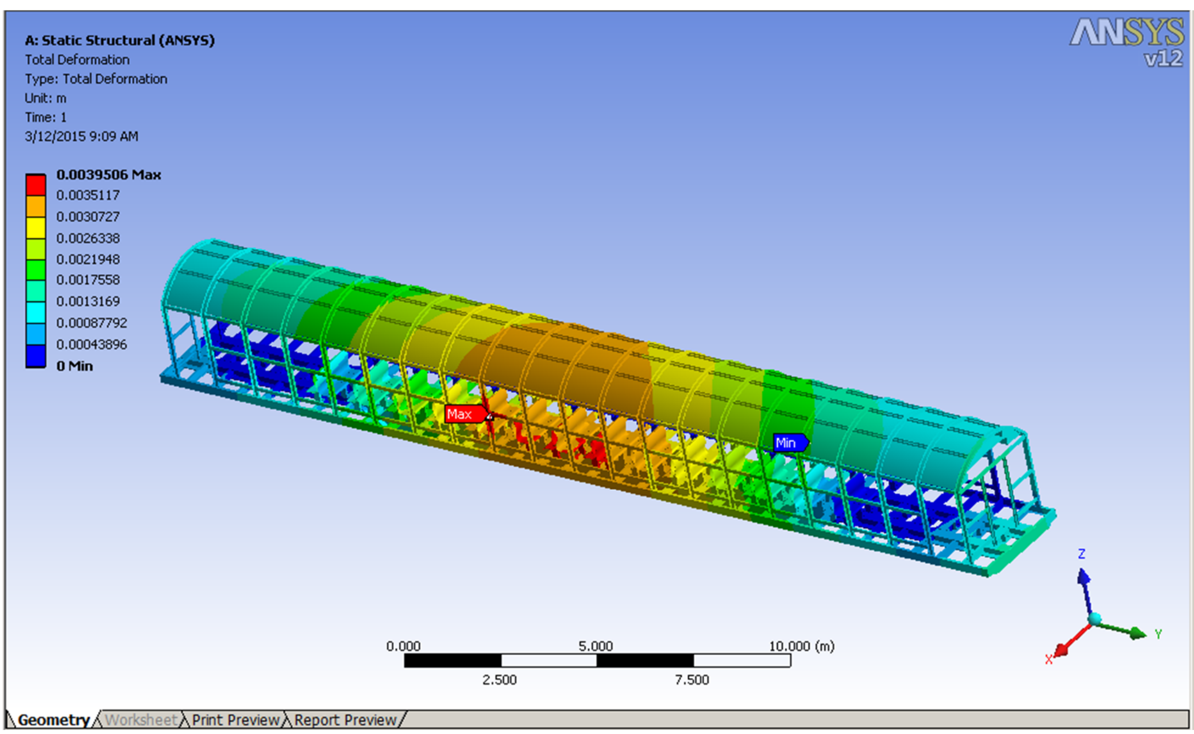


Fig 15b. Total deformation of passenger rail vehicle on under frame and roof

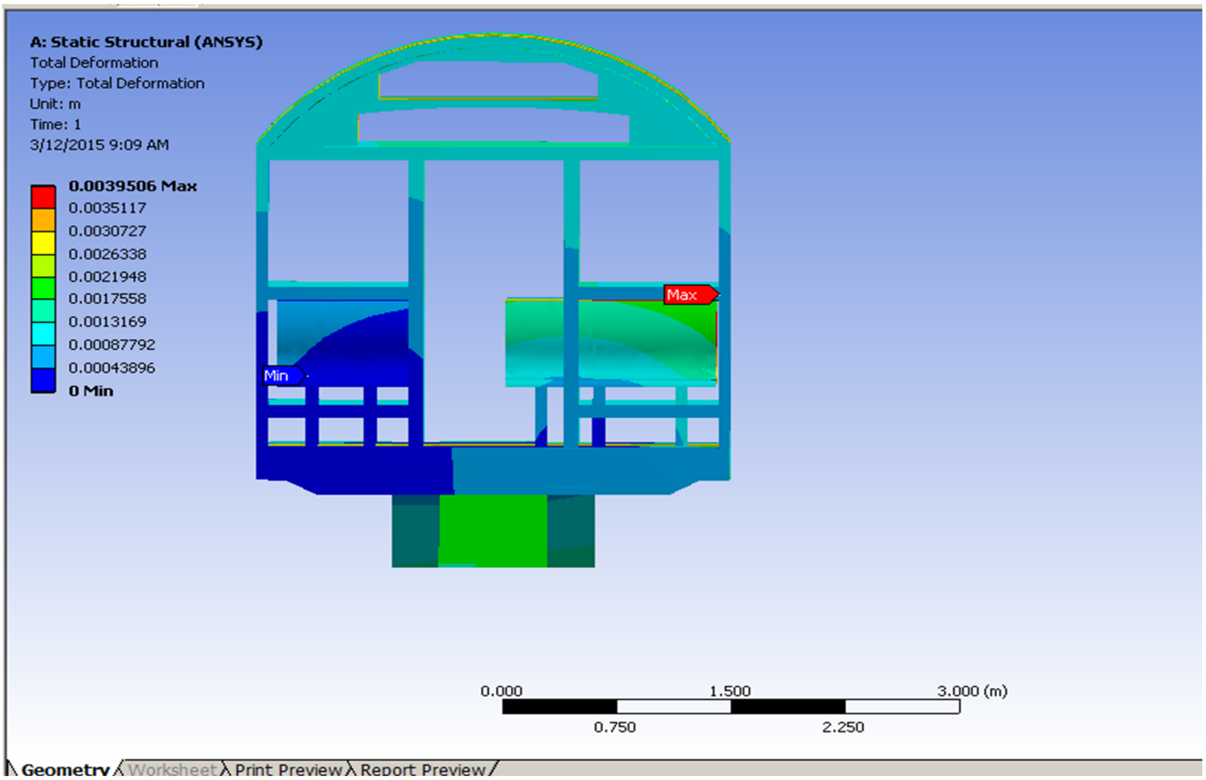


Fig 15c. Total deformation of Rail vehicle structure-end view

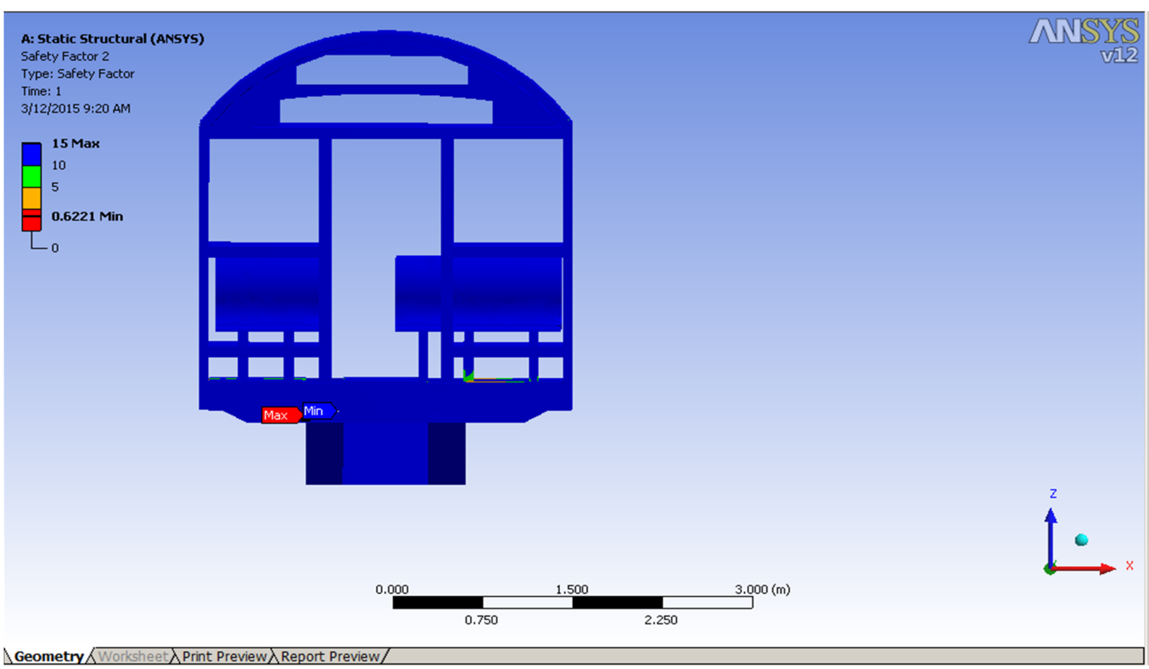


Fig 16b . Factor of safety of rail vehicle structure-end view

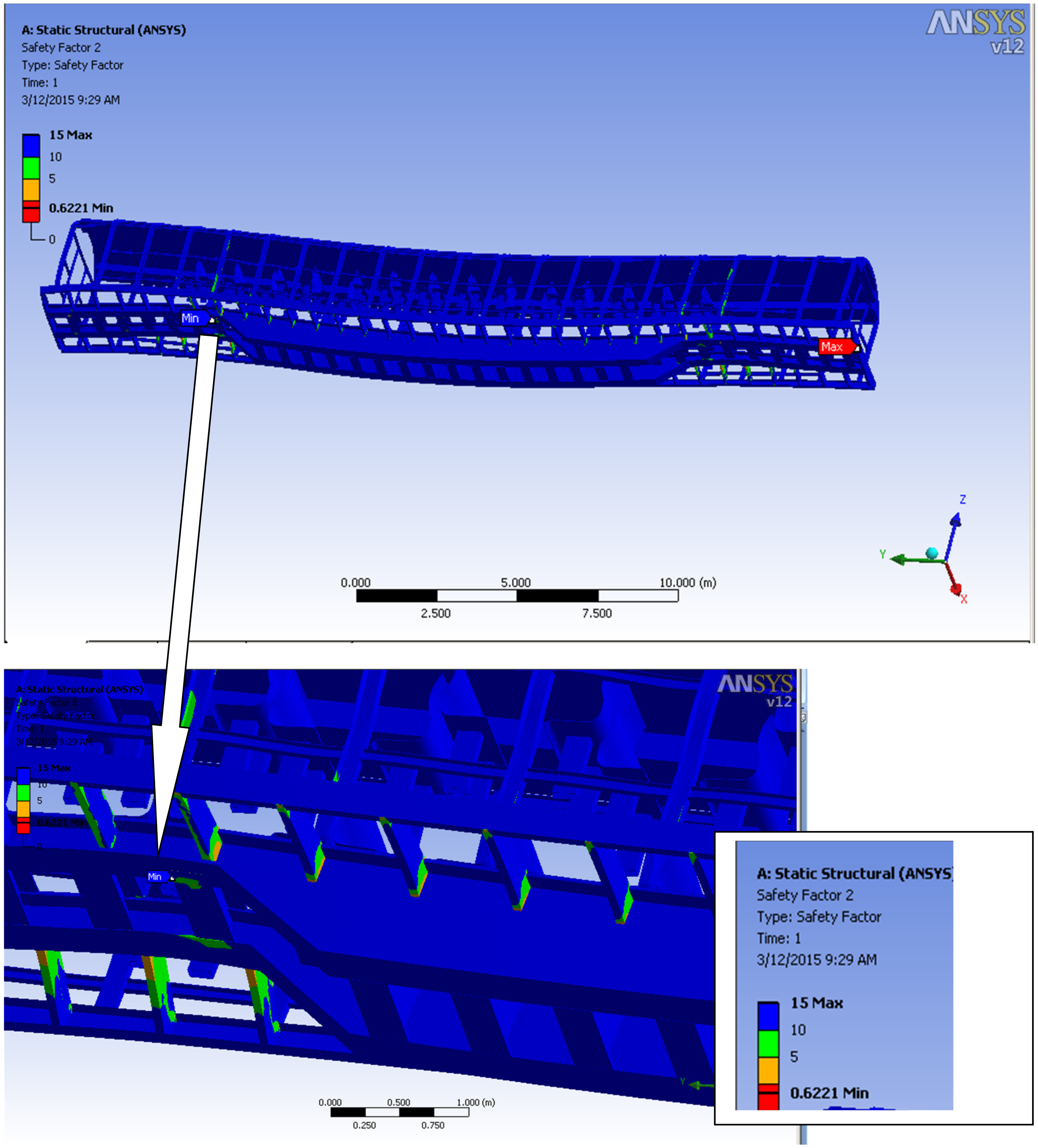


Fig 16a. Factor of safety of rail vehicle structure-under frame

4.2.2. Modal Analysis results

Modal analysis provides results which indicate the natural vibration characteristics of a component or structure. It involves determination of the natural frequencies and modes of vibration.In addition to studying vibration characteristics, the results obtained from modal analysis can be used in other dynamic analyses such as spectrum analysis. The vibrations of a rail vehicle car body are mainly caused by the design characteristics of the car body itself and the kind of design constraint applied to it.

Modal analysis is performed on FE models and can be added in the same run to all the other load cases. All fitted structures are designed with sufficient modal separation to avoid coupling. Since the modal analysis can be part of the same run as the linear static analysis, structural optimization can be performed while using design variables from the linear static solution in combination with the normal mode solution.

In rail vehicle modeling often a right handed triad is adopted

X – longitudinal translation, positive in the forward direction

Y – lateral translation, positive to the right

Z – vertical translation, positive downwards

T – rotation about X-axis (roll), positive clockwise

P – rotation about the Y-axis (pitch), positive clockwise

w – rotation about Z-axis (yaw), positive clockwise

Typical flexible modes that are interesting when focusing on vertical ride comfort are the first vertical bending mode, first torsion mode (torsional motion along the longitudinal axis) and the first so-called breathing mode, resulting from cross-sectional shear. Normally, the frequencies for the first flexible modes lie in the range of 8–15 Hz, however, depending on the vehicle configuration and settings. The higher the eigen frequency, the more complex the mode shapes. To avoid resonance between rail vehicle car body and bogie, the lowest possible bending frequency of the car body is determined by the eigen frequencies of bounce and first bending mode of the bogie.

According to present standards, humans are mainly sensitive to vertical vibrations in the interval of 5 to 15 Hz. The sensitivity reaches a maximum approximately at 8 Hz .Human beings are sensitive to lateral vibrations in the interval of 1 to 2 Hz. Therefore, to secure good ride comfort, it is of importance not to let the eigen frequencies of the rail vehicle car body to coincide with this sensitivity range. As mentioned before, the first vertical bending mode has the largest influence on vertical ride comfort. A theoretical vehicle model with rigid and flexible car body modes is used. When the structural stiffness of the car body is decreased, so are the bending frequencies, and there is a risk of significant vibrations in the rail vehicle car body.

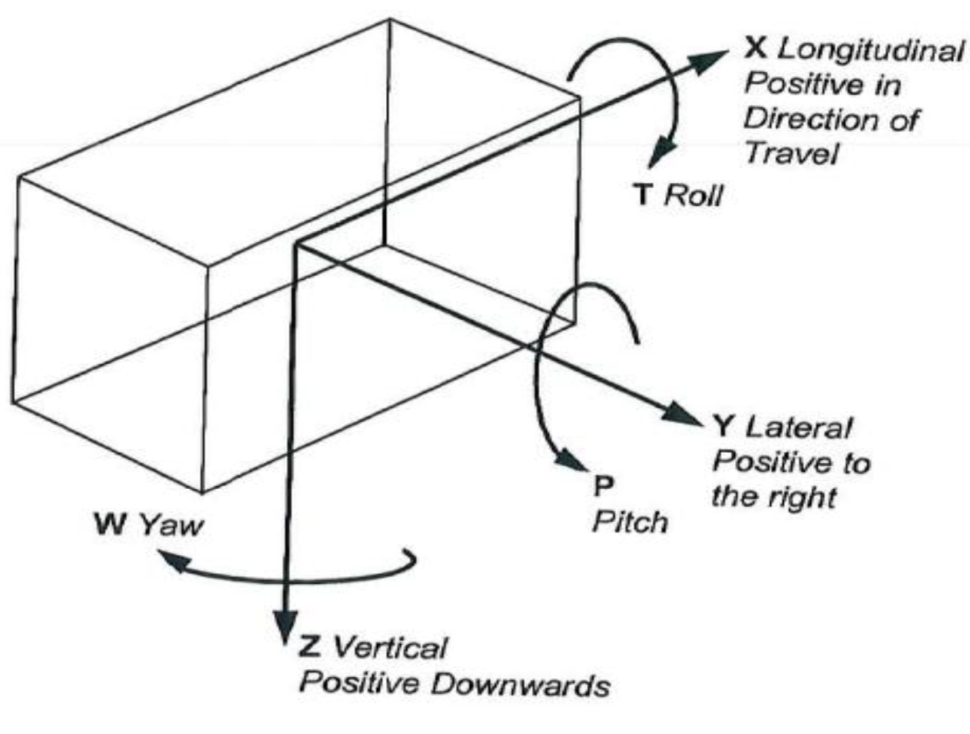


Fig 17 . Axis system for vibration modes

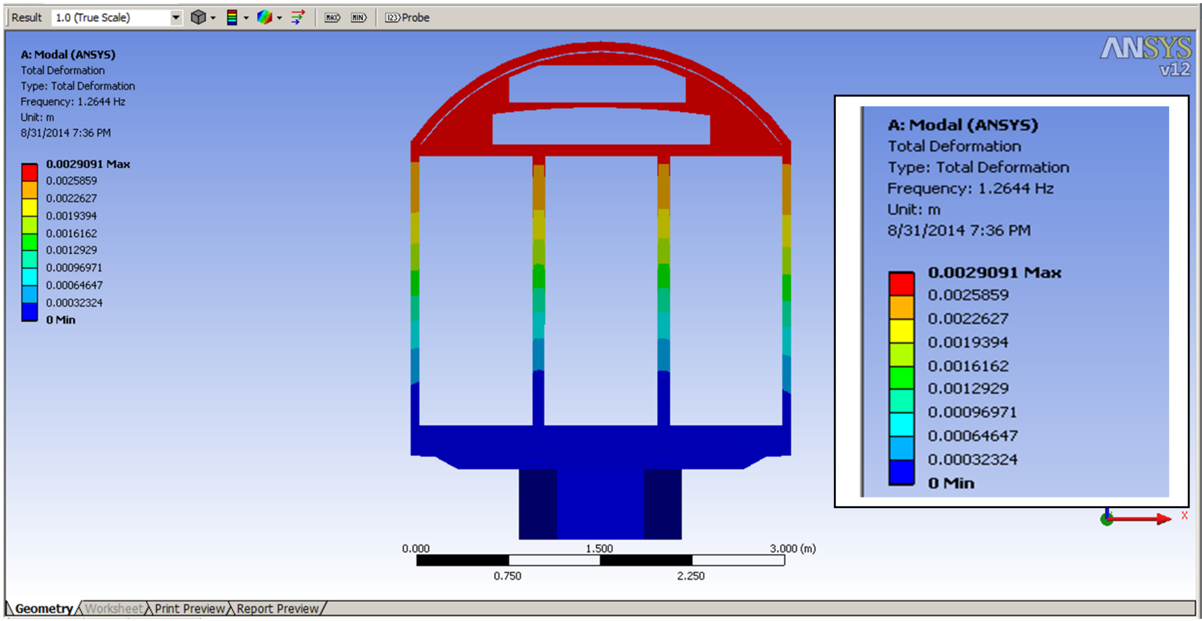
When the rail vehicle car body structure flexibility is modeled in the flexible multi body algorithms using experimentally identified modal characteristics, standard modal analysis techniques can be used to determine the rail vehicle car body modal parameters such as the natural frequencies, mode shapes, and modal damping coefficients.

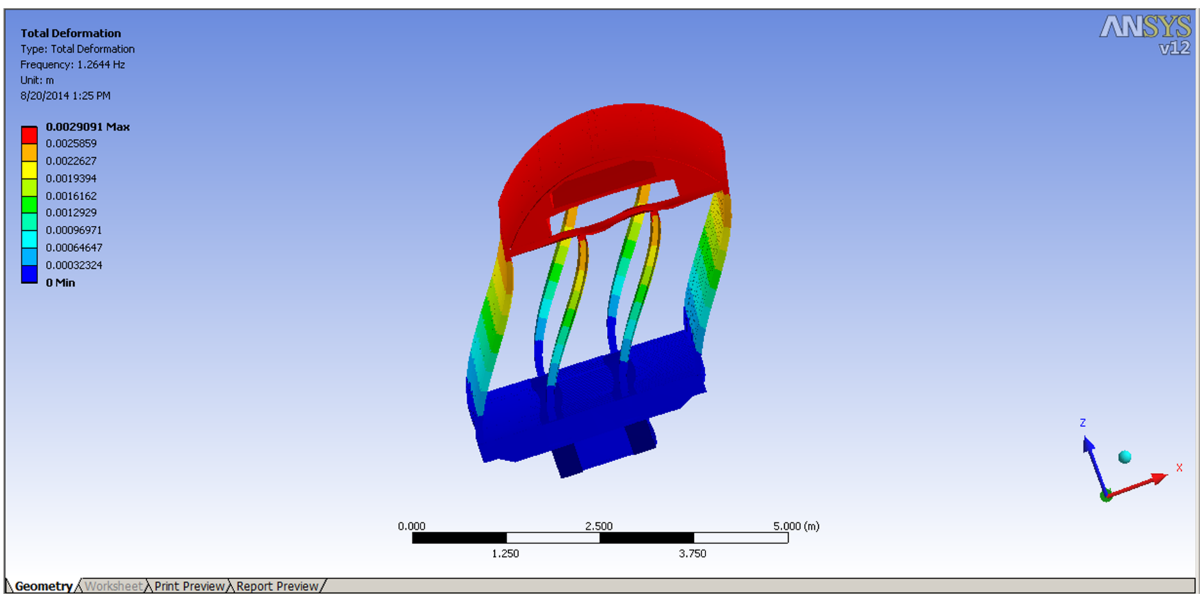
A detailed rail vehicle car body structure is idealized using the FEA software ANSYS (Version 12.0) and the FE model is shown in Fig 8. Finer meshes are used to prevent any possible stress concentration effects. The stress-strain histories at the critical areas of the car body structure are calculated. The modal frequency of the structural body is also obtained using modal analysis techniques.

Results for the Modal analysis of rail vehicle car body structure are listed below. The six lowest mode shapes and undamped frequencies of the studied passenger rail vehicle car body is shown in the table below.

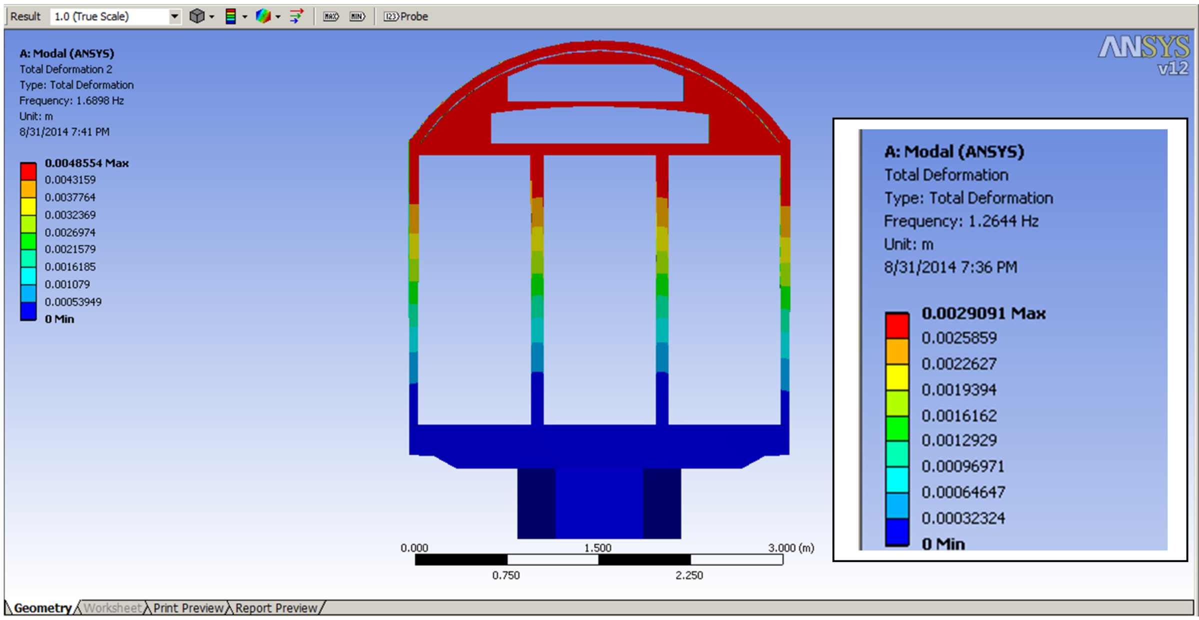
Table 11. Modal frequency and displacement of rail vehicle car body

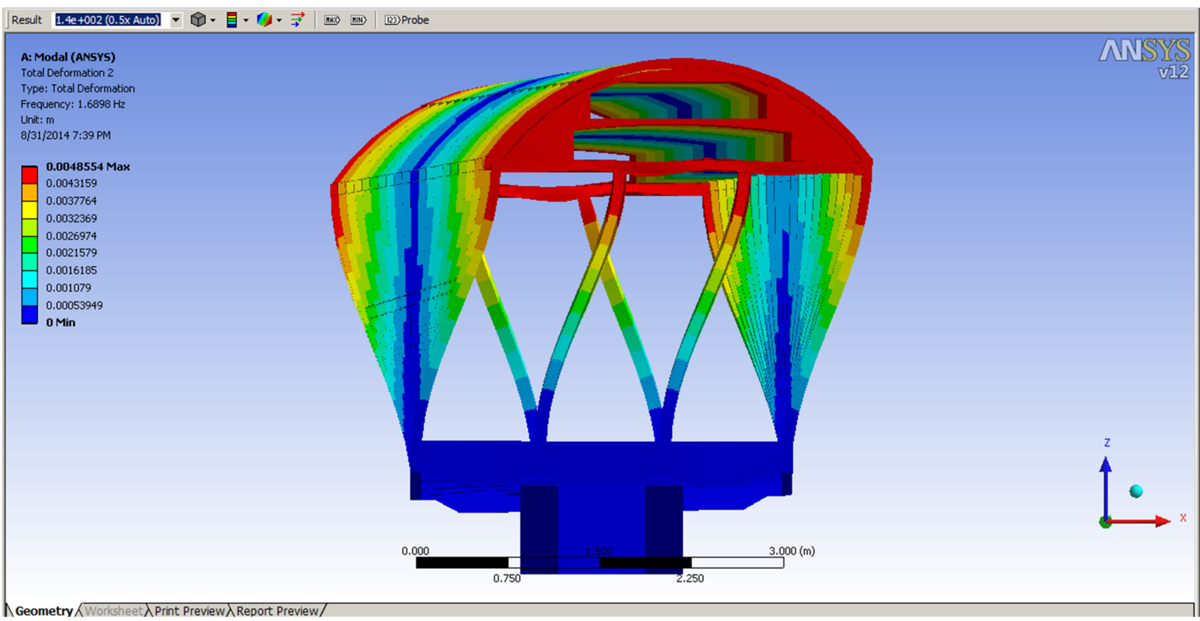
No.	**Mode shape**	**Frequency (Hz)**	**Max.deformation (m)**
1	1st lateral bending	1.2644	0.0029
2	Torsion 1	1.6898	0.0048
3	longitudinal bending	3.501 Hz	0.0028
4	vertical bending	5.84821Hz	0.0042
5	Torsion 2	10.625Hz	0.0064
6	2nd Lateral bending 2	11.047Hz	0.0071



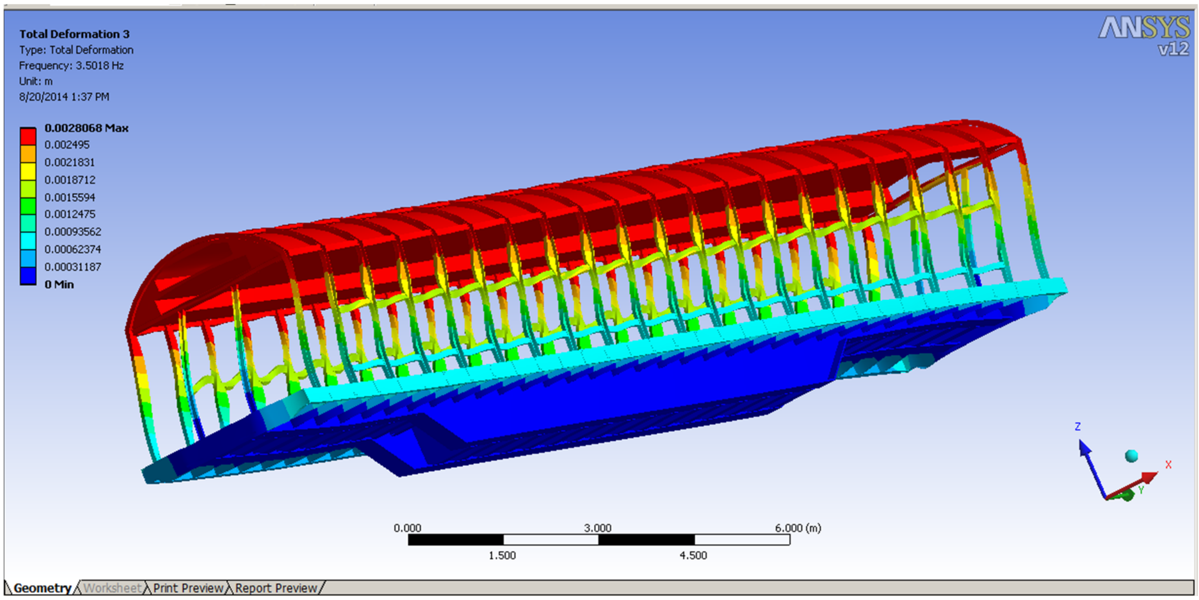


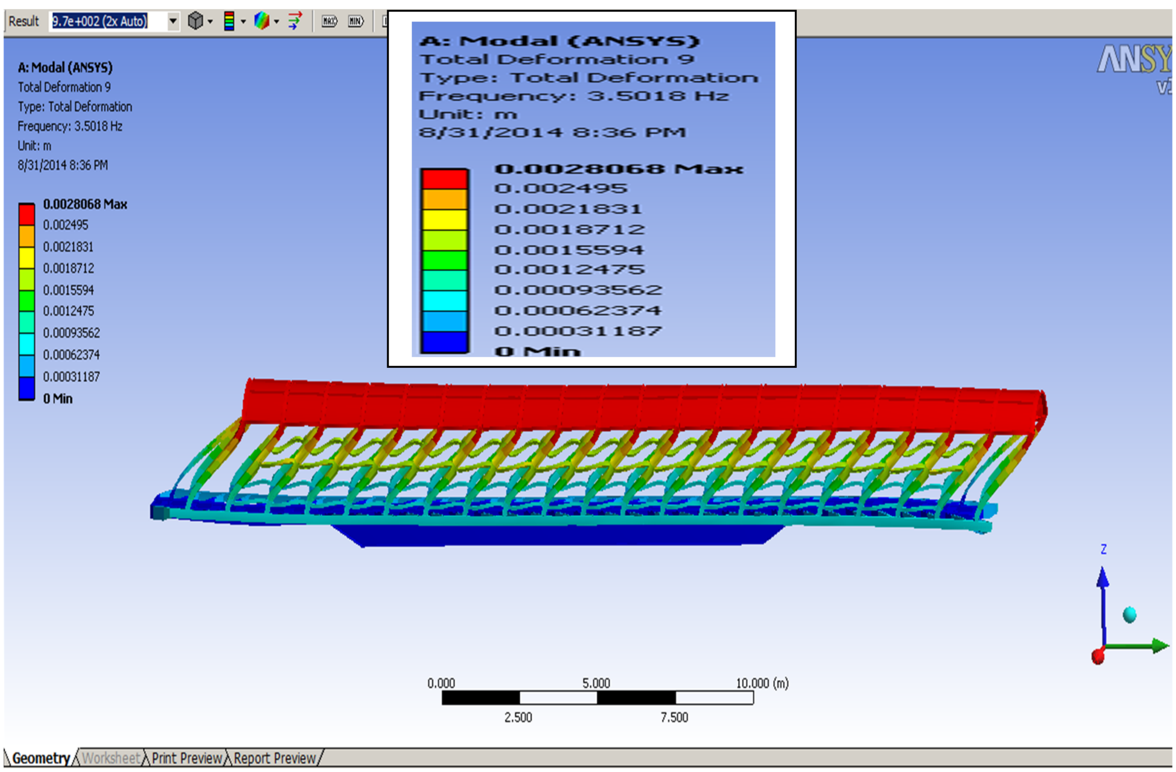
a. First Lateral bending mode shape(1.2644Hz)



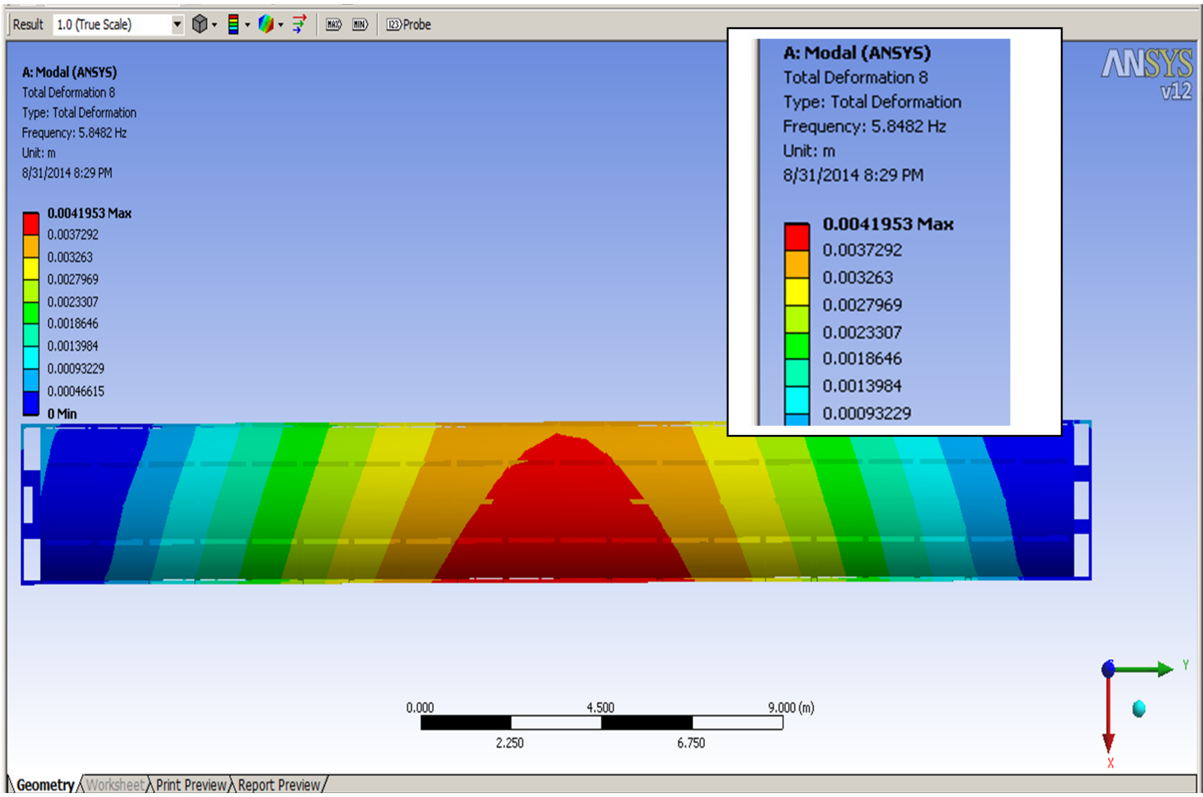


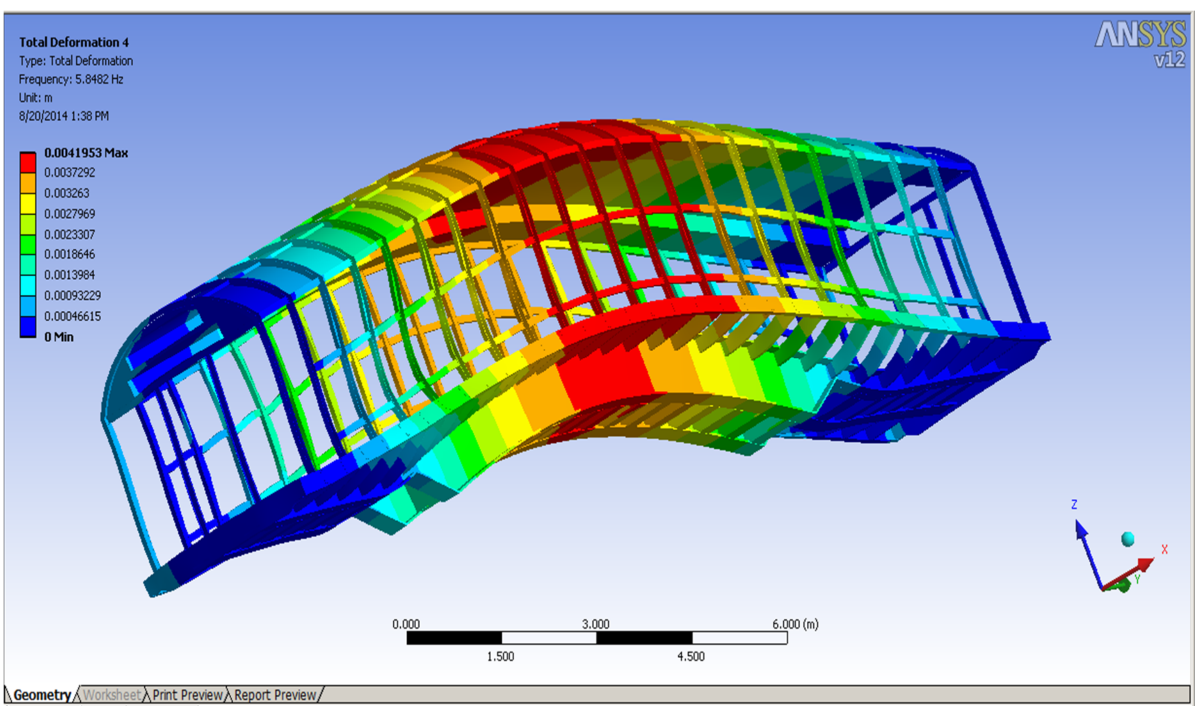
b. First torsion mode(1.6898Hz)



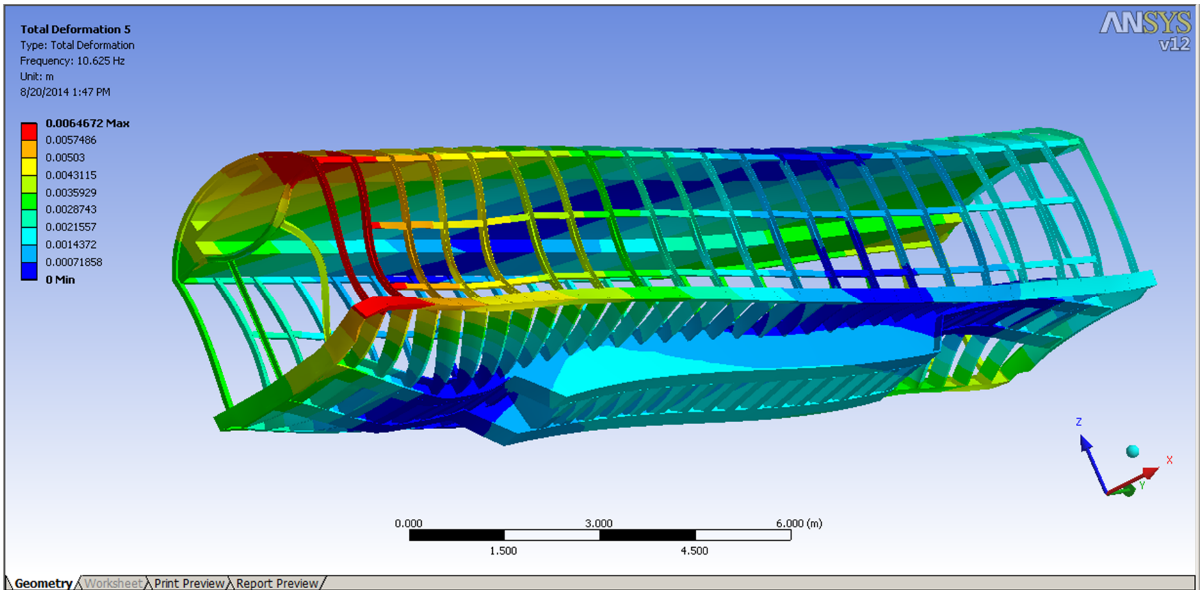


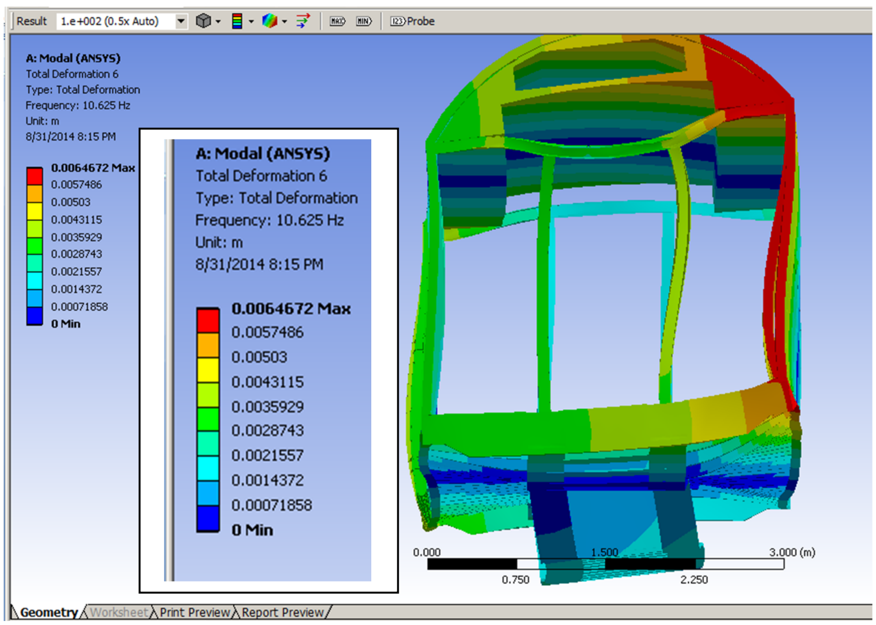
c. Longitudinal mode shape(3.5018Hz)



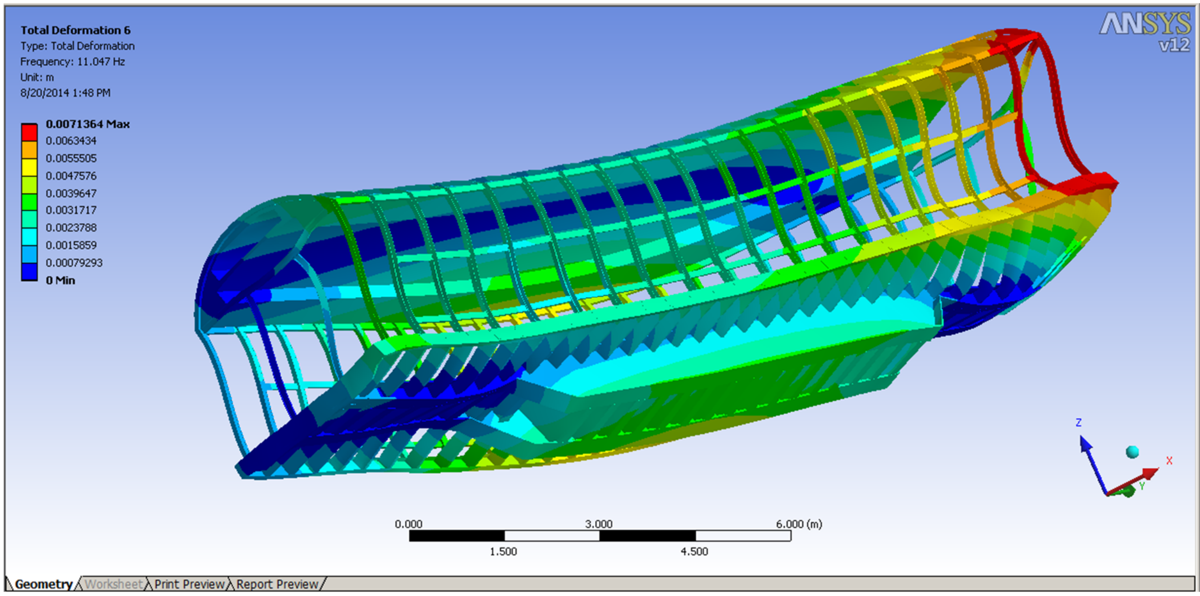


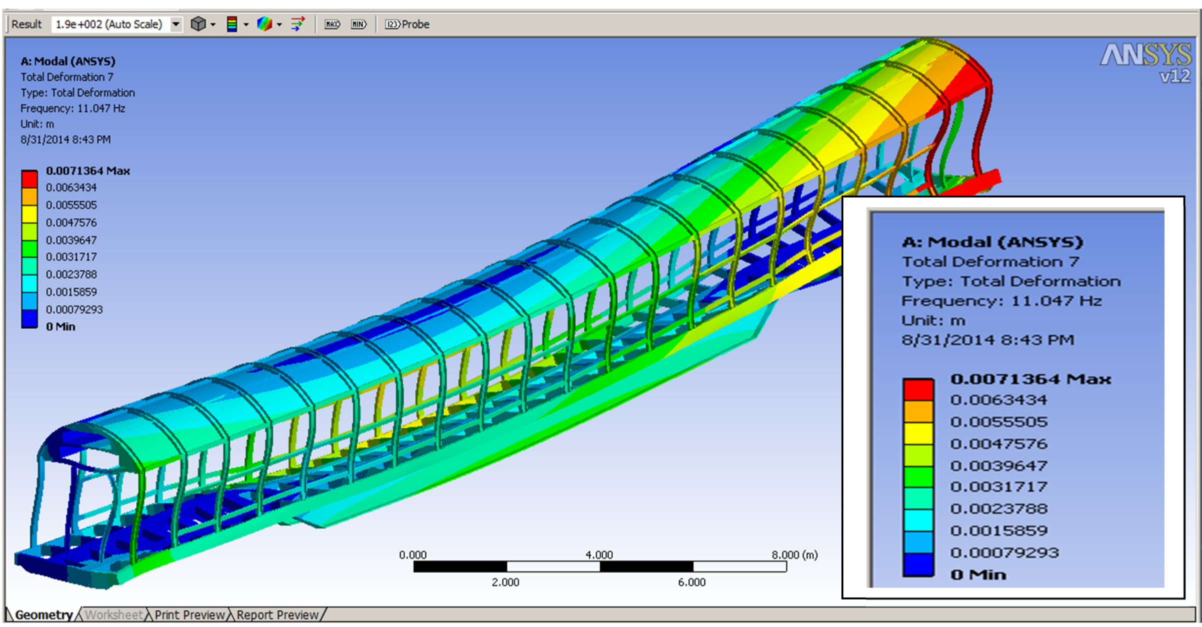
d. First vertical bending (5.8482Hz)





e. 2^{nd} Torsion mode (10.625Hz)





f. Second lateral bending mode(11.047Hz)

Fig 18. The first 6 Mode shapes and frequency of the rail vehicle

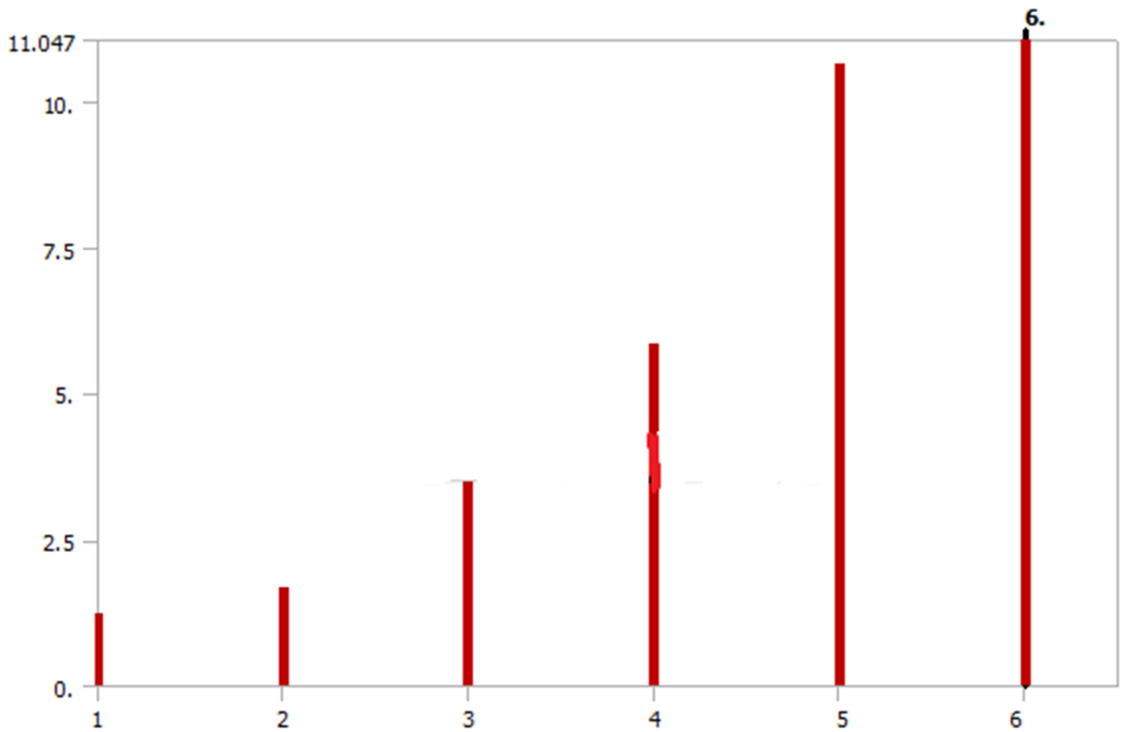


Fig 19. Response of passenger rail vehicle towards compressive & vertical load

CHAPTER FIVE

CONCLUSION AND RECOMMENDATION

5.1 CONCLUSION

The main conclusions drawn from the study are :

- The structural flexibility of the car body must be taken into account when predicting vertical vibration comfort.
- The rail vehicle car body dynamical properties depend on the excitation amplitude. That is direct connection the property of the track, the suspension and the stiffness of the rail vehicle structure.
- Passengers and car body interact significantly.
- The center sill, particularly, part of the under frame where there is change in cross-section is affected the most by static loads. The center sill section's yield stress is 460 Mpa(St 36) where as the load at that particular crossection is 401.8Mpa.
- The maximum deformation (3.9mm) of the rail vehicle structure is within the allowed range of 1/700-1/1000 for structural beams and members. Hence; mathematically ,the design is valid.

Passengers boarding this rail vehicle car are exposed to a vibration frequency ranging from 1.264-11.04Hz.This result is the natural frequency of the vehicle itself and is inevitable. Especially, the vertical bending mode having frequency of 5.8482Hz is the primary concern as it is within the sensitivity range of human beings in vertical ride comfort.

Rigid vibration modes of the first lateral bending (1.2644Hz) and first torsion(1.6898Hz) are also in the sensitive frequency range. However, this is considered little concern compared to the vertical bending mode.

5.2. RECOMMENDATIONS

Rail vehicle bodies shall be designed as stiff as possible to resist the incoming compressive, vertical, quasi static, vibration stress and as the same time provide comfort to the passenger on board.

The materials used in this design may differ from person to person. However, this passenger rail vehicle structure is made out of steel of different profile that is believed to be available on the market at any cost. However, this increases the dead load, operational cost, and efficiency of passenger rail vehicle. It is recommended to use more strong and light weight structures which are aesthetically, structurally, and operationally viable.

From a ride comfort point of view it is preferable to increase the structural stiffness of the car body in order to reduce the risk of car body oscillation resonance.

The following measures can be taken to increase the car body stiffness and to increase the natural frequency the car body.

- Make the car body as short as possible. However, this conflicts with the desire to maximize the number of passengers in relation to the number of bogies.
- Make the car body crossection as big as possible. However, the crossection must not interfere with the prescribed car body gauge profile.
- Use stiffer car body materials or design resulting in high car body stiffness. This is however contradictory with the requirement of lightweight car bodies.
- A comparison of the highly stressed elements and the ultimate strength of material used for construction shows that some reinforcement has to be made on certain parts of the rail vehicle structural components especially on the cross bearer and center sill. These will alleviate the fatigue life. An alternative material having better strength used in the construction may also work well with an additional cost.

5.3. FUTURE WORK

- Fatigue analysis of passenger rail vehicle car body
- Crashworthiness analysis of passenger rail vehicle car body structure
- Dynamic analysis of passenger rail vehicle car.
- Optimization of passenger rail vehicle structural members.

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