“Evolution of empirical relationship between high level design parameters with performance criteria of a ladder type chassis frame.”

Major Project Report

Submitted in Partial Fulfillment of the Requirements for
The Degree of

MASTER OF TECHNOLOGY
IN
Mechanical Engineering

(CAD/CAM)

By

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May 2007

CERTIFICATE

This is to certify that the Major Project Report titled

"Evolution of empirical relationship between high level design parameters with performance criteria of a ladder type chassis frame."

submitted by **Sourabh Tiwari (05MME018)** towards the partial fulfillment of the requirements for Master of Technology (Mechanical) in the field of CAD/CAM of Nirma University of Science and Technology is the record of work carried out by him under our supervision and guidance. The work submitted has in our opinion reached a level required for being accepted for examination. The results embodied in this major project work to the best of our knowledge have not been submitted to any other University or Institution for award of any degree or diploma.

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submitted by **SOURABH TIWARI (05MME018)** towards the partial fulfillment of the requirements for Master of Technology (Mechanical) in the field of CAD/CAM of Nirma University of Science and Technology is found satisfactory and is approved for the degree of M.Tech (CAD/CAM).

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1)

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3)

4)

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I want to show my gratitude to those who have helped me throughout my project presentation.

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ABSTRACT

The greatest challenge for any automotive company is to come out with a product having a competitive edge that requires acceptable performance but at a lower cost in the shortest possible time. Reducing development time and cost are thus a priority for all the companies to improve their competitive edge. Product optimization methods are continuously being improved to reduce the cost of the product and also reduce the development time & effort. This dissertation is aimed at reducing the development time & effort for a ladder type chassis frame.

This dissertation is carried out to obtain relationship between high level design variables having dominating effect and performance criteria of a ladder type chassis frame. The performance criteria selected are those having prime importance in chassis frame design. The chassis frame selected for analysis is a simplified model of actual ladder type chassis frame. Design of Experiments was used to create twelve frame models, within design envelope. The analysis was performed using Lumped Parameter Model (Mathematical Model) and Finite Element Analysis software (OptiStruct, I-DEAS). After getting sufficient correlation, the equations obtained in Lumped Parameter Model (Mathematical Model) were used to perform Weight Optimization of chassis frame within design envelope. The work shows the possibility of reducing effort, time and cost required for chassis frame design in early design stages. This although will not give exact solution but can give a near optimal initial design of chassis frame, thus reducing the effort for optimization during the detailed design phase.

Keywords: design variables, performance criteria, chassis frame, Design of Experiments, design envelope, Lumped Parameter Model, Finite Element Analysis, Weight Optimization.

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Nomenclature

β	=	Combined Lateral stiffness
γ	=	Combined tire vertical stiffness
ε	=	Combined spring stiffness
K_S	=	Frame Torsional rigidity
a_i	=	Height of i^{th} cross-section of Chassis frame
b_i	=	Width of i^{th} cross-section of Chassis frame
t_i	=	Thickness of i^{th} cross-section of Chassis frame
ll_i	=	Position of i^{th} cross-section of Chassis frame
L_i	=	Length of i^{th} cross-section of Chassis frame
t	=	Pure torsion
tb	=	Torsion due to bending
b	=	Cantilever bending
X	=	optimisation design variables.
m	=	no. of inequality constraints.
n	=	no. of variables.

1 INTRODUCTION

In the era of globalization, the customers are becoming more conscious about quality, life and performance of the equipment. The market is growing but simultaneously competition is also increasing. Government regulations are also becoming more stringent. To cope with the market and regulations, Automakers need to come out with improved products but with reduction in development time and cost.

In earlier days design evaluation was done using prototype testing. But it has many disadvantages like high cost, longer lead time and limited scope of modification. Nowadays Computer Aided Engineering (CAE) is used for virtual prototyping and analysis. Sufficient confidence level is made using CAE software for the product performance, prior to prototype testing. While this is a welcome step and has resulted in substantial reduction in development time, further improvement is possible by ensuring that the initial design chosen is closer to the optimal design. This will potentially result in less iteration using costly CAE resources and can further reduce the development time.

1.1 Need and scope of the project:

The formulation of relationship between the design variables, sectional properties and performance criteria can be used to evaluate the responses without using tedious and costly CAE analysis.

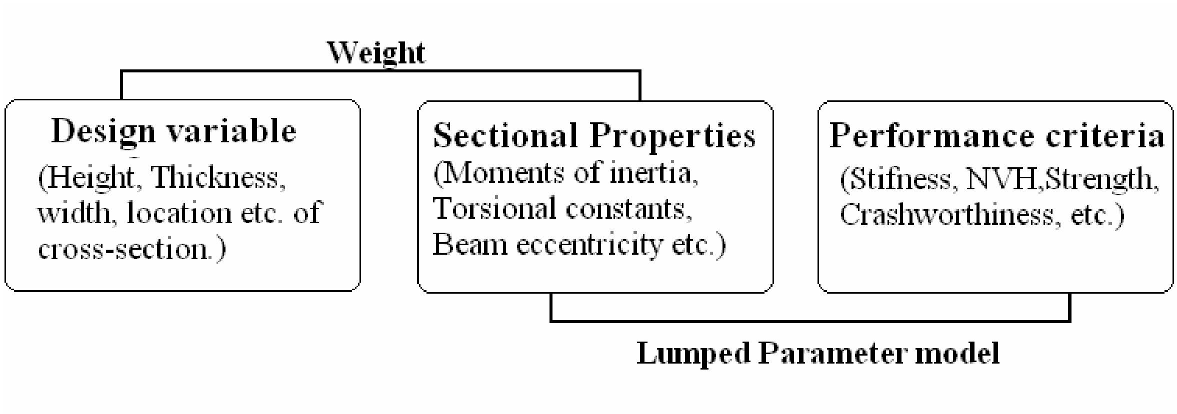


Figure 1.1 Basic Idea of the project

The chassis frame studied in this project is a simple ladder frame with two long-members and seven cross-members. The frame is simplified for the sake of convenience in analysis. It is having its structure close to actual frame. Analysis is done both analytically and using FE simulations. Finally Optimization was performed using the equations obtained by the LPM.

1.2 Chassis Frame

Chassis Frame is the basic structural unit which carries the payload, supports various components of vehicle and provides strength and rigidity to automobile. With growing emphasis on weight saving, power, payload carrying capacity and ride and handling aspects frame design and analysis needs greater attention.

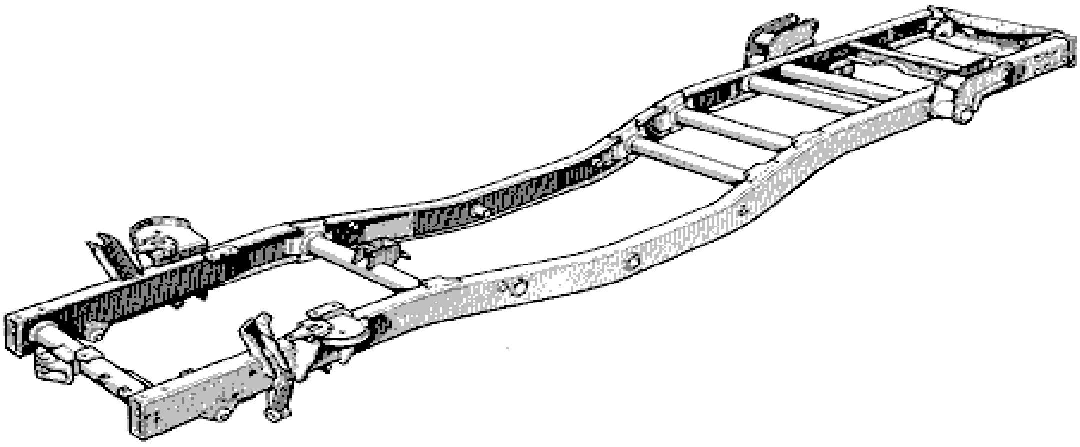


Figure 1.2 Ladder type Chassis frame model

Chassis frame is a working, mechanical part of the whole truck vehicle. It is the link in the transmittal of motive force from the powertrain to the wheels and back to the payload. Chassis frame of an automobile is basically composed of long-members and cross-members. The load carrying members are the long-members and are joined at significant points by the cross-members. The design, shape, complexity and arrangement of the frame structure vary as per the packaging, design & performance requirements. Long member vary greatly in length and dimension depending on truck application. Likewise cross-member varies in design weight complexity and cost depending on the cross-member purpose and location.

Ladder type chassis frame is the simplest kind of chassis frame. It is not the most effective arrangement to resist parallelograming or twist. The cruciform or X – type member is highly effective in resisting these motions but due to space and economic reasons its use is limited.

2 LITERATURE REVIEW

1. Finite element dynamic analysis of an automotive frame. (SAE-730506)

By V.J. Borovoski, R.L. Steury and J.L. Lubkin, FORD MOTOR COMPANY

In this paper authors have used the basic Finite Element Model to predict the dynamic response characteristics of an automotive frame. They developed and validated an efficient and accurate model for an automotive frame. NASTRAN was used for the dynamic analysis and following factors were evaluated:

- Refinement in the element mass representation.
- Inclusion of shear deformation.
- Allowance for flexibility of major frame joint.
- Torsional behavior of short open cross-section.
- Flexural Inertial properties of welded double channel cross-section.

After reasonable estimates and suitable assumptions the predicted natural frequency and measured values of mode shapes were found within 4% of measured value.

Inference: Finite Element model can be used as a design evaluation tool for automotive frame, prior to prototype construction with proper assumptions and boundary conditions.

2. Stresses and deflections in Truck Long-member Attachments.

By D.W. Sherman, Dana Corp. ***(SAE-690175)***

In this paper Authors have discussed the major principles and characteristics of structural distortions. He explained the reason of failure of frame rails at a point other than maximum bending moment. A half inch flange hole in the top and bottom flanges of a three inch flange, nine inch deep section, can reduce the strength by 11.6%. Author had used graphical illustrations to support his view point. They had summarized all the discussions under following points:

- Structural nature of frame parts.
- Primary input forces.
- Action of flat surfaces.
- Stress concentration.
- Side load generation and influence.

Inference: Placement of cross-member should be done with proper considerations like deformation; load values etc. and should be attached to the web of the long-members.

3. A new concept of Light Weight Highway Tractor Design.

By C.V. CROCKETTE and D.J. LaBELLE **(SAE-590281)**

This paper explains the design of highway tractor, specifically for the job of pulling a trailer. In this paper following measures were tried to achieve:

- Ability to haul more weight.
- Ability to haul bulkier load.
- Decrease in operating cost.
- Driver comfort and safety.

The description of vehicle is done in terms of power train, frame suspension, Cab and Brakes. Author has used various illustrations to Visualize and support his explanations.

Inference: Channel frame is having good torsional flexibility for off highway use but has many disadvantages like excessive material requirement, low bending and torsion rigidity and excessive cost.

4. Dynamic testing and computer analysis of Automotive Frames.

By Jay K. HAY and J. Michael Blew ***(SAE-720046)***

This paper Discuses the method for accurate and detailed Dynamic Design and Evaluation of automobile frame. The work has been done in two phases:

- **Dynamic testing of existing prototype** This is done to obtain empirical data for special structural effects such as joint efficiencies, joint slippage and degree of frame member end wrapping constraints for input into computer model.

- **Formulation of detailed computer model and analysis using the static and dynamic beam Finite Element program.** An automobile (Truck) underframe is modeled and is dynamically tested using BEST-II. This software is a comprehensive structural analysis computer program. Finally the correlation between these data is studied. Computer simulation is used for determining the effect of change on dynamic performances by varying the geometry of frame.

Inference: Complete frame design, analysis and Optimization , using only blueprint information can be done by computer modeling techniques and a comprehensive structural analysis software.

5. Truck Frame Analysis Study.

By Oskara Michejda (SAE-710594)

This paper describes the requirements of a realistic mathematical model of a truck frame which can be used in optimizing its design. The Discussions includes the effect of 3-dimensional forces, variable cross-sections, off shear-centre loading and joint flexibility.

The mathematical analysis of deformation due to engine torque-rear axle and due to uneven terrain is discussed by the author in detail. In the end author concludes that the calculation of forces should be based on elastic interaction of all vehicle components, after verifying the validity of assumptions in the derivation provided.

Inference: Stress analysis of frame should consider vertical gravity forces with dynamic stress history, horizontal lateral bending due to maximum force and twisting of frame.

6. Frame Rigidity – How much and where.

By Richard L. Exler, Chrysler Corp. ***(SAE- 640017)***

This paper defines the data in the area of frame flexibility versus rigidity and the contributions of various frame construction concepts these qualities. Ladder frame is dealt in particular in this paper. The design of cross-member and their attachments to the longitudinal member or long-member is given prime importance. Author has discussed types of cross-member and cross-member attachments, long-member joints and types of attachment gussets. He has also included elastic theory applied to the chassis frame and comparison of frame rigidity and flexibility. Author had conducted various tests and finally he concludes that:

- Degree of frame rigidity basically depends on operational conditions, other components and installation characteristics and economics of the required construction. It also reflects the experience and good engineering judgment.
- Test and documentation results must be used to compare the results of ultimate vehicle performance.
- Maximum stiffness must be located in the areas of the frame that encompasses the other major chassis components like suspension, steering and engine, and cab mountings.

Inference: Frame rigidity at any location should be a function of permissible deformation, loading and attachments. Product cost (economy) and past experience do play an important role.

7. Development of Aluminum Frame for Heavy-Duty Trucks

By Kenji KARITA, Yoichiro KOHIYAMA, Toshihiko KOBIKI, Kiyoshi OOSHIMA and Mamoru HASHIMOTO (Technical review-2003)

This technical report describes the development targets and technical features of a prototype aluminum chassis frame for heavy-duty trucks. It also gives an overview of a super-lightweight bulk truck that has a cab-and chassis configuration incorporating an aluminum frame and is fitted with an aluminum tank that was made by Mitsubishi Materials Corporation and has a class largest capacity of 17 m3.This report gives brief detail of variable-section extrusion technology that was used to mold the long-members of the aluminum frame. Finally Collaborative development with Mitsubishi Materials Corporation and Mitsubishi Aluminum Co., Ltd. yielded an aluminum frame that satisfies the weight-reduction target and has sufficient strength and rigidity equivalent to those of a standard steel frame.

Inference: Use of Aluminum instead of steel as truck frame reduces tensile strength by half and vertical elastic modulus by one-third. Channel section cross-member, which are highly restraint to bending, can be combined to form box section, which are highly restraint to torsion.

8. A Design Concept for an Aluminum Sport Utility Vehicle Frame

By Michael W. Danyo, Christopher S. Young, Henry J. Cornille and Joseph Porcari, Ford Motor Company ***(SAE-2003-01-0572)***

One of the principle techniques of reducing the weight of a vehicle (without affecting size or function) is the use of alternative materials. This study explores the use of aluminum in auto/truck frames. The objective of the study was to assess the capability of an aluminum frame to achieve equivalent performance to the 2002 Ford Explorer frame, but at a 40% weight reduction. Preliminary analysis of this study tells that an aluminum frame with the gage required achieving the desired stiffness needed for ride, handling and NVH (noise, vibration and harshness) will probably have excellent energy absorption during frontal impacts. Finally the authors conclude that:

- 40% weight reduction is possible with Aluminum frame without sacrificing the performance attributes like stiffness, durability, and safety.
- MIG welding should be minimized to control distortion. This may be best achieved by use of stampings in the long-members with joining achieved principally by self-piercing rivets and adhesive bonding.

Inference: Aluminum can be used for frame weight reduction with slightly larger packaging and having same performance.

9. An Analysis of Idling Vibration for a Frame Structured Vehicle

By Hiroshi Takata, Mitsuo Iwahara and Akio Nagamatsu

(SAE- 2003-01-1611)

In this paper author has used Finite Element Model to evaluate the idling vibration characteristics by entering the engine exciting forces to the Frame of Sports Utility Vehicles. More focus was given on first order vertical bending mode of frame as it is having significant effect. The path for the engine to excite vibration to a vehicle body through an engine mount, which has a greatest effect, is studied. Souma's Method was used to identify the exciting force through an engine mount. Finally authors conclude that:

- It is reasonable to use Souma's method to simulate engine excitation forces.
- The natural modes and the natural frequency for the body are approximately same even in the vehicle condition as per correlation analysis.
- In the same method, the vibration characteristics of the frame significantly change in the vehicle condition.

Inference: The idling vibration level of the vehicle is lowered by decreasing the frequency of the first-order frame bending mode.

10. Stress analysis of a truck chassis with riveted joints

By Cicek Karaoglu .and N. Sefa Kuralay

Elsevier Science*-Finite Elements in Analysis and Design* ***38 (2002) 1115–1130***

In this paper stress distribution of truck chassis frame analyzed using Finite Element package ANSYS v5.3. The long-member's parameter like thickness, connection plate thickness and connection plate length were varied to obtain reduction in the magnitude of stress near the riveted joint of the chassis frame. The thickness of long-member, connection plate, joint area and length of connection plate are varied and there stress patterns were analyzed by the author. The conclusions of this paper are:

- Increasing the thickness of long-member reduces stress and increases weight. So, it is better to go for local plates at joints.
- This causes increase in stresses at connection plate which can be reduced by increasing its thickness.
- Increase in connection plate length decreases stresses in long-member and connection plate.

Inference: In long-members of frame, stress can be reduced by increasing local plate thickness or by increasing length of connecting plate.

11. Effect of the cross-sectional shape of hat-type cross-sections on crash resistance of an "S"-frame.

By Heung-Soo Kim and Tomasz Wierzbicki

Elsevier Science- Thin-Walled Structures 39 (2001) 535–554

In this paper author describes the design aspects for crash characteristics and weight efficiency of the front part of an automobile frame. The specific energy absorption which is the measure of weight efficiency and crashworthiness is used to access the structural performance. Reinforcement of frame members by internal stiffeners is studied and its advantages are described in this paper. Non linear Finite Element code PAM-CRASH is used for the purpose of simulation. The following collusions were made as per the report:

- The model with a diagonally positioned internal stiffener and suitable triggering dents can absorb up to 200% more energy than the typical double-hat/double-cell profile member.
- The specific energy absorption can be increased by 2.84 times by using the concept of aluminum foam-filling with 3 MPa foam.
- After 30 Computer simulations, finally two designs of ‘S’- frame were found optimum.

Inference: Use of stiffeners and/or reinforcements in “S”-frame can remarkably increase energy absorption and specific energy absorption of frame.

12. Selection of Frames, Springs, and Axles for Utility Vehicles.

By A.R. Kaduk, G. Mladsi and E.D. Clise ***(SAE-690563)***

The objective of this paper is to design a set of criteria for the selection of proper utility vehicle components specifically Cassis Frame, Springs and Axles. Author has laid down various parameters for designing the components of Utility Vehicles and concluded that:

- The springs should provide tolerable ride characteristics, and also function as a stable base for the operating derrick.
- The axles must be of sufficient size to safely support the vehicle during on or off highway travel, and at the work site.
- Author has emphasized on the need of distinction between Standard Truck and Utility Vehicles.

Inference: For frame selection on utility vehicles, resisting bending moment and Moment of Inertia must be considered.

13. A torsional strength analysis of Truck Frames using Open Section members.

By Kunihiro Takahashi, Nissan Motors Co. ltd. (SAE-710595)

In this paper Authors have discussed conventional method of Truck Frame design based on Torsional Strength and its disadvantages. He has given a new concept using open section members. To examine load transmission experimentally between members, torsional vibration was applied to commercial type frames and the stress distribution was measured in detail by the “Vibration Method”. The authors conclude that:

- Proposed method, using open sections, gives results within permissible limits.
- The stress values obtained in conventional methods are 2-20 times smaller or larger than actual results in frames with C-type cross-member.
- In conventional methods transmission of the torsional moment is not taken into account which has significant influence on stress value. This is taken into account in C-type of frame.
- The restrained torsional moment of cross-member is equal to the sum of that of adjoining long-member(s) and is not effected by length of members.
- Vibration method is useful for detailed measurement of the stress distribution.

Inference: Torsional moment of cross-member has significant effect on stress value and is independent of cross-member length.

14. Cumulative Fatigue Damage Analysis of a Light Truck Frame.

By M.R. Mitchell and R.M. Wetzel ***(SAE-750966)***

In this report a case study of fatigue analysis of a light weight truck frame is discussed. The objective of this paper is to determine whether an existing frame design can safely accept a 10% increase in load. The analysis includes an experimental stress strain analysis, proving ground test data and experimentally determined properties of frame material. The conventional analysis is done on the basis of three common, single parameter damage methods. Authors explain a new method based upon stress and strain and compares it with traditional method. Finally the new method was found to be adequate. The original design of the light weight truck frame performed adequately under 10% increase in load which was proven by 3-years of successful field service.

Inference: Strain based cumulative fatigue damage procedures are more accurate than stress based cumulative fatigue damage procedures in frame design.

15. Frame Design Analysis under Complete Vehicle Boundary conditions.

By Wayne A. McClelland, Jay K. Hay, and Albert L. Klosterman, Structural dynamics corporation. ***(SAE-741142)***

This paper presents a comprehensive approach to frame design analysis that incorporates state-of-the art computer modeling and laboratory testing techniques. This paper comprises the study of frame and other component's behavior due to dynamic loading of the entire vehicle. The frame modeling includes open section beam modeling, cutouts, joint flexibilities and attachment considerations. Fourier analysis and constrained modal testing are described by the author. Authors conclude that this approach of frame dynamic analysis can be applied to various design and development stages of truck frame which are as follows:

- Frame concept studies
- Prototype development
- Troubleshooting of field complaints
- Background of future model design

Author has used various Illustrations and equations to describe the methodology. He has given concepts for assembling and solving the total vehicle governing equations for static, periodic, transient and random in service loading.

Inference: The approach of frame dynamic analysis is accurate and convenient and this can be applied to various design and development stages of frame.

16. Fatigue Life estimation on HSLA Chassis Frame.

By H. Shirasawa, J. Jizaimaru, T. Mizoguchi and N. Tada ***(SAE-810358)***

In this paper fatigue life of HSLA chassis frame was estimated by fatigue test performed on several model specimens with five kinds of welded joints of 2.6mm thick HSLA steel sheets. Authors have discussed about Scatter in fatigue life based on method of strain measurement based on following measures:

1. Availability of dynamic strain.
2. Effect of ratio of overhung bead.
3. Correlation between strain-measured bead and fractured bead.
4. He also discussed the relation between profile of welded bead and fatigue life. Finally author concludes the following:
5. Apart from toe radius of welded bead, fatigue life is influenced by ways of crack initiation and propagation.
6. Fatigue life variation is significant with joint geometry and loading condition.

Inference: Fatigue life is affected by toe radius, cracks and joint geometry. In chassis frame, local strain is an important parameter in fatigue life evaluation.

17. Truck Frame long-member Buckling stresses.

By Lewis F. McNitt, Midland Ross Corp. ***(SAE-690176)***

This paper provides information obtained from testing various specimens representing truck frame assemblies, which can be used for their effective design. The test specimen was having 15 ft overall length and is having different cross-member spacing (3-types) and cross-member attachments. The test specimen was subjected to pure bending and combination of bending and torsion. Also it is tested for the benefits which could be obtained by the proper design of structures through which the loads are imposed on a given truck frame long-member. Based upon the tests conducted on the frame specimens following conclusions were made:

- By the selection of proper loading system, proper cross-member spacing, 50% more load can be made to carry by the specimens.
- The best cross-member spacing was found to be 36 inch between centerlines of the cross-members, which were riveted only to the web of long-member.
- High vertical load inputs should be introduced to the long-members, through shear attachments. This avoids local flange radius failures.

Inference: Cross-members should be close enough to avoid buckling and high stresses in long-members. Vertical loads should be input to long-members at shear centre.

18. A general formulation for Topology Optimization .

By R. J. Yang, T. J. Walsh and P.A. Schilke, Ford motor Co. ***(SAE-942256)***

This paper provides a generic approach to solve multiple objective function and multiple constraints, using topology Optimization . In this paper MSC/NASTRAN finite element code is employed for response analysis. In this study layout Optimization is performed using topology Optimization based on density formulation. Authors have given the formulation for generalized topology design. He has given example of a simplified frame structure and a truck frame cross-member to demonstrate the general approach. The mathematical distribution of these examples gives various layout designs with different problem formulation.

Inference: The optimum design layout obtained from Topology Optimization method can be used in the early design stages of frame.

19. The studies of Crash Characteristics according to Chassis Frame types.

By Cheon-Hong Jeong, Nak-Seung Jung, In-Ho Choi and Seog-ju Cha, , Hyundai Motor Company ***(SAE- 2001-01-0119)***

This paper discusses the some parameters that have a major effect on the amount and pattern of intrusion into the occupant compartment during the frontal and offset crash test. He describes the importance of performance requirements of vehicle structure to satisfy with OFFCAP (40% offset frontal crash test of 40mph) and NCAP (full frontal crash test of 35mph). Finally authors conclude:

- In frontal offset crashworthiness in the same body structure, #-Frame is better than T-Frame.
- This can be improved by reinforcing dash and floor member.
- Author has designed a vehicle which consists of optimized body, chassis structure and material selections by controlling major parameters of frontal crash performance.

Inference: #-frame should be used for better crashworthiness of vehicle.

20. The effect of Forming on the Crashworthiness of vehicles with Hydroformed Frame Long-members. (SAE- 1999-01-3208)

By T. Dutton, S. Iregbu, R. Sturt, A. Kellict, B. Cowell and K. Kavikondala

This paper describes the use of forming simulation output data from a hydroformed frame long-member as initial material properties for crash simulation of the component. It is considered that up to 70% of Impact energy is absorbed by the chassis rail. The objectives of this paper are:

- To develop a method to utilize the data generated by forming analysis into crashworthiness analysis model.
- To determine the relative importance of parameters like thickness, work hardening and residual stresses, on crashworthiness results.

LS-DYNA was used for the simulation of crash and forming operation. It was found that compared to nominal material properties, formed rail show remarkable change in energy absorption, peak force and stroke of long-members. In the end authors conclude that forming effects plays a significant role in the vehicle crashworthiness predictions.

Inference: Forming process affects the yield strength and plastic strain relationship of frame long-members and other components. Thus they give far stiffer response.

21. An objective approach to Highway truck Frame Design.

By William J. Sidelko, Ford Motor Co. ***(SAE-660162)***

This paper gives a fundamental approach to design Highway truck frames considering all factors affecting the basic vehicle package. This method is applicable to all types of frames at any phase of frame design. Author has discussed various parameters and loading conditions which are to be considered for the design of vehicle frame design. Initial design of long-member is done on the basis of simple beam theory.

This paper also includes the significant features of a method to acquire actual vehicle road data to serve as a means for detailed stress analysis and laboratory tests conducted. Goodman Diagram was used to determine the adequacy of the long-member. Finally, deflection and stress conditions of a prototype vehicle were measured for comparison of the theoretical and experimental behavior of the frames.

Inference: There are four general modes of load acting on a chassis frame:

1. Vertical bending   2. Torsion
3. Lateral bending   4. Local wrapping.

22. Application of Computer Aided Engineering in the Design of Heavy-duty Truck Frames. (SAE- 1999-01-3760)

By Carlos Cosme, Amir Ghasemi and Jimmy Gandevia, Western Star Trucks, Inc.

The aim of this paper is to develop a process by which design changes to a truck frame could be quickly evaluated such that concurrent design and analysis would be done. This paper discusses the integration of computer aided design and engineering software codes (Pro/Engineer, ADAMS, and ANSYS) to simulate the effect to the truck frame. The MSS vehicle model was developed using ADAMS which includes:

1. 100 rigid bodies   2. 180 force elements

3. 45 joint elements 4. 415 degrees-of-freedom

ADAMS was also used for the simulation of vehicle handling, roll stability, ride performance and durability or performance of a full truck and trailer.

Inference: The changes in the design of a frame could be quickly evaluated by using combined effort of CAD and CAE codes with custom software routines.

23. Introduction of Formula SAE Suspension and Frame Design.

Edmund F. Gaffney III and Anthony R. Sallinas (SAE- 971584)

This paper includes the approach of Formula SAE suspension and frame design based on the experience of the design team at UM-Rolla. Author has discussed the basic design parameters of suspension and has given some examples also. In frame section he has discussed how to achieve a compromise with the FSAE design constraint.

Finally he has given design methodology used by UM-Rolla for 1996 race car. In this paper, Authors have made following conclusions:

- Apart from competitiveness on the race track suspension of FSAE cars must perform well in static events also like cost analysis, sales presentation and engineering design.
- Suspension should be designed for time constrained manufacturability and it should be cost effective and complexity should be minimized.
- For the dynamic events like skid pad, acceleration event, autocross, endurance race and fuel economy, designer should concentrate on geometry so that vehicle should not loose ground for all normal driving loads.
- Several Iterations must be done in order to achieve satisfactory compromise between performance envelope and design constraints.
- Time constraint is an important aspect in FSAE car design thus basic engineering concepts must be used to design the car. By increasing complexity, car may not perform well if there is no sufficient time to manufacture and test.

Inference: Stiffness of frame is an important design consideration but if too much material is added to frame in order to provide more stiffness the performance of vehicle would be degraded.

24. Substructure design using a Multi-Domain Multi-Step Topology Optimization approach.

By Zheng-Dong Ma, Hui Wang, Noboru Kikuchi, Christophe Pierre and Basavaraju Raju

(SAE- 2003-01-1303)

This paper discusses approach of multi domain and multi step topology Optimization (MMTO). This approach can be utilized to simplify the architecture/topological structures of a structure obtained in the Optimization process. This will result in increased manufacturability of the design. This paper include following three examples:

- Simplified truck frame design problem for desired vibration characteristics of the frame.
- Sandwich beam design problem with the numerical and experimental relations.
- Crash energy management problem.

Authors conclude in the end that this approach can be applied in realistic engineering design problems for developing lightweight and high performance structures in next generation ground vehicle.

Inference: MMTO can be used for frame Optimization .

25. Model Flexibility and Part Integration Concept for next generation Pickup and SUV Frames.

Abraham Tijerin and Jonathan Wortle ***(SAE- 2005-01-1026)***

This paper describes the work done to achieve the goals of manufacturer to enhance the performance of vehicle at low investment by utilizing various technologies. These goals include the variation in wheelbase, engine and transmission type, cab and box configuration to meet the customer requirements. The ease of manufacture and performance should not be sacrificed. To demonstrate the concept two prototype frames were developed for benchmarking and to measure component level characteristics. Authors have given the results in terms of intelligence, flexibility and innovation of this approach. He finally concludes that:

- It is possible to combine different materials in the same chassis frame without the galvanic corrosion.
- Design flexibility of the X-Structure concepts allows the new chassis frame structures to be tailored to specific vehicles more easily than conventional one.

- The concept shows potential to handle up to 70% of the frontal crash energy of the full vehicle.
- The built in torsional stiffness gives acceptable riding and handling when driving on sinuous roads.

Inference: By use of modal flexibility fresh changes can be made economically, speedily and efficiently.

26. Crashworthiness topology Optimization : with X-frame and deformation decomposition based on homogenized density method.

Yuseorg Jeong ***(SAE- 2004-01-1176)***

This paper discusses the design Optimization of X- frame based on homogenization based density method. Optimal structure is obtained using crash simulation and analytical math model. In the end authors conclude that:

- Decomposition based Optimization can be used for offset crash.
- The topology can vary with analysis condition like velocity and mass.
- Updation of optimal mesh based on Visco-elasto-plastic buckling gives close results.

Inference: X frame absorbs more energy than #-frame in both full frontal and offset frontal tests.

27. Heavy vehicle suspension frame durability analysis using Virtual Proving Ground. (SAE- 2004-01-1176)

Ramesh Edara, Shah Shih,Nasser Tamini, Tim Palmer and Arthur Tang.

This paper discusses the analysis of Trailer suspension frame with tires on virtual proving ground to predict the structural components durability. Authors have used LS-DYNA for contact analysis between the tire and road surfaces to predict the spindle loads as well as component loads and component stress-strain time histories.

Inference: Using vehicle suspension virtual prototype and virtual proving ground simulation, it is possible to estimate the suspension system durability performance can be estimated before physical prototype testing.

28. Simultaneous Topology and Performance redesign by large admissible perturbations for automotive structure design.

Danet Suryatma, Michael M. Bernitsas Gerald F. Budnick

and W. Joe Vitous ***(SAE- 2004-01-1176)***

The paper explores the use of large admissible perturbations (LEAP) theory to solve topology and performance redesign problems of automotive body frame. Authors conclude that this method reduces computational time for redesign significantly. The results are accurate and the algorithm eliminates the need of sensitivity analysis.

Inference: Leap algorithm can be used for solving size or topology redesign problems with static, modal dynamic etc. and there combinations.

29. The effect of frequency constraints on optimum design of automotive structures.

by Mohamed E. M. El sayeed. (SAE- 900831)

The paper discusses the study of the effect of frequency requirements on the optimum design of automotive. Authors have given a case study in order to demonstrate this methodology. By performing this study he was able to achieve weight reduction to optimum mass.

Inference: Weight Optimization can be done without sacrificing the performance requirements.

30. Stiffness and strength of square thin-walled beams.

by Kuang-Huei Lin. ***(SAE- 840734)***

Inference: For square thin walled beams, actual stiffness and strength tends to deviate from their theoretical predictions based on classical beam theories when the size upon thickness ratio exceeds 50.

31. Beam cross-section properties determined by Boundary Element Analysis. (SAE- 840734)

by Gordon H.Holze, C. Paul Pulver and Yoseph Gebre-Giorgis.

The paper gives the analytical formulation of method to calculate cross sectional properties (Area, moments of inertia and centroid location) of a beam.

Inference: Beam models are useful for analysis in early product development stages.

32. Application of thin walled beam theory in the analysis of automobile structures.

by Moisey B. Shkolnikov (SAE- 840731)

The paper describes the methods for analytical predictions of critical loads and stress areas under overall static torsion of a vehicle using Vlasov's Thin –Walled Beam theory.

Inference: Overall vehicle torsion is a major operational load with respect to the strength analysis of bodies.

33. Effective Computer Aided Engineering in the automotive product development stages.

by Robert G.Dumbensky (SAE- 2001-01-0764)

Inference: There are four distinct stages of product development namely:

1. Concept development
2. Pre-production engineering stage
3. Production Engineering stage
4. Failure analysis and redesign stage

3 CHASSIS FRAME CONSTRUCTION

Chassis frame has two primary load carrying members, commonly called long-members and are joined together at appropriate points by cross-members. The long-members and cross-members form an integral structure for the support of all chassis equipment and payload. Although simplest frame have straight long-members in the plan and elevation views, high powered Vee-type engines frequently require more complex shapes at the engine area.

3.1 Long-member:

Long-member of the chassis frame is the unit designed for carrying payload, supporting components and other parts of system.

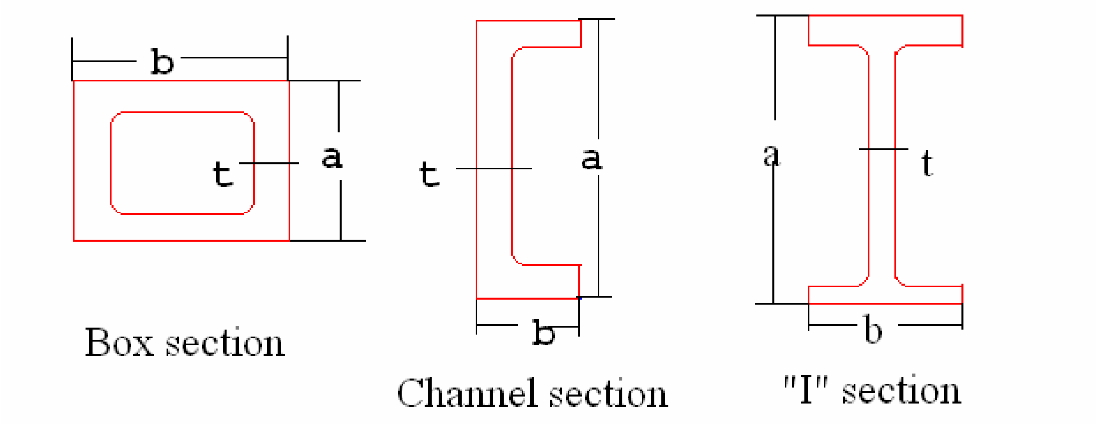


Figure 3.1 Sections for long-member

3.2 Cross-members:

The design of cross-member basically originates from past experience and packaging conditions. The design requirements of cross-members are:

- Provide support to the vehicle components.
- Separate the long-member and sustain the dimensional integrity of the frame structure.
- Provide resistance to twisting, torsion and parallelograming forces applied to the frame.

The channel, I, and hat sections are used to provide attachment or clearance for a component. Rectangular sections are best for providing torsion and bending rigidity.

Tubular sections are best to resist torsion. For resisting cantilever loads in vertical and horizontal directions I - section is preferable.

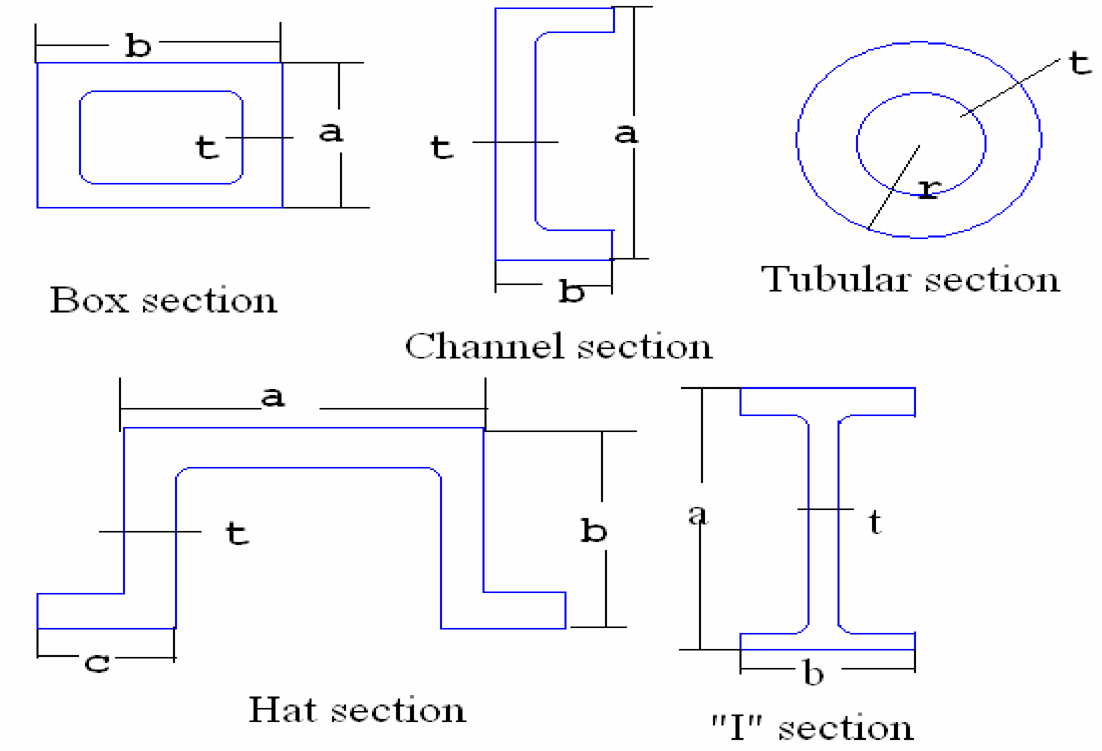


Figure 3.2 Sections for cross-members [7]

	Channel section (Base)	Hat section	Circular tube section	Closed hat section
Bending moment resistance factor (For equal tensile and compressive stress)	1.0	0.838	0.872	0.635
Torsional rigidity factor (For equal shear stress)	1.0	0.63	45.33	17.0

Table 3.1 Relative torsional and beaming resistance qualities [7]

3.3 Functions of frame

- To support the chassis components and body of automobile.
- To withstand static and dynamic loads without undue deflection or distortion.
- To retain alignment of axles and driveline.
- To transmit the steering, driving and braking forces from the respective axles to the mass of the vehicle.

3.4 Types of loads in chassis frame

- Weight of vehicle, passengers and payload.
- Bump loads.
- Impact load due to road obstacles.
- Loads due to road camber, side and cornering force while taking a turn.
- Engine torque and braking torque.
- Sudden impact load due to collision.

3.5 Materials used for frame construction

Steels are used for the construction of frames

- Mild sheet steel, carbon sheet steel and sheet nickel alloy steel (0.25-0.25%C, 0.35-0.75%Mn, 0.3%Si, 0.05%P, 0.5%S).
- **Long-members**: quenched and tempered steel with a minimum yield strength of 110,000 psi. Manufactured by forming in multiple stand rolling mill. The material is fully killed fine grain alloy steel having good forming characteristics. In the heat treated condition.
- **Cross-member:** Rimmed steel with carbon range of 0.09-0.16% by weight.

It is having yield and ultimate strengths around 32,000 and 45,000 psi. One of the key techniques for weight reduction of a vehicle (without affecting size or function) is by using alternative materials. Aluminium’s inherent material characteristics give a good option in achieving fuel-efficient vehicles.

Aluminium is having less ductility and formability than steels. Thus forming is a consideration in the design as in any stamped aluminium automotive component

3.6 Frame specification

The basic length of a frame depends upon following factors:

- Overall length regulations.
- Overall wheelbase and axle spacing.
- Permissible axle loadings.

As per SAE report J691 frame width should be 39 inch to accommodate four tires, two brakes and suspension components of a vehicle with overall width limitation of 96 inch. Also SAE report J696 proposes 40 inch frame height and 7-9 inch high fifth wheel for a 10.0×20 tire.

E.g. specifications of a typical ladder frame:

- Heavy duty 2 x 4 inch tubular steel ladder frame chassis with crumple zones
- Parallel ladder frame design with large diameter outriggers and scuttle hoop
- Door hinge, steering column and windshield post pick-up points incorporated onto main chassis hoop
- TIG welded to precise tolerances
- Headers and side pipes with hardware kit

3.7 Types of Frame Joints

The long-members and cross-members are joined by welding, bolting or riveting.

Welded joints: Welded joints are most effective because they provide good rigidity. But it is not an economical for production in very large volumes and also not suitable for heat treated high alloy steels.

Bolted joints: Bolted attachments are extensively used where there is low volume and wide component usage variation in frame is present. Also use of bolted joints makes easier for the service removal of the frame members.

Riveted joints: These are most widely used joints and especially on frames that are completely assembled and delivered by the supplier. Cold riveting is preferred over hot riveting due to the contraction associated with the later.

Bolted and riveted joints have drawback that they get loosen due to operational vibration and this results in increased flexibility of frame members. This produces undesirable motion between members and further hole elongation. To determine the level of joint flexibility for frame joints, several methods might be employed:

- Static Tests of joint
- Finite Element Analysis of joint
- Dynamic Impedance testing of joint in conjunction with simple beam.

More accurate results can be obtained from dynamic testing than static tests. Using FE modeling quick and less costly results can be obtained. Also nonlinearity of joint can be studied by this approach.

3.8 Types of Frame Attachments

The attachment of cross-member to long-member is of basically three types and is selected depending upon loading and accommodation of components. They are:

Flange attachment: In this cross-member is connected to the flange of the long-member. It is most effective for overall frame rigidity since maximum gripping span in the vertical plane can be obtained from this arrangement. The load, in this arrangement is completely transmitted in shear from long-member to cross-member, and vice versa.

Web attachment: In this cross-member is connected to the web of the long-member. In flange attachments flange hole reduces the vertical carrying capacity significantly. Compared to flange attachments web attachments are better for vertical carrying capacity.

Flange and Web attachment: In this cross-member is connected to combination of web and flange of the long-member. They are widely used to transmit the load of web mounted component, such as spring bracket directly to the cross-member, thus reducing the twist effects on the long-members.

In practice, the light, high volume models employ flange method, while reverse is generally true for the heavy, low volume models.

4 LOADING OF CHASSIS FRAME

Modes of loading in chassis frame (of prime interest)

There are four main types of loading which acts upon the frame, they are:

4.1 Vertical Bending [22]

This results in vertical deflection of frame. Vertical deflection is the function of moment of inertia of the frame rails. It should be limited to 1 inch of deflection in 360 inch of span. **I** – sections can perform better than Channel sections under vertical deflection, but due to higher fabrication cost, it is not preferred.

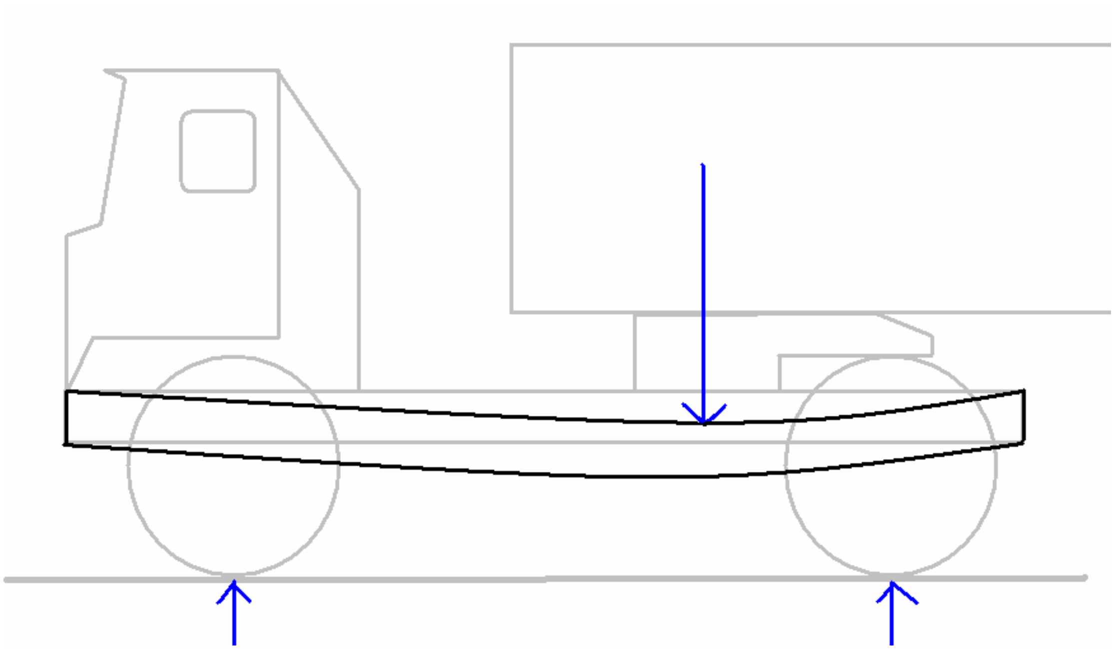


Figure 4.1 Vertical bending [22]

4.2 Torsion or "beam walking" [22]

It is the rotation of frame or one of its members about its longitudinal axis. Channel section does not have its shear centre coincident with its centroid. Thus any load not passing through shear centre produces rotation which is resisted by cross-members. This kind of deflection occurs due to out of phase loading when one wheel drops into pothole or strikes a bump. This is resisted by the longitudinal beam and lateral Torsional capabilities of the long-members.

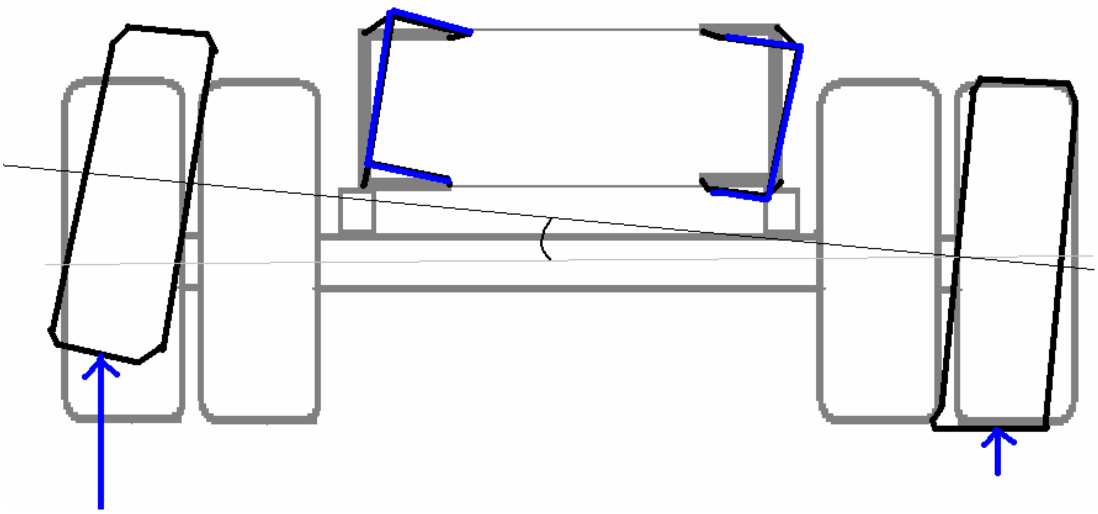


Figure 4.2 Torsion in frame [22]

4.3 Axial and lateral bending [22]

This is caused by the unbalanced longitudinal loads generated during steering, unequal braking or any end forces due to docking or collision actions. This results in parallelograming and lateral deflection of the frame. It has been found experimentally that lateral bending causes all other types of motion simultaneously. For example during low speed turning of vehicle, frame experiences high lateral loads and is subjected to combination of structural deflections: parallelograming, wrapping and rail twist.

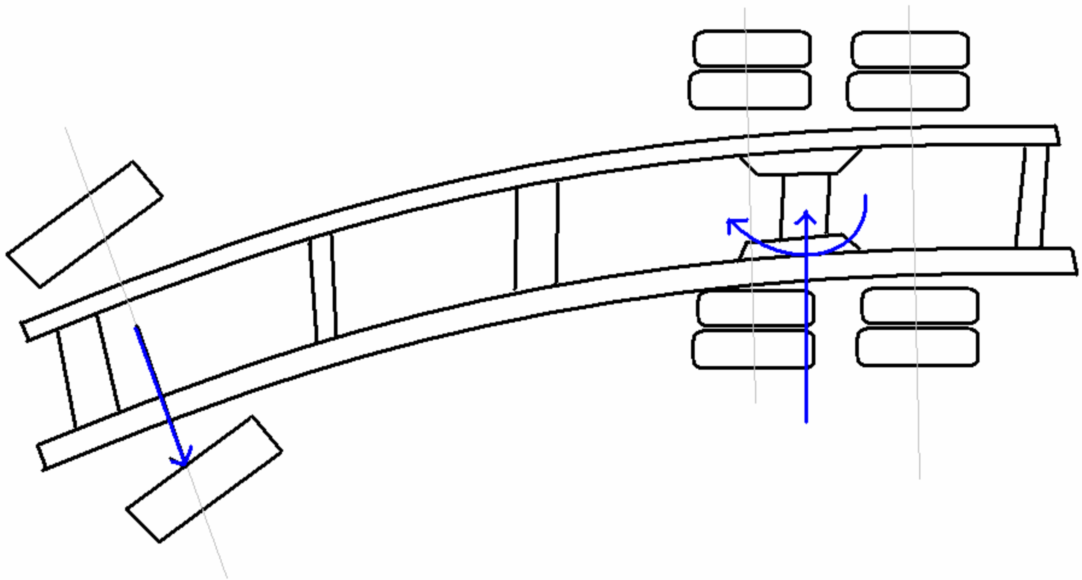


Figure 4.3 Axial and lateral bending [22]

The forces that lead to parallelograming are generally small compared to the overall vehicle capacity. "X" type of cross-member assembly at the rear axle section of the frame would provide a large measure of stiffness and resistance to parallelograming. But this makes the frame highly rigid and high stress thus generated at attachments causes localised failures.

4.4 Twist or local wrapping of individual long-members [22]

This is due to mounting of cantilevered components. Vibration of these components and resulting deflection magnitude and frequency effects on long-member stress levels causes serious problems. It can be overcome by proper placement of cross-members.

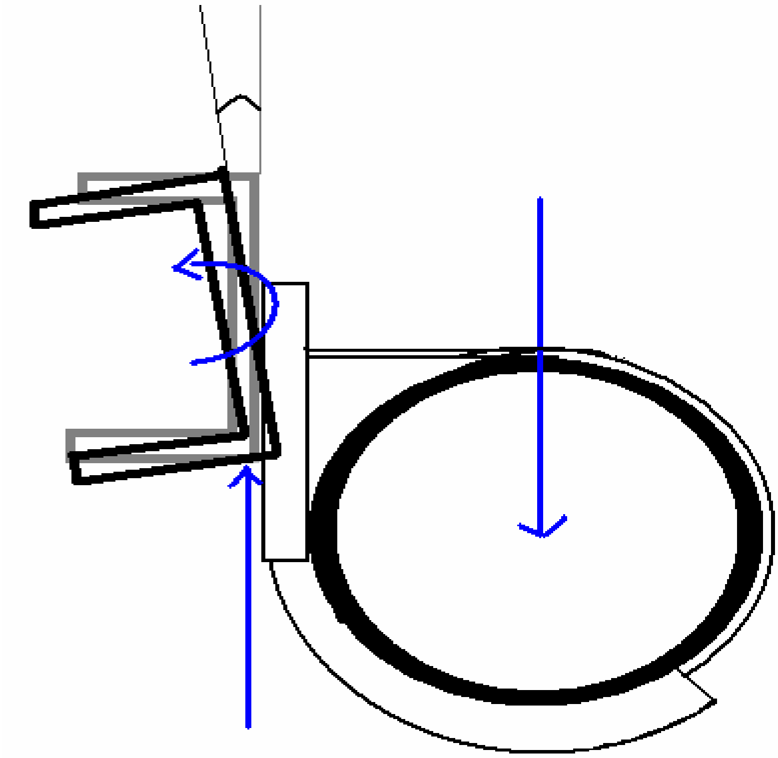


Figure 4.4 Long-member Twist

These attachments represents hang on auxiliary equipments and play limited role in overall vehicle structural functions. Some of the major attachments are as follows:

1. Battery box assembly	2. Exhaust stack assembly
3. Fuel tank assembly	4. Air cleaner assembly
5. Mud flap assembly	6. Fifth wheel mount plate
7. Rear air tank	8. Hanger brackets
9. Shift tower	10. Steering gear

5 CHASSIS FRAME DESIGN

A designer should aim for a frame design providing a balance of structural integrity, minimum weight, minimum inter-changeability, ease of service and maximum service life. Frame design should assure that contact of the wheels on the ground is maintained all the time and control of the vehicle is not lost. General criteria for frame design are:

- It should be rigid from front to rear of the cab to desired extent.
- It should permit controlled flexure from a point behind the cab to just ahead of the rear spring. This area absorbs the beaming loads imposed by the payload.
- The frame must be rigid that encompasses the rear suspension.

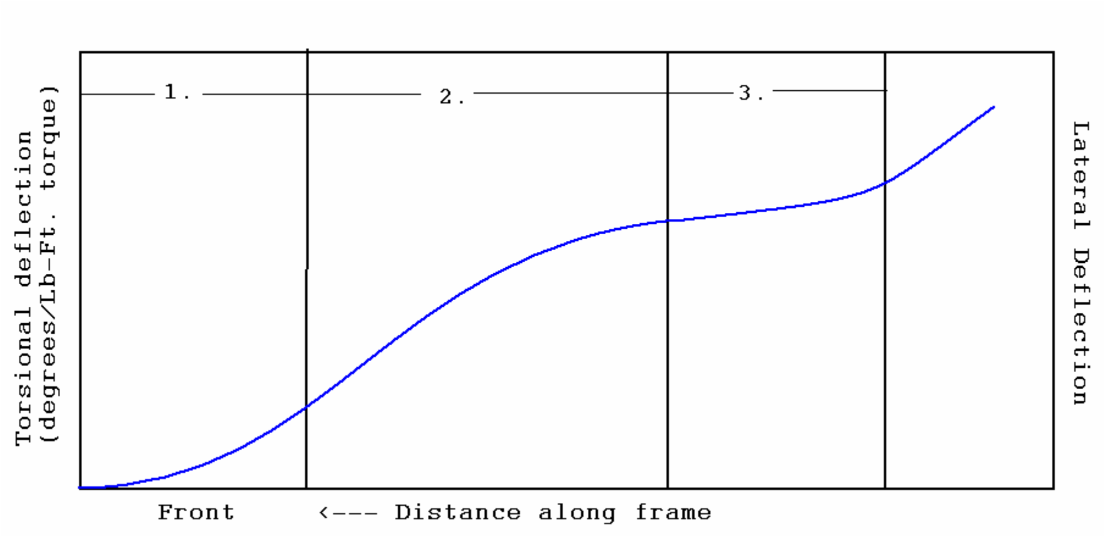


Figure 5.1 Itemized frame design Criteria [7]

For designing frame both "Theory of elasticity" and "Strength of material" approach is required. The functional limitations of frame are also important and are to be considered during the design procedure. Based upon the Literature following properties are found to be considered for frame design:

1. Stiffness
2. Peak stress (Strength)
3. Natural frequency
4. Extreme durability
5. Fatigue Resistance
6. Ease to repair.
7. Manufacturing and joining properties.

5.1 Performance requirements:

The Chassis frame is expected to perform under following parameters:

Criteria	Sub-criteria	SUV	Reference	Heavy truck	Reference
Stiffness	Bending	Y	[9],[25]	Y	[22]
	Torsion	Y	[9]	Y	[22]
	Lateral			Y	[22],[9]
	Twist or local wrapping			Y	[22],[5]
Strength	Static load	Y	[9],[25]	Y	[22]
	Fifth wheel load			Y	[22]
	One wheel lift/drop			Y	[5]
	Long-member web buckling			Y	[22]
NVH	Natural frequencies	Y	[9]	Y	[5]
	Mode shapes	Y	[9]	Y	[23],[5]
Endurance/Durability	Fatigue			Y	[23],[22]
Crashworthiness	Energy Absorption	Y	[9],[26]		
Safety factor	Residual bending moment			Y	[22]
Point mobilities		Y	[9]		
No. of components		Y	[26]		
Ease to manufacture, join and repair		Y	[9],[26]		

Table 5.1 Performance requirements of chassis frame

5.2 Properties selected

The properties shown in previous table is having many properties which are not of our prime interest. Thus the properties selected for the analysis are:

- Vertical Bending Stiffness
- Torsional Stiffness
- Lateral Bending Stiffness
- NVH (modal analysis)
- Bending
- Torsion
- Maximum deflection under One wheel Lift/drop (Strength Analysis).
- Crashworthiness

Above properties are discussed in detail in further chapters.

Chassis frame is an integral part of the vehicle. Its performance is not only affecting the system but also affects the overall performance of the vehicle.

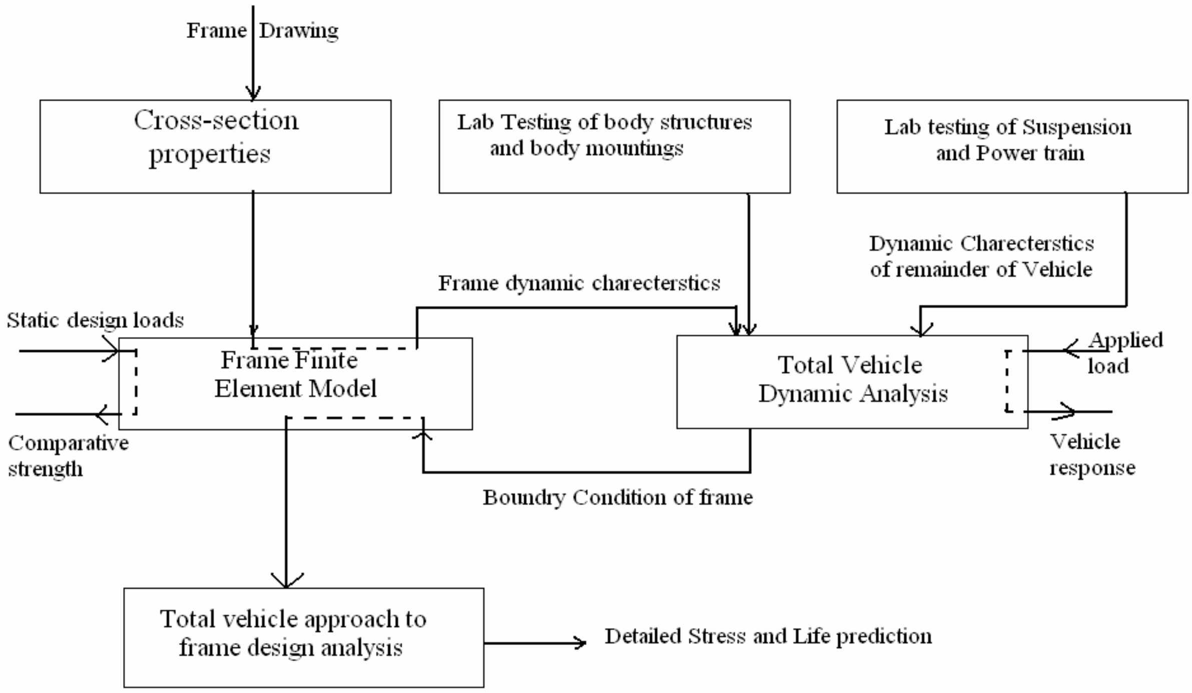


Figure 5.2 Total vehicle approach of frame design analysis [16]

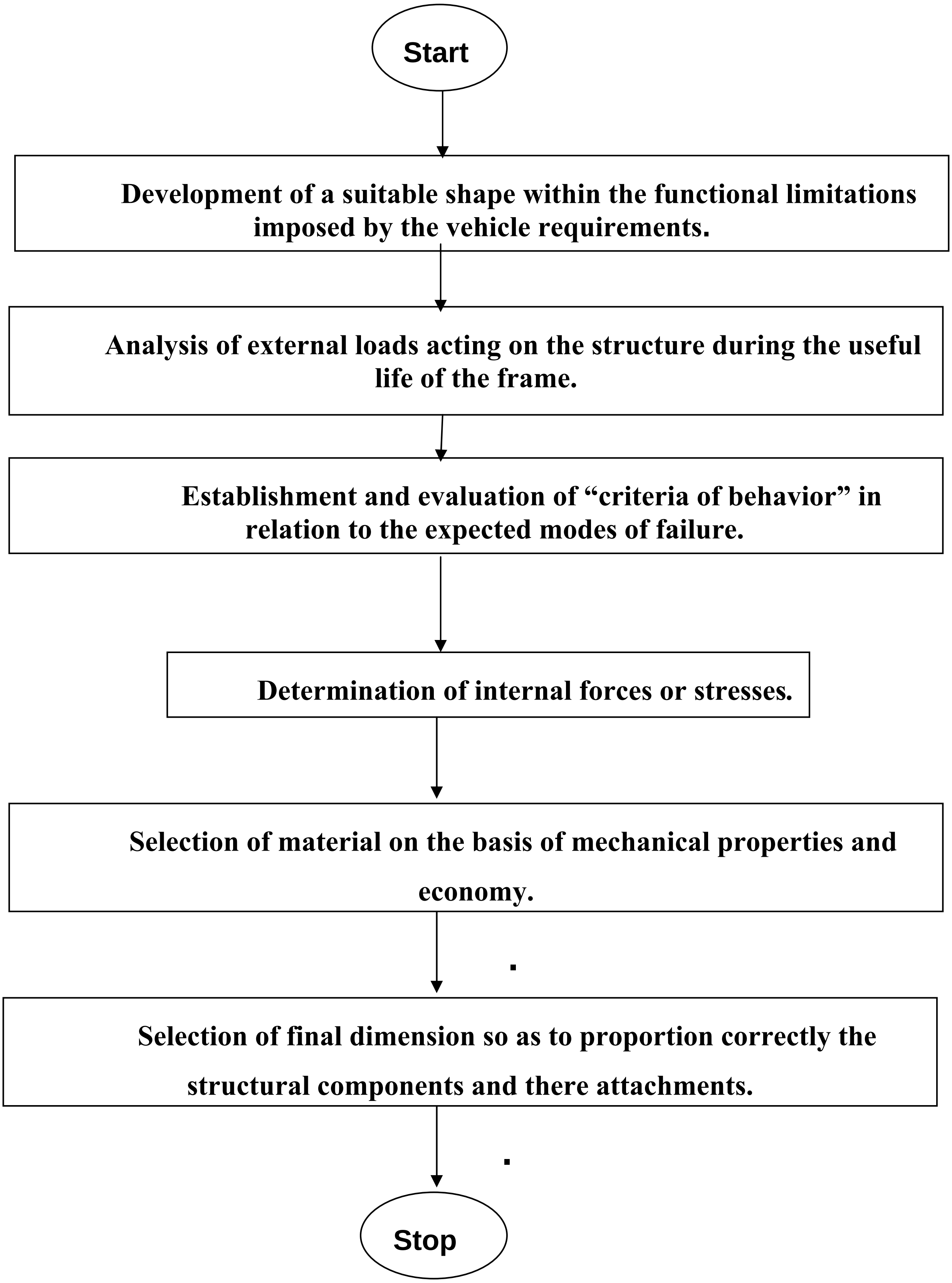


Figure 5.3 Flowchart for frame design [22]

5.3 Finite element analysis

Frame design can be virtually tested using Finite element method. Finite element model has following advantages:

- **Predictive capability of analysis:** with building the physical model we can predict the response of the component in simulated environment and loading.
- **Speed and cost effectiveness:** Most of the time spent initially in construction of Finite element model itself. This involves discretisation and refinement of elements. After this is accomplished, most structural modifications are analyzed with fraction of effort.
- **Storage:** The component does not exist as hardware and all the information is in electronic storage devices.
- **Versatility:** Same model can be used for varied application as deflection, stress, buckling and elasto-plastic analysis.

5.4 Optimization

Light weight and design efficient structures are becoming more important aspect in the vehicle design process. In order to obtain optimum design with reduced weight, cost without sacrificing the performance requirements of a chassis frame, Optimization is done. Generally designer performs iterative cycles to produce an optimum design. By utilising Optimization techniques trial and error in design modifications can be eliminated. The optimal structural design makes a significant success with topological Optimization in the field of linear static and vibration analysis. The structural Optimization problem can be divided into size, shape and topology Optimization .

Size Optimization : Size Optimization deals with Optimization of cross-sectional properties with maintaining structural geometric boundaries and material.

Shape Optimization : Shape Optimization deals with Optimization of structural geometric boundaries.

Topology Optimization : Topology Optimization deals with Optimization of material distribution. Topological changes give the designer guidelines on required structural stiffening. The stiffening can be in the form of beads or any type of metal reinforcements.

This result is a stronger structure that posses the desired performance or characteristics.

Optimization processes are usually performed separately or most of the time sequentially. Generally shape or topology Optimization is carried out first and than followed by size Optimization . Rarely an Optimization methodology is developed to allow for simultaneous solution of more than one type of Optimization problem. One of the important aspects of Optimization is the requirement of very high computational resources and lack of ready tools for modifying shapes etc. Continuous effort is also seen in literature to improve the Optimization methods that require lower number of iterations to reach optimum. However this dissertation focuses only on improving the initial design in order to reduce Optimization effort.

6 METHODS OF ANALYSIS OF CHASSIS FRAME

In chassis frame analysis three static stiffness cases (Vertical bending, Lateral bending and Torsion) were determined using simplified analytical models and FE simulations. Strength analysis (One wheel lift) and Frame vibration analysis (Free-free case) was also done. For crash analysis simulations were done using Inertia relief method. Finally a number of analytical iterations (Weight Optimization) were performed to obtain most effective design parameters.

Methods selected for Performance analysis:

- Lumped Parameter Model (analytical model).
- Finite Element Model (with beam elements).
- Finite Element Model (with shell elements).

6.1 Lumped parameter model (Simplified Analytical model):

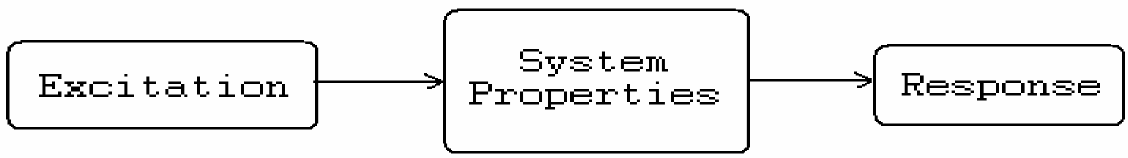


Figure 6.1 Mathematical model of truck frame simulates system properties

This is a standard technique of estimating performance using simplified analytical models that use functional elements of the system. This method utilizes concepts of “Strength of Material”, especially “Beam Theory”. It is having the advantage of being relatively quick and in expensive. But it is having difficulty in getting results of sufficient accuracy for complex structures like Chassis frame. Also internal force or stress distribution for complex beam structures can be obtained by using Computer Program, but is not possible with this method. The results obtained by this method can be used for improving the design concepts but cannot be used for detailed design Optimization . However with sufficient effort this method can be very effectively used to carry out Optimization during the concept design phase where detailed geometry information may not be available for FE simulations.

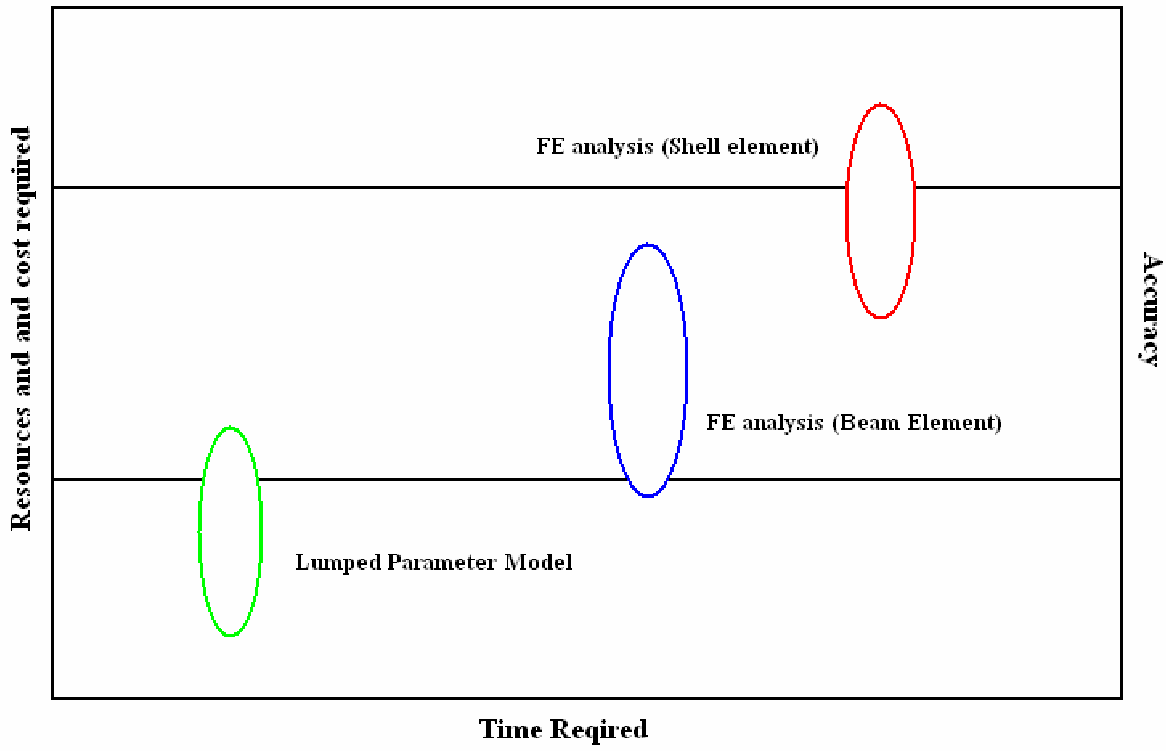


Figure 6.2 Comparison of various methods

In previous sections the loads on the chassis frame was considered. Now considering those loads various performance parameters are analyzed mathematically further sections.

6.2 Finite element model (With beam elements):

Chassis frame typically comprise of beam like members. The beam assumption for frame components helps to reduce the FE model size, computational time and resource utilization. Once the beam model representation of chassis frame is developed than a number of load cases for a specific design and many design variations can be quickly and economically evaluated before the design is committed to prototype buildup. The beam cross-sectional properties are selected based on the component cross-sectional properties.

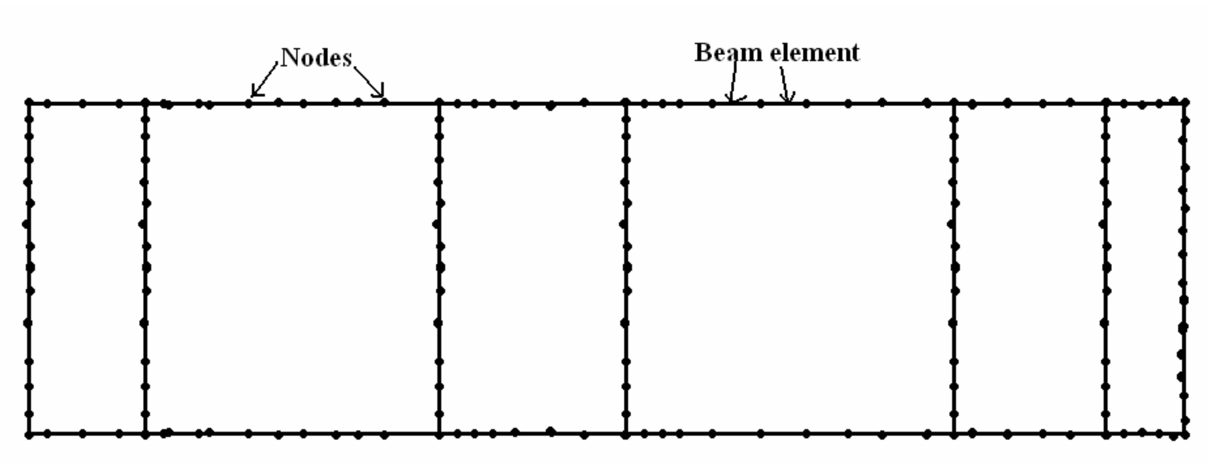


Figure 6.3 FE Beam Element model of Cassis Frame

Beam elements should be used due to following reasons:

- Automotive components are basically frame composed of beams. They comprise of continuously varying cross sections with slots holes and seams.
- Prohibitive amount of work is to be done to model the structure using shell or plate elements. For this, great many data points are to be gathered.
- Every data point in shell or plate element has six degrees of freedom. Thus more no. of data point increases the size of stiffness matrix and finally increases the cost of solution.

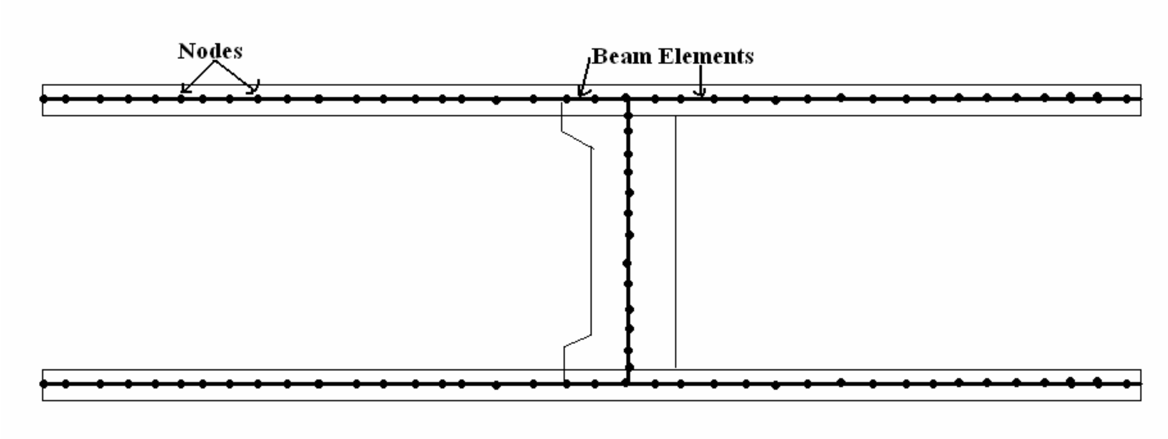


Figure 6.4 Finite element model of H-frame using beam elements [16]

The FE model analysis is practical because the frame is having series of varying cross section beams, which give close resemblance with actual frame.

6.3 Finite element model (With Shell elements):

This is a very detailed and costly approach to Finite Element Analysis. The mesh is of interconnected flat shell elements. The frame is modeled as shell elements with deformable material properties. Linear material properties are generally used. The welds between various long-members and cross-member or long-member and various frame components are generally shown as rigid one dimensional element.

It has following disadvantages also:

- The model formulation and computer running costs can be significantly more expensive and time consuming than prototype testing.
- Different design concepts and major design changes cannot be studied without the expenditure of time and money with the original design study.
- Dynamic analyses using these techniques are probably very expensive.

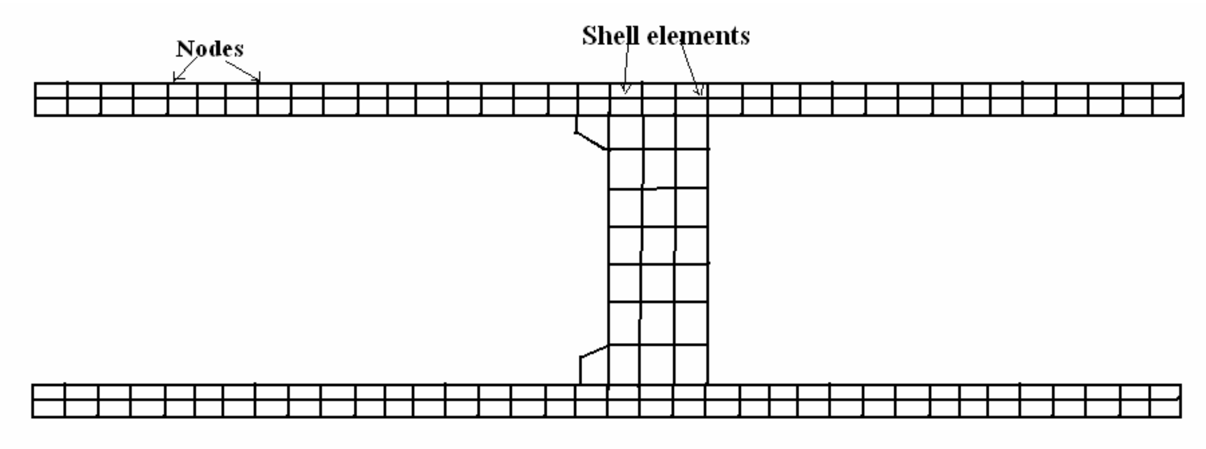


Figure 6.5 FE Shell element representation of H-frame

The non structural masses and the payload are often modeled as mass elements at the measured centre of gravity locations. Nevertheless the analysis is time consuming but it gives extremely accurate deflection and stress results.

6.4 Ladder type chassis frame model:

The ladder frame selected for the analysis consists of two long-members and seven cross-members. The long-member is channel section having three transitions in area for entire length.

The cross-member is box section and is selected for different dimensions of cross sectional areas. Again the length of placement of each cross-member and width of frame is also variable.

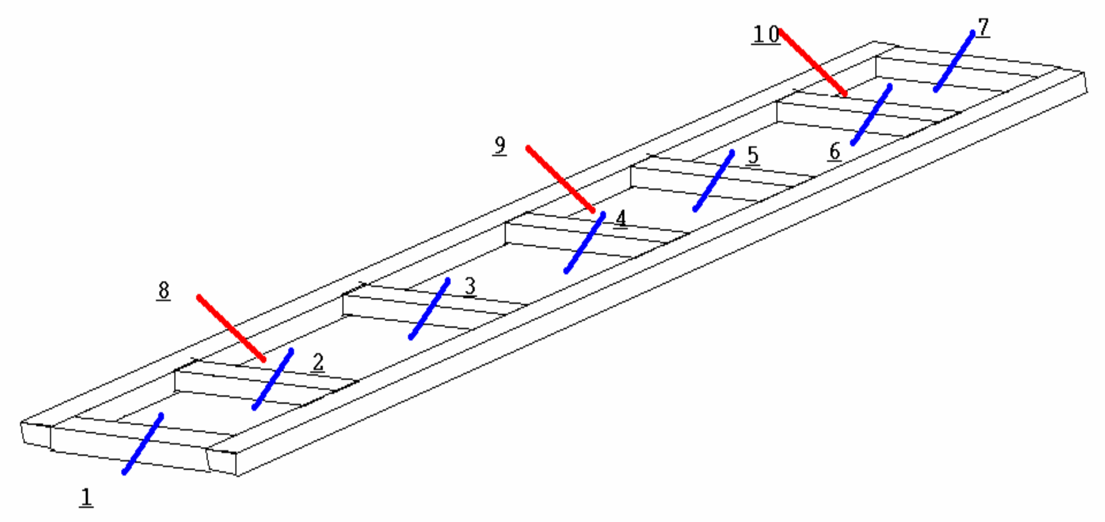


Figure 6.6 Model of ladder frame

- Dimensions per section : 3
- Cross-member dimensions : 3 X 7 = 21
- Long-member dimensions : 3 X 3 = 9
- Position of cross-members (excluding first one) : 6
- Length of constant area in long-member : 3
- Width : 1
- Total no. of variables (21+9+6+3+1) : 40

6.5 Modelling:

In last fifteen years Computer Aided Designing (CAD) systems have replaced drawing boards as the method for design. They enable the designers to quickly create realistic vehicle components, their assemblies, and design drawings for manufacturing. These systems are having features like parametric solid modelling and assembly management. CAD systems provide data to be used as in put for CAE analysis.

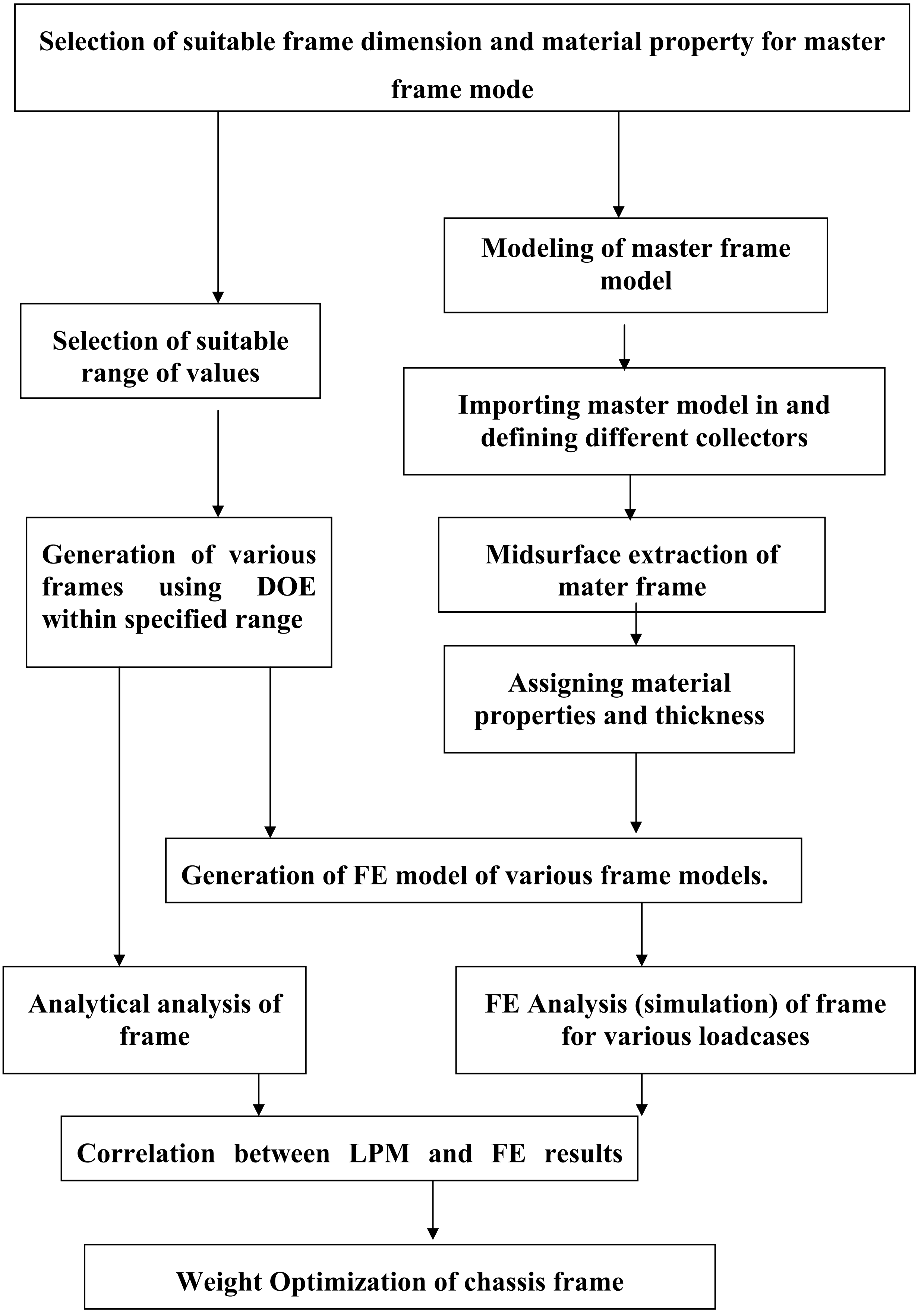


Figure 6.7 Flowchart showing the Work plan for Analysis

6.6 Meshing:

Typical FE analysis is composed of determination of displacement, natural frequencies and stresses of the component. For analysis the CAD model needs to be converted into discretised representation of structure. For this purpose the CAD model is imported in Meshing software and mesh is created using one, two or three dimensional elements, interconnected to each other at nodes. Element carry the mass, stiffness and damping properties of the structure. Before analysis boundary conditions are defined by the user.

In this analysis CAD model Pro/ENGINEER is imported in HyperMesh using neutral file (*.iges or *.step) and its mid-surface is extracted for creation of shell elements. Mesh using shell and rigid elements is created that reflects the current CAD model configuration. The element properties (thickness, material etc.) are defined in various collectors and boundary conditions are defined.

Quality Checks

- Aspect Ratio : less than 5:1
- Warpage : 15° (max.)
- Skewness : 45° (max.)
- Quad angle : 45° to 135°
- Tri angle : 20° to 120°
- Jacobian : 0.6 (max.)
- No. of trias : 5% (maximum)

6.7 Analysis:

For analysis, suitable solver (depending upon the type of analysis) is selected and FE model is imported using appropriate neutral file. During solution, solver creates equations relating to mass, stiffness and damping properties of structure. The correctness of solution obtained by the solver depends upon the judgement and accuracy of work carried out in the previous stages. Results of analysis can be generated using images of displacement or stress-strain contours and animations.

Material Properties: Material (Steel)

- Modulus of elasticity (E) : 2.1×10^5 N/mm^2
- Modulus of rigidity (G) : 8.1×10^4 N/mm^2
- Poisson's ratio : 0.3
- Density : 7.9×10^{-6} Kg/mm^3

One of FE frame model consists of following information in it's ***.out** file:

FINITE ELEMENT MODEL DATA INFORMATION:

Total # of Nodes	:	6584
Total # of Elements	:	6377
Total # of Rigid Elements	:	6
Total # of Rigid Element Constraints	:	552
Total # of Degrees of Freedom	:	39504
Total # of Non-zero Stiffness Terms	:	1062708

Element Type Information

CQUAD4 Elements	:	6363
CTRIA3 Elements	:	14

Load and Boundary Information

FORCE Sets	:	4
SPC Sets	:	4

Material and Property Information

PSHELL Cards	:	12
MAT1 Cards	:	1

7 DESIGN OF EXPERIMENTS (DOE)

Experimental design is a research study in which the researcher has control over the selection of participants in the study, and these participants are randomly assigned to treatment and control groups. *DOE is a methodology that defines an optimal set of experiments in the design space in order to obtain the most information as possible with the highest accuracy at the least cost.* An orderly procedure that results in the most information with a minimum of changes (variables) constitutes a designed experiment. A properly designed experiment allows relatively simple statistical interpretation of the results, which may not be possible otherwise.

Tools: Optimus, MATLAB toolbox, Excel plug-in, SAS, S+, SPSS

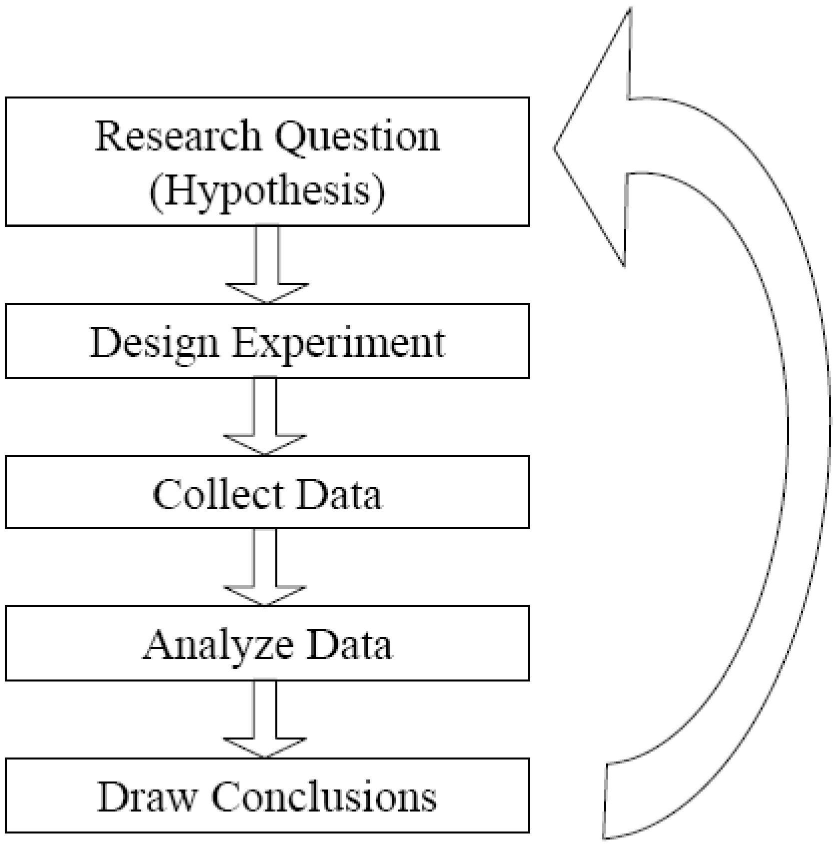


Figure 7.1 Experiment Process flowchart [32]

Design of experiment (DOE) is not a simple one step process but is actually a series of steps which must follow a certain sequence for the experiment to yield an improved understanding of product or process performance.

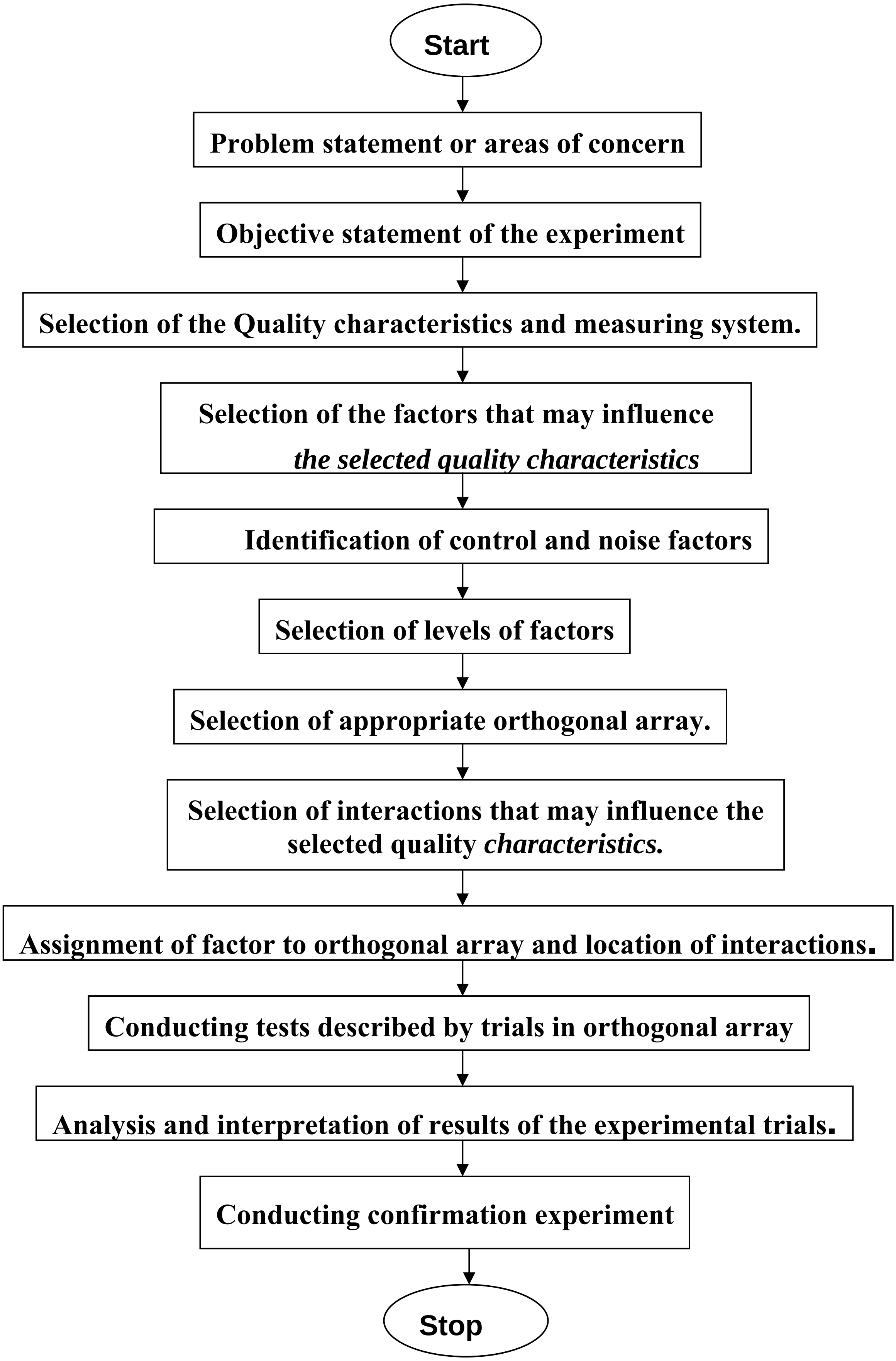


Figure 7.2 Major steps of Design of Experiments [33]

The steps in DOE process are generically the same regardless of experiment design. Positive experimental results are dependent of planning of experiment not the analysis.

Experimental design process is a theory concerning the minimum number of experiments necessary to develop an empirical model of a research question and a methodology for setting up the necessary experiments.

List of DOE Methods reported in literature:

1. Full Factorial design
2. Fractional Factorial design
3. Random and Latin Hypercube design
4. Placket-Burman design
5. Box-Behnken design
6. Taguchi design
7. Nested design
8. Split Plots
9. John’s ¾ design
10. Simplex centroid design
11. Simplex lattice design
12. D-Optimal design
13. G-Optimal design
14. A-Optimal design

Important methods are explained in the following sections:

7.1 Full factorial design

A **Full factorial experiment** is an experiment whose design consists of two or more factors, each with discrete possible values or "levels", and whose experimental units take on all possible combinations of these levels across all such factors. Such an experiment allows studying the effect of each factor on the response variable, as well as the effects of interactions between factors on the response variable.

When there are many factors, many experimental runs will be necessary, even without replication. For example, experimenting with 10 factors at two levels each produces 2^{10}=1024 combinations. At some point this becomes infeasible due to high cost or insufficient resources. In this case, fractional factorial designs may be used.

No. of Experiments for current design: $\mathbf{3^{40}}$

7.2 Fraction Factorial Method

Fractional factorial designs are experimental designs consisting of a carefully chosen subset (fraction) of the experimental runs of a full factorial design. The subset or fraction is chosen so as to exploit the sparsity-of-effects principle to access information about the most important features of the problem studied, while using considerably fewer resources than a full factorial design.

Factorial experiments can be used when there are more than two levels of each factor. As with any statistical experiment, the experimental runs in a factorial experiment should be randomized to reduce the impact that bias could have on the experimental results.

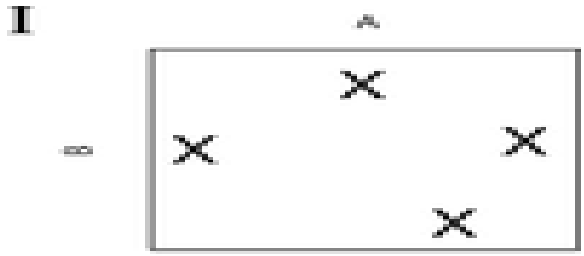
In practice, this can be a large operational challenge However, the number of experimental runs required for three-level (or more) factorial designs will be considerably greater than for their two-level counterparts. Factorial designs are therefore less attractive if a researcher wishes to consider more than levels.

No. of Experiments for current design: $\mathbf{3^{(40-5)}}$

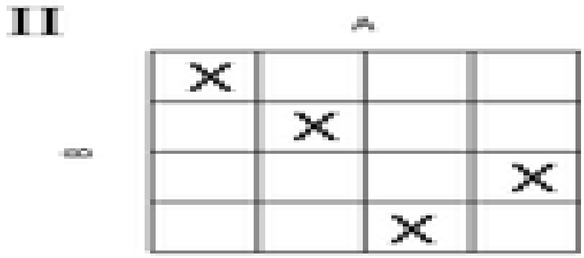
7.3 Latin Hypercube

In the context of statistical sampling, a square grid containing sample positions is a Latin square if (and only if) there is only one sample in each row and each column.

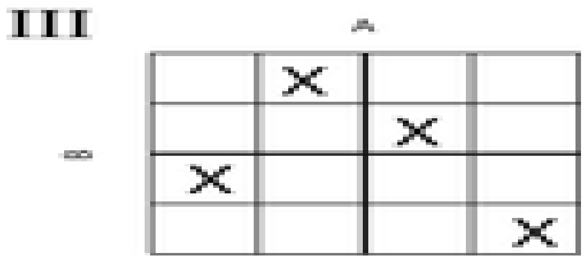
Random Sampling



Latin Hypercube sampling



Orthogonal Sampling



A **Latin hypercube** is the generalization of this concept to an arbitrary number of dimensions, whereby each sample is the only one in each axis-aligned hyperplane containing it. Any no. of experiments can be selected in this method.

No. of Experiments for current design: **User defined**

7.4 Central composite design

Central composite design is an experimental design, useful in response surface methodology, for building a second order (quadratic) model for the response variable without needing to use a complete three level factorial.

No. of Experiments for current design: **Cannot be applied for discrete levels**

7.5 Box-Behnken designs

Box-Behnken designs are used to generate higher order response surfaces using fewer required runs than a normal factorial. This method and the central composite plan essentially suppress selected runs in an attempt to maintain the higher order surface definition.

No. of Experiments for current design: **Cannot be applied for discrete levels**

7.6 Plackett-Burman designs

Plackett-Burman designs are saturated fractional factorial designs. This class includes designs with N (the number of runs) being a multiple of 4. These are orthogonal designs and may be used for studies involving up to (N-1) two-level factors. The designs for which "N runs" are not integral powers of 2, are sometimes referred as non-geometric Plackett-Burman designs.

No. of Experiments for current design: **For 2 levels**

Plackett-Burman Design in 12 Runs for up to 11 Factors (Hadamard matrices) is shown below.

	Pattern	*X*1	*X*2	*X*3	*X*4	*X*5	*X*6	*X*7	*X*8	*X*9	*X*10	*X*11
1	+++++++++++	+1	+1	+1	+1	+1	+1	+1	+1	+1	+1	+1
2	-+-+++---+-	-1	+1	-1	+1	+1	+1	-1	-1	-1	+1	-1
3	--+-+++---+	-1	-1	+1	-1	+1	+1	+1	-1	-1	-1	+1
4	+--+-+++---	+1	-1	-1	+1	-1	+1	+1	+1	-1	-1	-1
5	-+--+-+++--	-1	+1	-1	-1	+1	-1	+1	+1	+1	-1	-1
6	--+--+-+++-	-1	-1	+1	-1	-1	+1	-1	+1	+1	+1	-1
7	---+--+-+++	-1	-1	-1	+1	-1	-1	+1	-1	+1	+1	+1
8	+---+--+-++	+1	-1	-1	-1	+1	-1	-1	+1	-1	+1	+1
9	++---+--+-+	+1	+1	-1	-1	-1	+1	-1	-1	+1	-1	+1
10	+++---+--+-	+1	+1	+1	-1	-1	-1	+1	-1	-1	+1	-1
11	-+++---+--+	-1	+1	+1	+1	-1	-1	-1	+1	-1	-1	+1
12	+-+++---+--	+1	-1	+1	+1	+1	-1	-1	-1	+1	-1	-1

Figure 7.3 Hadamard matrices

DOE selected: In this dissertation work Latin Hypercube was selected for creating twelve Design of experiments for 40 variables. The design of experiment was performed to get the values of various parameters of twelve ladder type chassis frame. This work was carried out in Altair Hyperstudy software. The table in appendix A shows the list of values of various parameters.

8 ANALYSIS OF CHASSIS FRAME:

Analysis was done for selected performance criteria using both FE analysis and Lumped Parameter Model.

8.1 Vertical bending stiffness:

It is the force required for the unit deflection of the component. Basically long-member is responsible for the vertical bending stiffness. Since long-member in our frame is composed of three variations of cross section, stiffness is summation of individual stiffness values in series.

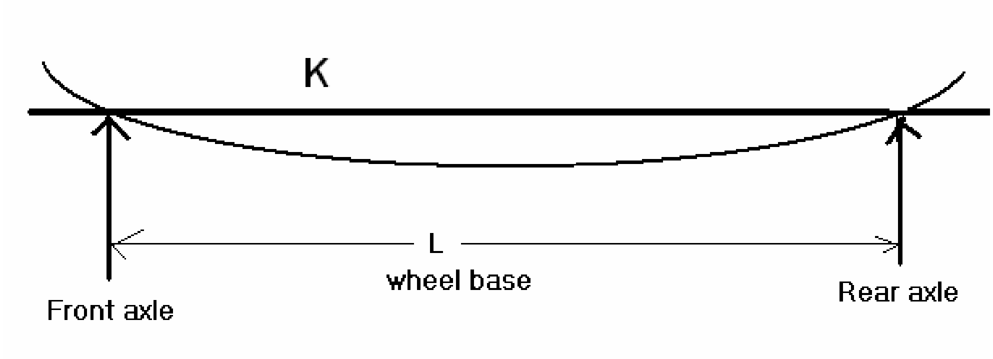


Figure 8.1 Vertical bending stiffness

$$K = \frac{48 E}{\left(\frac{l_1^3}{I_1} + \frac{l_2^3}{I_2} + \frac{l_3^3}{I_3} \right)}$$

Shear centre

In standard simple "beam theory" it is often assumed that the centroidal axis and elastic axis (Shear centre or Torsion centre axis) are coincident. But in case of beams with open cross-sections there is difference in location to the axis. This difference is called beam eccentricity and has the effect of coupling in all bending and torsional deflections occurring in the beam.

If the vertical load is not applied at Shear centre, this will result in bending and torsion of the section. Thus it is advisable to apply vertical loads at shear centre to avoid twisting of beam. In the simulations, shear node was connected with long-member by rigid elements and load was applied at shear centre.

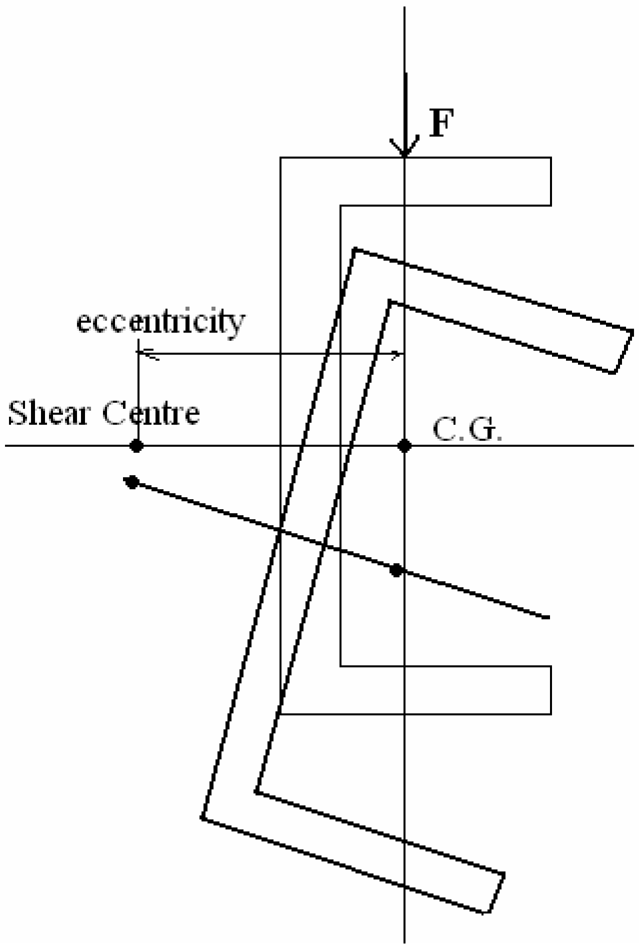


Figure 8.2 Illustration of coupled bending and torsion

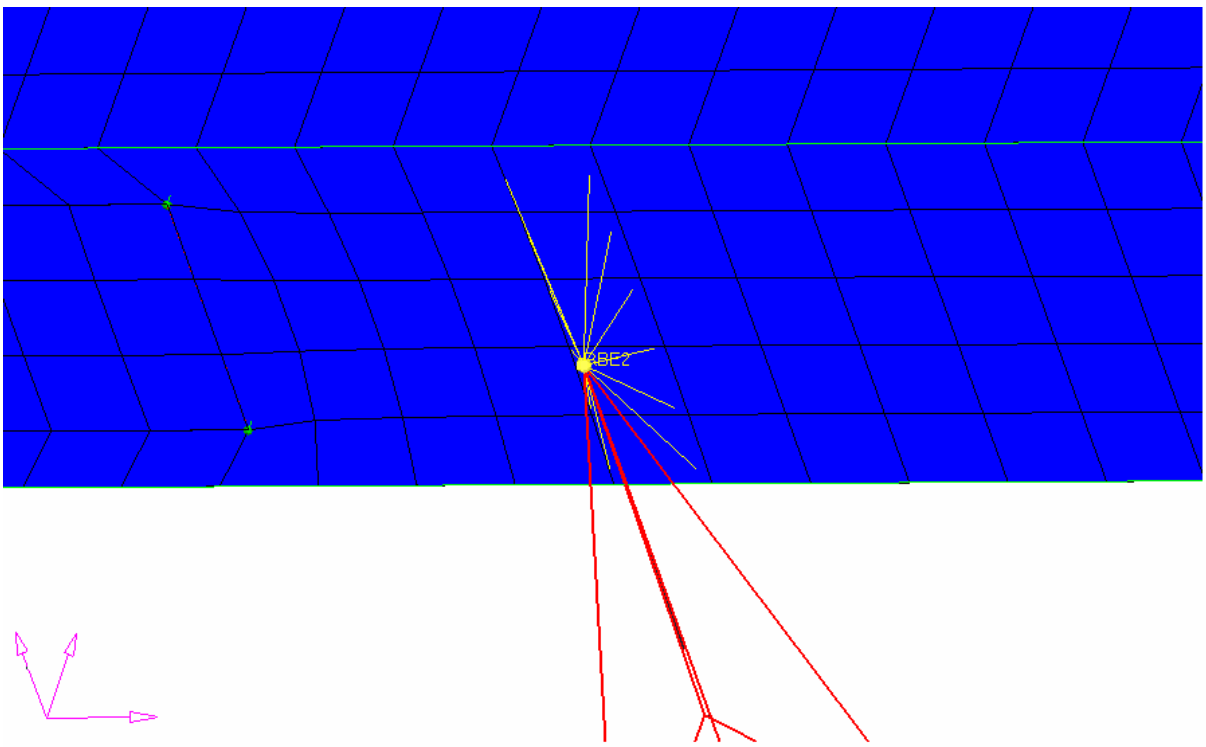


Figure 8.3 Application of force at the shear centre of the Long-member of the chassis frame

8.1.1 Conjugate beam theory

Conjugate beam method (or Funicular analogy or method of elastic weights is a special case of the moment of area method. This method is applied for beams having non-uniform flexural rigidity. This method is directly used only for Simply Supported Beams.

Conjugate beam theorem: the theorem states that,

"*The slope at any section of a loaded beam, relative to original axis of the beam is equal to the shear in the conjugate beam at the corresponding section.*"

"*The deflection at any section of a loaded beam, relative to original position is equal to the bending moment at the corresponding section of the conjugate beam.*"

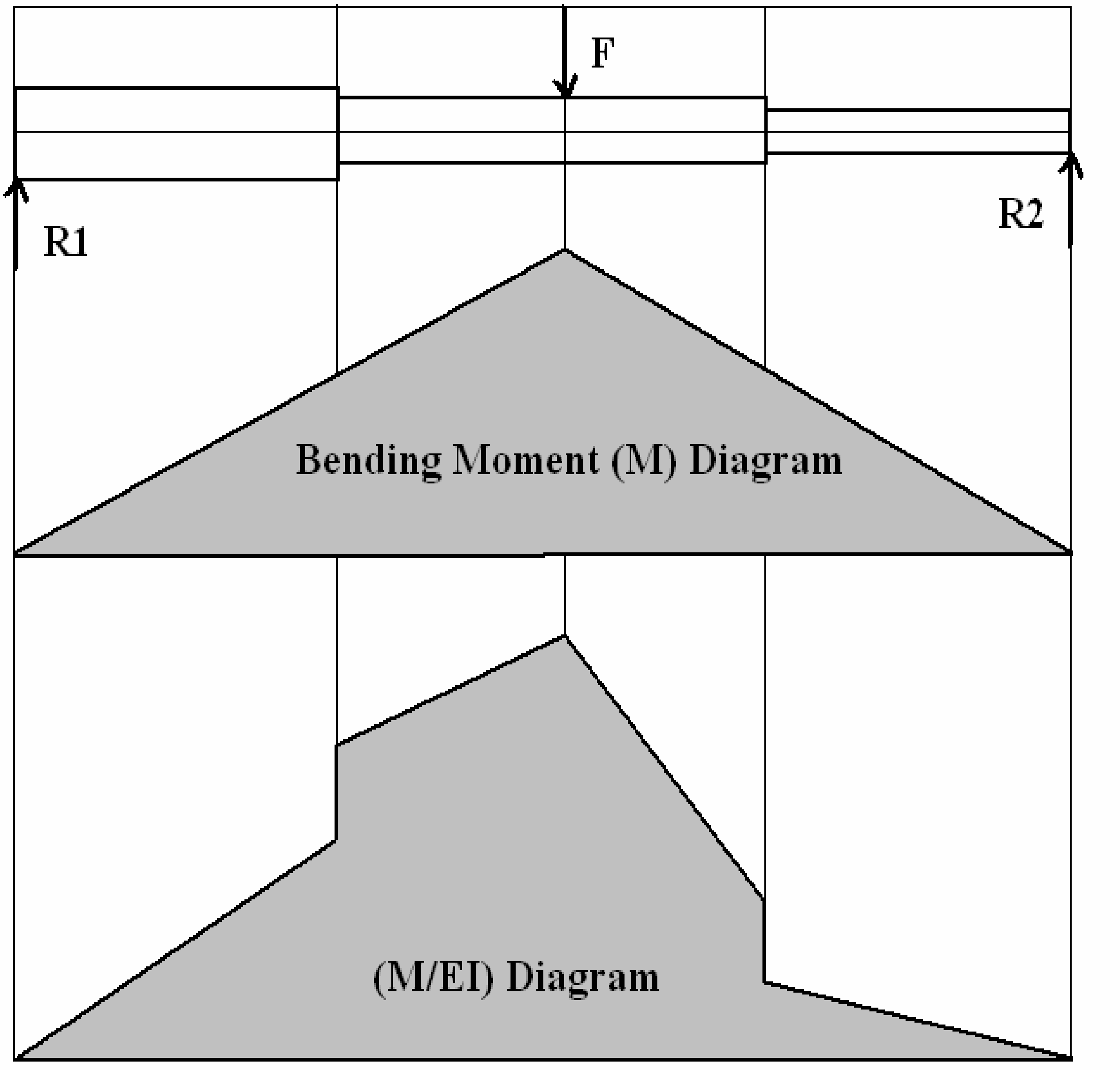


Figure 8.4 Bending moment diagram of beam and loading of conjugate beam

We know that, Shear $S_x = \int_0^x \frac{M}{EI} dx$

Thus Bending Moment of Conjugate beam

$$M_x = \int_0^x S_x dx = \int_0^x \int_0^x \frac{M}{EI} = \int_0^x \int_0^x \frac{d^2 y}{dx^2} = \int_0^x \frac{dy}{dx} = y(deflection)$$

8.1.2 Boundary conditions

The constraints defined for the FE model are at the four wheelbase locations. To avoid incorrect results due to over constraining, minimum no. of constraints were applied to the model. Forces of 100N each were applied to the frame at two centres of the chassis frame long-members. All loads were applied on the shearcentres of the sections.

8.1.3 Discussions on results

The LPM analysis was done using conjugate beam method and spring method. Based on the results for vertical bending stiffness, we can summarize the correlation in following points:

- The variation of cross-section of long-member at the junction is not accounted in the LPM
- The LPM only considers the Long-member of the chassis frame and is irrespective of selection of type of cross-member selected.
- Spring method does not give good correlation but its pattern is similar to the FE simulation results.
- Conjugate beam method gives perfect correlation with the FE results when the LPM results are divided by 10.
- The correlation obtained for vertical bending stiffness varies from 85% to 95% in the analysis.

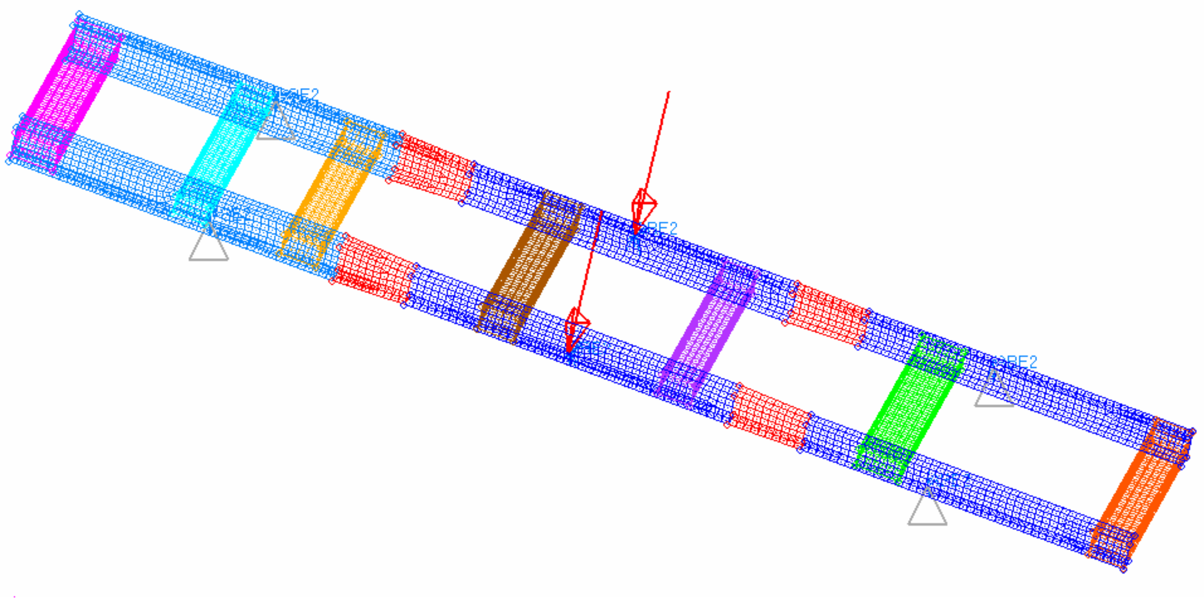


Figure 8.5 Boundary conditions applied to chassis frame for vertical bending analysis

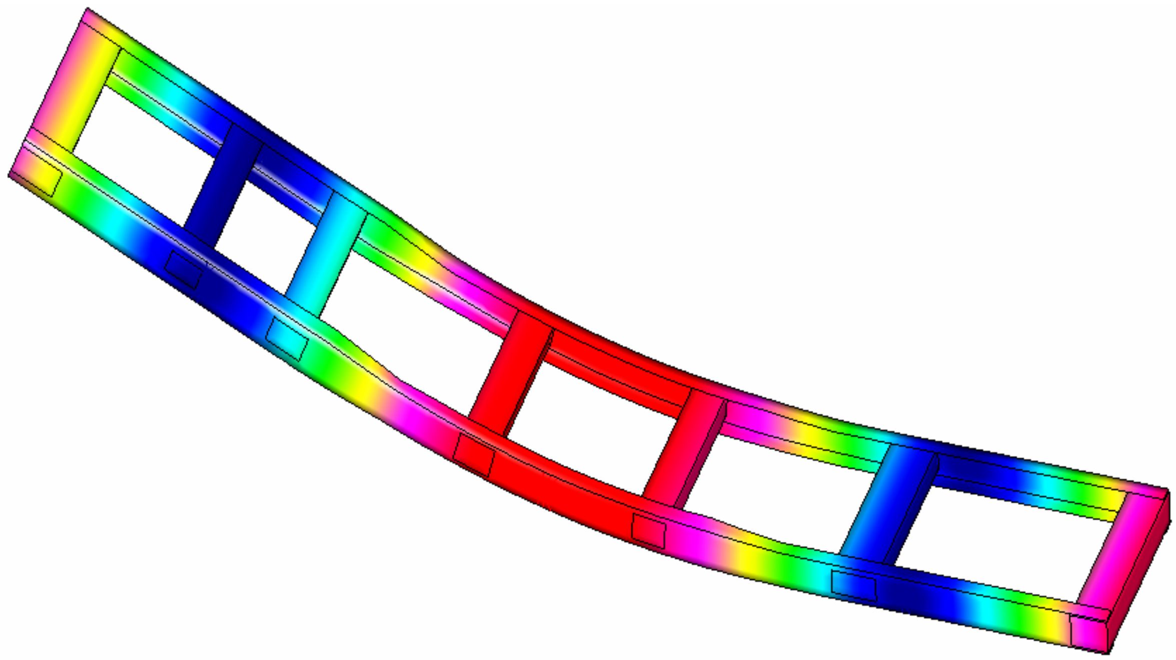


Figure 8.6 Contoured plot of deflection of chassis frame in vertical bending analysis

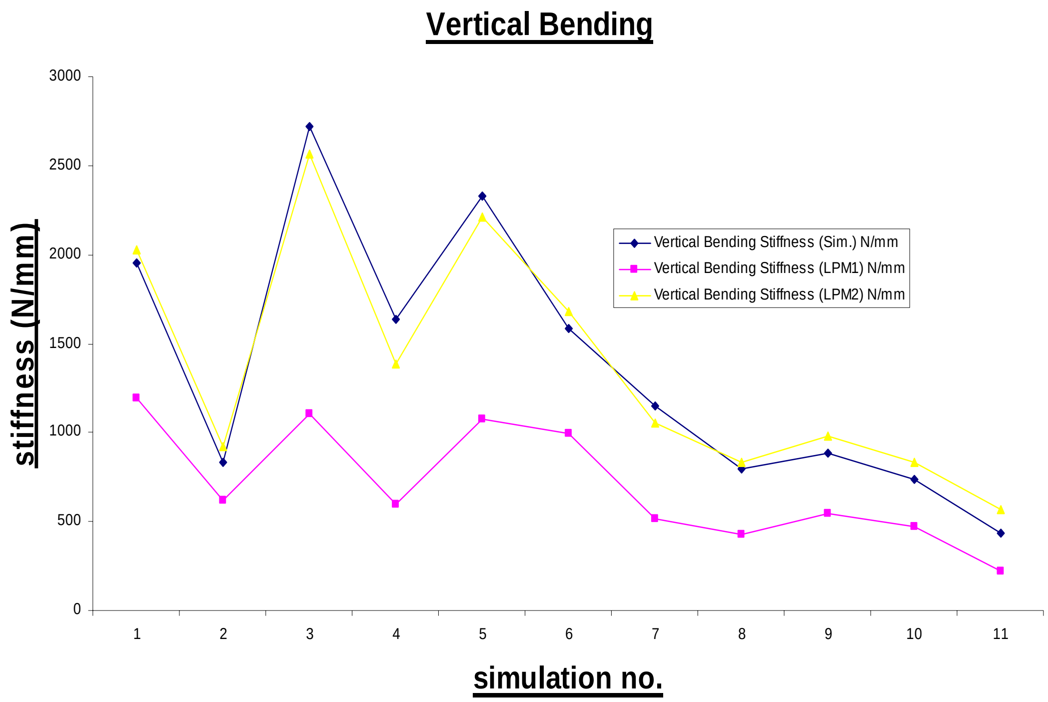


Figure 8.7 Stiffness plot of various chassis frame models in vertical bending analysis

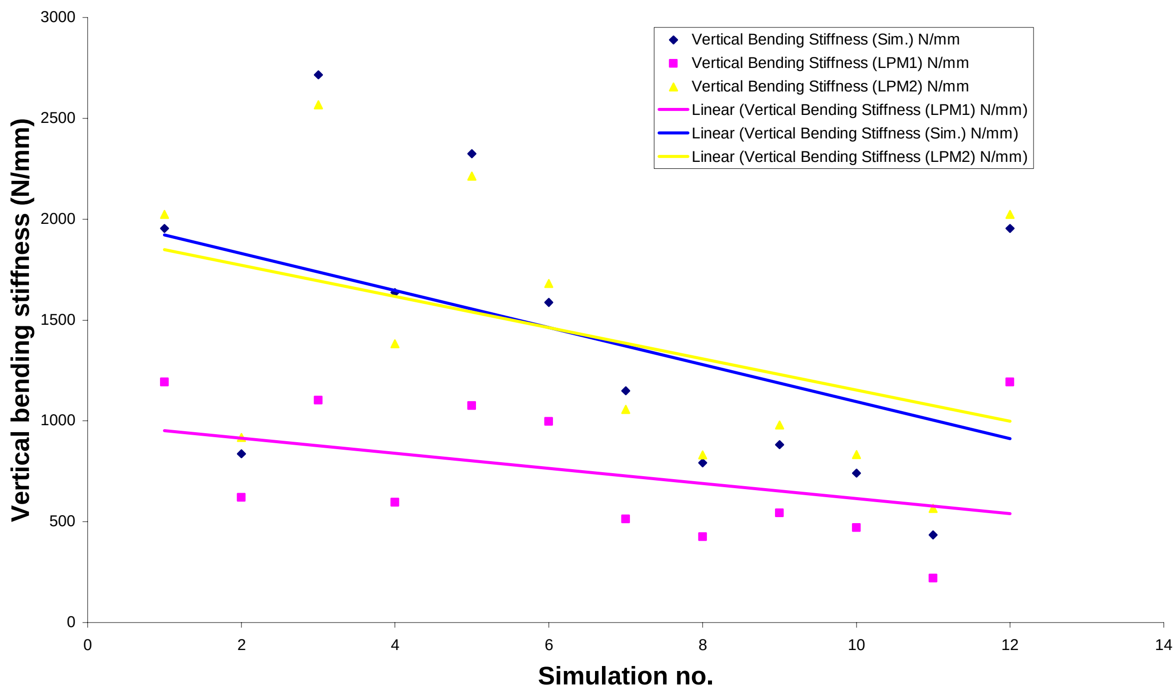


Figure 8.8 Scatter plot of various chassis frame models in vertical bending analysis

8.2 Torsional bending stiffness [7]:

It is the torque require by a component for unit angular twist. This formulation provides a guide to the optimum member selection. Although this cannot provide accurate results, but selection of proper cross-members and long-members can be done using these formulations. In this, two deflection modes are applied separately and overall stiffness is derived using proper summation.

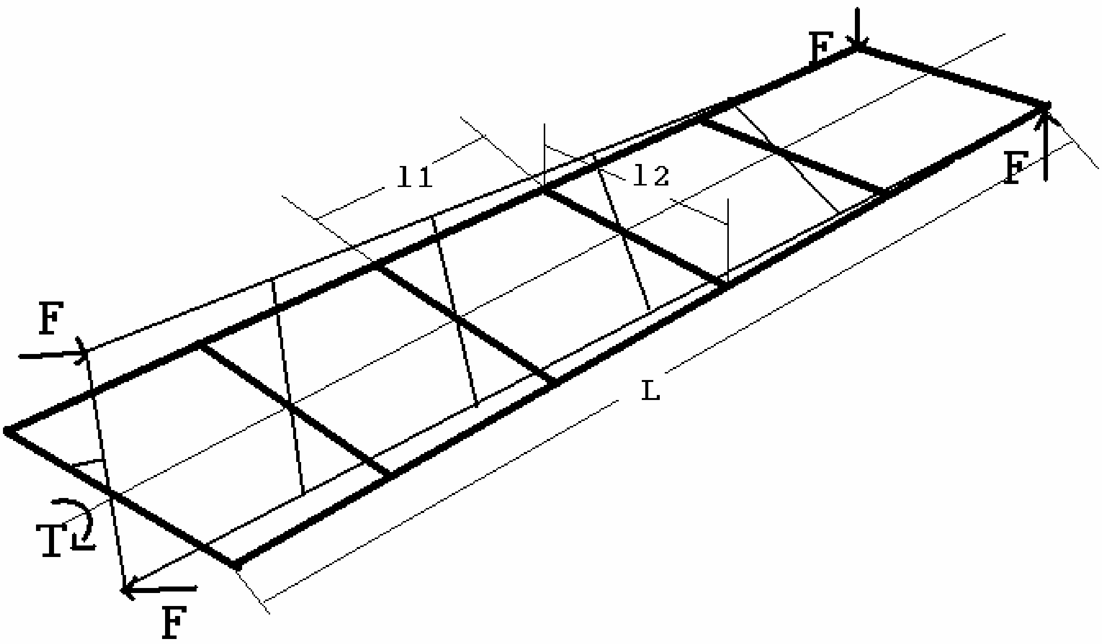
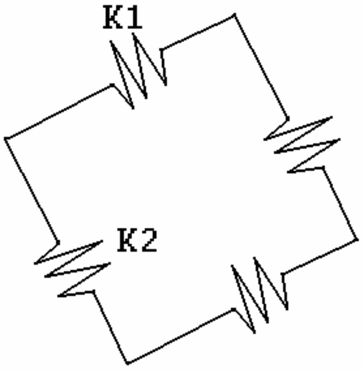


Figure 8.9 Torsion in chassis frame

Simplified representation of chassis frame torsion as a summation of individual member deflection is shown.



$$k_1 = \frac{1}{K_1} + \frac{1}{K_2}$$

$$\frac{1}{K} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} + \frac{1}{k_3} + \frac{1}{k_4} + \frac{1}{k_5} + \frac{1}{k_6}$$

Figure 8.10 Stiffness in parallel and in series

For pure twisting of members:

From figure below we get,

$$\frac{T_t}{\alpha_t} = \frac{G}{5l_1}\left(J_1 + J_2\right) \qquad \text{since, } \begin{matrix} l_1\theta_1 = l_2\theta_2 \\ \alpha_1 = 10\theta_1 \end{matrix}$$

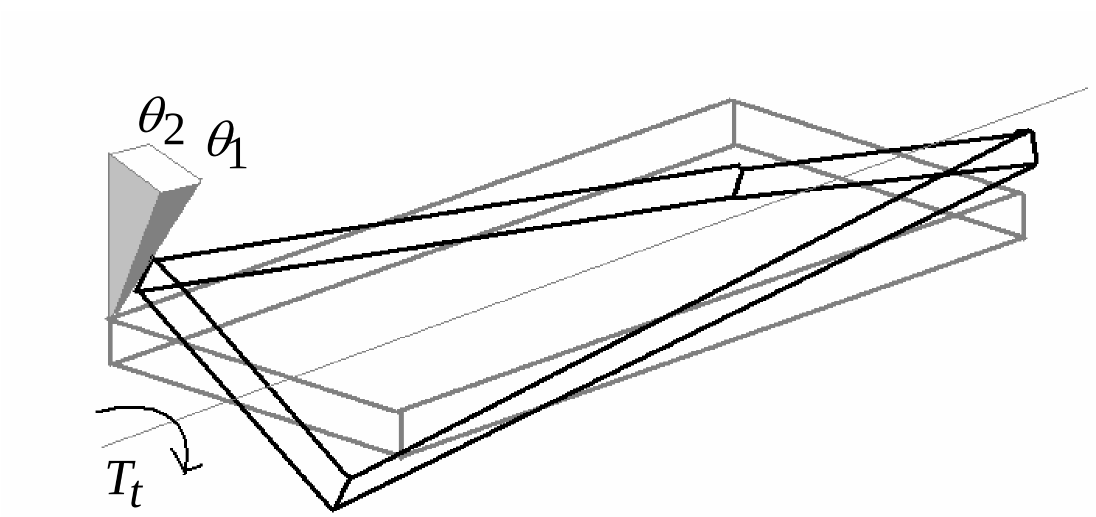


Figure 8.11 Pure twisting of member

For cantilevered bending and twisting of members:

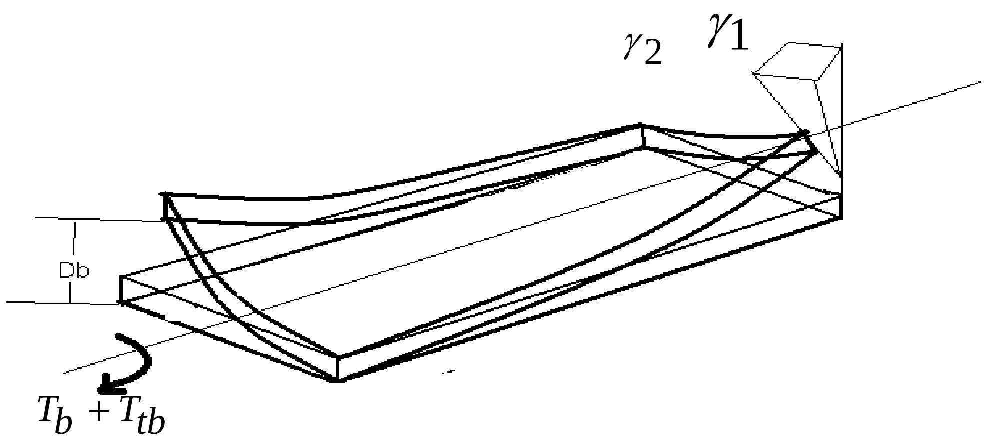


Figure 8.12 cantilevered bending and twisting of members

$$\frac{T_b}{\alpha_b} = \frac{3l_2^2 E}{10}\left(\frac{I_1}{l_1^3} + \frac{I_2}{l_2^3}\right) \quad \text{and} \quad \frac{T_{tb}}{\alpha_b} = \frac{3G}{20l_1}\left(J_1 + J_2\right)$$

For complete frame,

$$\frac{T}{\alpha} = \frac{T_t}{\alpha_t} + \frac{T_{tb}}{\alpha_b} + \frac{T_b}{\alpha_b} = \frac{7G}{20\,l_1}(J_1 + J_2) + \frac{3l_2^2 E}{10}\left(\frac{I_1}{l_1^3} + \frac{I_2}{l_2^3}\right)$$

This formula can be used to quickly asses the specific frame stiffness for various member sections and cross-member spacing, related to chassis length and width.

8.2.1 Boundary conditions

The constraints defined for the FE model of the chassis frame, for Torsional Stiffness calculation are at the two wheelbase locations in rear side. To avoid incorrect results due to over constraining, minimum no. of constraints were applied to the model. Forces of 100N each were applied to the frame at two wheelbase locations in front side of the chassis frame long-members in reverse directions. All loads were applied on the shearcentres of the sections.

8.2.2 Discussions on results

The LPM analysis was done using above discussed theory. Based on the results for Torsional stiffness, we can summarize the shortcoming in correlation in following points:

- The variation of cross-section of long-member at the junction is not accounted in the LPM
- The LPM only considers only the four loops of Long-member and cross-member of the chassis frame and is irrespective of wheelbase locations.
- This method gives considerable correlation and its pattern is also similar to the FE simulation results.
- Localized deformations at long-member and cross-member joints were not considered in the analysis.
- . The correlation obtained for Torsional stiffness varies from 72% to 97% in the analysis.

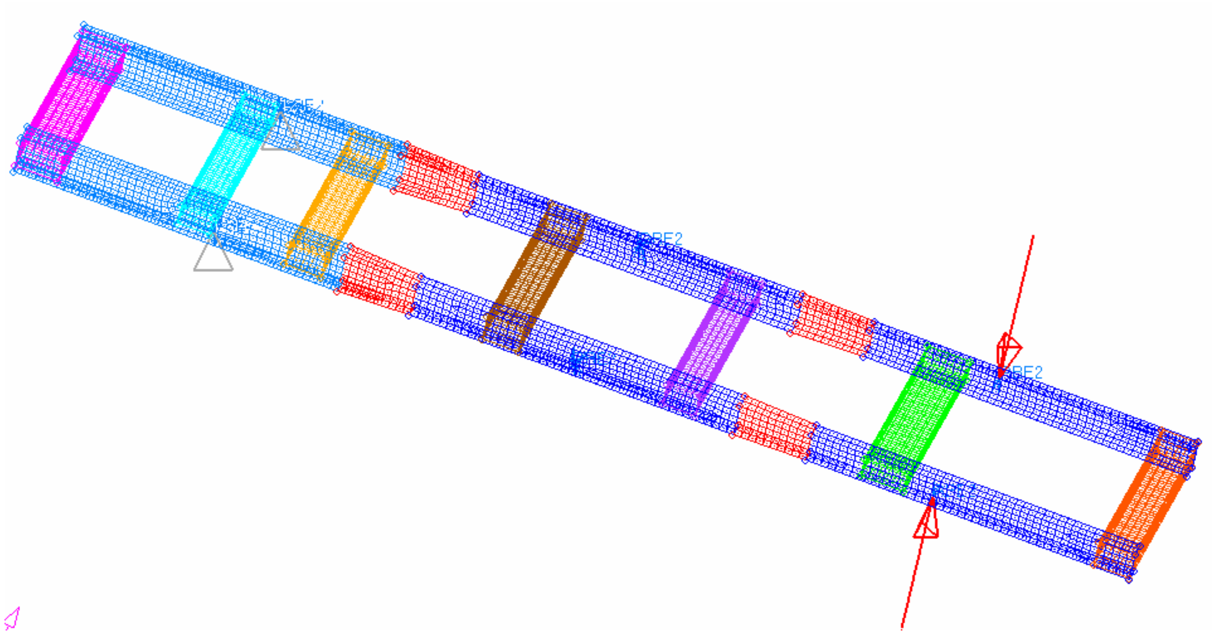


Figure 8.13 Boundary conditions applied to chassis frame for Torsional analysis

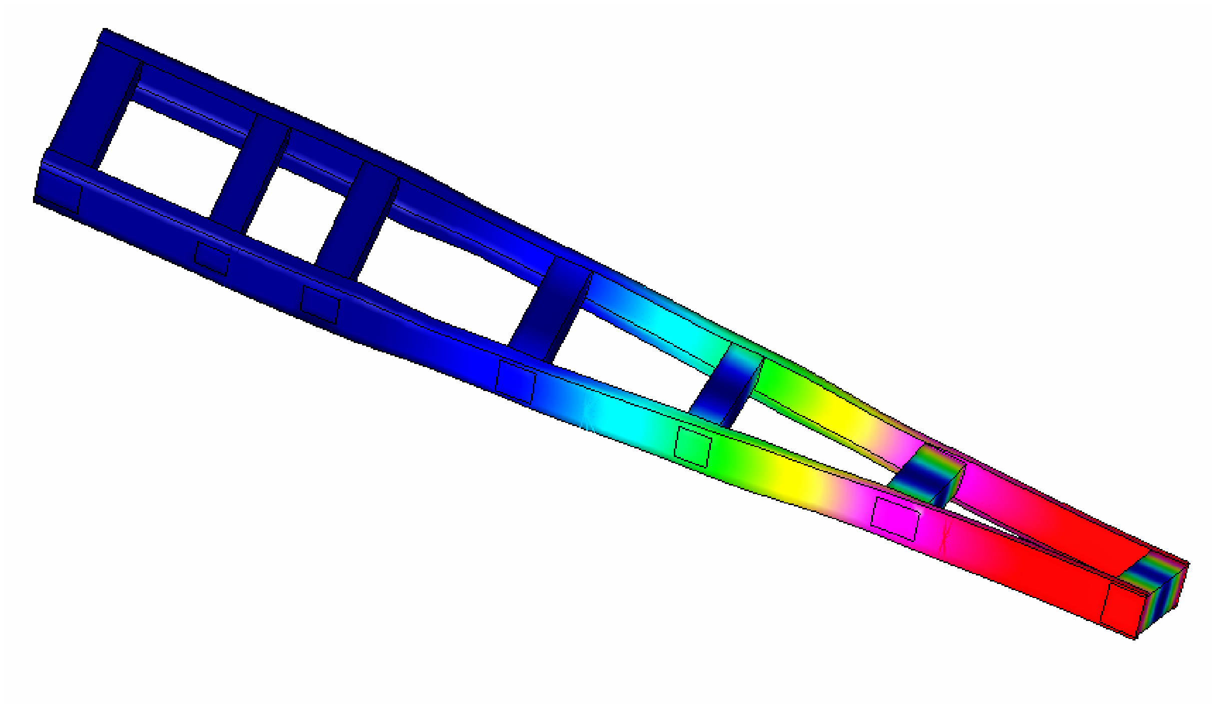


Figure 8.14 Contoured plot of deflection of chassis frame in Torsional analysis

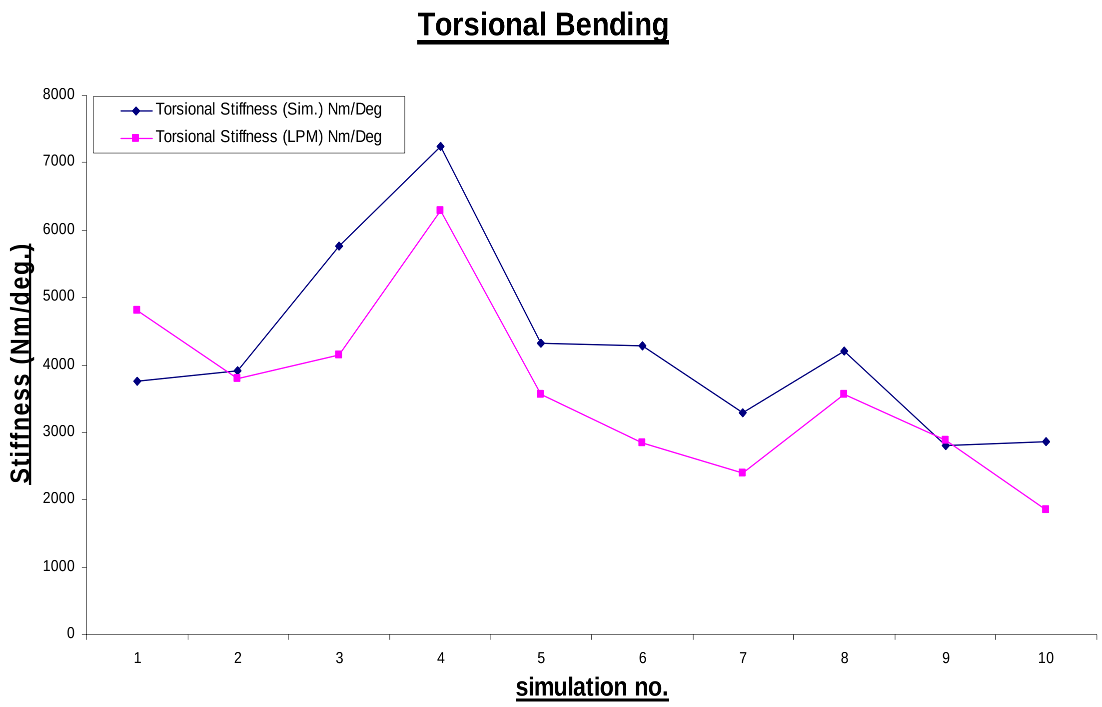


Figure 8.15 Stiffness plot of various chassis frame models in Torsional analysis

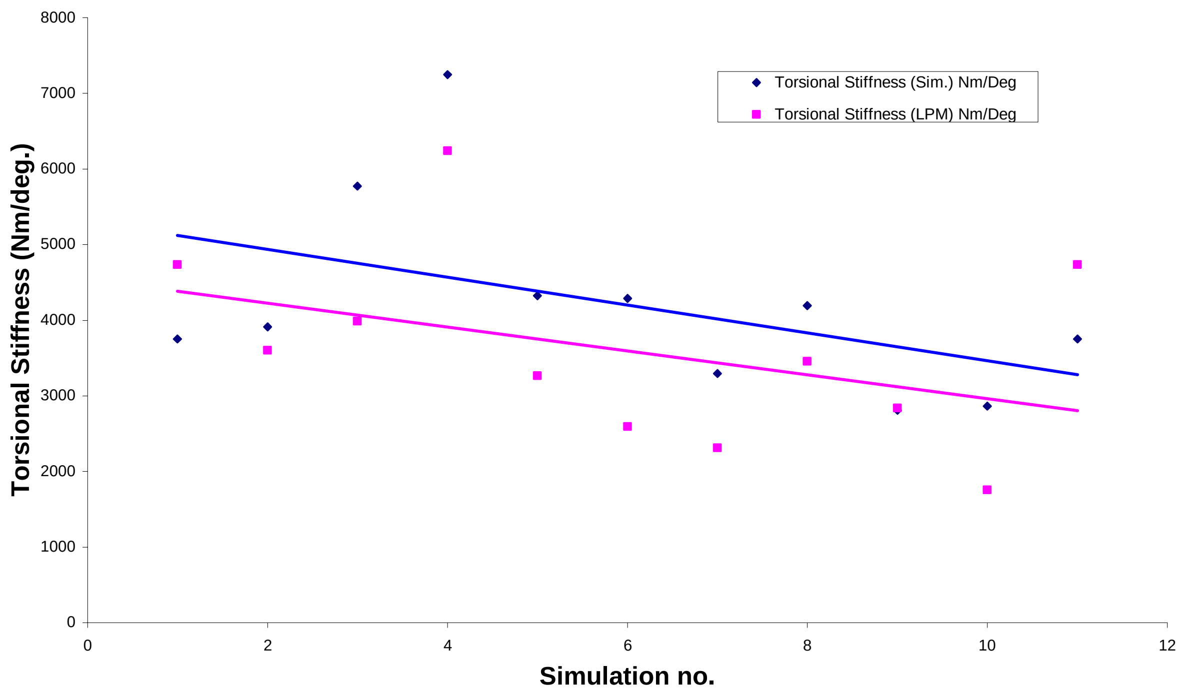


Figure 8.16 Scatter plot of various chassis frame models in Torsional analysis

8.3 Lateral Bending Stiffness:

The analysis of ladder type chassis frame for lateral bending stiffness was done for two different cases. The major difference in these cases is the location of application of resultant load based upon there design and packaging.

8.3.1 Lateral Bending Stiffness for Heavy Vehicles

The forces that lead to lateral bending in (heavy vehicles) are small compared to the overall capacity of the vehicle. Some of these are the unequal side to side driving or braking loads transmitted to the frame, or any end forces due to docking. The tandem rear axle almost is stationary with respect to front axle providing a cantilever action to the chassis frame and whole vehicle.

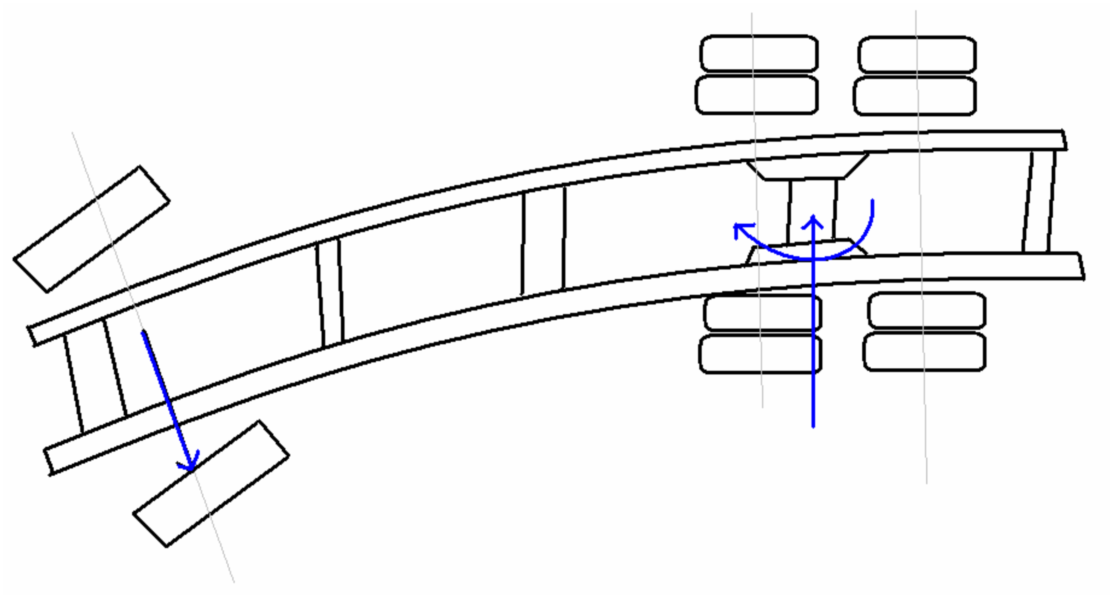


Figure 8.17 Axial and lateral bending (heavy vehicles)

8.3.1.1 Boundary conditions

The constraints defined for the FE model of the chassis frame, for lateral bending Stiffness calculation are at the two wheelbase locations in rear side and centre of first cross-member centre. To avoid incorrect results due to over constraining, minimum no. of constraints were applied to the model. Forces of 100N each were applied to the frame at two wheelbase locations in front side of the chassis frame long-members. All loads were applied at the shearcentres of the sections.

8.3.1.2 Discussions s on results

The LPM analysis was done using above discussed theory. Based on the results for Lateral bending stiffness, we can summarize the correlation in following points:

- The variation of cross-section of long-member at the junction is not accounted in the LPM
- The LPM only considers only the four loops of Long-member and cross-member of the chassis frame and is irrespective of wheelbase locations.
- This method gives reasonable correlation and its pattern is also similar to the FE simulation results.
- Localized deformations at long-member and cross-member joints were not considered in the analysis.
- First and last cross-member also contributes to the stiffness of the frame in lateral bending significantly, but was ignored in the analysis.

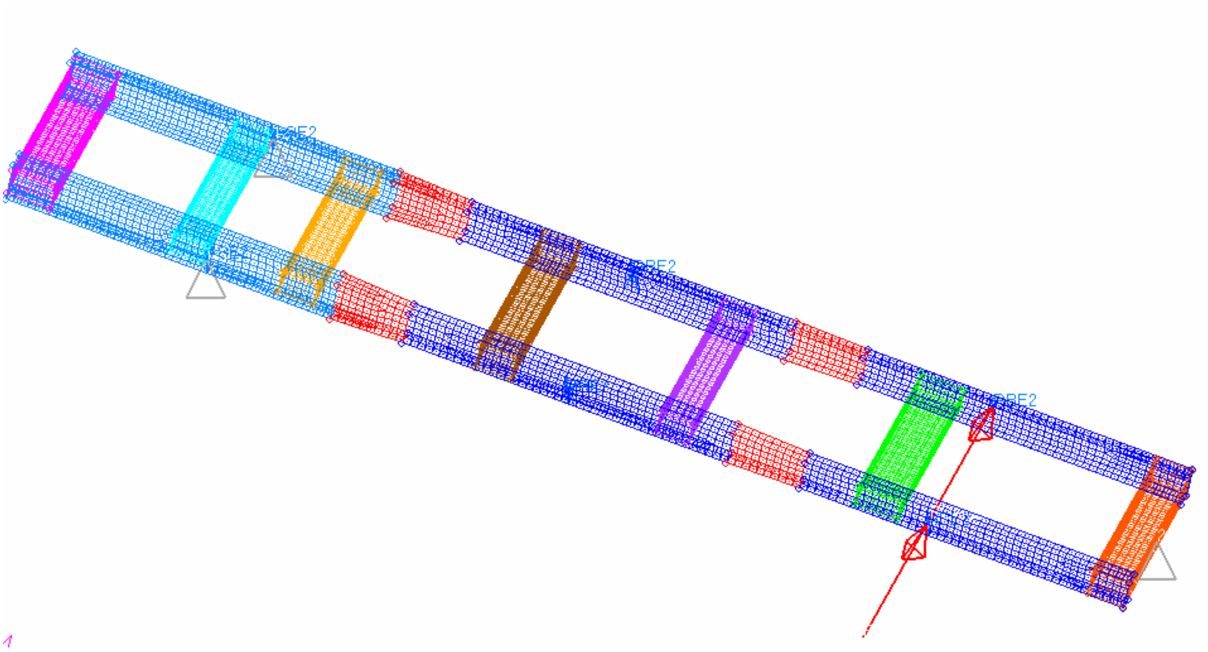


Figure 8.18 Boundary conditions applied to chassis frame for Lateral bending analysis (heavy vehicles)

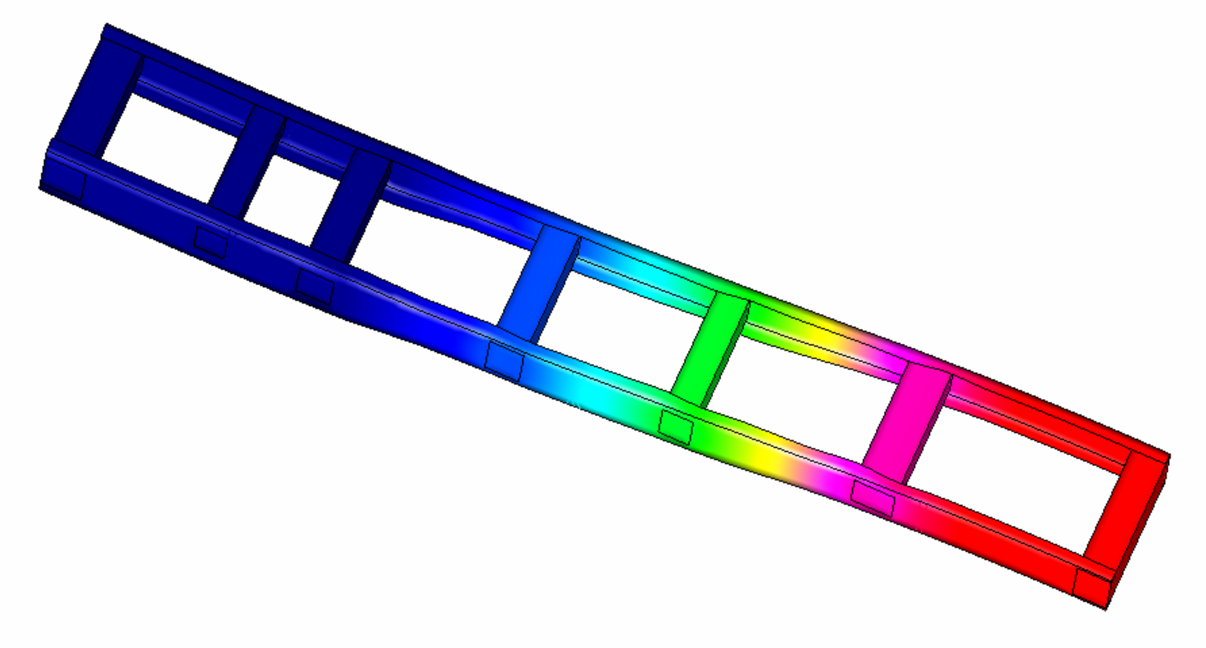


Figure 8.19 Contoured plot of deflection of chassis frame in Lateral Bending analysis (heavy vehicles)

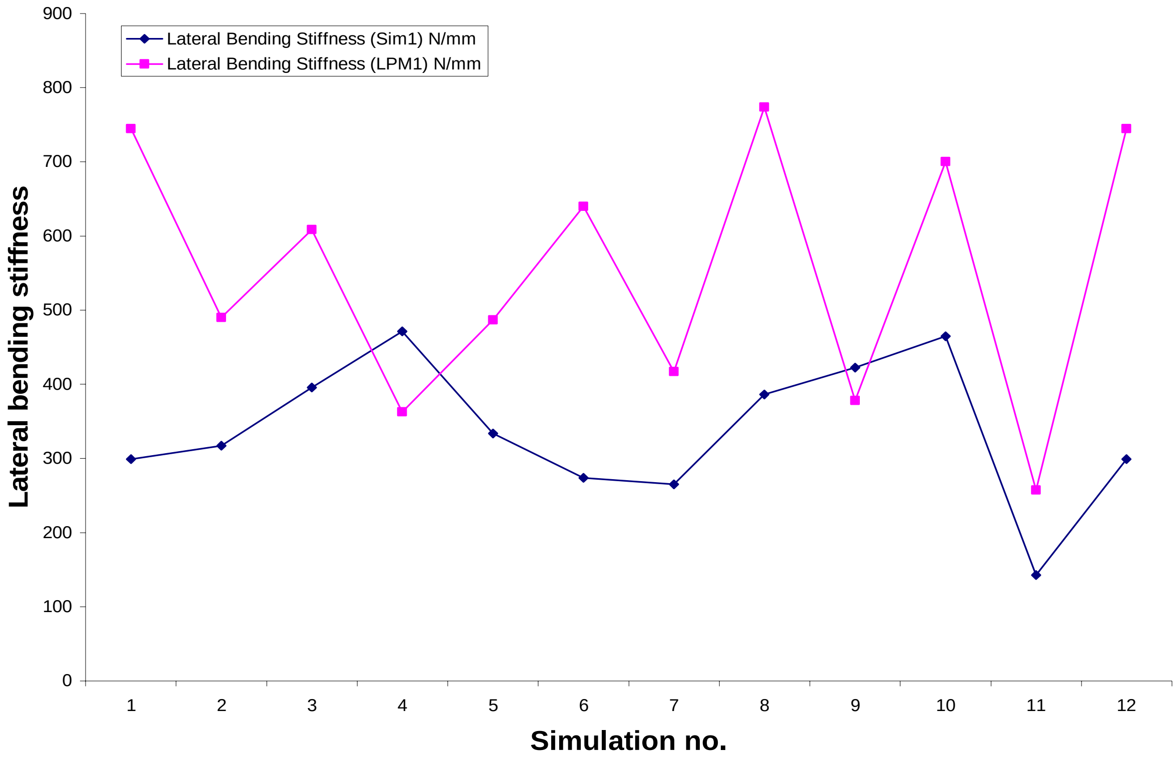


Figure 8.20 Stiffness plot of various chassis frame models in

Lateral bending analysis (Heavy vehicles)

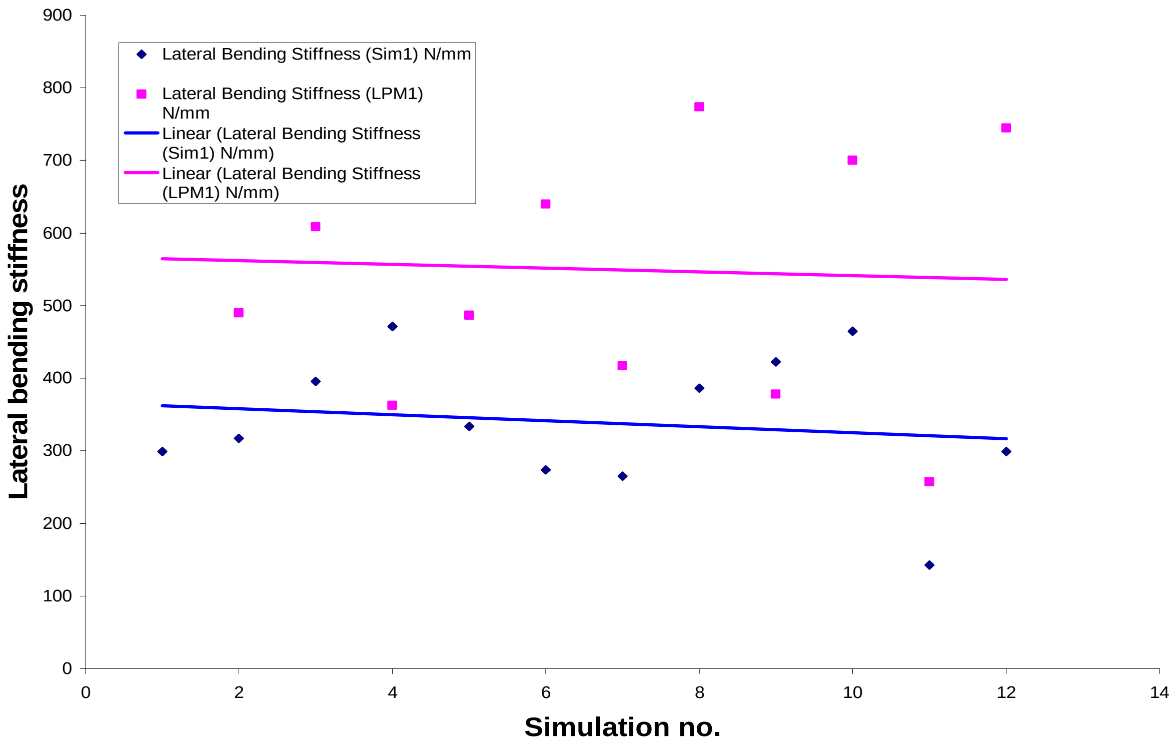


Figure 8.21 Scatter plot of various chassis frame models in

Lateral bending analysis (Heavy vehicles)

8.3.2 Lateral Bending Stiffness for Formula SAE cars [27]

This is a case which is applied for lateral bending loads in Formula SAE cars. These loads are induced in the frame for various reason such as load camber, side wind loads and centrifugal forces caused by cornering. The sideways force act along the length of the car and will be resisted at the tires. This causes a lateral load and resultant bending.

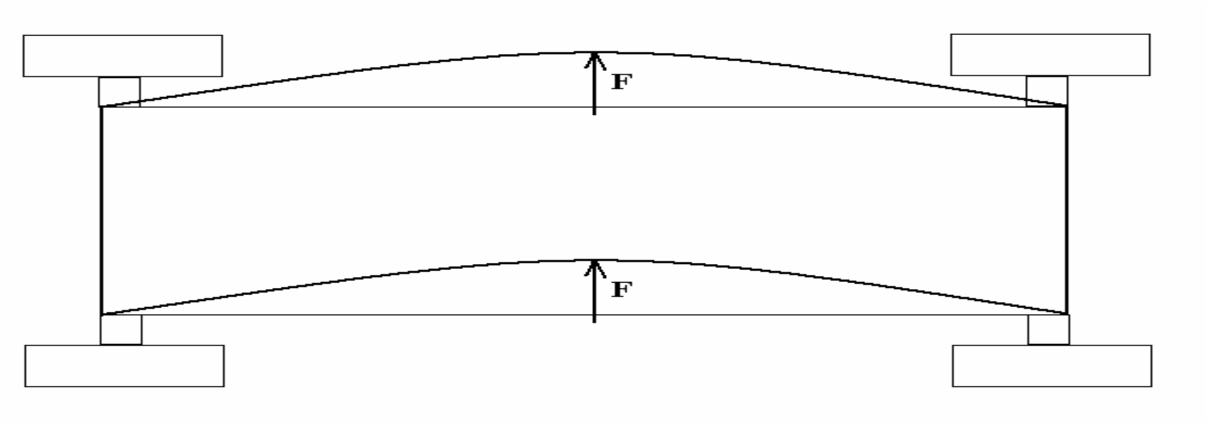


Figure 8.22 Lateral bending in Formula SAE cars

8.3.2.1 Boundary conditions

The constraints defined for the FE model of the chassis frame, for lateral bending Stiffness calculation are at the four wheelbase locations of chassis frame. To avoid incorrect results due to over constraining, minimum no. of constraints were applied to the model. Forces of 100N each were applied to the frame at centre locations of the chassis frame long-members. All loads were applied at the shearcentres of the sections.

8.3.2.2 Discussions s on results

The LPM analysis was done using above discussed theory. Based on the results for Lateral bending stiffness, we can summarize the correlation in following points:

- The variation of cross-section of long-member at the junction is not accounted in the LPM
- The LPM only considers only the four loops of Long-member and cross-member of the chassis frame and is irrespective of wheelbase locations.
- This method gives reasonable correlation and its pattern is also similar to the FE simulation results.
- The correlation obtained for lateral bending stiffness varies from 45% to 87.5%.

- . The correlation obtained for vertical bending stiffness varies from 45% to 87.5% in the analysis.
- Localized deformations at long-member and cross-member joints were not considered in the analysis.
- First and last cross-member also contributes to the stiffness of the frame in lateral bending significantly, but was ignored in the analysis.

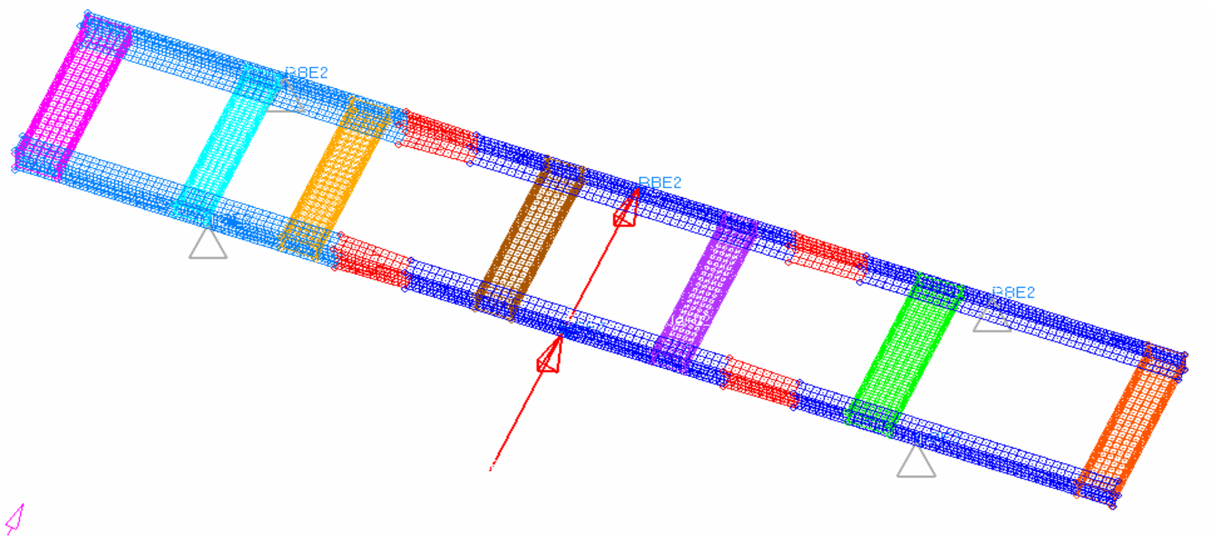


Figure 8.23 Boundary conditions applied to chassis frame for Lateral bending analysis (Formula SAE cars)

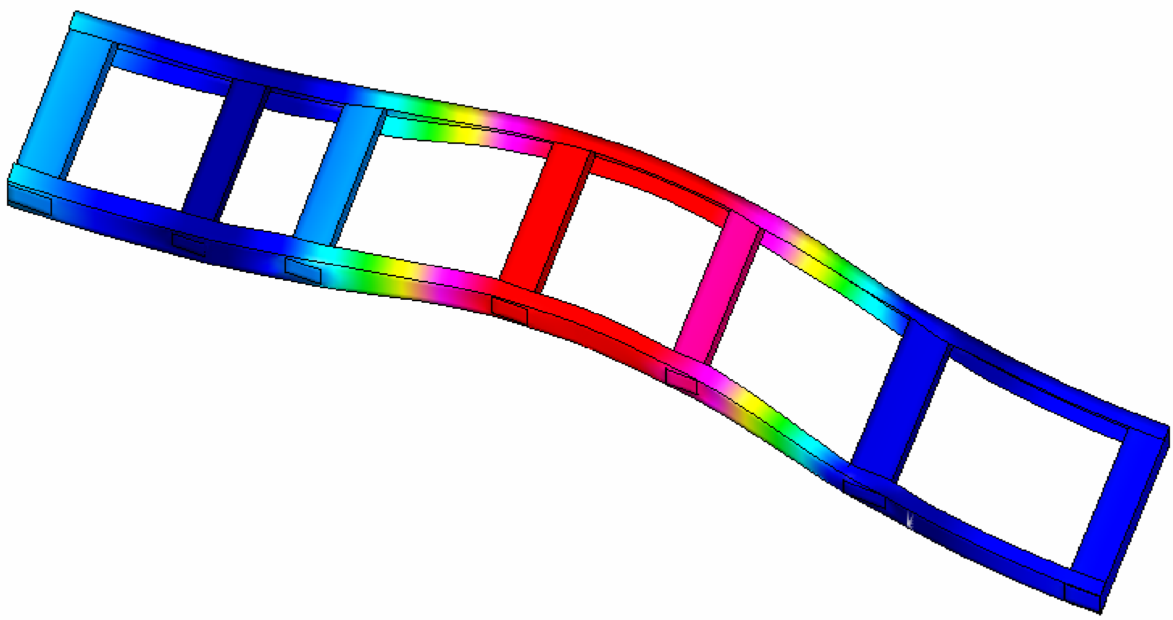


Figure 8.24 Contoured plot of deflection of chassis frame in Lateral bending analysis (Formula SAE cars)

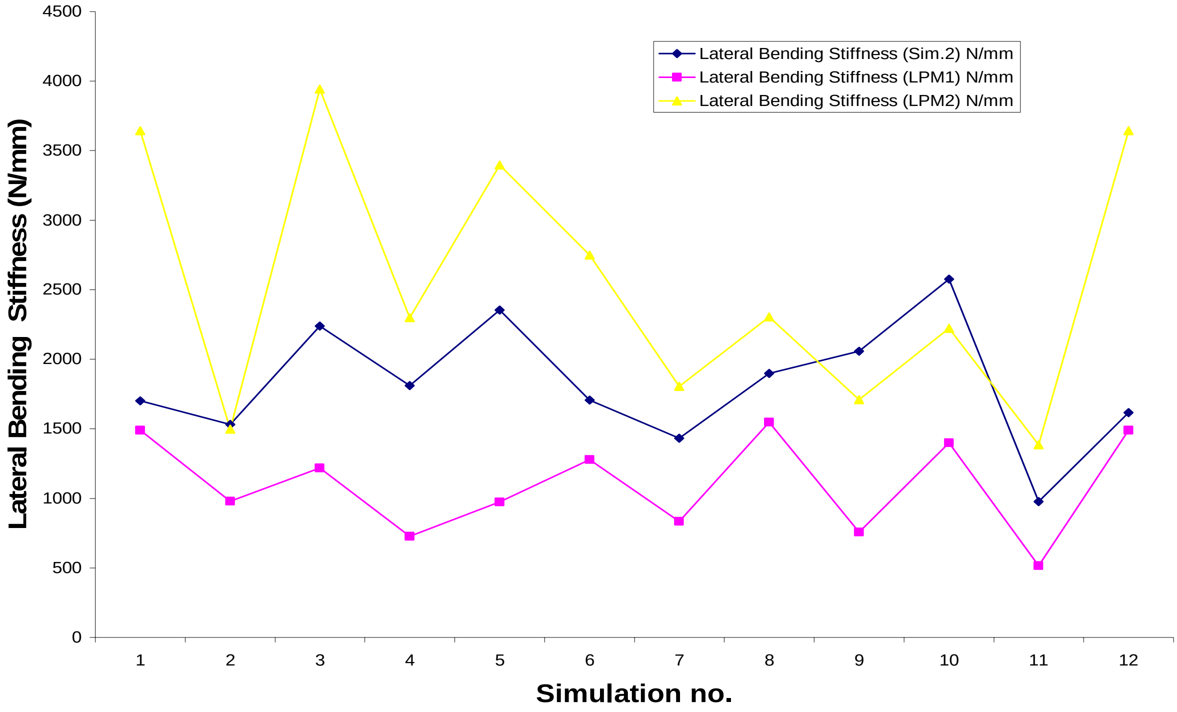


Figure 8.25 Stiffness plot of various chassis frame models in Lateral bending analysis (Formula SAE cars)

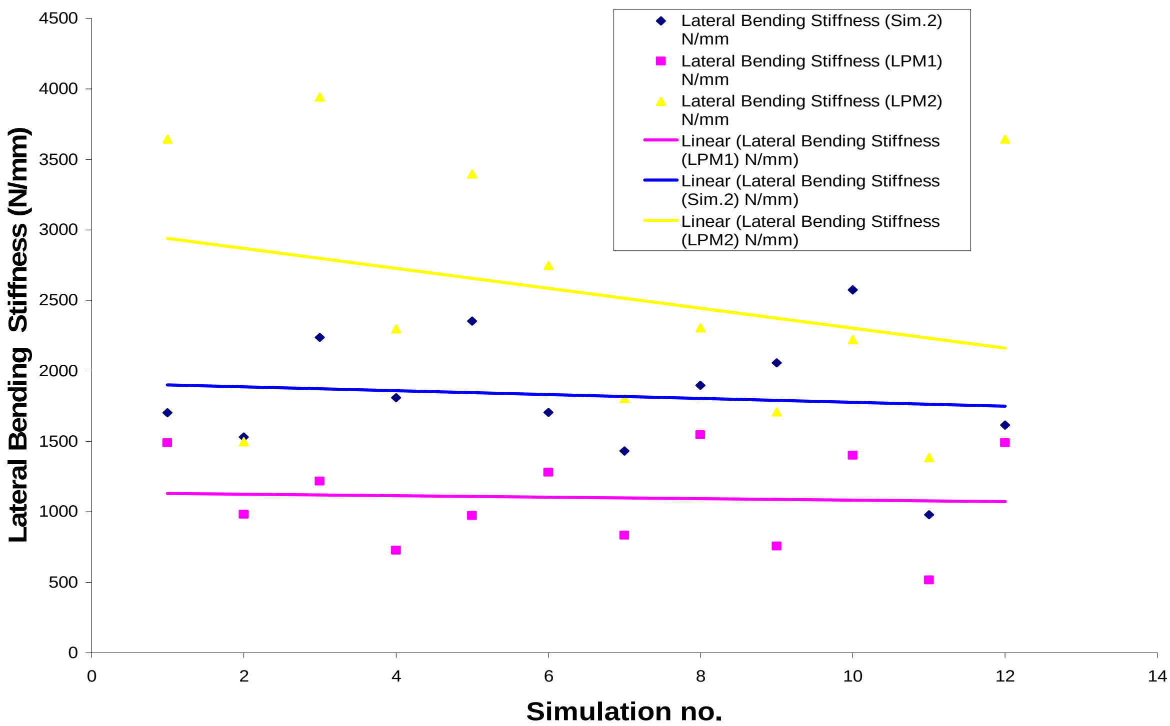


Figure 8.26 Scatter plot of various chassis frame models in Lateral bending analysis (Formula SAE cars)

8.4 Maximum deflection under One wheel Lift (Strength Analysis) [6]

The stresses produced in frame by lateral bending and twisting due to uneven terrain may be expected to be large and this can be the basis of truck designed for highway and off-highway conditions.

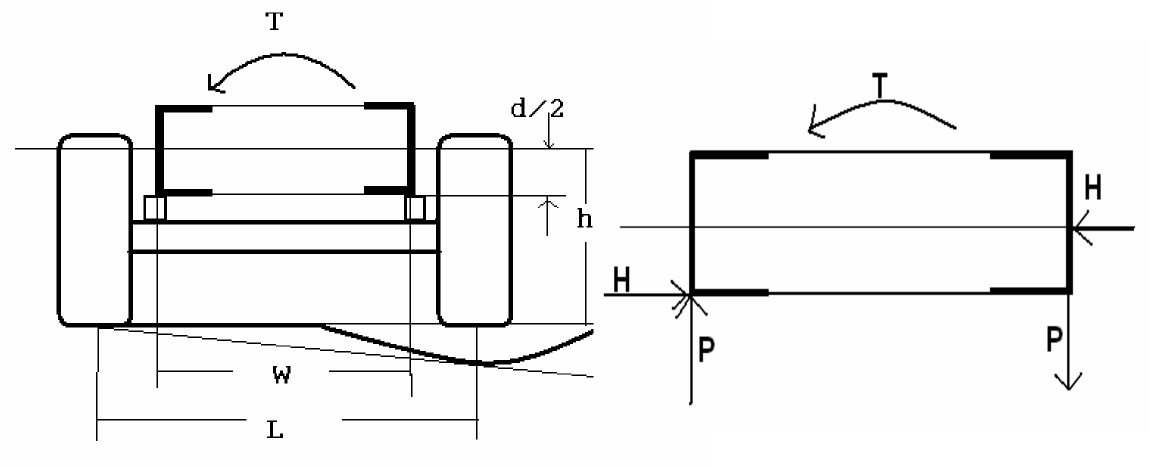


Fig. 8.7 Maximum deflection under One wheel Lift

$$F = \frac{\gamma L \left\{\beta h\left[(2h + \alpha L)(2k_s + \varepsilon w^2) - 2k_s d\right] + 2\varepsilon k_s w^2\right\}}{\Omega}\alpha$$

$$P = \frac{k_s \gamma L^2(\varepsilon w^2 - \beta hd) + \beta\gamma L^2 h \frac{d}{2}\left(2k_s + \varepsilon w^2\right)}{w\Omega}\alpha$$

$$H = \frac{\alpha}{\frac{1}{\beta h} + \frac{2h + \alpha L}{\gamma L^2}} \qquad T = \frac{k_S \gamma L^2(\varepsilon w^2 - \beta hd)}{\Omega}\alpha$$

where,

$$\Omega = \gamma L^2(2k_S + \varepsilon w^2) + 2\varepsilon k_S w^2 + \beta h\left[(2h + \alpha L)(2k_S + \varepsilon w^2) - 2k_S d\right]$$

It is observed that relatively small terrain twist produces considerable lateral horizontal forces which tend to turn the vehicle. But the frame torque is found to be small. Twisting angle of the frame generally accounts to 50% of the terrain twist. This means that under normal riding conditions without slip of the front wheels and without any rotation of rear axle, the Torsional stresses in the frame, induced by the twist of the vehicle, are rather low. Much larger stresses may be expected due to bending of the frame in the horizontal plane.

8.4.1 Boundary conditions

The constraints defined for the FE model of the chassis frame, for Strength analysis are at the three wheelbase locations. To avoid incorrect results due to over constraining, minimum no. of constraints were applied to the model. Force of 100N was applied to the frame at one of the wheel location in front side of the chassis frame long-member. All loads were applied at the shearcentres of the sections.

8.4.2 Discussions on results

- Maximum stresses were developed in the cross-members which were subjected to both twisting and bending.,
- This case can be taken as worst case strength determination of chassis frame.

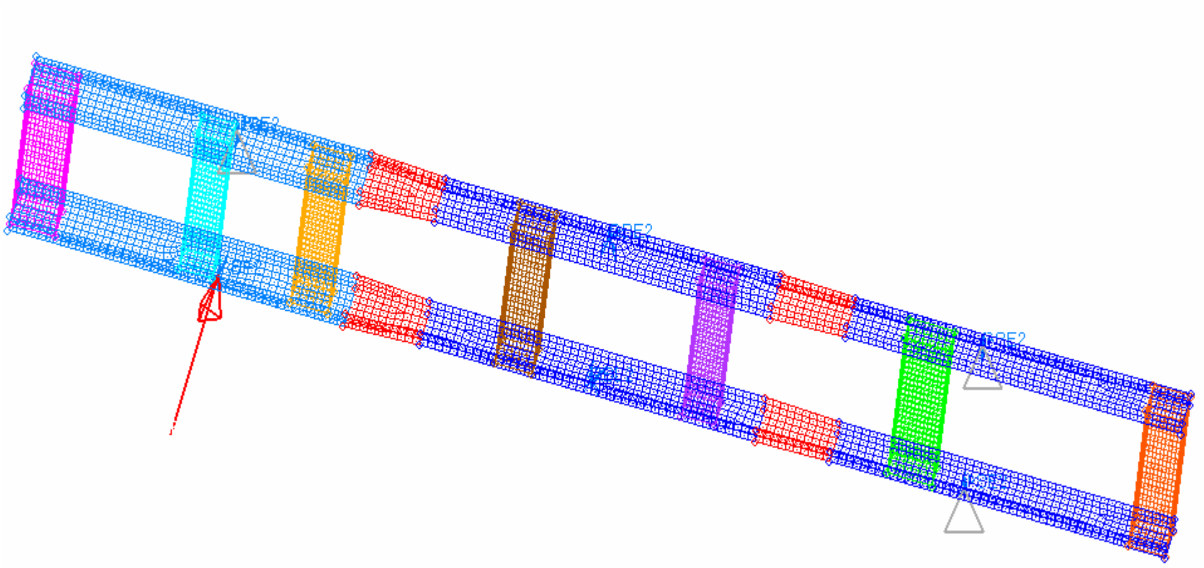


Figure 8.27 Boundary conditions applied to chassis frame for Strength analysis

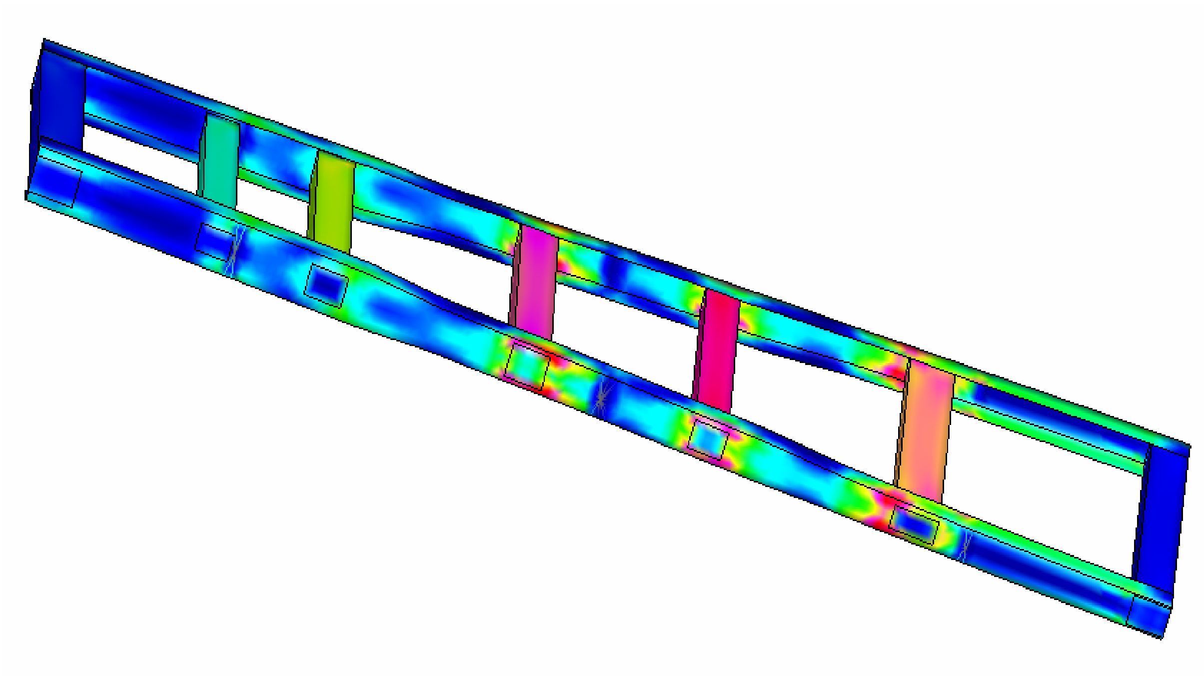


Figure 8.28 Contoured plot of Stress variation of chassis frame in Strength analysis

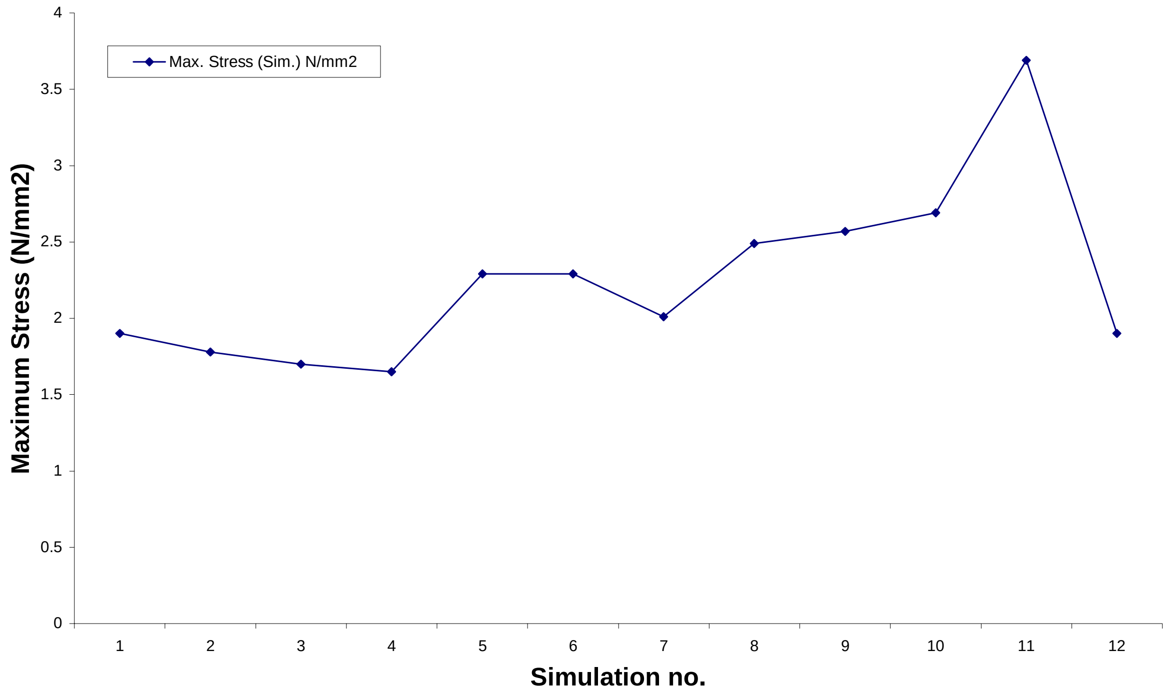


Figure 8.29 Maximum stress plot of various chassis frame models in vertical bending analysis

8.5 Vibration Analysis

In dynamic case the loads are influenced by operating terrain conditions and speed of vehicle. The forces can be tremendous and may cause frame to twist and weave. Thus forces must be isolated and transferred to a point on the frame where the resulting deflections will cause no or minimum damage

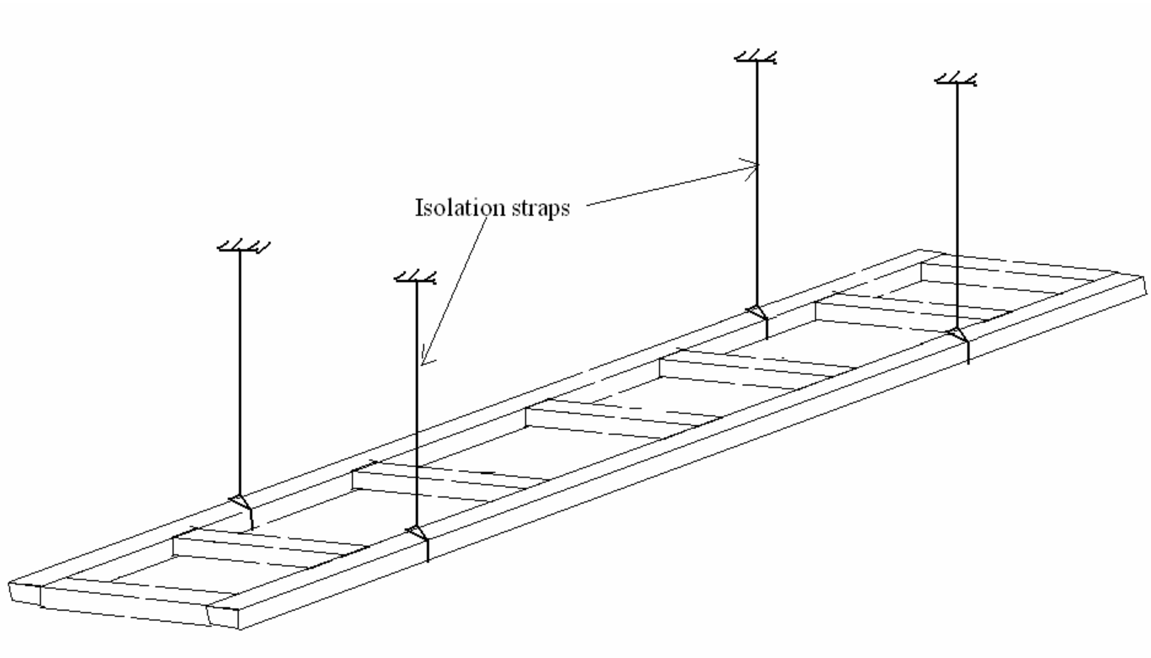


Figure 8.30 Frame supported in "free-free" condition [5]

To determine the dynamic behavior of a frame, it is supported in a “free-free” condition. This is achieved by hanging the frame from isolation straps (weak rubber band) as shown in figure. First step of Dynamic analysis is to determine the frequency response. Excitation is given using electro-hydraulic or electro-mechanical exciter and transducers are used to measure instantaneous value of excitation force and vibration motion is also measured at numerous other points. Finally the natural frequencies and respective mode shapes are obtained.

In this analysis, one eigenvalue load collector “EIGRL” was created for ten modes of vibrations. Initial six modes are shows rigid body motion of the frame. Rest four odes are of actual interest.

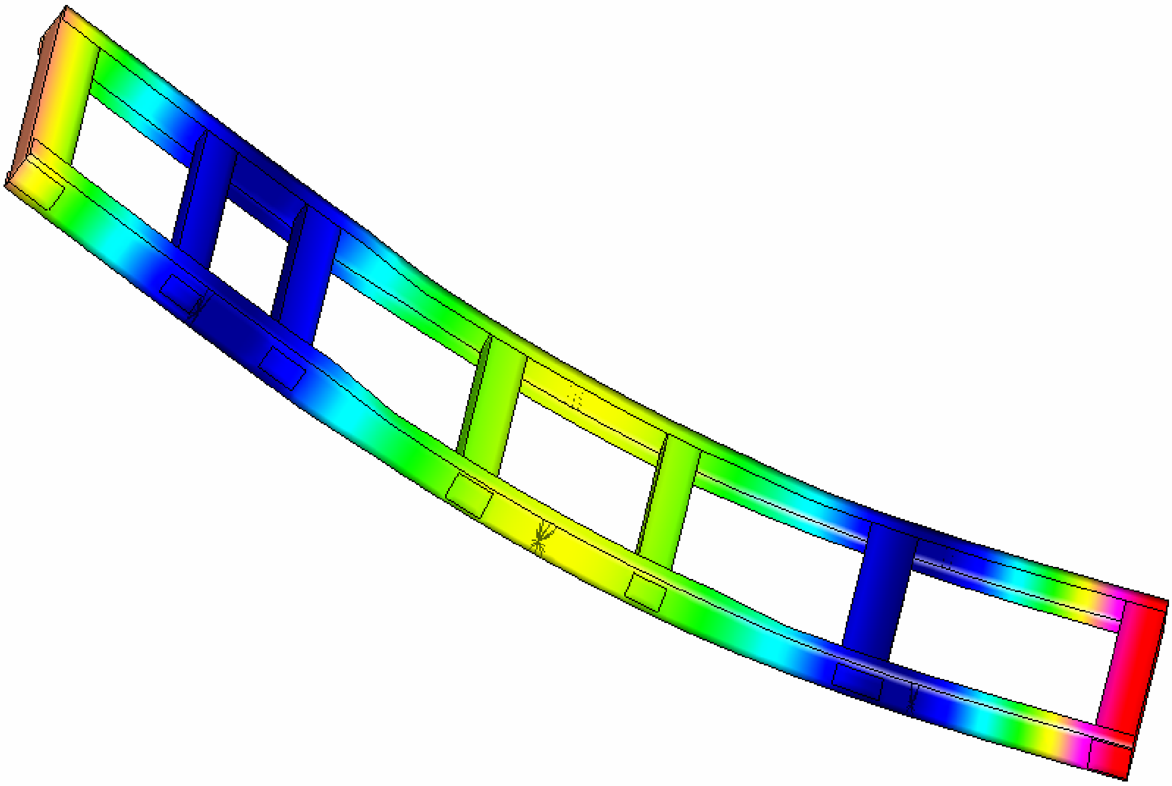


Figure 8.31 Contoured Eigenvector plot of chassis frame in Free- Free Vibration analysis (Vertical beaming)

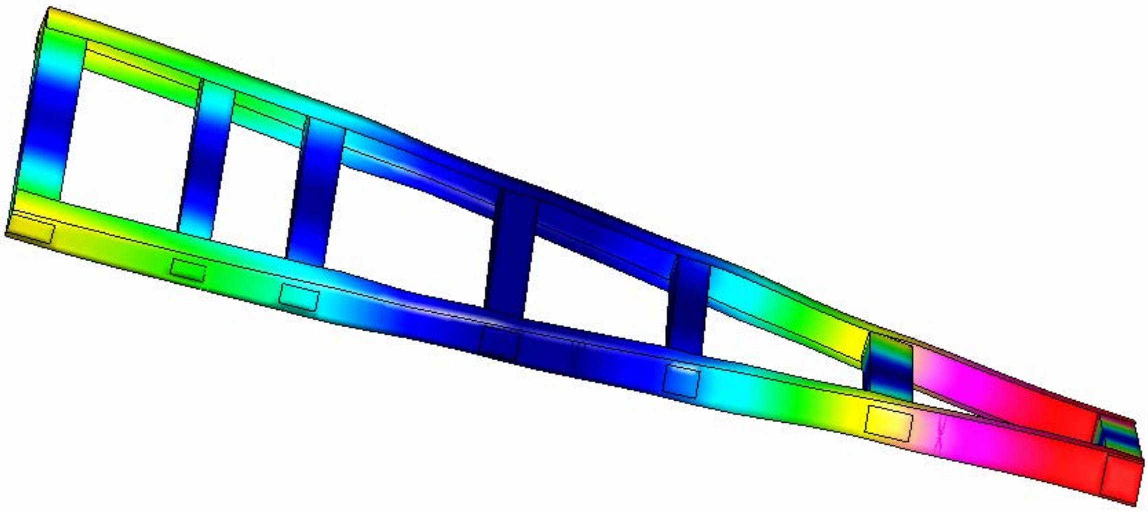


Figure 8.32 Contoured Eigenvector plot of chassis frame in Free- Free Vibration analysis (Torsional beaming)

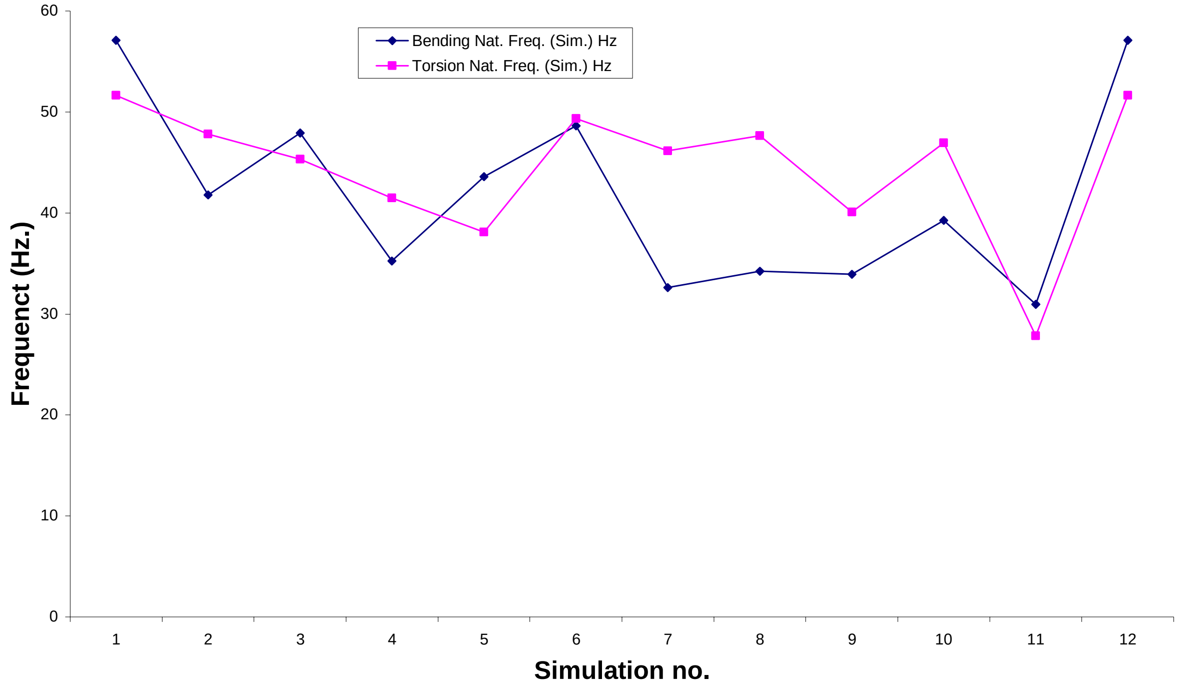


Figure 8.33 Frequency plot of various chassis frame models in Free- Free Vibration analysis (Torsional and vertical beaming)

8.6 Crashworthiness

Crashworthiness performance is emerging as an uncompromising factor in vehicle design. Most Automakers have adapted air bag as standard specification to reduce passenger's injury, especially at head and chest, passenger could survive in an accident.

During offset crash, passenger's lower leg injury becomes more severe. Few automakers adapt knee airbag which is a costly affair and also unacceptable due to layout constraints. Since passengers compartment is to be made stiff enough to avoid penetration during offset crash, it is likely that vehicle will endure higher deceleration and thus increases passenger's injury.

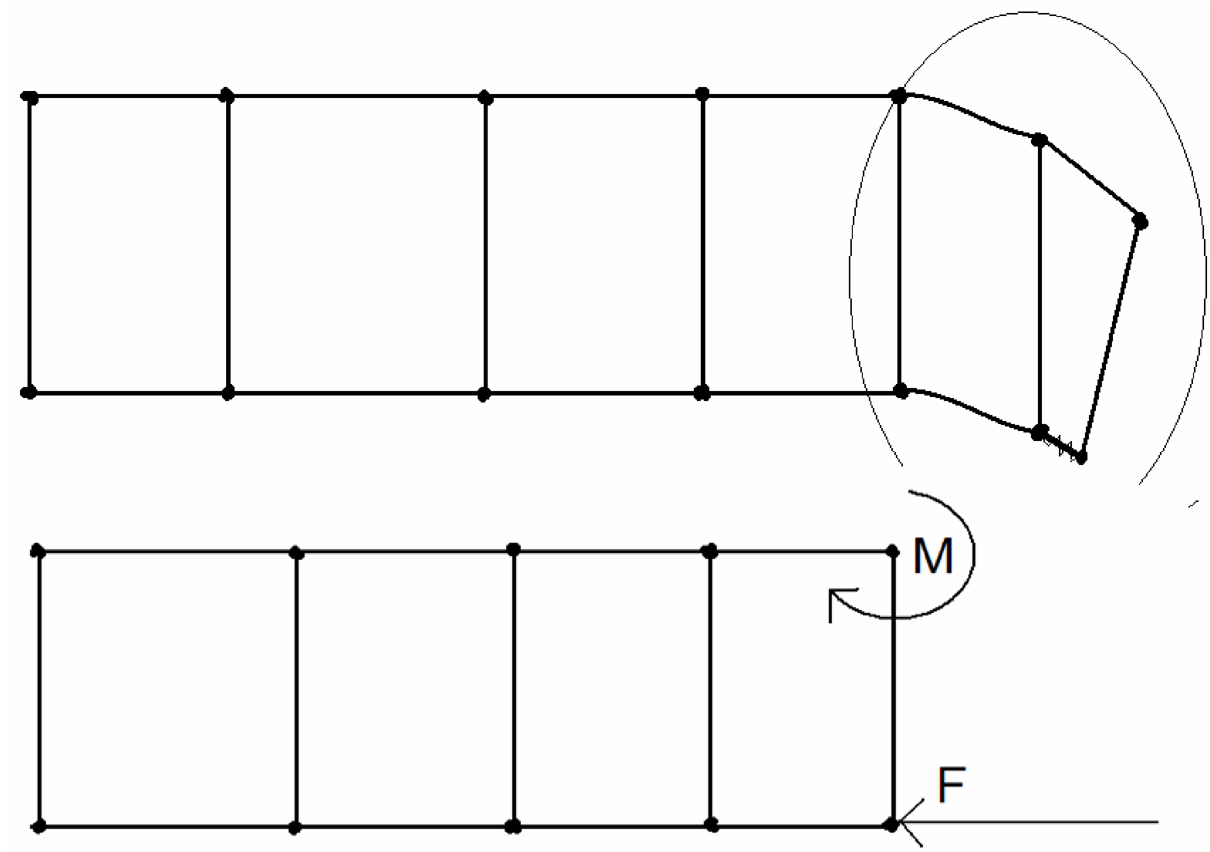


Figure 8.34 Simplification of chassis frame for Crash analysis

The plastic deformation of chassis frame occurs in front portions of chassis frame. For analysis the front portion of the chassis frame was eliminated and was replaced with a force and moment. To get the response of chassis frame in actual case for the rest of the chassis frame, inertia relief method was used.

8.6.1 Inertia relief method

A simple, but in many cases effective, means to introduce some information about the system dynamics into those static shapes consists in introducing is called inertia relief. It consists in decoupling the static shapes from the rigid body motion of the system by way of the inertia matrix.

In an inertia relief analysis, the applied loads are balanced by a set of translational and rotational accelerations. These accelerations provide body forces distributed over the structure in such a way that the sum total of the applied forces on the structure is zero. This provides the steady-state stress and deformed shape in the structure as if it were freely accelerating due to the applied loads. Boundary conditions are applied only to restrain rigid body motion. Because the external loads are balanced by the accelerations, the reaction forces corresponding to these boundary conditions are zero.

Without accounting for inertia relief effects, the static analyses of such structures would either encounter singularity in the solution process or yield unrealistic displacements. Several commercial structural analysis and Optimization software perform inertia relief calculation through a user-controlled switch. Inertial effects could also influence the designs obtained from structural Optimization procedures. In this paper, several structural Optimization case studies are presented to illustrate the influence of inertia relief on optimal designs. In the present case analysis was performed using the commercial software, Altair OptiStruct.

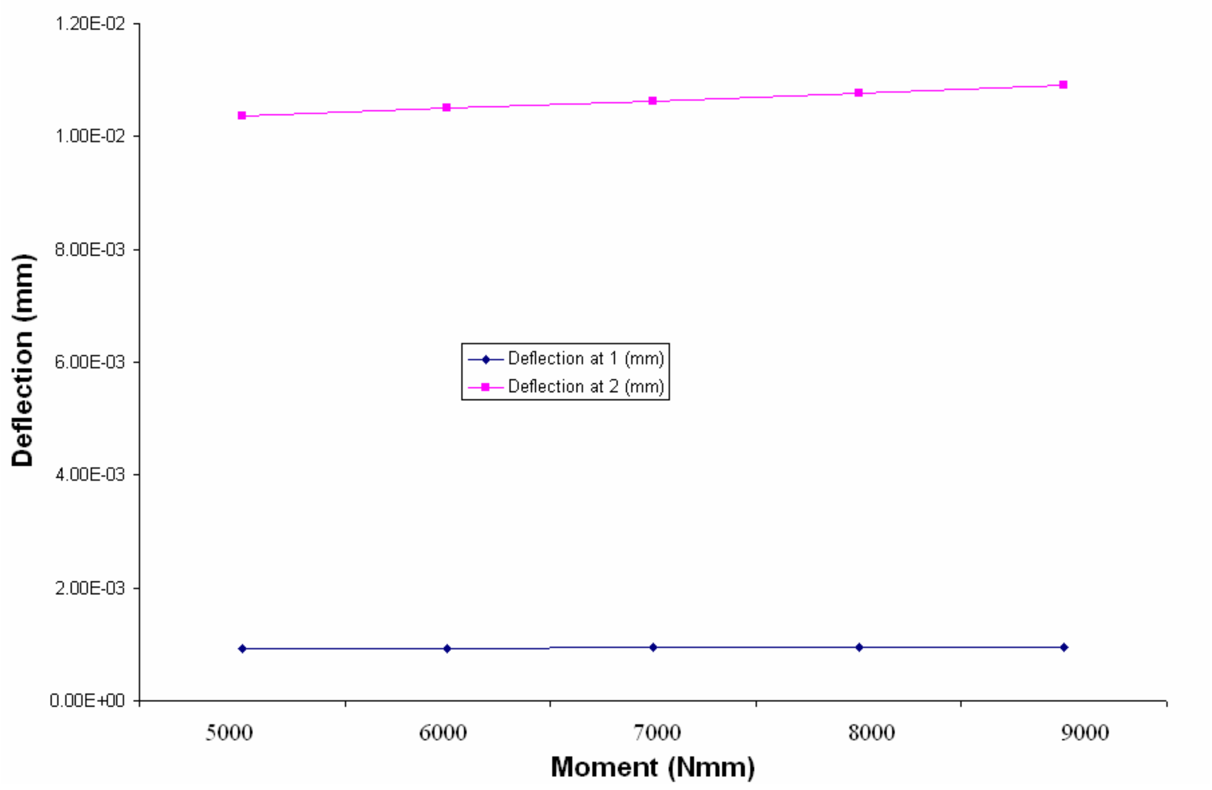


Figure 8.35 Deflection plot of the simplified chassis frame model (for moment variation)

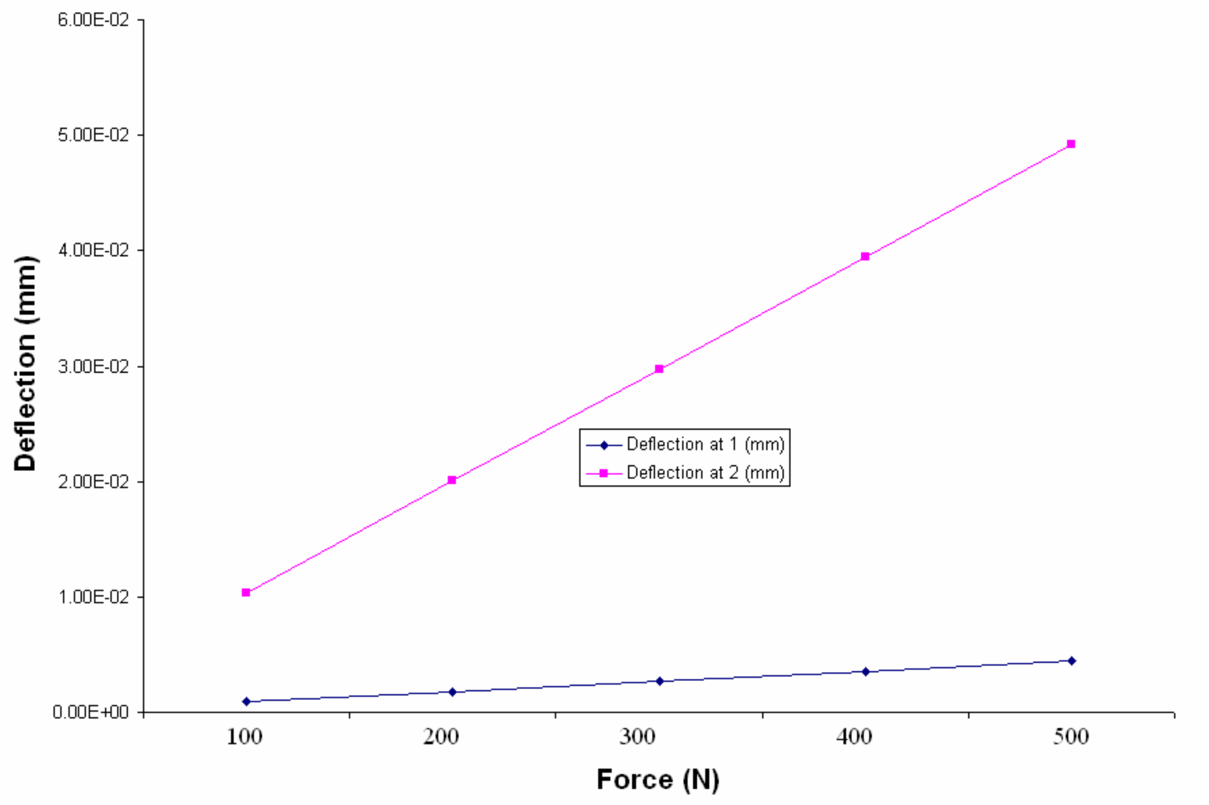


Figure 8.36 Deflection plot of the simplified chassis frame model (for Force variation)

9 OPTIMIZATION

The environmental issue is also an essential factor considering pollution and cost reduction, automakers are putting significant effort on weight reduction while keeping competitive performance. Weight reduction causes reduction in pollution, fuel economy and improvement in acceleration performance.

For resizing or performance Optimization almost any property and material information of an element can be used as design variables. The examples of Optimization design variables are element thickness, moment of inertia and their combination, cross-sectional area, modulus of elasticity mass etc.

Structural Optimization is applied in real life structures. But it is important to recognise and select optimum design space and efficient design criteria. Due to some uncertainty in early design stages they need to be relaxed to achieve better Optimization results.

To minimise weight, which is a function of sizing variables limits and constraints are applied and the problem is solved.

Objective function:

Minimise, $\textbf{Weight}\ (W) = f\,(X)$

Subjected to;

1. Inequality constraints, $g_j(X) \le 0 \qquad j=1,2,\text{--},m.$

The inequality constraints are constraints derive from performance requirements. They typically include the limiting values of stresses, joint deflections and vibration frequencies. In present case it is the three stiffness values;

- **Vertical bending stiffness** $\ge K_{vb}$
- **Lateral bending stiffness** $\ge K_{lb}$
- **Torsional stiffness** $\ge K_t$

2. Side constraints, $\mathbf{X^{l}_{i} \leq X_{i} \leq X^{U}_{i}}$ $\quad$ ***i=1,2,---,n.***

Side constraints are the simple limits imposed on the design variables to provide practical limits on member sizes. Thus it is used to define the envelope of variation that can be done in the structure. In present case, they are the Design variables of the chassis frame.

- Height of the cross section $\mathbf{a^{l}_{i} \leq a_{i} \leq a^{U}_{i}}$ $\quad$ ***i=1,2,---,n.***
- Width of the cross section $\mathbf{b^{l}_{i} \leq b_{i} \leq b^{U}_{i}}$
- Thickness of the cross section $\mathbf{t^{l}_{i} \leq t_{i} \leq t^{U}_{i}}$
- Location of the cross-member $\mathbf{ll^{l}_{i} \leq ll_{i} \leq ll^{U}_{i}}$
- Length of the Long-member Sections (Constant area) $\mathbf{L^{l}_{i} \leq L_{i} \leq L^{U}_{i}}$
- Width of the frame $\mathbf{W^{l}_{i} \leq W_{i} \leq W^{U}_{i}}$

The formulation of chassis frame problem contains 90 constraints and 40 design variables. Based upon above formulation, weight Optimization was performed for three different cases. The Optimization was done using MS Excel as solver.

9.1 Optimization 1

Table 9.1 Results of Optimization 1

		Initial_Values	Target Values	Optimised_Values
Weight Optimization	**Kg**	**75.1**		**48.8**
Performance				
Vertical Bending Stiffness (Sim.)	**N/mm**	836.1		1368.2
Vertical Bending Stiffness (LPM)	**N/mm**	**918.1**	**1400**	**1401.6**
Torsional Stiffness (Sim.)	**Nm/Deg**	3913.3		2327.2
Torsional Stiffness (LPM)	**Nm/Deg**	**3798.1**	**3800**	**3799.99**
Lateral Bending Stiffness (Sim.)	**N/mm**	1530.2		732.3
Lateral Bending Stiffness (LPM)	**N/mm**	**980.4**	**930**	**936.5**

Weight optimisation of chassis frame for above case shows significant reduction in the weight (54%) without sacrificing the three performance criteria. The resulting frame model was analyzed using FE simulations. This gives 97.6% correlation in vertical bending stiffness, 62.23% correlation in vertical bending stiffness, 78.2% correlation in lateral bending stiffness

9.2 Optimization 2

Weight optimisation of chassis frame for this case shows significant reduction in the weight (58%) but the vertical bending stiffness is improved with sacrificing torsional and lateral bending stiffness. The resulting frame model was analyzed using FE simulations. This gives 74% correlation in vertical bending stiffness, 51% correlation in vertical bending stiffness, 44% correlation in lateral bending stiffness

Table 9.2 Results of Optimization 2

		Initial_Values	Target Values	Optimised_Values
Weight Optimization	**Kg**	**75.1**		**31.7**
Performance				
Vertical Bending Stiffness (Sim.)	**N/mm**	836.1		1474.3
Vertical Bending Stiffness (LPM)	**N/mm**	**918.1**	**2000**	**2000.0**
Torsional Stiffness (Sim.)	**Nm/Deg**	3913.3		1489.9
Torsional Stiffness (LPM)	**Nm/Deg**	**3798.1**	**2900**	**2910.4**
Lateral Bending Stiffness (Sim.)	**N/mm**	1530.2		324.3
Lateral Bending Stiffness (LPM)	**N/mm**	**980.4**	**750**	**755.4**

9.3 Optimization 3

Weight optimisation of chassis frame for this case shows reduction in the weight (25%). The vertical bending stiffness is improved by 67% with improvement in torsional stiffness by 46% and lateral bending stiffness by 26%. The resulting frame model was analyzed using FE simulations. This gives 99.5% correlation in vertical bending stiffness, 60% correlation in vertical bending stiffness, 79% correlation in lateral bending stiffness

Table 9.3 Results of Optimization 3

		Initial_Values	Target Values	Optimised_Values
Weight Optimization	**Kg**	**75.1**		**56.3**
Performance				
Vertical Bending Stiffness (Sim.)	**N/mm**	836.1		2790.4
Vertical Bending Stiffness (LPM)	**N/mm**	**918.1**	**2800**	**2805.1**
Torsional Stiffness (Sim.)	**Nm/Deg**	3913.3		4177.9
Torsional Stiffness (LPM)	**Nm/Deg**	**3798.1**	**7000**	**6999.99**
Lateral Bending Stiffness (Sim.)	**N/mm**	1530.2		1687.8
Lateral Bending Stiffness (LPM)	**N/mm**	**980.4**	**1330**	**1339.7**

10 CONCLUSION AND FUTURE WORK

The characteristics of automotive structure vary in geometry, material and design criteria. The primary aim of this study was to develop a methodology.The study performed in present context gives an important tool to achieve desired performance requirements within design space while minimizing weight. This study has also demonstrated the reduced effort for optimization as number of FE simulations have been drastically reduced.

10.1 Conclusions

1. Analysis of ladder type chassis frame using Finite Element Analysis and Lumped Parameter Model (Mathematical Model) estimates the possibility creation of Lumped Parameter Model (Mathematical Model) for all performance criteria of chassis frame.

2. After creating twelve numerical models of ladder type chassis frame and getting very close correlation for Vertical bending stiffness of ladder type chassis frame, the assumption is proven that cross-member size or type has no or negligible effect on Vertical bending stiffness. The correlation obtained for vertical bending stiffness varies from 85% to 95% in the analysis.

3. Analysis shows that for creating Lumped Parameter Model (Mathematical Model) of Torsional bending stiffness of ladder type chassis frame should consider pure twisting, Cantilever bending and twisting due to foreshortening due to cantilever bending. The correlation obtained for Torsional stiffness varies from 72% to 97% in the analysis.

4. Analysis results for lateral bending case shows that the lateral stiffness is a function of design variables of both long-members and cross-members. The correlation obtained for lateral bending stiffness varies from 45% to 87.5% in the analysis.

5. Analysis done for one wheel lift case (Strength analysis) ladder type chassis frames shows that maximum stress is developed in the cross-member due to combined effect of bending and twisting. Analysis done for crashworthiness of ladder type chassis frame (with removing front portions with nonlinear deformations) for offset frontal crash using inertia relief method shows that the deformation is more affected by force than moment variation.

6. The analysis done for crashworthiness of ladder type chassis frame shows the possibility of simplifying dynamic problems into static problems with suitable assumptions. The Lumped Parameter Model mathematical model will also be a function of weight
7. Variation in the results obtained for different performance criteria of twelve models of ladder type chassis frame justifies the use of Design of Experiments in analysis.
8. Reasonable correlation obtained in three stiffness analysis (Vertical bending stiffness, Lateral bending stiffness, Torsional stiffness) shows that the performance criteria of ladder type chassis frame can be predicted using Lumped Parameter Model (Mathematical Model). These results tough not very accurate but can give a preliminary idea of selecting chassis frame design variables.
9. The result for three Weight Optimizations shows high reduction in weight, from 25-57% for the three cases optimized.
10. Reasonable correlation, in optimized frame performance criteria shows the possibility to perform Optimization of ladder type chassis frame for reducing weight without sacrificing the performance criteria. This Optimization completely eliminates the involvement of CAE (Computer Aided Analysis) software.

10.2 Future work

- Creation of suitable Lumped Parameter Model (Mathematical Model) for rest of the performance criteria (Strength, Noise Vibration Harshness, Crashworthiness) of ladder type chassis frame.
- Refinement of existing Lumped Parameter Model (Mathematical Model) by eliminating assumptions to obtain better results.
- Creation of Lumped Parameter Model (Mathematical Model) for more realistic chassis frame by adding more complexities like cranking of long-member, brackets etc.
- Multi-Disciplinary Optimization code generation to maximize all the performance criteria of chassis frame within design envelope.

ANNEXURE-A

Table A DOE results for 12 Frame Models using Latin Hypercube

	Sim1	Sim2	Sim3	Sim4	Sim5	Sim6	Sim7	Sim8	Sim9	Sim10	Sim11	Sim12
lb	2000	2200	1800	1800	1800	2000	1800	2000	2200	2200	2200	2200
W	900	1000	1000	1200	1200	1000	1000	900	1200	900	900	1000
a1	60	60	80	80	70	80	80	70	70	80	60	60
b1	100	120	120	120	100	90	100	100	100	120	100	90
t1	2.5	3.75	3.75	3.25	3.25	3.25	3.75	3.75	3.75	3.75	2.5	2.5
ll1	0	0	0	0	0	0	0	0	0	0	0	0
a2	60	60	80	60	70	60	60	80	70	60	80	80
b2	90	100	100	120	100	120	100	100	100	90	120	120
t2	2.5	3.75	3.75	3.25	3.75	3.75	3.25	2.5	3.25	2.5	3.25	2.5
ll2	450	500	450	400	400	400	400	500	400	500	450	500
a3	70	70	60	70	70	70	60	60	70	80	60	80
b3	90	100	120	100	120	120	90	120	90	120	100	100
t3	2.5	3.75	2.5	3.25	2.5	3.25	3.25	3.75	3.75	3.25	3.75	3.25
ll3	800	850	850	850	750	750	850	850	800	800	750	800
a4	80	70	80	80	80	70	70	60	70	60	80	70
b4	100	120	100	120	120	120	120	90	100	90	100	100
t4	3.75	3.25	3.25	2.5	2.5	3.25	3.75	3.75	3.25	2.5	2.5	2.5
ll4	1300	1300	1450	1350	1300	1300	1300	1450	1450	1350	1350	1350
a5	60	70	80	80	70	60	60	80	80	60	80	80
b5	120	100	90	100	90	120	120	100	90	100	120	90
t5	3.25	2.5	3.25	3.75	2.5	3.25	3.25	3.75	3.75	2.5	3.75	3.75
ll5	1750	1700	1750	1700	1750	1750	1750	1800	1800	1800	1700	1800
a6	70	80	80	70	70	70	60	70	80	80	60	80
b6	100	100	90	90	120	120	100	90	120	120	100	90
t6	3.25	3.25	3.25	3.75	2.5	3.75	3.25	2.5	3.25	3.75	2.5	2.5
ll6	2200	2300	2350	2350	2200	2350	2200	2300	2300	2200	2300	2200
a7	60	70	80	80	70	60	80	60	80	70	80	70
b7	90	120	90	100	90	100	90	120	90	90	90	100
t7	3.75	2.5	3.25	3.25	2.5	2.5	3.75	3.25	3.25	3.25	3.75	3.25
ll7	2760	2780	2910	2900	2960	2800	2910	2930	3010	2910	2710	2850
a8	100	100	120	90	90	120	90	120	100	100	120	90
b8	50	40	50	45	40	50	40	45	50	45	40	50
t8	2.5	3.75	3.25	3.75	3.75	2.5	3.25	3.25	3.25	3.25	3.75	2.5
L8	950	1000	900	950	1000	900	950	1000	1000	950	900	1000

a9	120	100	120	100	120	120	100	90	100	90	100	90
b9	50	40	45	40	45	45	40	50	40	50	45	45
t9	3.75	3.25	3.75	3.25	3.25	3.25	2.5	2.5	3.75	3.25	2.5	3.75
L9	1000	1000	1100	1100	1050	1000	1050	1100	1100	1100	1000	950
a10	120	120	100	120	120	90	90	90	90	90	90	100
b10	40	40	50	40	45	50	40	50	50	40	50	45
t10	3.75	3.25	3.75	3.25	3.75	3.25	2.5	3.25	2.5	3.75	3.25	3.75
L10	900	900	1000	950	1000	1000	1000	950	1000	950	900	1000

ANNEXURE-B

Optimization No. 1

Table B.10 Results of Optimization No. 1

		Initial_Values	Target Values		Optimised_Values
Weight Optimization	**Kg**	**75.1**			**31.7**
Performance					
Vertical Bending Stiffness (Sim.)	**N/mm**	836.1			1474.3
Vertical Bending Stiffness (LPM)	**N/mm**	**918.1**	**2000.0**		**2000.0**
Torsional Stiffness (Sim.)	**Nm/Deg**	3913.3			1489.9
Torsional Stiffness (LPM)	**Nm/Deg**	**3798.1**	**2900.0**		**2910.4**
Lateral Bending Stiffness (Sim.)	**N/mm**	1530.2			324.3
Lateral Bending Stiffness (LPM)	**N/mm**	**980.4**	**750.0**		**755.4**
Overall Frame			**Range of values**		
			min	**max**	
Frame Length	**L**	2900			3000
Wheel base	**lb**	2200	1800	2500	1858.921
Frame width	**W**	1000	800	1200	800
Young's Modulus	**E**	210000			210000
Modulus of rigidity	**G**	81000			81000
Density	**kg/mm3**	7.9E-06			7.90E-06
Section 1					
Rectangle c/m	**a1**	60	60	100	60
	b1	120	60	100	60
	t1	3.75	2	5	2
Area	**A1**	1293.8			464.0
c/m position	**lr1**	63.8			32.0
	ll1	0.0			0.0
MOI	**Ix**	803408.2			260458.7
MOI	**Iy**	2410752.0			260458.7
Polar MOI	**J**	3214160.2			520917.3
Section 2					
Rectangle c/m	**a2**	60	60	100	60
	b2	100	60	100	60

	t2	3.75	2	5	2
Area	**A2**	1143.75			464
c/m position	**lr2**	533.75			532
	ll2	500	350	500	500
MOI	**Ix**	1537392.6			260458.7
MOI	**Iy**	684580.1			260458.7
Polar MOI	**J**	2221972.7			520917.3
Section 3					
Rectangle c/m	**a3**	70.0	60.0	100.0	60.0
	b3	100.0	60.0	100.0	60.0
	t3	3.8	2.0	5.0	2.0
Area	**A3**	1218.8			464.0
c/m position	**lr3**	903.8			932.0
	ll3	850.0	700.0	900.0	900.0
MOI	**Ix**	976416.0			260458.7
MOI	**Iy**	1711181.6			260458.7
Polar MOI	**J**	2687597.7			520917.3
Section 4					
Rectangle c/m	**a4**	70.0	60.0	100.0	116.3
	b4	120.0	60.0	100.0	60.0
	t4	3.3	2.0	5.0	2.0
Area	**A4**	1192.8			689.0
c/m position	**lr4**	1363.3			1343.5
	ll4	1300.0	1300.0	1500.0	1311.5
MOI	**Ix**	1008213.8			1254716.1
MOI	**Iy**	2342867.0			449758.7
Polar MOI	**J**	3351080.8			1704474.8
Section 5					
Rectangle c/m	**a5**	70.0	60.0	100.0	60.0
	b5	100.0	60.0	100.0	60.0
	t5	2.5	2.0	5.0	2.0
Area	**A5**	825.0			464.0

c/m position	**lr5**	1752.5			1804.9
	ll5	1700.0	1700.0	1900.0	1772.9
MOI	**Ix**	684218.8			260458.7
MOI	**Iy**	1189218.8			260458.7
Polar MOI	**J**	1873437.5			520917.3
Section 6					
Rectangle c/m	**a6**	80.0	60.0	100.0	60.0
	b6	100.0	60.0	100.0	60.0
	t6	3.3	2.0	5.0	2.0
Area	**A6**	1127.8			464.0
c/m position	**lr6**	2353.3			2232.0
	ll6	2300.0	2200.0	2400.0	2200.0
MOI	**Ix**	1172865.6			260458.7
MOI	**Iy**	1660089.4			260458.7
Polar MOI	**J**	2832955.0			520917.3
Section 7					
Rectangle c/m	**a7**	70.0	60.0	100.0	60.0
	b7	120.0	60.0	100.0	60.0
	t7	2.5	2.0	5.0	2.0
Area	**A7**	925.0			464.0
c/m position	**lr7**	2842.5			2972.0
	ll7	2780.0			2940.0
MOI	**Ix**	798177.1			260458.7
MOI	**Iy**	1841927.1			260458.7
Polar MOI	**J**	2640104.2			520917.3
Section 8					
C-section l/m	**a8**	100.0	100.0	180.0	100.0
	b8	40.0	100.0	180.0	40.0
	t8	3.8	2.0	5.0	2.0
	e8	12.8			13.4
Area	**A8**	646.9			352.0
c/s position	**l8_start**	0.0			0.0

	l8_end	1000.0	1000.0	1250.0	1250.0
	L8	650.0			679.5
MOI	**Ix**	942485.4			531669.3
MOI	**Iy**	154360.4			90709.3
Polar MOI	**J**	1096845.7			622378.7
Section 9					
C-section l/m	**a9**	100.0	100.0	180.0	174.4
	b9	40.0	100.0	180.0	48.9
	t9	3.3	2.0	5.0	2.0
	e9	13.0			14.8
Area	**A9**	563.9			536.5
c/s position	**l9_start**	1000.0			1250.0
	l9_end	2000.0	2100.0	2300.0	2100.0
	L9	1000.0			850.0
MOI	**Ix**	830044.7			2278308.8
MOI	**Iy**	137534.7			226841.3
Polar MOI	**J**	967579.4			2505150.1
Section 10					
C-section l/m	**a10**	120.0	100.0	180.0	108.9
	b10	40.0	100.0	180.0	40.0
	t10	3.3	2.0	5.0	2.0
	e10	12.3			13.1
Area	**A10**	628.9			369.8
c/s position	**l10_start**	2000			2100
	l10_end	2900	3000	3300	3000
	L10	550			329.4605
MOI	**Ix**	1282210.4			649711.8
MOI	**Iy**	159538.5			97149.7
Polar MOI	**J**	1441749.0			746861.4

Optimization No. 2

Table B.2 Results of Optimization No. 2

		Initial_Values	Target Values		Optimised_Values
Weight Optimization	**Kg**	**75.1**			**56.3**
Performance					
Vertical Bending Stiffness (Sim.)	**N/mm**	836.1			2790.2
Vertical Bending Stiffness (LPM)	**N/mm**	**918.1**	**2800.0**		**2805.1**
Torsional Stiffness (Sim.)	**Nm/Deg**	3913.3			4177.9
Torsional Stiffness (LPM)	**Nm/Deg**	**3798.1**	**7000.0**		**7000.0**
Lateral Bending Stiffness (Sim.)	**N/mm**	1530.2			1687.8
Lateral Bending Stiffness (LPM)	**N/mm**	**980.4**	**1330.0**		**1339.7**
Overall Frame			**Range of values**		
			min	**max**	
Frame Length	**L**	2900			3000
Wheel base	**lb**	2200	1800	2200	1800
Frame width	**W**	1000	900	1200	1031.16399
Young's Modulus	**E**	210000			210000
Modulus of rigidity	**G**	81000			81000
Density	**kg/mm3**	7.90E-06			7.90E-06
Section 1					
Rectangle c/m	**A1**	60	60	80	60
	B1	120	80	120	80
	T1	3.75	2.5	3.75	2.5
Area	**A1**	1293.75			675
c/m position	**Lr1**	63.75			42.5
	Ll1	0			0
MOI	**Ix**	803408.2			400156.3
MOI	**Iy**	2410752.0			626406.3
Polar MOI	**J**	3214160.2			1026562.5
Section 2					
Rectangle c/m	**A2**	60.0	60.0	80.0	60.0
	B2	100.0	80.0	120.0	80.0
	T2	3.8	2.5	3.8	2.5
Area	**A2**	1143.8			675.0

c/m position	**Lr2**	533.8			532.5
	Ll2	500.0	350.0	500.0	500.0
MOI	**Ix**	1537392.6			626406.3
MOI	**Iy**	684580.1			400156.3
Polar MOI	**J**	2221972.7			1026562.5
Section 3					
Rectangle c/m	**A3**	70.0	60.0	80.0	60.0
	B3	100.0	80.0	120.0	80.0
	T3	3.8	2.5	3.8	2.5
Area	**A3**	1218.8			675.0
c/m position	**Lr3**	903.8			924.9
	Ll3	850.0	700.0	900.0	882.4
MOI	**Ix**	976416.0			400156.3
MOI	**Iy**	1711181.6			626406.3
Polar MOI	**J**	2687597.7			1026562.5
Section 4					
Rectangle c/m	**A4**	70.0	60.0	80.0	60.0
	B4	120.0	80.0	120.0	80.0
	T4	3.3	2.5	3.8	2.5
Area	**A4**	1192.8			675.0
c/m position	**Lr4**	1363.3			1351.3
	Ll4	1300.0	1300.0	1500.0	1308.8
MOI	**Ix**	1008213.8			400156.3
MOI	**Iy**	2342867.0			626406.3
Polar MOI	**J**	3351080.8			1026562.5
Section 5					
Rectangle c/m	**A5**	70.0	60.0	80.0	60.0
	B5	100.0	80.0	120.0	80.0
	T5	2.5	2.5	3.8	2.5
Area	**A5**	825.0			675.0
c/m position	**Lr5**	1752.5			1796.9
	Ll5	1700.0	1700.0	1900.0	1754.4

MOI	**Ix**	684218.8			400156.3
MOI	**Iy**	1189218.8			626406.3
Polar MOI	**J**	1873437.5			1026562.5
Section 6					
Rectangle c/m	**a6**	80.0	60.0	80.0	60.0
	b6	100.0	80.0	120.0	80.0
	t6	3.3	2.5	3.8	2.5
Area	**A6**	1127.8			675.0
c/m position	**lr6**	2353.3			2242.5
	ll6	2300.0	2200.0	2400.0	2200.0
MOI	**Ix**	1172865.6			400156.3
MOI	**Iy**	1660089.4			626406.3
Polar MOI	**J**	2832955.0			1026562.5
Section 7					
Rectangle c/m	**a7**	70.0	60.0	80.0	80.0
	b7	120.0	80.0	120.0	80.0
	t7	2.5	2.5	3.8	2.5
Area	**A7**	925.0			775.0
c/m position	**lr7**	2842.5			2962.5
	ll7	2780.0			2920.0
MOI	**Ix**	798177.1			776614.6
MOI	**Iy**	1841927.1			776614.6
Polar MOI	**J**	2640104.2			1553229.2
Section 8					
C-section l/m	**a8**	100.0	90.0	120.0	120.0
	b8	40.0	40.0	50.0	50.0
	t8	3.8	2.5	3.8	2.8
	e8	12.8			16.9
Area	**A8**	646.9			593.8
c/s position	**L8_start**	0.0			0.0
	L8_end	1000.0	1000.0	1250.0	1180.8
	L8	650.0			580.8

MOI	**Ix**	942485.4			1297583.8
MOI	**Iy**	154360.4			234647.9
Polar MOI	**J**	1096845.7			1532231.7
Section 9					
C-section I/m	**a9**	100.0	90.0	120.0	120.0
	b9	40.0	40.0	50.0	50.0
	t9	3.3	2.5	3.8	3.8
	e9	13.0			16.6
Area	**A9**	563.9			796.9
c/s position	**L9_start**	1000.0			1180.8
	L9_end	2000.0	2100.0	2300.0	2300.0
	L9	1000.0			1119.2
MOI	**Ix**	830044.7			1712329.1
MOI	**Iy**	137534.7			304223.6
Polar MOI	**J**	967579.4			2016552.7
Section 10					
C-section I/m	**a10**	120.0	90.0	120.0	120.0
	b10	40.0	40.0	50.0	50.0
	t10	3.3	2.5	3.8	3.8
	e10	12.3			16.6
Area	**A10**	628.9			796.9
c/s position	**l10_start**	2000.0			2300.0
	L10_end	2900.0	3000.0	3300.0	3000.0
	L10	550.0			100.0
MOI	**Ix**	1282210.4			1712329.1
MOI	**Iy**	159538.5			304223.6
Polar MOI	**J**	1441749.0			2016552.7
s					

Optimization No. 3

Table B.3 Results of Optimization No. 3

		Initial_Values	Target Values		Optimised_Values
Weight Optimization	**Kg**	**75.1**			**48.8**
Performance					
Vertical Bending Stiffness (Sim.)	**N/mm**	836.1			1368.2
Vertical Bending Stiffness (LPM)	**N/mm**	**918.1**	**1400.0**		**1401.7**
Torsional Stiffness (Sim.)	**Nm/Deg**	3913.3			2327.2
Torsional Stiffness (LPM)	**Nm/Deg**	**3798.1**	**3800.0**		**3800.0**
Lateral Bending Stiffness (Sim.)	**N/mm**	1530.2			732.3
Lateral Bending Stiffness (LPM)	**N/mm**	**980.4**	**930.0**		**936.5**
Overall Frame			**Range of values**		
			min	**max**	
Frame Length	**L**	2900			3000
Wheel base	**lb**	2200	1800	2200	1985.24691
Frame width	**W**	1000	900	1200	900
Young's Modulus	**E**	210000			210000
Modulus of rigidity	**G**	81000			81000
Density	**kg/mm3**	7.90E-06			7.90E-06
Section 1					
Rectangle c/m	**a1**	60	60	80	60
	b1	120	90	120	90
	t1	3.75	2.5	3.75	2.5
Area	**A1**	1293.75			725
c/m position	**lr1**	63.75			47.5
	ll1	0			0
MOI	**Ix**	803408.2			441510.4
MOI	**Iy**	2410752.0			830260.4
Polar MOI	**J**	3214160.2			1271770.8
Section 2					
Rectangle c/m	**a2**	60.0	60.0	80.0	60.0
	b2	100.0	90.0	120.0	90.0
	t2	3.8	2.5	3.8	2.5
Area	**A2**	1143.8			725.0
c/m position	**lr2**	533.8			532.5
	ll2	500.0	350.0	500.0	500.0

MOI	**Ix**	1537392.6			830260.4
MOI	**Iy**	684580.1			441510.4
Polar MOI	**J**	2221972.7			1271770.8
Section 3					
Rectangle c/m	**a3**	70.0	60.0	80.0	60.0
	b3	100.0	90.0	120.0	90.0
	t3	3.8	2.5	3.8	2.5
Area	**A3**	1218.8			725.0
c/m position	**lr3**	903.8			947.5
	ll3	850.0	700.0	900.0	900.0
MOI	**Ix**	976416.0			441510.4
MOI	**Iy**	1711181.6			830260.4
Polar MOI	**J**	2687597.7			1271770.8
Section 4					
Rectangle c/m	**a4**	70.0	60.0	80.0	60.0
	b4	120.0	90.0	120.0	90.0
	t4	3.3	2.5	3.8	2.5
Area	**A4**	1192.8			725.0
c/m position	**lr4**	1363.3			1359.0
	ll4	1300.0	1300.0	1500.0	1311.5
MOI	**Ix**	1008213.8			441510.4
MOI	**Iy**	2342867.0			830260.4
Polar MOI	**J**	3351080.8			1271770.8
Section 5					
Rectangle c/m	**a5**	70	60	80	60
	b5	100	90	120	90
	t5	2.5	2.5	3.75	2.5
Area	**A5**	825			725
c/m position	**lr5**	1752.5			1820.4
	ll5	1700.0	1700.0	1900.0	1772.9
MOI	**Ix**	684218.8			441510.4
MOI	**Iy**	1189218.8			830260.4
Polar MOI	**J**	1873437.5			1271770.8
Section 6					
Rectangle c/m	**a6**	80.0	60.0	80.0	80.0
	b6	100.0	90.0	120.0	90.0
	t6	3.3	2.5	3.8	2.5

Area	**A6**	1127.8			825.0
c/m position	**lr6**	2353.3			2247.5
	ll6	2300.0	2200.0	2400.0	2200.0
MOI	**Ix**	1172865.6			851718.8
MOI	**Iy**	1660089.4			1021718.8
Polar MOI	**J**	2832955.0			1873437.5
Section 7					
Rectangle c/m	**a7**	70.0	60.0	80.0	80.0
	b7	120.0	80.0	120.0	80.0
	t7	2.5	2.5	3.8	2.5
Area	**A7**	925.0			775.0
c/m position	**lr7**	2842.5			2962.5
	ll7	2780.0			2920.0
MOI	**Ix**	798177.1			776614.6
MOI	**Iy**	1841927.1			776614.6
Polar MOI	**J**	2640104.2			1553229.2
Section 8					
C-section l/m	**a8**	100.0	90.0	120.0	114.7
	b8	40.0	40.0	50.0	40.0
	t8	3.8	2.5	3.8	2.5
	e8	12.8			12.8
Area	**A8**	646.9			474.3
c/s position	**l8_start**	0.0			0.0
	l8_end	1000.0	1000.0	1250.0	1250.0
	L8	650.0			742.6
MOI	**Ix**	942485.4			904655.3
MOI	**Iy**	154360.4			123229.3
Polar MOI	**J**	1096845.7			1027884.5
Section 9					
C-section l/m	**a9**	100.0	90.0	120.0	119.6
	b9	40.0	40.0	50.0	47.2
	t9	3.3	2.5	3.8	2.5
	e9	13.0			15.7
Area	**A9**	563.9			522.4
c/s position	**l9_start**	1000.0			1250.0
	l9_end	2000.0	2100.0	2300.0	2100.0
	L9	1000.0			850.0

MOI	**Ix**	830044.7			1122911.6
MOI	**Iy**	137534.7			186796.8
Polar MOI	**J**	967579.4			1309708.3
Section 10					
C-section l/m	**a10**	120.0	90.0	120.0	120.0
	b10	40.0	40.0	50.0	40.0
	t10	3.3	2.5	3.8	2.5
	e10	12.3			12.6
Area	**A10**	628.9			487.5
c/s position	**l10_start**	2000.0			2100.0
	l10_end	2900.0	3000.0	3300.0	3000.0
	L10	550.0			392.6
MOI	**Ix**	1282210.4			1007265.6
MOI	**Iy**	159538.5			127890.6
Polar MOI	**J**	1441749.0			1135156.3

REFERENCES

1. S. Ravishankar and C. Anil Kumar, TELCO, Pune ”*Crash Test Facility at Telco*” **(SAE-990031)**

2. V.J. Borovoski, R.L. Steury and J.L. Lubkin, FORD MOTOR COMPANY “*Finite element dynamic analysis of an automotive frame*.” **(SAE-730506)**

3. D.W. Sherman, Dana Corp. *“Stresses and deflections in Truck Long-member Attachments.*” **(SAE-690175)**

4. C.V. CROCKETTE and D.J. LaBELLE. ”*A new concept of Light Weight Highway Tractor Design.”* **(SAE-590281)**

5. Jay K. HAY and J. Michael Blew “*Dynamic testing and computer analysis of Automotive Frames.*” **(SAE-720046)**

6. Oskara Michejda **“***Truck Frame Analysis Study.”* **(SAE-710594)**

7. Richard L. Exler, Chrysler Corp. *“Frame Rigidity How much and where.”* **(SAE-640017)**

8 Kenji KARITA, Yoichiro KOHIYAMA, Toshihiko KOBIKI, Kiyoshi OOSHIMA and Mamoru HASHIMOTO .”*Development of Aluminum Frame for Heavy-Duty Trucks”* **(Technical review-2003)**

9. Michael W. Danyo, Christopher S. Young, Henry J. Cornille and Joseph Porcari, Ford Motor Company *“A Design Concept for an Aluminum Sport Utility Vehicle Frame”* ***(*SAE-2003-01-0572)**

10. Hiroshi Takata, Mitsuo Iwahara and Akio Nagamatsu *“An Analysis of Idling Vibration for a Frame Structured Vehicle.”* **(SAE- 2003-01-1611)**

11. Cicek Karaoglu .and N. Sefa Kuralay *“Stress analysis of a truck chassis with riveted joints.”***Elsevier Science-**Finite Elements in Analysis and Design (**38 (2002) 1115–1130)**

12. Heung-Soo Kim and Tomasz Wierzbicki. “*Effect of the cross-sectional shape of hat-type cross-sections on crash resistance of an “S”-frame.”* Elsevier Science- Thin-Walled Structures (**39 (2001) 535–554)**

13.A.R. Kaduk, G. Mladsi and E.D. Clise . *"Selection of Frames, Springs, and Axles for Utility Vehicles."* **(SAE-690563)**

14.Kunihiro Takahashi, Nissan Motors Co. ltd. "*A torsional strength analysis of Truck Frames using Open Section members"* **(SAE-710595)**

15. M.R. Mitchell and R.M. *Wetzel "Cumulative Fatigue Damage Analysis of a Light Truck Frame."* **(SAE-750966)**

16. Wayne A. McClelland, Jay K. Hay, and Albert L. Klosterman, Structural dynamics corporation. *"Frame Design Analysis under Complete Vehicle Boundary conditions."* **(SAE-741142)**

17 H. Shirasawa, J. Jizaimaru, T. Mizoguchi and N. Tada .*" Fatigue Life estimation on HSLA Chassis Frame."* **(SAE-810358)**

18. Lewis F. McNitt, Midland Ross Corp. "*Truck Frame long-member Buckling stresses*." **(SAE-690176)**

19. R. J. Yang, T. J. Walsh and P.A. Schilke, Ford motor Co. "*A general formulation for Topology Optimization ".* **(SAE-942256)**

20. Cheon-Hong Jeong, Nak-Seung Jung, In-Ho Choi and Seog-ju Cha,Hyundai Motor Company "*The studies of Crash Characteristics according to Chassis Frame types"*. **(SAE-2001-01-0119)**

21 T. Dutton, S. Iregbu, R. Sturt, A. Kellict, B. Cowell and K. Kavikondala The effect of *Forming on the Crashworthiness of vehicles with Hydroformed Frame Long-members* .. **(SAE- 1999-01-3208)**

22. William J. Sidelko, Ford Motor Co. *"An objective approach to Highway truck Frame Design."* **(SAE-660162)**

23. Carlos Cosme, Amir Ghasemi and Jimmy Gandevia, Western Star Trucks, Inc. "*Application of Computer Aided Engineering in the Design of Heavy-duty Truck Frames*." (SAE- 1999-01-3760)

24. Edmund F. Gaffney III and Anthony R. Sallinas *Introduction of Formula SAE Suspension and Frame Design*. **(SAE- 971584)**

25. Zheng-Dong Ma, Hui Wang, Noboru Kikuchi, Christophe Pierre and Basavaraju Raju "*Substructure design using a Multi-Domain Multi-Step Topology Optimization approach.*(SAE- 2003-01-1303)

26. Abraham Tijerin and Jonathan Wortle "*Model Flexibility and Part Integration Concept for next generation Pickup and SUV Frames.*" **(SAE- 2005-01-1026)**

27. William B. Rileyand Albert R. George *"Design, Analysis and Testing of Formula SAE car."*

28.John Fenton "*Vehicle body layout and analysis*"

29. "*An Introduction to modern Vehicle Design*" SAE publication

30. William T. Thomson *"Theory of Vibration with Applications"*

31. Kripal Singh *"Automobile engineering"*

32. Philip J. Ross "*Taguchi Techniques for Quality Engineering"*

33. Robert F. Brewer *"Design of Experiments for process Improvement and Quality assurance."*