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DYNAMIC ANALYSIS AND DESIGN OPTIMIZATION OF LADDER CHASSIS

KANDI ASHOK
P.G Scholar, QIS College of Engineering & Technology, Ongole.
Dr. D. ELAYARAJA
Professor, Department of M.E, QIS College of Engineering & Technology, Ongole.

Abstract - The chassis is the main structure of the automobile. All other systems like transmission, suspension etc.., are attached to it. The chassis provide strength as well as flexibility to automobile. When the vehicle travels along the road, the chassis is subjected to excitation from the engine and transmission system due to road profile. Due to excitations, the chassis begins to vibrate. If the natural frequency of the vibration concedes with the frequency of external excitation, resonance occurs. The resonance leads to excessive deflection and failure of the chassis. The chassis type used in sports utility vehicle and trucks are ladder type. This project work will address the ladder type chassis failure by conducting the dynamic analysis of the chassis. The chassis design will be modified and optimized for Indian road conditions in such a way that the vibration of the chassis will be within the limit. Harmonic analysis will be conducted on the original and optimized chassis to check the response under a harmonic force.

Keywords - Chassis, Static Stiffness, Dynamic analysis of Road loads, Optimization.

I. INTRODUCTION

Ladder Chassis: The chassis is the framework of any vehicle. The suspension, steering, and drive train components such as engine, transmission, and final drive components are mounted to the chassis. The chassis would have to be strong and rigid platform to support the suspension components. Furthermore, the constructions of today are vehicles require the use of many different materials. Chassis of go-kart is not much different from normal car chassis; in fact, it is much less complicated. This type of frame is common for the type of perimeter frame where the transversely (lateral) connected members are straight across. Figure below shown as ladder frame sample where viewed with the body removed. The frame resembled a ladder viewed from top.

Lane Change Maneuver: The review and analysis of the use of "lane change" maneuvers to evaluate body stiffness and vehicle performance. A successful lane change is defined as any vehicle maneuvers in which steering is first applied in one direction (either to the right or to the left) to displace the vehicle laterally and then the steering is reversed in the other direction while maintaining directional control and recovering the original direction of travel at a lateral displacement of approximately one lane width(i.e., approximately 12 feet).Numerous investigations of vehicle directional performance have included some form of this type of maneuver. However, no generally accepted methodologies or procedures for conducting lane-change maneuvers have evolved. Three basic reasons exist for selecting particular maneuvers:

- The geometry of the manoeuvre emphasizes an important facet of vehicle control or performance.
- The manoeuvre is representative of manoeuvres frequently performed on the highway.
- Vehicle performance obtained in the manoeuvre varies significantly from one

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vehicle to another (i.e., the manoeuvre discriminates between vehicles).

$$\frac{1}{R(t)} = \frac{r(t)+\beta'(t)}{V}$$

1/R = the path curvature (a positive polarity means a right turn).

R = the yaw rate

β = the side slip angle

V = velocity

The variables r and β constitute the two modes of rotation illustrated in Figure. As shown in Figure, the yaw rate is the rate of change of the orientation of the vehicle's x-body axis (an axis out the front window) with respect to an axis fixed on the road (for example, the centerline of the road). The sideslip angle, β, is the angle between the vehicle's x-body axis and the velocity vector of the vehicle's centre of mass, that is, the direction of the tangent to the path of the centre of mass. From Equation (1) it can be seen that the crucial reversal in path curvature, which is required in a successful lane change, occurs at the time when,

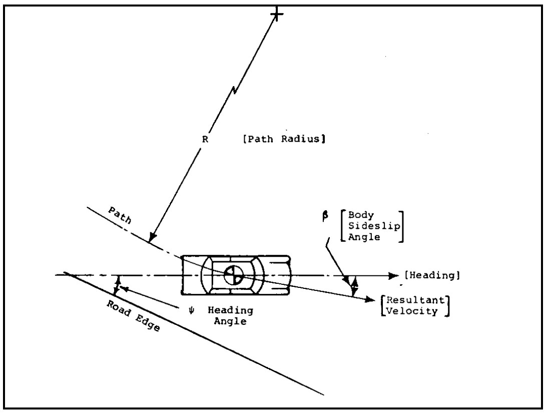

Figure: Lane Change Maneuver

Static Stiffness: Global car body stiffness is a crucial design attribute in vehicle design. Accurate Frame structural identification, including global static stiffness identification is therefore of high importance. Increasingly CAE techniques are used in this regard. Nevertheless, experimental car chassis structural identification is required to verify and update structural finite element models. In automotive industry different tests are performed, starting from static deformation tests, experimental modal analysis to operational testing on laboratory test benches and therefore the road.

Global dynamic stiffness characterization is an elementary a part of this and is decided by an experimental modal analysis test. These dynamic tests are used for target verification, troubleshooting and finite element model updating. For Frame testing, measurements are performed under so-called free-free boundary conditions, which suggest that the car frame is decoupled from the environment. The sensible realization of this condition is well defined and realized by hanging the structure or mounting it on very soft springs.

Main advantages of this sort of testing are the great consistency with which these free-free boundary conditions are often realized and therefore the relatively low influence of small changes within the test set-up, resulting in high

The global static stiffness is measured on a static deformation test bench. Different load cases are available. This work is going to be limited to global static bending and global static torsional stiffness determination. During a worldwide bending test, forces are applied at the front seat locations, while the body is constrained at front and rear shock towers. The static bending stiffness results from the ratio of the applied load to the utmost deflection along the rocker panel and tunnel beams.

For global static torsion stiffness, a static moment is applied to the body-in-white at the front shock towers, whereas the rear shock towers are constrained.

The force method simulates a static test conducted on a rig during which a flash applied on the auto body. During this method, forces that are equal in magnitude and opposite in direction are applied at the front suspension mounting locations in the model, while keeping the rear mounting locations constrained. The total angular deflection of the structure (θ) is based on the vertical deflections and thus the width. The torque T, is represented by the vertical force applied at the mounting locations. The vertical deflections at the left-front and right-front locations are measured.

The displacement method, vertical displacement is applied at one location, for instance the left-front location, instead of applying equal and opposite forces at both the front mounting locations. The displacement is gradually increased until a predetermined maximum value. The applied displacement Δ, will produce a torque about the longitudinal axis (X-axis). The reaction force can be retrieved and used in estimating the torsional stiffness. Generally, the location where no displacement is applied (front right in this case) is unconstrained. This is done to simulate a static test in which jack-screw actuators are used to apply displacement at one location, while the other location is resting on a knife edge. The predicted torsional stiffness from this method is typically lower than the predictions from the force method as the right front location is unconstrained.

Stiffness is defined as force required to supply unit displacement. It depends on Geometry as well as material properties.

$$\text{Stiffness } K = F \text{ (Force)}/D \text{ (Displacement)}$$

$$\text{Tensile Stiffness} = K_{tension} = AE/L$$

$$\text{Bending Stiffness} = K_{bending} = 3EI/L^3$$

$$\text{Torsional stiffness} = K_{torsion} = GJ/L$$

II. METHODOLOGY

Static Bending rigid condition: Chassis is constrained at the Rear shock points and Front shock points and the load is applied at the center of the chassis to bend it.

Static Torsion rigid condition: Chassis is constrained at the Rear shock points and the load is applied at the front shock points in order to introduce torsional load.

Normal modal Analysis: Modal analysis is the process of extracting modal parameters like frequency and mode shape of the system. Mode of a structure is defined as mode shape also called Eigen vector and frequency of vibration also called Eigen value. Global Modes are predicted for the chassis FE model like bending, torsion and lateral modes.

The natural frequency of a system is the frequency the system vibrates at which it is disturbed. The natural frequency of a system depends on the elasticity and shape.

If the vibrating system is driven by an external force at the frequency at which the amplitude of its motion is greater or close to the natural frequency of the system is called resonant frequency.

A mode shape is the dynamic property of the structure, it represents a pattern of structural deflection model that corresponding to each natural frequency. Different mode shapes will be associates with different frequencies.

Dynamic Analysis: Road loads are measured at the mounting points and are applied at the chassis mounting points under 20 to 50 Hz, where the global modes are observed. Modal frequency response analysis method is used to measure the displacements under application of road loads at 20 to 50 Hz, here the dynamic stiffness is calculated at the mounting points.

III. MODELLING & ANALYSIS OF EXISTING CHASSIS

The main objective of the project is to find the global stiffness of the car frame subjected to considered maneuvers by CAE simulations. The stiffness of car frame varies with maneuvering conditions and it affects the ride comfort of the passengers.

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Design of Chassis: The chassis parts that are required are modeled in SOLIDWORKS which is one of the best CAD software's, where the solid and surface modeling of the complex features is easy to model in solid works and its user friendly. The CAD models are generated and exported in the is IGES format and imported to the preprocessing software ANSA where the meshing is done on the CAD model. The different parts of the CAD generated parts are assembled in the preprocessing software's using the welds and it is solved using NASTRAN and post processed in Metapost, the three steps in analyzing the CAD are: Preprocessing: The CAD geometry design like the physical bounds of the CAD models are defined. The IGES model is meshed; the meshed part is uniform or non-uniform. The CAD physical meshing is defined. The Boundary conditions are defined on the FE Model. This CAD with assigned boundary conditions involves specifying the behavior of chassis and properties of CAD model at the boundaries of the problem. For transient dynamic problems, the problem initial conditions are defined for the solver.

The NASTRAN solving is initiated and the dynamic equations of equilibrium are solved in step by step iterations as a transient and steady-state. Finally, after the solving of the CAD model, the postprocessor metapost is used for visualization of the response of the structures. The CAD modeling in the solid works are described as follows, First go to sketcher for making the design of the cross section of the extruding parts, the cross section of the part is designed and the dimensions are set depending in the requirements given by the designer. The designed cross section is extruded along the distance in the direction perpendicular to the cross section and the chassis parts are modeled.

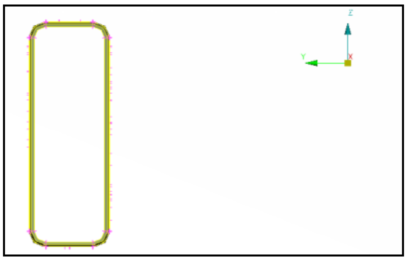

Figure: Cross Section of the Rail member

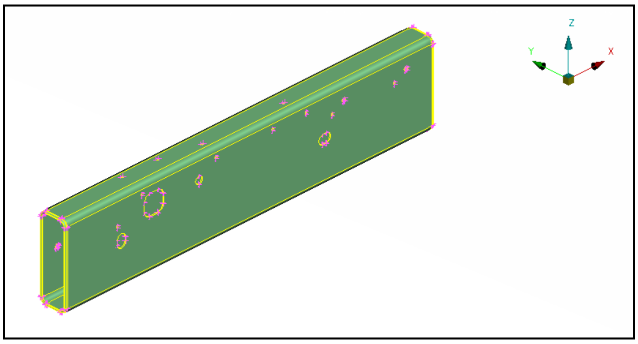

Figure: Extrusion of the cross section

The chassis cross member is designed using the multiple cross sections at the multiple stages of the member design along the Y-axis and the cross sections are connected using the generative profile to get the better design, then the holes and the other design modifications are modeled and on the surface of the generated cross member.

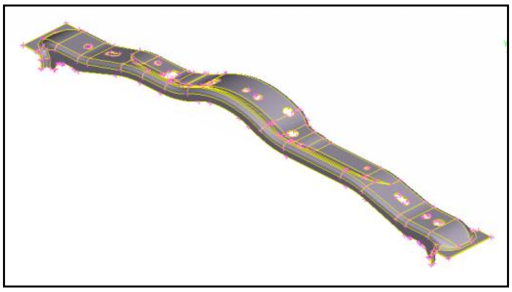

Figure: Cross member CAD design

The CAD model of the complete chassis is shown below:

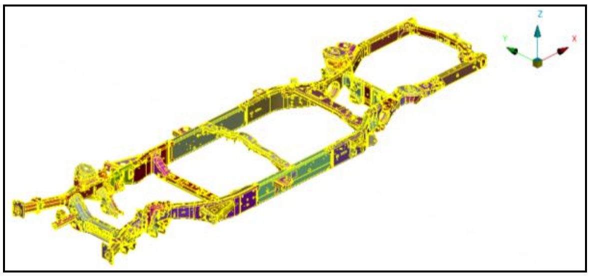

Figure: Chassis CAD Model

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Modelling of Existing Chassis: Chassis is modeled using PSHELL elements with average element size of 18 mm and connections are defined using Spot welds represented as RBE3 – HEXA – RBE3 and Seam Welds represented as Node – to – Node RBE2.

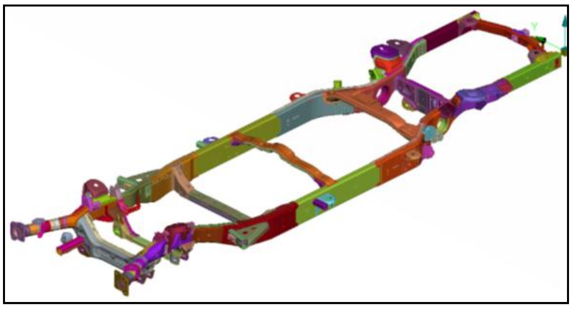

Figure: Chassis FE Model

Material Properties: Material of chassis is Mild Steel E = 206 GPa. Density = 7.83e-09 Poisson's Ratio = 0.3

Chassis Static Stiffness Analysis: Stiffness improvement calculations are carried out in CAE as follows; Lane change load data is converted from Time domain to Frequency domain using FFT (Fast Fourier Transform). Numerical analysis is carried out on the chassis model; the static stiffness is measured using the bending and static rig.

- Static bending stiffness of chassis is observed at **2.94 KN/mm.**
- Static torsion stiffness of chassis is observed at **20.65 KN-mm/deg**.

Static Bending rigid condition:

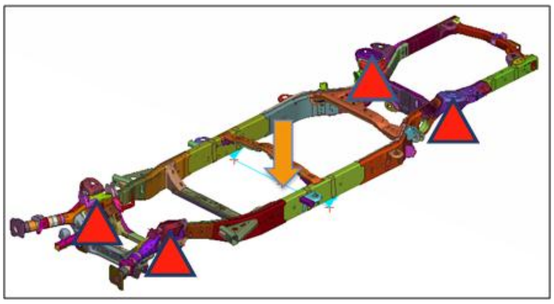

Figure: Chassis Bending Rigid Setup

Front shock LT DOF (3) is constrained
Front shock RT DOF (23) is constrained
Rear shock LT DOF (13) is constrained
Rear shock LT DOF (123) is constrained
1KN Load applied at the center of the chassis

Static Torsion rigid condition:

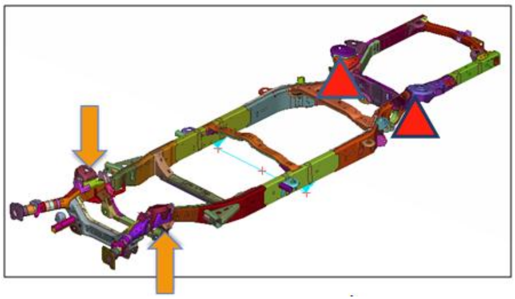

Figure: Chassis Torsion Rigid Setup

+1KN Load applied at Front shock LT
-1KN Load applied at Front shock LT
Rear shock LT DOF (123) is constrained
Rear shock LT DOF (13) is constrained

Modal Analysis of Existing chassis: Normal Mode Study is carried out on the chassis the global modes are observed to be at the below mentioned frequencies.

Global Bending Mode:

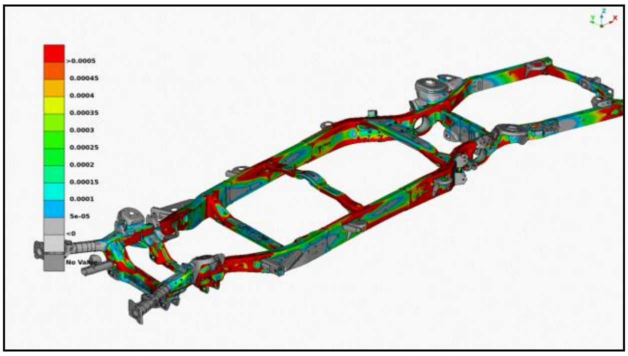

Figure: Baseline Bending Mode

- Global Bending Mode at **25.9 Hz.**

Global Lateral Mode:

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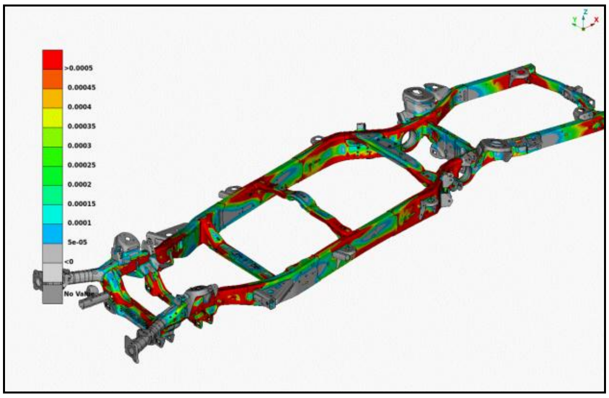

Figure: Baseline Lateral Mode

- Global Lateral Mode at **27.5 Hz.**

Global Torsion Mode:

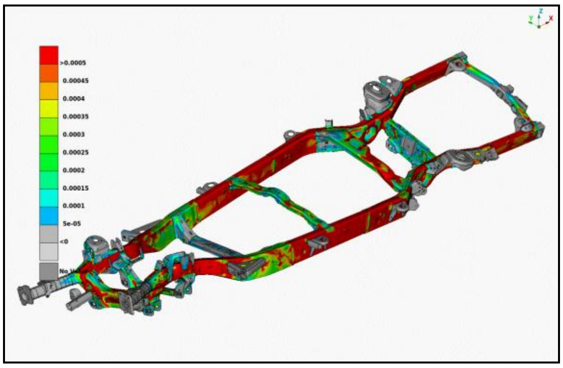

Figure: Baseline Torsion Mode

- Global Torsion Mode at **24.3 Hz.**

Dynamic Stiffness Analysis of Existing Chassis:

In dynamic analysis the loads which vary with respect to time is used. Here the chassis of sedan model is constructed to perform dynamic analysis. The time domain loads collected during physical during under lane change maneuvering condition at varies location of sedan model is obtained. The lane change maneuvers loads are applied at attachment points to find response of the chassis.

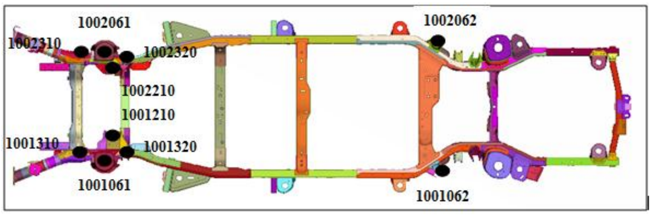

Figure: Chassis Mounting Points

Dynamic response for Existing chassis mounting points:

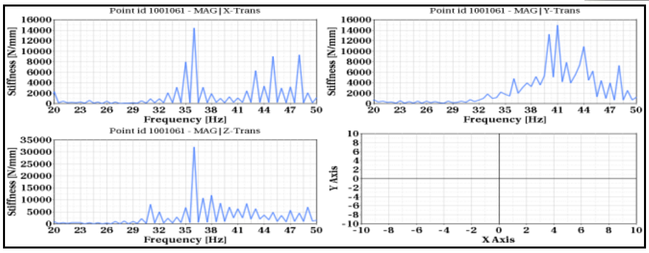

Figure: Engine Mount LHS (1001061)

Rear Sub frame mount LT dynamic response is shown below

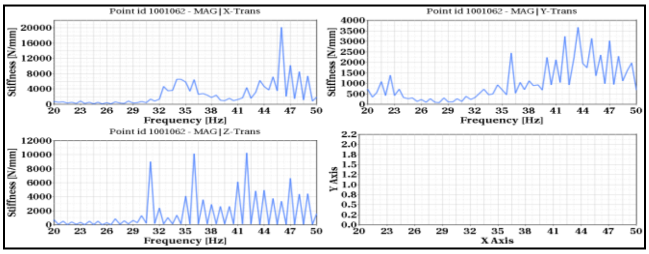

Figure: Rear Sub frame Mount LHS (1001062)

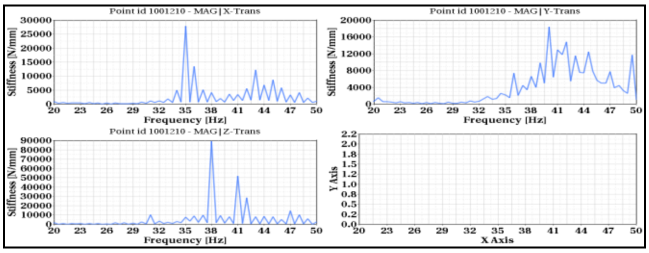

Figure: Front Sub frame Mount LHS (1001210)

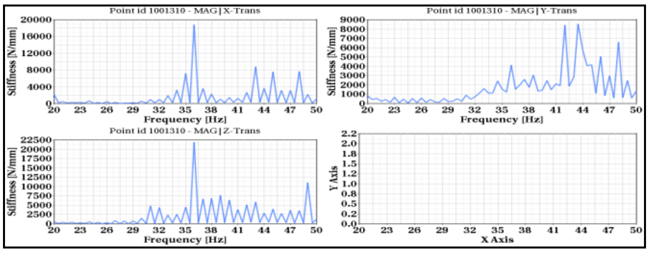

Figure: Front Upper Link LHS (1001310)

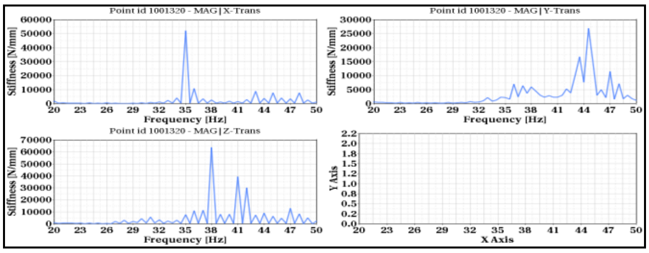

Figure: Rear Upper Link LHS (1001320)

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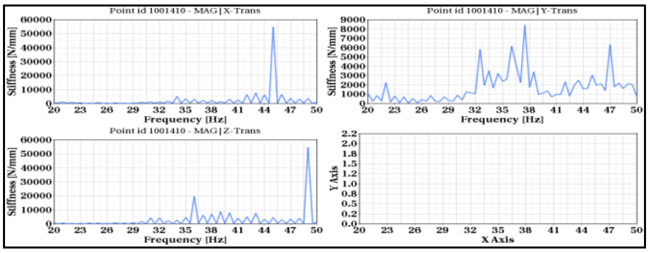

Figure: Front Lower Link LHS (1001410)

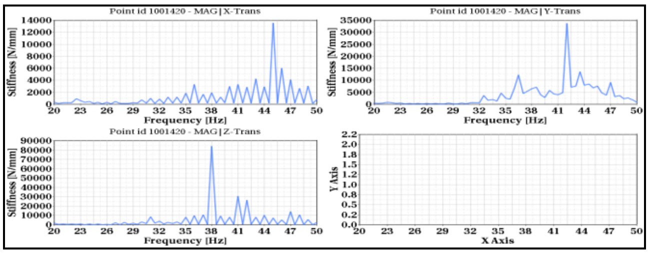

Figure: Rear Lower Link LHS (1001420)

IV. MODELLING & ANALYSIS OF MODIFIED CHASSIS

Global Bending strain energy plots show strain energy concentration at the mid rail between the Engine mount and the mid cross member, where it shows a need of locations for strengthening the structure by introducing the bulk heads.

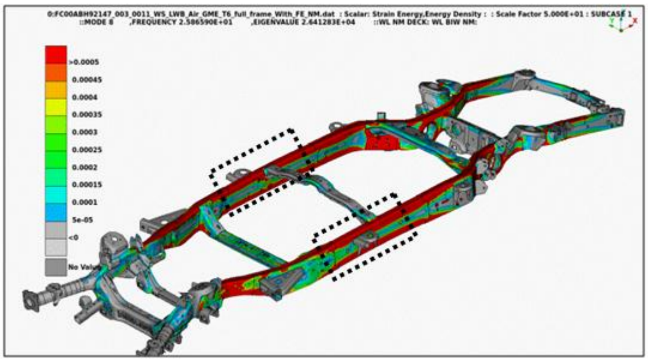

Figure: Bending Mode Failure

Bending mode deflections are majorly observed at the mid portion of the chassis / mid rail, with introduction of bulk heads at the mid rail area will improve the bending stiffness. The bulk heads placement and its thickness are mentioned below.

Chassis mid rail bulk heads are shown below:

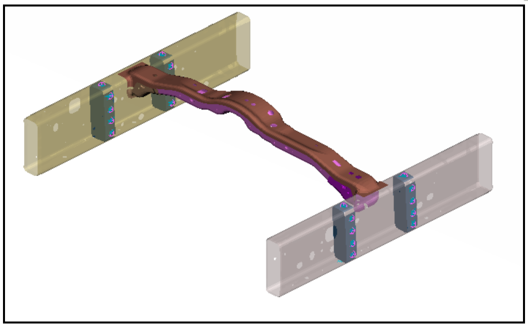

Figure: Mid Rail Bulk heads from Normal Mode Analysis

The mid rail bulk heads are optimized to a thickness of **4.5 mm.**

Global Lateral strain energy plots show strain energy concentration at the Front rail between the Engine mount and the Front cross member, where it shows a need of locations for strengthening the structure by introducing the bulk heads.

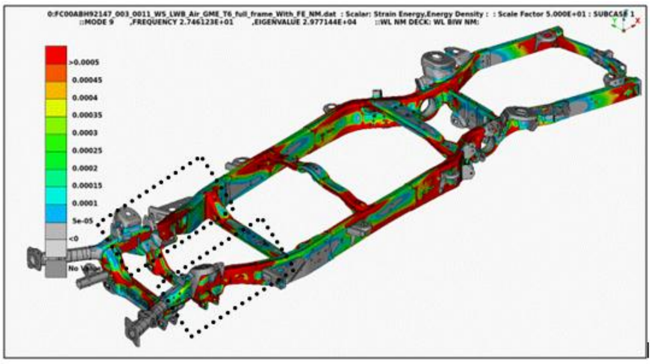

Figure: Lateral Mode Failure

Lateral mode deflections are majorly observed at the front portion of the chassis / front rail, with introduction of bulk heads at the front rail area will improve the lateral stiffness. The bulk heads placement and its thickness are mentioned below.

Chassis front rail bulk heads are shown below:

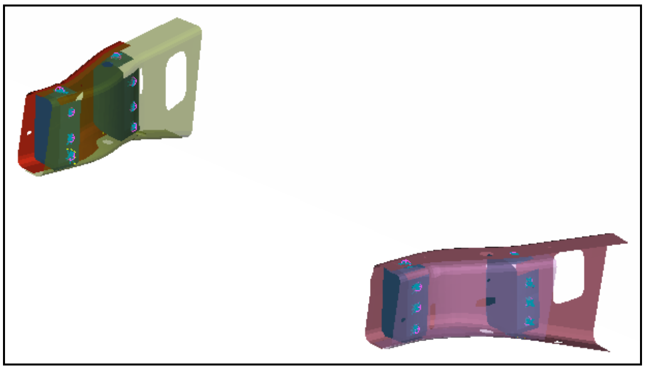

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Figure: Front Rail Bulk heads from Normal Mode Analysis

The front rail bulk heads are optimized to a thickness of **4 mm.**

Global Torsion strain energy plots show strain energy concentration at the front rail and rear rail between the Engine mount, front cross member and the rear shock mount and rear cross member, where it shows a need of locations for strengthening the structure by introducing the bulk heads.

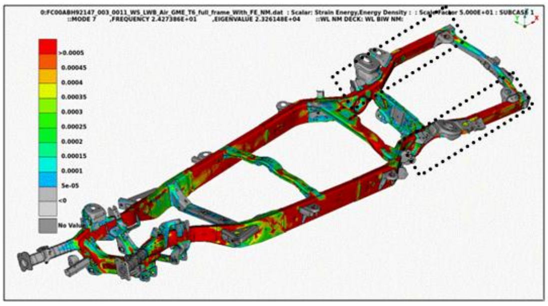

Figure: Torsion Mode Failure

Torsion mode deflections are majorly observed at the front and rear portion of the chassis / front and rear rail, with introduction of bulk heads at the front rail area will improve the torsional stiffness. The torsional and lateral stiffness are interrelated so, the bulk heads introduced in the front rail with lateral mode are used to satisfy the torsional mode. The bulk heads placement and its thickness are mentioned below.

Chassis rear rail bulk heads are shown below:

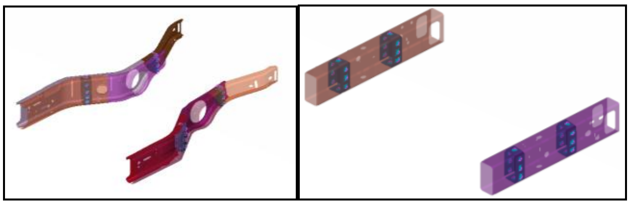

Figure: Rear Rail Bulk heads from Normal Mode Analysis

The rear rail bulk heads are optimized to a thickness of **5.0 mm.**

Modified Chassis:

Bulk heads are introduced at the Front, mid and rear rails to improve the stiffness and modal performance of the chassis. Gauge optimization is carried out using Optistruct to find out the thickness of the bulk heads,

- The front rail bulk heads are optimized to a thickness of 4 mm.
- The mid rail bulk heads are optimized to a thickness of 4.5 mm
- The rear rail bulk heads are optimized to a thickness of 5 mm.

Below figure shows the bulk head locations in the chassis depending on the Global modal analysis contours.

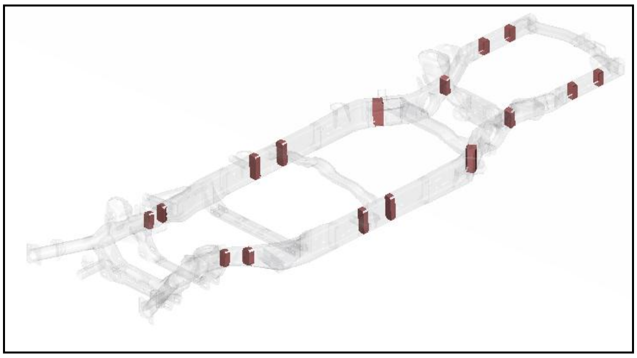

Figure: Bulk heads from Normal Mode Analysis

Modal Analysis of Modified Chassis:

The Normal modes improvement with introduction of Bulk heads is shown below:

Existing Chassis bending mode is observed at 25.9 Hz and the modified chassis model shows mode at 26.5 Hz.

Modified bending mode is shown as below:

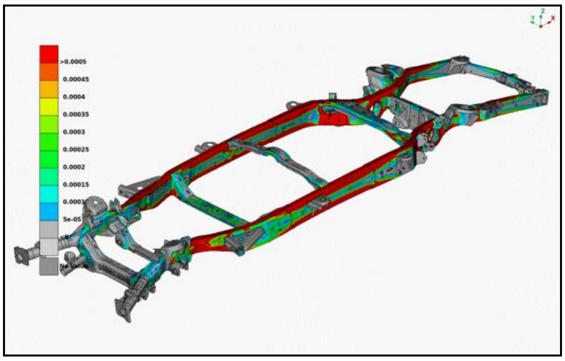

Figure: Modified Bending Mode

Existing Chassis lateral mode is observed at 27.5 Hz and the modified chassis model shows mode at 29 Hz.

Modified lateral mode is shown as below:

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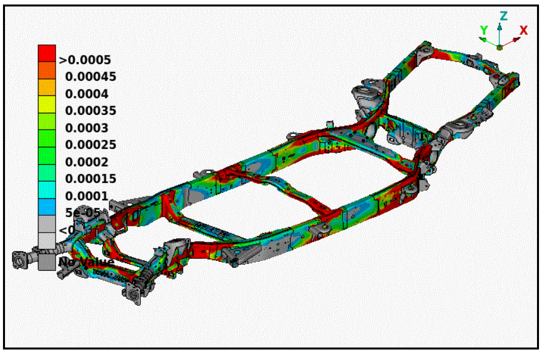

Figure: Modified Lateral Mode

Existing Chassis Torsion mode is observed at 24.3 Hz and the modified chassis model shows mode at 27 Hz.

Modified Torsion mode is shown as below:

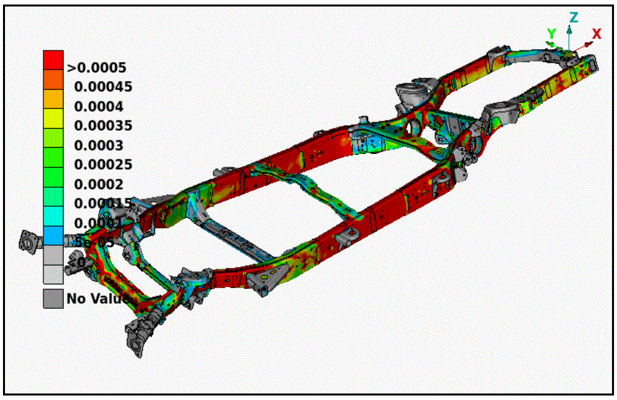

Figure: Optimal Torsion Mode

With implementation of the bulk heads the global modes are improved, bending mode improves from 25.9 Hz to 26.5 Hz by 0.6 Hz; torsion mode improves from 24.3 Hz to 27.0 Hz by 2.7 Hz; lateral mode improves from 27.5 Hz to 29.0 Hz by 1.5 Hz.

V. MODIFIED CHASSIS ROAD LOAD RESPONSE:

Road loads are applied on the modified model with bulk heads and the comparison between the stiffness between modified model and Existing model are shown below.

Lane change load stiffness response at the attachment points mentioned above is depicted below; frequency of interest depends on the global mode of chassis frequency range from 20 to 50Hz. Modified chassis frequency response curves are mentioned in black color, as the Existing chassis curves are in blue color. The modified model with bulk heads shows better stiffness in Y and Z directions compared to Existing due to front rail bulk heads implementation.

Dynamic Analysis of Modified chassis:

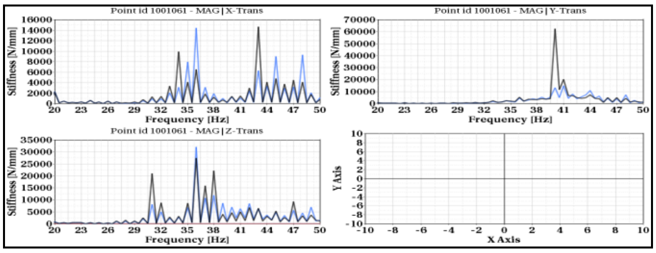

Figure: Modified Engine Mount LHS (1001061)

The Modified model with bulk heads shows better stiffness in Z directions compared to Existing due to rear rail bulk heads implementation.

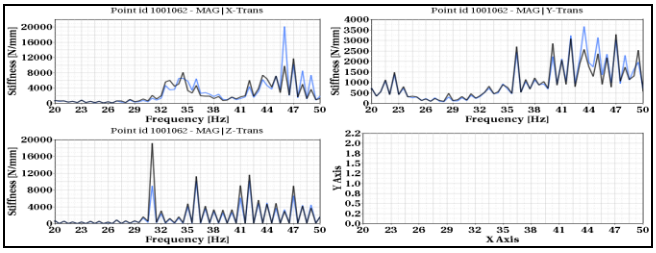

Figure: Modified Rear Sub Frame Mount LHS (1001062)

The Modified model with bulk heads shows better stiffness in Y and Z directions compared to Existing due to front rail bulk heads implementation.

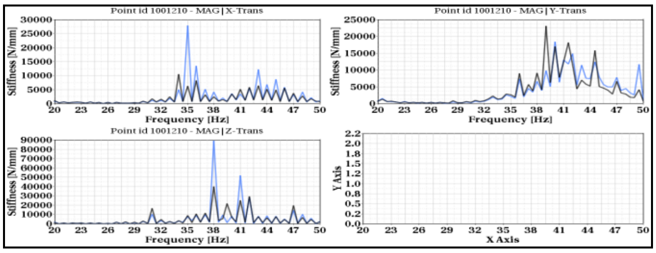

Figure: Modified Front Sub Frame Mount LHS (1001210)

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The Modified model with bulk heads shows better stiffness in X and Z directions compared to Existing due to front rail bulk heads implementation.

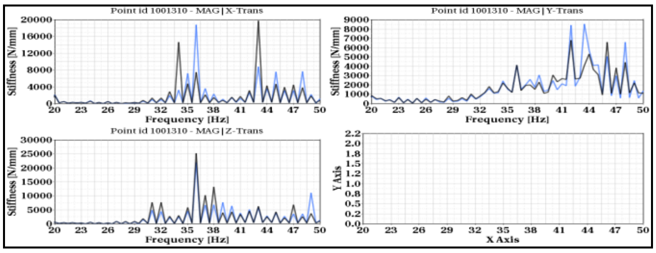

Figure: Modified Front Upper Link LHS (1001310)

The Modified model with bulk heads shows better stiffness in X, Y and Z directions compared to Existing due to front rail bulk heads implementation.

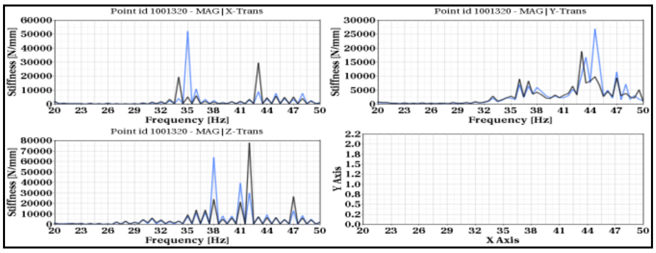

Figure: Modified Rear Upper Link LHS (1001320)

The Modified model with bulk heads shows better stiffness in Y and Z directions compared to Existing due to front rail bulk heads implementation.

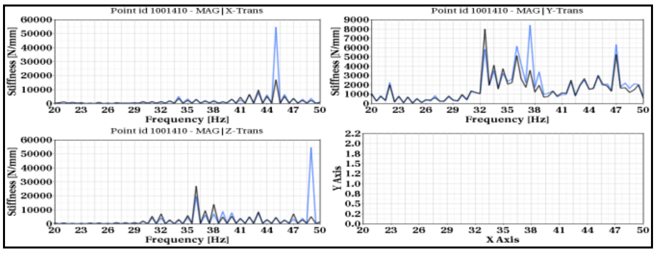

Figure: Modified Front Lower Link LHS (1001410)

The Modified model with bulk heads shows better stiffness in Y and Z directions compared to Existing due to front rail bulk heads implementation.

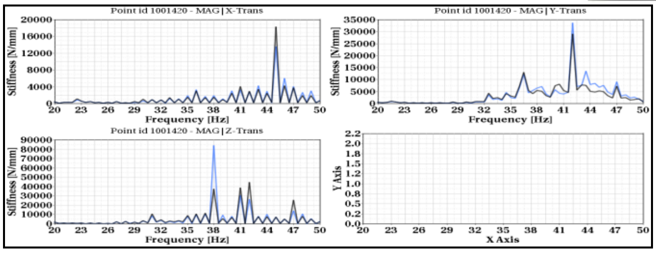

Figure: Modified Rear Lower Link LHS (1001410)

VI. RESULTS AND DISCUSSION

It is observed that by application of road loads on the Modified model shows stiffness improvement compared to the Existing model in all the directions for all the response points. Optimization is also carried out for the optimal weight of the chassis. After reviewed the literature review it is obvious that all the researchers worked on the ladder chassis for heavy vehicles like trucks and buses. Only few researchers have done dynamic analysis on the ladder chassis for passenger vehicles and design optimization carried.

COMPARISSION OF EXISTING AND OPTIMIZED CHASSIS STATIC STIFFNESS:

Static Bending and Torsion responses for the optimized chassis compared to Existing chassis are tabulated below:

Stiffness	Existing Chassis	Chassis Optimized with bulk heads	Difference between Optimized and Existing
Static Bending (KN/mm)	2.94	4.32	1.38
Static Torsion (KN-mm/ deg)	20.65	23.50	2.85

Static bending shows an improvement by 1.38 KN/mm and static torsion shows an improvement by 2.85 KN-mm/ deg

VII. CONCLUSION

In the existing model of chassis, we have observed high strain energy concentration at front, Mid, Rear Rail which intern affecting the mounting point stiffness over the global mode frequency range.

Prediction of high deflection region with Normal modes load case and the development of bulk heads is done to improve the stiffness of chassis where the static and dynamic stiffness improvement has been observed, gauge Optimization is also carried out for the optimal weight of the chassis.

VIII.FUTURE SCOPE

Chassis structure is further improved by subjecting it to the weight optimization studies and the crash loads and durability loads with MDO (Multi Disciplinary Optimization) studies.

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