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# Static and Harmonic Analysis of Truck Chassis with Tuned Mass Damper under Class-B Road Conditions

Mahendra varma.M[1], Lakshmana kishore.T[2]
*P.G student, Mechanical Engineering[1], Assistant Professor, Department of Mechanical Engineering[2]*
*[1,2]JNTUK University College of Engineering, Vizianagaram-535003, Andhra Pradesh, India*
*Email: mahi.ahabonisa@gmail.com[1], kishore.jntuk@gmail.com[2]*

**Abstract-**Chassis is major vital component in a vehicle structure. The main theme of this project involves static structural and harmonic analysis to investigate the key characteristics of a truck chassis .The static response includes identification of failure zone location and high stress concentration area and also determines the deformation of the truck chassis. Finite element analysis (FEA) is used to determine the dynamic response parameters of truck chassis such as the natural frequencies and mode shapes. Numerical analysis was done to validate with the FEA models. Design modification of the chassis model was done by placing tune mass damper to optimize the vibrations of the truck chassis and to reduce the chances of failure. The main area which is to be analyzed in the existing truck chassis model was the structural resonance at 53 Hz, experienced during the combine bending and torsion modes. Modifications of existing design of chassis to shift natural frequencies were proposed by placing tune mass damper which is a combination of spring mass damper system which carrying 3% of mass of the entire chassis mass in order to reduce the amplitude in turn to increase the frequency which leads to reduce the effect of vibration of chassis under good road conditions.

***Index Terms-***Heavy Truck Chassis, Tuned Mass Damper, Inertia Loads

## 1. INTRODUCTION

Vehicle truck Chassis frame is a key noted element which acts as base for all components like Body, Engine, Transmission system, Axles, wheels, Suspension system. The chassis frame acts as housing for components which are placed on it which are responsible for impact loads from the suspension and gravitational loads.

The truck chassis frame has two side rails and several cross beams of different thickness. The side rails are designed as C type beams; however crossed beams are attached to side rails by braces with bolts and nuts.

The chassis mainly classified according to type of frames

- Laddered frame chassis
- Space frame chassis
- Backbone chassis
- Tube design chassis
- Monocoque chassis

In India, two types of trucks which are used in the market those are mining trucks and normal market purpose trucks are used different type of operations.

## 2.PROBLEM IDENTIFICATION

When vehicle moving on the road, the vehicle truck chassis frame is subjects to the dynamic forces which are induced by the Bumps, Potholes, Roughness of the road and the impact loads which are due to suspension effect from the road irregularities. The vehicle truck chassis leads to vibrate in such various dynamic excitations. If any of the excitation frequencies value occurs simultaneously match with the natural Frequencies value of the mining truck vehicle chassis, so resonance events occurs and the chassis will undergo unwanted huge Oscillations that may leads to deflection and Failure. In this analysis, the factor that affects the heavy truck chassis is necessary to minimize these problems. Chassis failure is related to several factors such as the poor quality of roads on which vehicles are driven, vehicle speed, etc. Then, stress analysis is done to identify the dynamic nature and displacements of the chassis which is important to determine the failure zone and the location having the highest stress due to random vibrations that were considered as the critical location which may responsible for failure of chassis

## 3. OBJECTIVE OF THE WORK

### *3.1. Static analysis*

- To determine Von Misses stresses
- To determine Maximum principal stresses
- To determine Normal and Shear stresses
- To determine maximum deflection

### *3.2. Modal analysis*

- Modal analysis was done to know the mode shapes of the vehicle truck chassis for the class of first six natural frequencies and also deformation of chassis

### *3.3. Harmonic analysis*

- In order to obtain the resonance and frequency for the chassis by plotting the amplitude vs. frequency graph.

### *3.4. Simulation of road profile*

- To analyze displacement profile, stress and deformation of chassis frame when the vehicle truck chassis frame moving on the road profile.

## 4 .METHODOLOGY

- Creation of 3D CAD model: The 3D modeling of the chassis is done using CATIA V5 software and saved in a format such as .iges format.
- Material properties: The selected material properties are taken from ansys library. The material considered as structural steel.
- Meshing: meshing operation is done in ANSYS for geometry which is imported from CATIAV5 where the entire geometry is divided in to small elements.
- Boundary conditions: The loads are applied on the side rails of the chassis and supports were given according to the position.
- Solve: the solution is obtained from the given conditions as input in ANSYS such as static structural, dynamic analysis and harmonic response.
- Post processing: the results taken for analysis.

## 5 .CHASSIS AND SUSPENSION MODEL

The heavy vehicle truck chassis TATA LPT 3718tc truck taken and those specifications are shown in the followings tables.

| Material | Modulus of elasticity (G pa) | Density (kg/ $m^3$) | Tensile Strength ( M pa ) | Yield Strength (M pa) |
|---|---|---|---|---|
| Q460 CFD STEEL | 200 | 7850 | 510 | 460 |

**Table-1**Material properties of heavy vehicle chassis frame

| S. No. | Parameters | Value |
|---|---|---|
| 1 | Total length of truck the chassis | 9760mm |
| 2 | Width of the truck chassis | 2434mm |
| 3 | Wheel base | 6750mm |
| 4 | Front over hang | 1260mm |
| 5 | Rear over hang | 2420mm |
| 6 | Ground clearance | 250mm |
| 7 | Capacity(GVW) | 37000 kg |

**Table-2** Specifications of heavy vehicle chassis frame

### *5.1. Calculation for chassis frame:*

Truck Model =TATA LPT 3718TC
Max Capacity of Truck
=37 ton (Kerb Weight+ Payload)
=37000kg =37000*10= 370000N
Truck with capacity of 1.25%
=370000*1.25N
=462500N
Total load concentrating on the chassis frame
=462500N

The parts of chassis are made of 'C' channels with 285mm*65mm*7mm. Truck chassis having two beams. So the load impinging on each side rail is half of the total load impinging on the chassis frame.
Load impinging on the each frame = (total load impinging on the chassis/2)
= (462500/2) =231250N/Beam

### *5.2. Loading calculations:*

Continuous Beam with uniformly distributed load is chosen for this analysis. Load impinging on the full span of beam 231250 N. Length of the Beam is 9760 mm.
Uniformly Distributed Load is 231250/9760 =23.69N/mm.

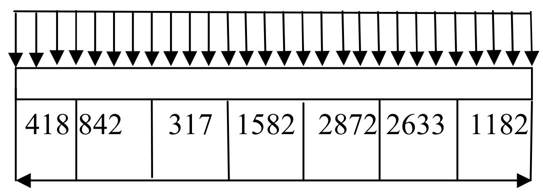


**Fig.1.**Total load acting on the beam

## 6. STATIC MODEL:

In this, geometrical modeling of chassis and its various components has been carried out in CATIA V5 software. Each and every dimensions of chassis frame taken from reference (Tata LPT 3718tc brochure).

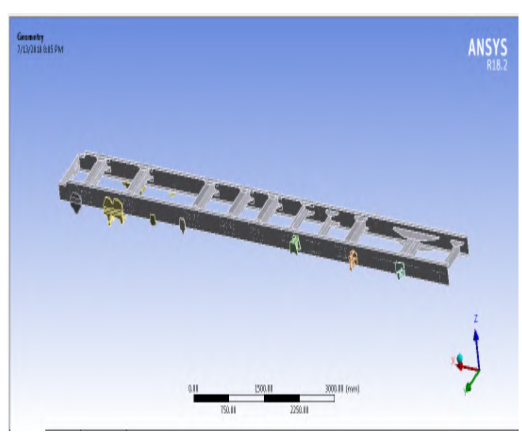


**Fig.2.** Assembled chassis frame in CATIA

## 7. MESHEDMODEL:

The method used for meshing is tetrahedrons and finer meshing is applied at certain portion. The number of Nodes is 161878 and number of elements is 88070. This will approaches result outcome precise and possible.

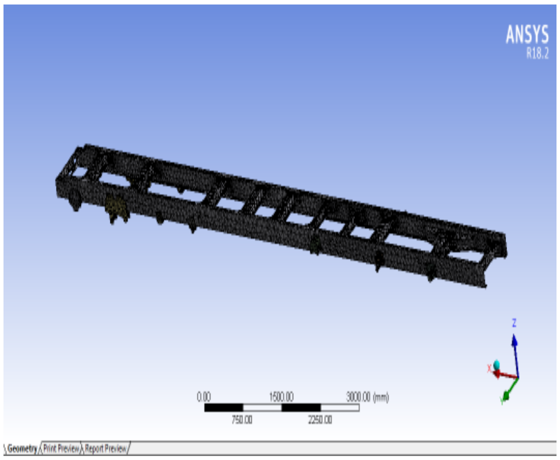


**Fig.3.** Meshed model of chassis frame

## 8. LOAD DISTRIBUTION

The load which is impinging on the chassis frame is considered as uniform distributed load which is obtained from total load divided by throughout length of the chassis frame. The total load impacted on upper area of chassis frame is 231250N/beam. Total six boundary conditions which are front two and last two cross member are fixed and force on side rail beam of chassis considered as 23.69N/mm.

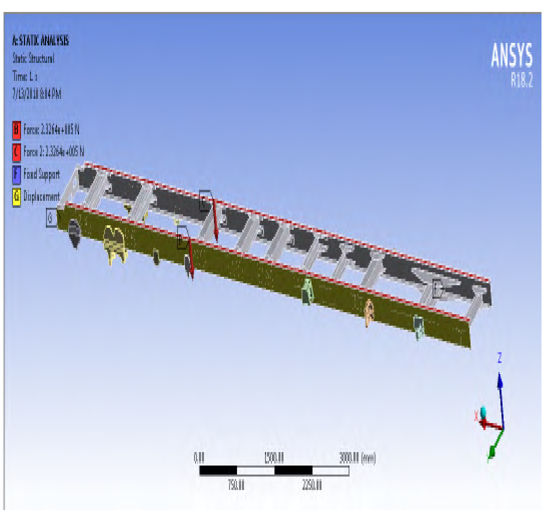


**Fig.4**.Load distribution of chassis frame

## 9. STRESS

The result of study shows that maximum von miss stress on entire vehicle truck chassis frame is shown in below figure, which is at rear of the chassis frame and near the rear wheel axle.

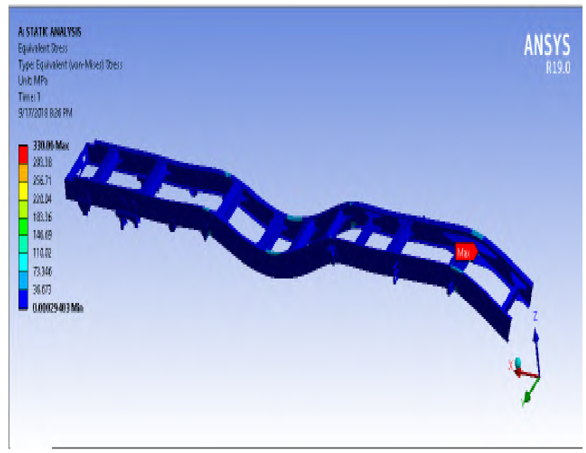


**Fig.5.**Stress distribution of chassis

## 10. DEFORMATION OF CHASSIS:

The result of this study shows that deformation of vehicle truck chassis frame is 1.39mm which is shown in below figure and also critical location was identified.

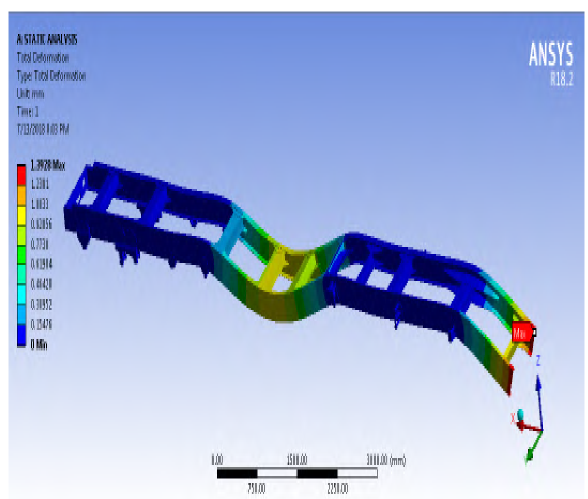


**Fig.6.**Displacement pattern in terms of maximum weight load

## 11. MODAL ANALYSIS:

The modal analysis is the basic technique used for analysis of dynamic character. Mode shapes and natural frequencies were examined through this approach. The key characteristics of every mode of the structure can be figured out through this analysis and the actual vibration response under this frequency range can be predicted. The outcome of modal analysis considered as reference input value for harmonic analysis.
In this paper Modal analysis is performed on the vehicle truck chassis and the natural Frequencies have been found out, the first eight natural frequencies are listed in the following Table 3.

| Mode No | Frequency(Hz) | Deformation(mm) |
|---|---|---|
| 1 | 53.001 | 0.003316 |
| 2 | 66.947 | 0.0074191 |
| 3 | 106.76 | 0.015422 |
| 4 | 119.08 | 0.0086682 |
| 5 | 121.41 | 0.00929 |
| 6 | 129.04 | 0.0080685 |
| 7 | 133.74 | 0.003636 |
| 8 | 146.33 | 0.0081813 |

**Table-3**Shows that the deformation of chassis at Natural frequencies

## 12. MODE SHAPES

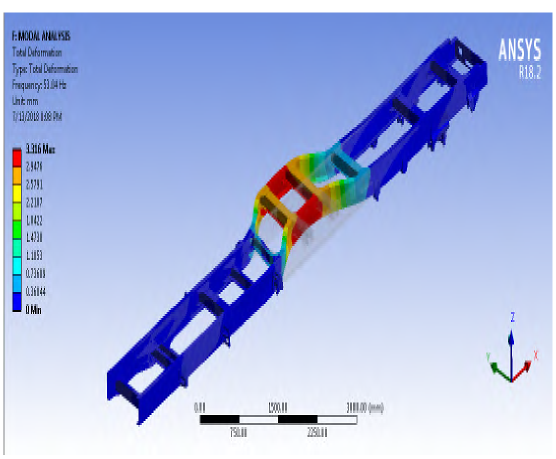


**Fig.7.**Mode1-natural frequency53.001 Hz

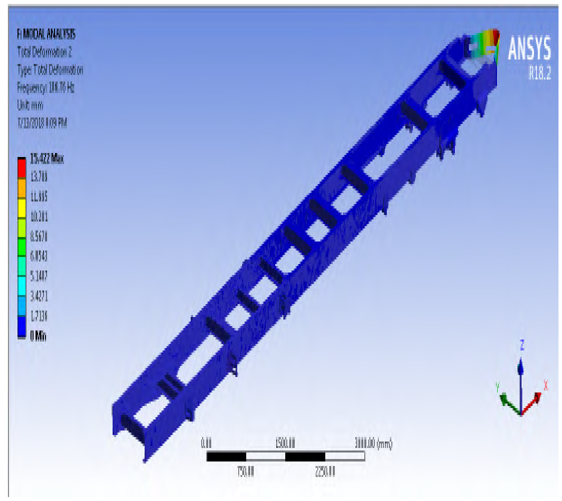


**Fig.8**. Mode2-Natural frequency 66.947Hz

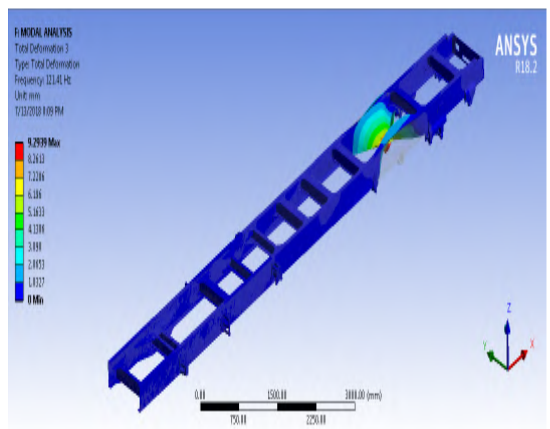


**Fig.9.**Mode3-Natural frequency 106.976Hz

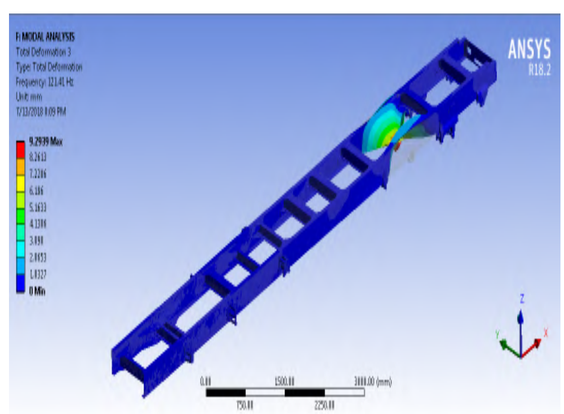


**Fig.10.**Mode 4-natural frequency 121.41 Hz

## 13. DESIGN MODIFICATION OF CHASSIS:

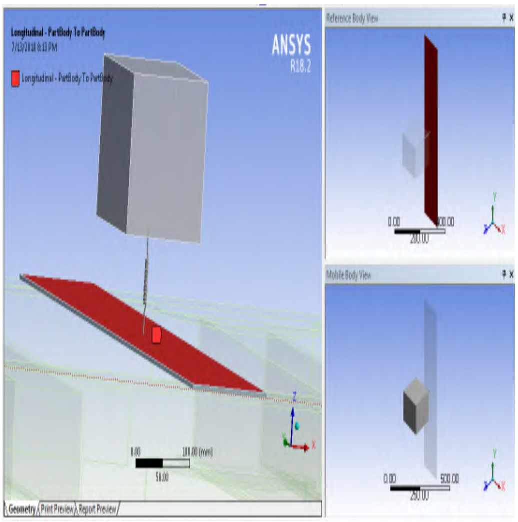


**Fig.11.**Design modification of chassis

Tuned mass damper is combination of spring and damper system. From the modal analysis we observed that at the frequency 53 Hz chassis amplitude was very high so it is essential to reduce the amplitude so we need to introduce tuned mass damper at that particular frequency mode the amplitude value decreases which was shown fig.10.

## 14. TUNED MASS DAMPER:

Tune mass damper is a vibration absorber system which is widely used for controlling unwanted vibration in mechanical engineering system. Present scenario TMD theory and its principle have been adopted to minimize vibrations of structure. Tuned mass damper device is a combination of mass, damping and spring is effective and stable structural vibration control device generally attached to primary vibration system for suppressing unwanted vibrations. When the basic structure is subjects to unwanted external disturbances the fundamental mode of the basic structure resonance with the natural frequency of the TMD is tuned. So that huge amount of vibrating energy is transmitted to TMD and then dissipated by the damping. Tuned mass damper is placed on a primary structure and its natural frequency is tuned to be very near to the dominant mode of the primary structure, considerable reduction in the dynamic responses of primary structures. A number of approaches are available currently which can be used for control vibration. Broadly classified as active, passive and hybrid methodologies, each of these has its advantages and inherent drawbacks. One of these control measures is the TMD - the tuned mass damper, which effectively attenuates the vibration energy at resonance.

### *14.1. Tuned mass damper design*

Mass of Chassis Mb=850 Kg
Mass of Tune Mass Damper Mt= (3to 5%) Of Mass of Beam
Mt=0.03 X 850
Finally Mass Of Tune Mass Damper Is 25.5kg.
Tuned Frequency Is 53 Hz.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{K}{Mt}}$$

Therefore Final K= 2824953 N/m.

Mass ratio $\mu = \frac{\text{Mass of TMD}}{\text{Mass of beam}}$ =0.03.

$$\partial_d = \sqrt{\frac{3\,X\mu}{8(1+\mu)^3}} = 0.1$$

$$\partial_d = \frac{C}{C_C}$$

$$C_C = 2\sqrt{km} = 16974$$

$$\partial_d \text{ x cc} = 1722\ Ns/m$$

## 15. HARMONIC ANALYSIS:

### *15.1. Frequency Vs amplitude graph*

In order to get vibration response at different location of the chassis harmonic analysis is carried out. Harmonic analysis is done in the frequency range of 0Hz to 200Hz at solution intervals of 200 and vibration at different location is measured in terms of frequency and amplitude.

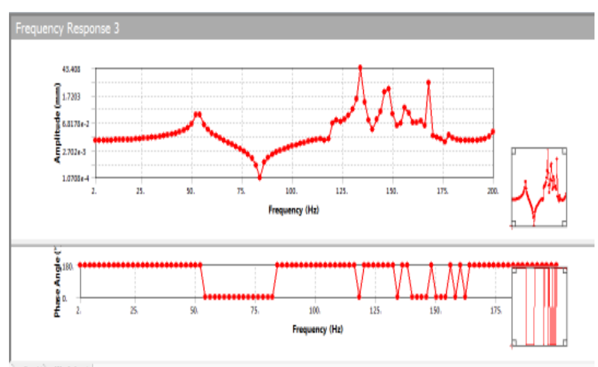


**Fig.12.** frequency @ 53Hz Vs Amplitude

### *15.2. Frequency Vs amplitude graph (modified chassis)*

From harmonic analysis of existing chassis the amplitude value is very high which is obtained at 53Hz frequency. Design modification chassis by placing tune mass damper, the amplitude response was decreased which is shown in below Fig.13.

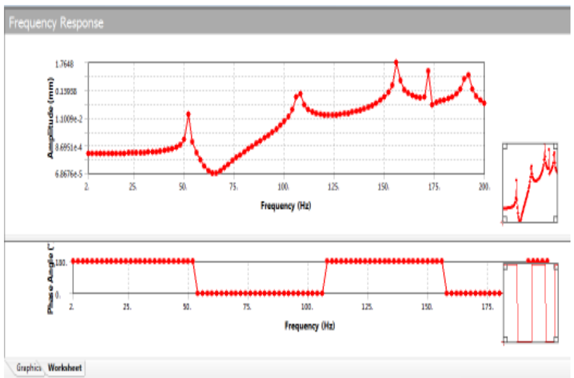

**Fig.13.** Frequency @53Hz Vs Amplitude

## 16. LOADS DUE TO ROADS PROFILE:

The excessive vibrations of chassis and suspension occur as the vehicle experiencing through uneven road profiles

A Two degree freedom suspension Quarter vehicle model is effectively used to study the dynamic response between vehicle and road roughness profile. The equations for 2DOF vehicle model are derived from the Newton's $2^{nd}$law of motion. In which sprung mass (Ms) and it is attached to unsprung mass (Mus) free to move vertically with respect to the sprung mass. The tire is modeled as spring of stiffness (Kus) and dampers of damping coefficients (Cu) the suspensions between the sprung mass and the un-sprung masses are modeled as linear spring of stiffness (Ks) respectively, and linear damper with damping coefficient (Cs) the sprung mass has 1-DOF body bounce. The Xs is the vertical displacement of the sprung mass at the centre of gravity, the un-sprung mass have 1-DOF due to vertical motions (Zus).The input to the wheels is the road excitations (xo) thus there are a total of two second order differential equations governing the motion of the sprung and un-sprung masses against excitations caused by the road roughness (h) as follows

Sprung mass:

$$M_S\ddot{Z}_S + K_S(Z_S - Z_{US}) + C_S(\dot{Z}_S - \dot{Z}_{US}) = 0$$

Un sprung mass:

$$M_{US}\ddot{Z}_S - K_S(Z_S - Z_{US}) - C_S(\dot{Z}_S - \dot{Z}_{US}) + K_W(Z_{US} - r) + C_w(\dot{Z}_{US} - \dot{r}) = 0$$

The parameters of truck: tire mass is 28.58kg, mass on spring is 288.9kg, suspension stiffness is 19960 N/m , tire stiffness is 155900N/m , suspension damping C is 1300 Ns/m . Then the two above equations are simulated in the Simulink of MATLAB software using different blocks. The Simulink environment is used to modelling, simulating and analysis of dynamic systems and supports non linear and linear systems. In this environment, we will be able to simulate and execute a system by blocks and use the results for different purposes. The output of the above system is the vertical acceleration of sprung mass, which can be used to obtain the dynamic force.

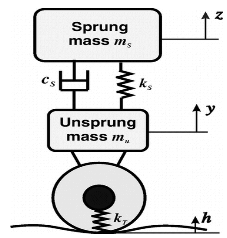


**Fig.14.** Quarter vehicle model

The boundary conditions are applied to the position of the connection between the spring elements and the chassis. Also, the unsprung mass of the truck has been distributed on the plates, between the spring elements. In addition, the displacement of the truck in the directions of the axial and the lateral axis at the mentioned boundaries should be constrained because the spring of the vehicle truck is linear. Also, because the truck does not rotate around the perpendicular axis of its body (yaw), the truck rotation, the motion of the longitudinal and lateral directions should be constrained. Now, by considering the mentioned boundaries, road roughness, considered as random load, can be applied to the truck.

## 17.PARAMETERS VALUES OF 2DOF SYSTEMS

| Parameter Symbol | Description | Numerical Value |
|---|---|---|
| $M_S$ | Sprung mass of the vehicle body | 288.9 kg |
| $M_{US}$ | Un sprung mass of the tire | 28.58 kg |
| Ks | Stiffness of the suspension spring | 19,960 N/m |
| $K_U$ | Stiffness of the tire | 155,900 N/m |
| Cs | Damping coefficient of suspension | 1300 Ns/m |

**Table-4**vehicle parameters for 2DOF suspension model

A random road of class-B is developed in time bound with parameters listed in the below table using MATLAB/SIMULINK

| SNO | Parameter | Random Road Profile Of Type-B (ISO-8606) |
|---|---|---|
| 1 | Variance | 4*10-3 |
| 2 | Velocity(m/s) | 60 |
| 3 | α (Depends on type of road surface) | 0.127 |

**Table-5** Parameters of Random profile road CLASSB

## 18. SIMULATION OF DIFFERENT ROAD CLASSES

| Road class | $\sigma\ (10^{-3}m)$ Roughnes s Variance | $\varphi(\Omega_0)(10^{-3}m^3),$ $= 1$ Power spectral density | $\alpha\ (ra$ $/m)$ |
|---|---|---|---|
| Class B(good | 4 | 4 | 0.12 7 |

**Table-6** Parameters of Random profile road CLASSB

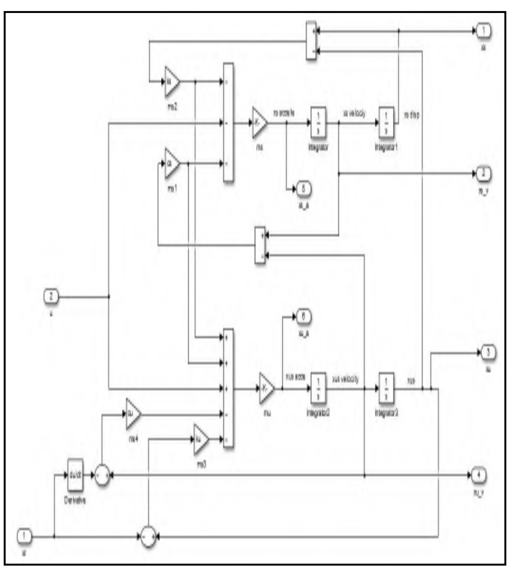

**Fig.15.** Simulation of quarter vehicle model in the SIMULINK environment of MATLAB software

## 19. RESULTS AND DISCUSSION

### *19.1. Static analysis*

| S NO | Parameter | Theoretical | FEM Analysis |
|---|---|---|---|
| 1 | DEFORMATION (mm) | 1.69 | 1.39 |
| 2 | MAX STRESS (Mpa) | 340 | 330 |

**Table-7** The maximum stress on the chassis and deformation

### *19.2.Harmonic analysis*

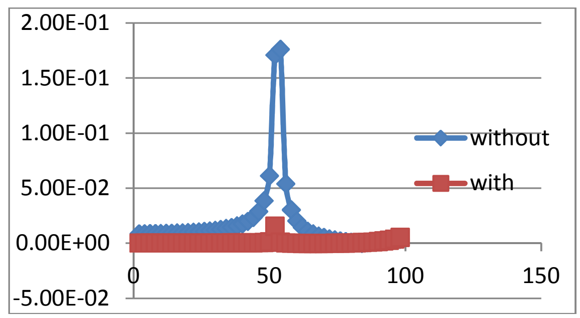


**Fig.16.** Frequency Vs amplitude graph for chassis with Tuned mass damper and chassis without Tuned mass damper

In harmonic frequency response analysis of chassis without and with TMD is shown in Fig.15, for the input is given in the frequency range 0-200 Hz. The response was computed at frequencies observed as peak values obtained.

From the graph, we observed that frequency vs. amplitude positions of chassis with tune mass damper and chassis without tune mass damper. from harmonic analysis we were creating artificial frequency range from 0 – 200 Hz. in that range truck chassis has highest amplitude value at 53 Hz so we need minimize that frequency by placing tune mass damper setup at that particular frequency, after placing TMD, amplitude value decreased this shows that our chassis is safe. Then there is a reduction in amplitude of considerable percentage at resonance condition due to installation of Tuned Mass Damper.

## 20. DYNAMIC ANALYSIS

Dynamic analysis is used to determine von misses' stress and displacement profile of chassis with respect to time factor.

### *20.1. Stress and deformation*

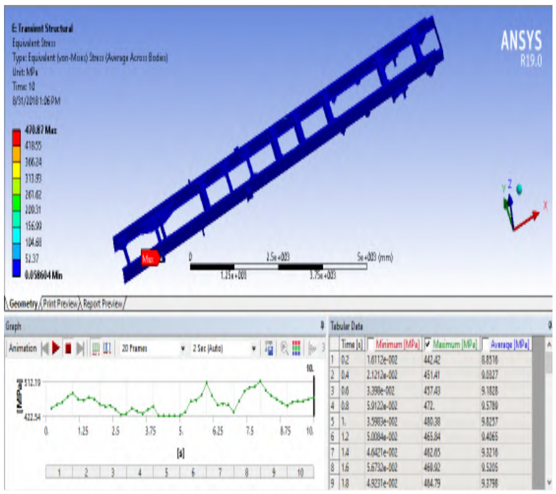

**Fig.17.** The location of critical stress of chassis in CLASS-B road

Dynamic force obtained by the quarter vehicle model used to simulate uneven road in CLASS –B road condition. In this case force is applied at front axle and then with time (l/v) where l is distance between two axles, v is the speed of the vehicle.

In this case maximum stress and deformation generated on the chassis in ISO road CLASS –B was 470 Mpa and 20.22mm. The location of critical stress and the zones of failure were identified.

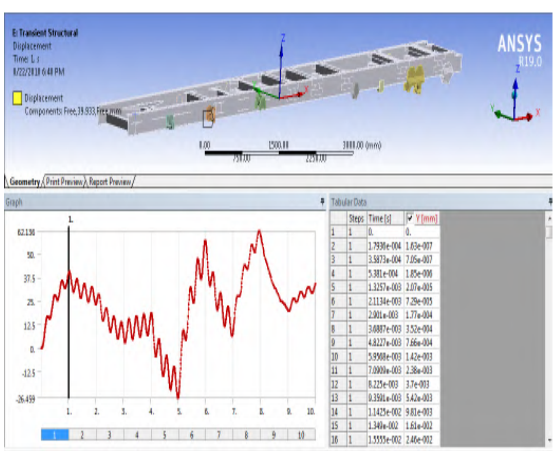

**Fig.18.**The Displacement of chassis under CLASS B road profile

### *20.2. Displacement profile*

The displacement profile of chassis under class b road condition which is shown in below figure

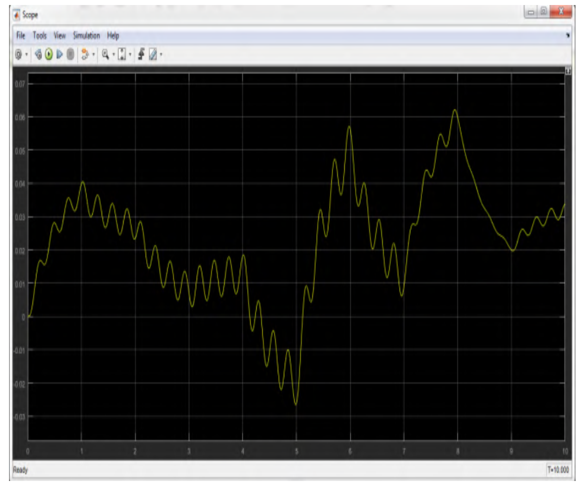

**Fig.19.** Displacement profile of chassis frame in CLASS-B road

## 21. CONCLUSION

The study aim is to analysis a heavy vehicle truck chassis structure using numerical method of FEM. In this regard, different elements of the chassis and weld connections are modeled in CATIA and then entered into ANSYS software and were meshed. Afterwards, static and dynamic forces such as weight force and forces caused by road roughness were examined and applied to the model. Road roughness was also simulated based on the ISO 8606and the dynamic forces applied to the chassis frame were extracted using simulation of a quarter-vehicle model during a motion of constant speed and by introducing tune mass damper the amplitude of chassis decreases with Considerable limit in turn to reduce the vibration effects and the displacement of chassis profile were observed.

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