High Cycle Fatigue Life Estimation of Automotive Chassis Under the Dynamic Loading Condition

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Abstract - *The chassis serves as a frame work for supporting different vehicle components and also the chassis should be rigid enough to withstand the shock, vibration and other stresses due to various road conditions. In this paper fatigue life assessment of automotive chassis is carried out the chassis of heavy-duty truck that is Mercedes-Benz Actros was used for analysis. Initially the static structural analysis is performed by taking different material to compare various stresses and to choose the best material is chosen based on the factor available for analysis. When the vehicle is in motion there will be having vibrations in order to analyze the natural frequencies the Modal analysis is carried out on chosen material then the harmonic analysis is carried out to check the response of the frequencies under dynamic loading condition and alternating stresses induced in the chassis is compared with endurance strength of the material and Fatigue life assessment by using Goodman approach. The chassis is modelled in Spaceclaim and FEA analysis is performed on the modelled chassis using the ANSYS Workbench 2022*

Key Words: chassis, vibration, stress, natural frequencies, ANSYS Workbench 2022

1. INTRODUCTION

The importance of the chassis in the automotive industry and the need to ensure its structural integrity and dynamic characteristics. The chassis plays a vital role in supporting the vehicle's body and components while also withstanding various forces and vibrations.

Purpose of the chassis in vehicle

- Carrying load of passengers or goods carried in vehicle's body.
- Supporting the weight of the body, engine, gearbox, and other components.
- Enduring the forces resulting from sudden braking or acceleration.
- Handling the stresses caused by bad road conditions.

Resonant vibrations acting on chassis

- Resonant vibrations occur when the excitation frequency matches the natural frequency of the chassis.
- This phenomenon can lead to ride discomfort, safety issues, and stability problems.
- Investigating the dynamic characteristics of the chassis is crucial to controlling vibration and noise and avoiding resonance.

1.1 Sources of vibration in a vehicle

The following are some sources of vibration in vehicles.

1.Road roughness: The road surface is not perfectly smooth and contains various irregularities such as bumps, potholes, cracks, and undulations. When a vehicle passes over these road irregularities, it experiences vertical accelerations and vibrations. Free vibrations occur when the vehicle encounters isolated irregularities, while forced vibrations result from persistent disturbances caused by obstacles or rough sections on the road.

2.Suspension system: The suspension system of the vehicle plays a crucial role in handling road-induced vibrations. The springs, shock absorbers, and other components in the suspension system are designed to dampen and absorb the impact of road irregularities to minimize the effects of vibrations on passengers and enhance ride comfort.

3.Tires: Tires also contribute to the transmission of road vibrations to the vehicle. The tire's sidewall flexibility and tread design can affect how vibrations are transmitted to the chassis and, ultimately, to the passengers.

4.Drivetrain: The engine, transmission, and drivetrain components can generate vibrations due to their moving parts and interactions. Engine vibrations, in particular, can be transmitted to the cabin, affecting passenger comfort.

5.Aerodynamic forces: At higher speeds, aerodynamic forces acting on the vehicle can cause vibrations and buffeting, especially in open-top or high-performance vehicles.

6.Braking and Acceleration: Sudden braking or acceleration can create transient forces that lead to vibrations in the vehicle chassis.

2. LITERATURE SURVEY

Mr. M Sani [1] presents a computational study on the modal transient response and stress analysis of a car chassis. The analysis aims to understand the behavior of chassis structure when subjected to loads and to assess whether the stress levels are within the allowable limits for the chosen material. Based on results obtained from stress and modal analysis, the study concludes that stress and modal analysis techniques are essential and significant in the design of automotive chassis structures. These computational analyses provide important information about the behavior of the chassis under different load conditions and help ensure that the chassis can withstand the expected stresses during vehicle operation.

Han Quan-li [2] did the optimization analysis of the frame of a heavy-duty truck. The study aims to improve the design of the truck's frame by employing a finite element model to study its characteristics under different loading conditions. The optimization process involves selecting design variables, state variables, and an objective function to achieve an optimal heavy truck design. The results of the frame optimization analysis will provide valuable insights into the structural behaviour of the heavy-duty truck's frame and allow for the identification of an optimal design that balances mass reduction with required strength and stiffness.

Dr. R. Rajappan [3] focuses on analyzing the static and dynamic load characteristics of a truck chassis using finite element (FE) models. The research aims to identify areas of high stress, analyze vibrations, determine natural frequencies, and study mode shapes using the FE method. Modal updating of truck chassis model is performed by adjusting chosen properties such as mass density and Poisson's ratio. The predicted natural frequencies and mode shapes are validated against previously published results. Based on findings, modifications to the FE truck chassis model are proposed to achieve several objectives.

3. PRESENT STUDY

The present study focuses on investigating the vibrations induced and fatigue life of the chassis due to vibrations induced of a heavy truck, specifically the Mercedes-Benz Actros. The chassis under consideration is of the ladder type, which is a common design used in heavy-duty trucks. The Mercedes-Benz Actros is a heavy-duty truck model introduced by Mercedes-Benz in 1996 at the Commercial Vehicle IAA in Hanover, Germany. The Vehicle specifications are as shown in Table-1

To study the induced vibrations in the chassis, the researchers likely employed various methods, such as finite element analysis (FEA) or experimental testing. The study is essential to ensure the structural integrity, safety, and performance of the truck during its operation.

Table -1: Vehicle specifications

Sl No.	Specifications	Standard ratings
1	Engine type	V6
2	Capacity	400 liters
3	Fuel type	Diesel
4	Maximum power	394HP at 1800 rpm
5	Maximum torque	1850 Nm at 1080 rpm
6	Cylinders	6 cylinders
7	GVW Weight (Kg)	26000

3.1 CAD model of truck chassis

The truck chassis chosen for the analysis is used in the Mercedes-Benz Actros. The detailed dimensions and 3D sketch of the truck chassis are given below Fig-1.

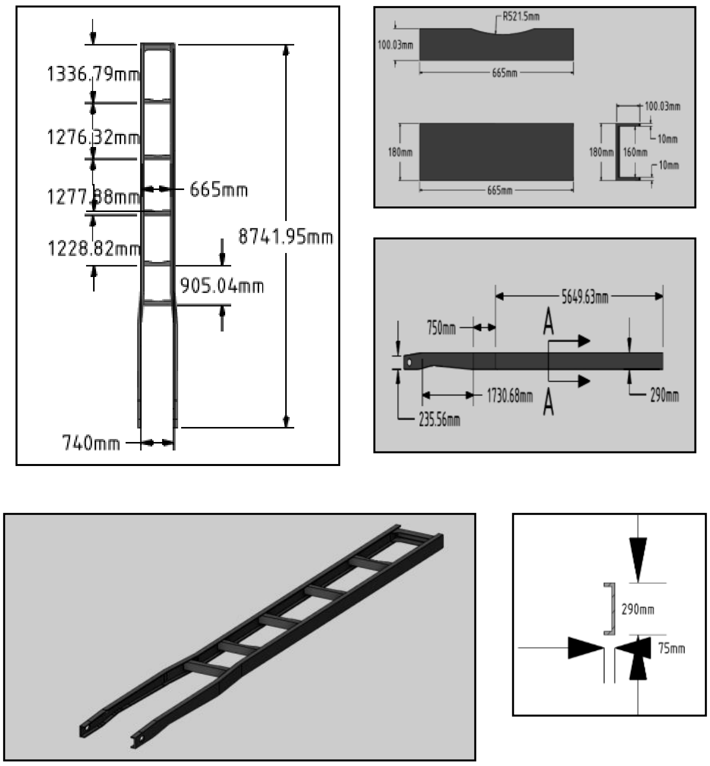


Fig-1: Detailed dimensions and 3D sketch of chassis

4.MATERIAL SELECTION

The study is carried out for two materials Aluminium T6 and High strength steel. The material properties of both the chosen materials are given in the below

Table-2: Material properties of materials used in analysis

Properties	Aluminum T6	High strength steel
Modulus of Elasticity, GPa	700	200
Density, kg/m3	2700	7500
Poison ratio	0.33	0.28
Yield strength, MPa	245	500
Percentage of elongation	12	15
Endurance strength, MPa	93	195

5. STATIC STRUCTURAL ANALYSIS

The analysis is performed for both the materials under the static loading condition. Stresses, deformation and Factor of safety is evaluated for both the material and based on safety margins, cost, reliability, availability and safety best material is chosen for further study.

5.1 Boundary conditions

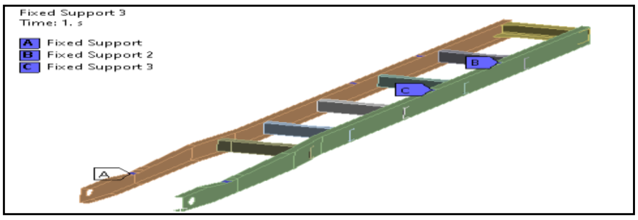


Fig-2: Fixed supports applied to chassis

The chassis is constrained at specific locations in all degrees of freedom with respect to the global coordinate system. This means that the chassis is prevented from moving or rotating at these constrained points. This kind of constraint is commonly known as a “fixed support” or “fully constrained” boundary condition as shown in Fig-2.

5.2 Loading conditions

This present Mercedes Benz Actros is built to take the maximum load of 26000kg. The force of 26000kg load is applied on the top surface of the chassis to check its structural integrity under the maximum loading capacity as shown in Fig-3.

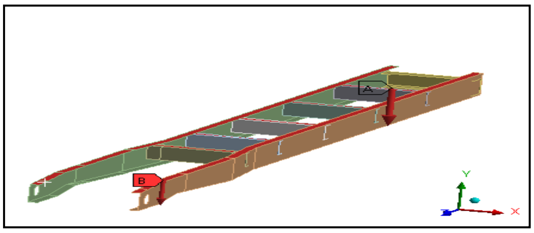


Fig-3: Load applied on chassis

The analysis is performed for both the materials under the static loading condition. Stresses, deformation and Factor of safety is evaluated for both the material and based on safety margins, cost, reliability, availability and safety as shown in Table-3 best material is high strength steel is chosen for further study.

Table-3: Comparison of two materials

Properties	Aluminum T6	High strength steel
Modulus of Elasticity, GPa	700	200
Yield strength, MPa	245	500
Stress, MPa	127.1	137.8
Deformation, mm	2.08	6.2
Weight of the chassis, kg	456	1321
Cost Per Kg	280	70
Total Cost	127680	92470
FOS for static	1.9	3.6

6. RESULTS AND DISCUSSIONS

6.1 Static structural analysis

This analysis is carried out to check the deformation and the stresses developed due to self-weight of the truck and its full loading capacity scenario. The developed stress is checked with the strength of the truck chassis material.

6.1.1 Equivalent stress- High strength steel

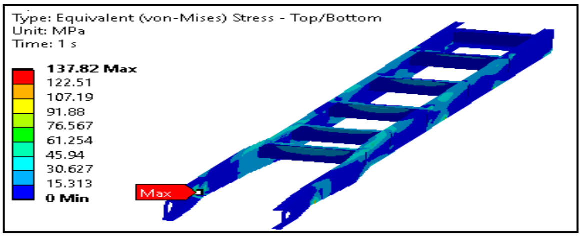


Fig- 4: Equivalent stress- High strength steel

The truck chassis observed equivalent stress of 137.8 MPa under the maximum load carrying capacity of 26000kg and its self-weight. The allowable strength of high strength steel is 500MPa. The allowable strength of the material is greater than the developed stress in the truck chassis as shown in above Fig-4. Hence it meets the requirement of design.

6.1.2 Total deformation – High strength steel

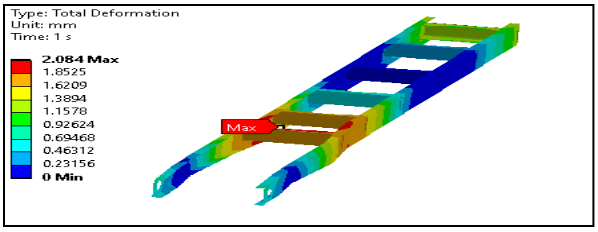


Fig.-5: Total deformation – High strength steel

The magnitude of the total deformation of 2.08mm is observed as shown in above Fig-5 on the cross channels of the truck chassis with respect to the global coordinate system predominantly due to gravity loads. The maximum allowable deformation of the given material i, e 24.28mm (as per length to span ratio L/360 where L=Total length of chassis) is greater than observed deformation. Hence it meets the design requirement.

6.1.3 Equivalent stress- Aluminium T6

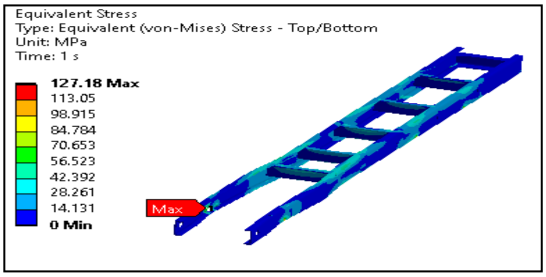


Fig-6: Equivalent stress- Aluminium T6

The truck chassis observed equivalent stress of 127.1 MPa under the maximum load carrying capacity of 26000kg and its self-weight as shown in Fig-6. The allowable strength of aluminium T6 is 245MPa. The allowable strength of the material is greater than the developed stress in the truck chassis.

6.1.4 Total deformation – Aluminium T6

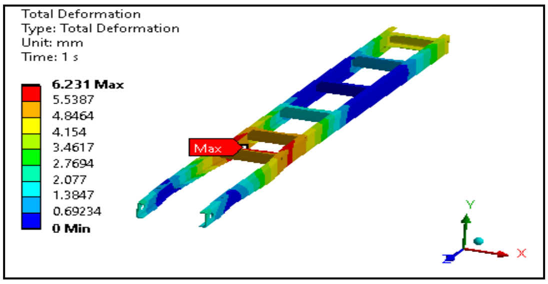


Fig-7: Total deformation – Aluminium T6

The magnitude of the total deformation of 6.2 mm is observed on the cross channels of the truck chassis with respect to the global coordinate system predominantly due to gravity loads as shown in above Fig-7. The maximum allowable deformation of the given material i, e 24.28mm is greater than observed deformation.

6.2 Modal Analysis

Performed an undamped free vibration analysis on the truck chassis to extract its eigenvalues and mode shapes. This analysis helps you understand the natural frequencies and corresponding mode shapes of the chassis. These mode shapes represent the different ways the chassis can vibrate without any external damping or forcing. It's crucial to analyze and interpret the mode shapes to understand how the chassis responds to various vibration modes. The mode shapes can help identify potential weak points or areas of high stress concentration that might need design adjustments.

Modal analysis was carried out and extracted 20 modes and frequencies as shown in Table-4 and critical frequencies were found in Y direction as it has more deformation in Y direction as compared to other 2 directions. Only critical direction was plotted in this paper.

Table-4: Natural frequencies of chassis

Mode No	Frequency in Hz
1	20.323
2	28.955
3	44.869
4	45.009
5	60.869
6	61.727
7	70.039
8	77.213
9	87.529
10	100.33
11	101.19
12	101.72
13	102.44
14	110.61
15	115.38
16	119.26
17	121.44
18	143.07
19	150.37
20	156.08

6.2.1 Primary dominant mode in Y direction

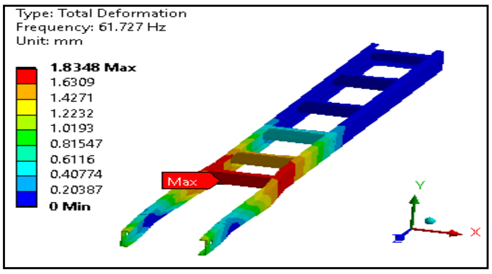


Fig-8: Primary dominant mode in Y direction

The sixth mode observed at the frequency of 61.7 Hz as shown in Fig- 8. The maximum deformation observed at the sixth mode is 1.8 mm This mode is primary dominant mode in Y direction where the more mass is participating in Y direction.

6.2.2 Secondary dominant mode in Y direction

The seventh mode observed at the frequency of 70.03 Hz as shown in above Fig-9. The maximum deformation observed at the seventh mode is 3.1 mm This mode is secondary dominant mode in Y direction where the more mass is participating in Y direction.

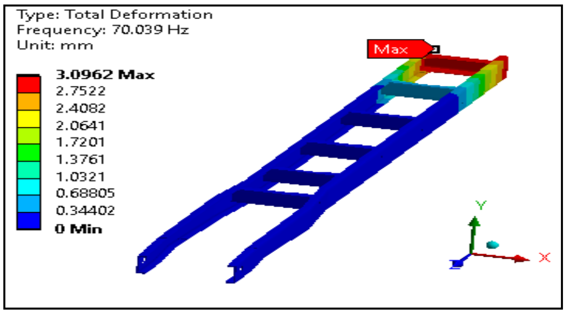


Fig-9: Secondary dominant mode in Y direction

6.2.3 Third dominant mode in Y direction

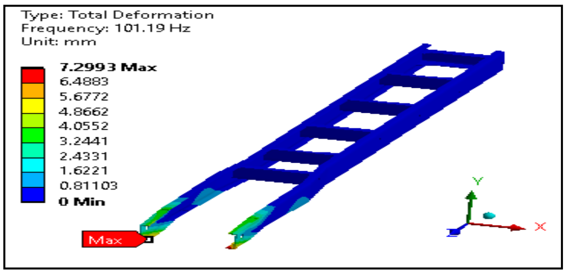


Fig-10: Third dominant mode in Y direction

The twelfth mode observed at the frequency of 101.2 Hz as shown in Fig-10. The maximum deformation observed at the twelfth mode is 7.3mm This mode is third dominant mode in Y direction where the more mass is participating in Y direction.

6.3 Harmonic Response Analysis

The Harmonic response analysis helps to analyse how a structure will respond to repetitive dynamic loading. It is linear dynamic analysis which will determines response of system at specific frequencies. The alternating stresses are induced in the truck chassis due the rotation of engine and those stresses are very important in fatigue life estimation. Harmonic analysis is carried out to extract the alternating stresses. The results are plotted for the worst case in worst direction.

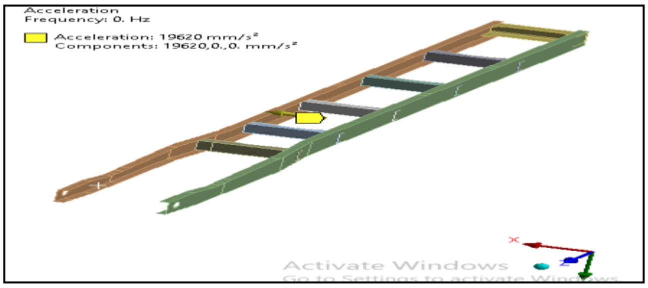
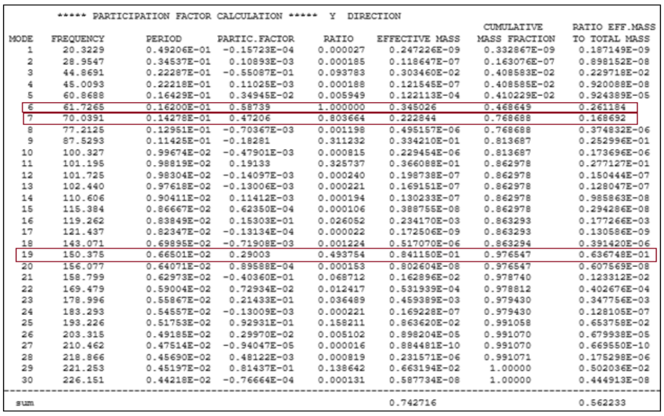


Fig-11: Harmonic excitation through global acceleration method

Above Fig-11 shows that global acceleration method is used for giving the harmonic excitation in the dynamic analysis to compute the alternating stresses. Harmonic analysis is carried out in all three direction and critical results are observed in the vertical direction that is Y direction. Only worst-case direction is plotted in this paper. As the frequencies found in this direction has higher mass participation as shown in Fig-12 under dynamic loading conditions.

6.3.1 Mass participation factor in Y direction


***** PARTICIPATION FACTOR CALCULATION ***** Y DIRECTION

MODE	FREQUENCY	PERIOD	PARTIC.FACTOR	RATIO	EFFECTIVE MASS	CUMULATIVE MASS FRACTION	RATIO EFF.MASS TO TOTAL MASS
1	20.3229	0.49206E-01	-0.15723E-04	0.000027	0.247226E-09	0.332867E-09	0.187149E-09
2	28.9547	0.34537E-01	0.10893E-03	0.000185	0.118647E-07	0.163076E-07	0.898152E-08
3	44.8691	0.22287E-01	-0.55087E-01	0.093783	0.303460E-02	0.408583E-02	0.229718E-02
4	45.0093	0.22218E-01	0.11025E-03	0.000188	0.121545E-07	0.408585E-02	0.920088E-08
5	60.8688	0.16429E-01	0.34945E-02	0.005949	0.122113E-04	0.410229E-02	0.924389E-05
6	61.7265	0.16200E-01	0.58739	1.000000	0.345026	0.468649	0.261184
7	70.0391	0.14278E-01	0.47206	0.803664	0.222844	0.768688	0.168692
8	77.2125	0.12951E-01	-0.70367E-03	0.001198	0.495157E-06	0.768688	0.374832E-06
9	87.5293	0.11425E-01	-0.18281	0.311232	0.334210E-01	0.813687	0.252996E-01
10	100.327	0.99674E-02	-0.47901E-03	0.000815	0.229454E-06	0.813687	0.173696E-06
11	101.195	0.98819E-02	0.19133	0.325737	0.366088E-01	0.862978	0.277127E-01
12	101.725	0.98304E-02	-0.14097E-03	0.000240	0.198738E-07	0.862978	0.150444E-07
13	102.440	0.97618E-02	-0.13006E-03	0.000221	0.169151E-07	0.862978	0.128047E-07
14	110.606	0.90411E-02	0.11412E-03	0.000194	0.130233E-07	0.862978	0.985863E-08
15	115.384	0.86667E-02	0.62350E-04	0.000106	0.388755E-08	0.862978	0.294286E-08
16	119.262	0.83849E-02	0.15303E-01	0.026052	0.234170E-03	0.863293	0.177266E-03
17	121.437	0.82347E-02	-0.13134E-04	0.000022	0.172506E-09	0.863293	0.130586E-09
18	143.071	0.69895E-02	-0.71908E-03	0.001224	0.517070E-06	0.863294	0.391420E-06
19	150.375	0.66501E-02	0.29003	0.493754	0.841150E-01	0.976547	0.636748E-01
20	156.077	0.64071E-02	0.89588E-04	0.000153	0.802604E-08	0.976547	0.607569E-08
21	158.799	0.62973E-02	-0.40360E-01	0.068712	0.162896E-02	0.978740	0.123312E-02
22	169.479	0.59004E-02	0.72934E-02	0.012417	0.531939E-04	0.978812	0.402676E-04
23	178.996	0.55867E-02	0.21433E-01	0.036489	0.459389E-03	0.979430	0.347756E-03
24	183.293	0.54557E-02	-0.13009E-03	0.000221	0.169228E-07	0.979430	0.128105E-07
25	193.226	0.51753E-02	0.92931E-01	0.158211	0.863620E-02	0.991058	0.653758E-02
26	203.315	0.49185E-02	0.29970E-02	0.005102	0.898204E-05	0.991070	0.679938E-05
27	210.462	0.47514E-02	-0.94047E-05	0.000016	0.884481E-10	0.991070	0.669550E-10
28	218.866	0.45690E-02	0.48122E-03	0.000819	0.231571E-06	0.991071	0.175298E-06
29	221.253	0.45197E-02	0.81437E-01	0.138642	0.663194E-02	1.00000	0.502036E-02
30	226.151	0.44218E-02	-0.76664E-04	0.000131	0.587734E-08	1.00000	0.444913E-08
sum					0.742716		0.562233

Fig-12: Mass participation factor in Y direction

The Fig-12 shown above likely presents a list of vibration modes in the Y direction, along with their corresponding frequencies and the percentage of mass participation in the Y-direction for each mode. The "Primary mode in Y direction" with a frequency of 61.7Hz. Harmonic analysis is important for understanding how the chassis behaves under specific frequencies that are close to its natural frequencies.

6.3.2 Stress plot for primary dominant frequency in Y direction

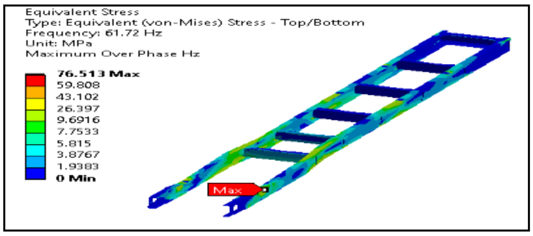


Fig-13: Stress plot for primary dominant frequency in Y direction

The equivalent stress of 76.5MPa is observed at the 61.7Hz frequency as shown in Fig-13 which is primary dominant mode in Y direction. The developed stress is well below the endurance strength of the material i, e 195MPa

6.3.3 Stress plot for second dominant frequency in Y direction

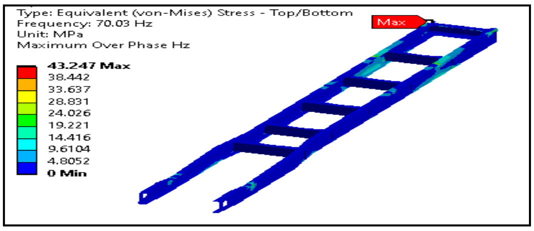


Fig-14: Stress plot for second dominant frequency in Y direction

The equivalent stress of 43.2 MPa is observed at the 70 Hz frequency which is second dominant mode in Y direction as shown in above Fig-14. The developed stress is well below the endurance strength of the material i, e 195MPa.

6.3.4 Stress plot for third dominant frequency in Y direction

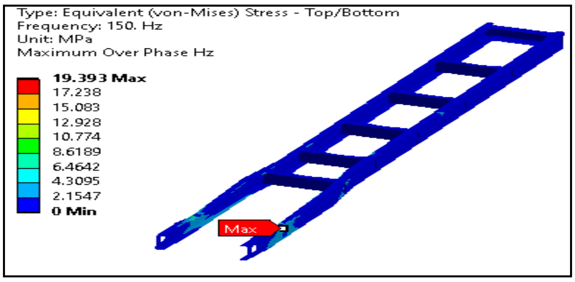


Fig-15: Stress plot for third dominant frequency in Y direction

The equivalent stress of 19.4 MPa is observed at the 150 Hz frequency which is third dominant mode in Y direction as shown in above Fig-15. The developed stress is well below the endurance strength of the material i, e 195MPa.

6.4 Fatigue Life Assessment

The fatigue life is defined as the number of stress or loading cycle a material sustains before failure of that material occur. This fatigue life is affected by cyclic stresses, material properties, defect in the material, design geometry, surface quality, etc.

Fatigue life assessment is performed for two scenarios by using Goodman plot.

6.4.1 Case 1: Maximum alternating stress location

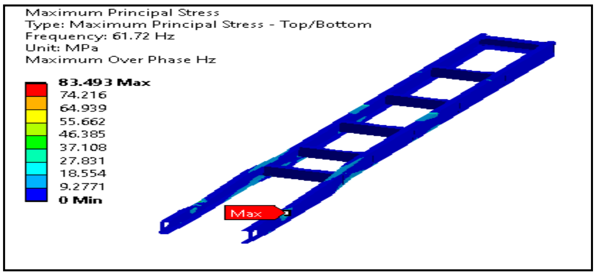


Fig-16: Maximum alternating stress location

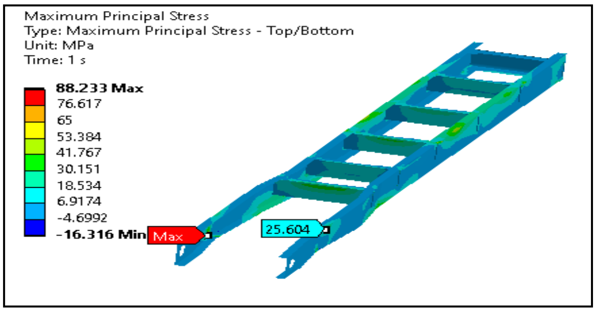


Fig-17: Mean stress at maximum alternating stress location

Maximum principal stress from harmonic analysis is taken for fatigue life assessment as alternating stress and for same location mean stress is extracted from static structural analysis. Maximum principal stress of 83.5MPa as shown in above Fig-16 is observed in the harmonic analysis in Y direction at 61.7Hz frequency which is dominant frequency in Y direction at the same location mean stress of 25.6MPa as shown in Fig-17 is plotted from static structural analysis and both the values are plotted in the Goodman plot for fatigue life assessment.

Ultimate strength of high strength steel = 650MPa

Endurance strength =0.3* ultimate strength

Endurance strength =195MPa

Mean Stress = 25.6MPa

Alternating Stress= 83.5MPa

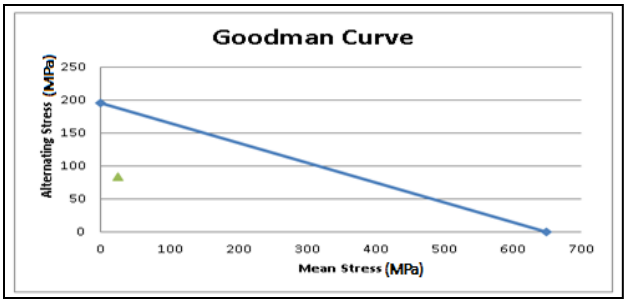


Fig-18: Goodman diagram for fatigue life assessment – Case 1

A point lies in the safe region of the Goodman line as shown in above Fig-18. Hence it meets the design requirement for high cycle fatigue. It sustains infinite life.

6.4.2 Case 2: Maximum mean stress location

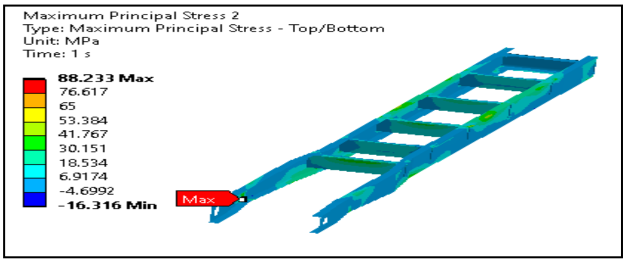


Fig-19: Maximum mean stress location

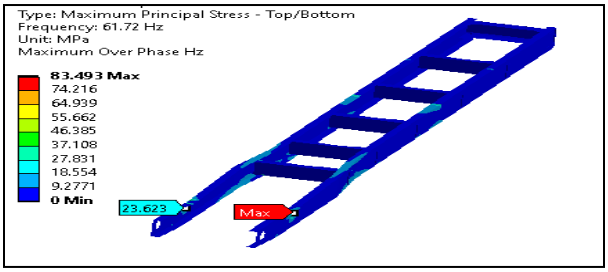


Fig-20: Alternating stress at maximum mean stress location

Maximum principal stress from static structural analysis is taken for fatigue life assessment as mean stress and for same location alternating stress is extracted from harmonic analysis. Maximum principal stress of 88.2 MPa as shown in Fig-19 is observed in the static structural analysis and at the same location alternating stress is extracted from harmonic analysis as shown in above Fig-20 and both the values are plotted in the Goodman plot for fatigue life assessment.

Ultimate strength of high strength steel = 650MPa

Mean Stress = 88.3MPa

Alternating Stress= 23.6MPa

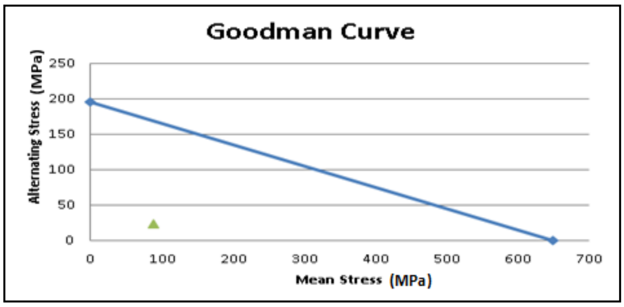


Fig-21: Goodman diagram for fatigue life assessment – Case 2

A point lies in the safe region of the Goodman line as shown in above Fig-21 Hence it meets the design requirement for high cycle fatigue. It sustains infinite life.

7. CONCLUSIONS

The present work dealt with fatigue life analysis of truck chassis initially the material comparison study was carried out between High strength steel and Aluminium T6. Based on stress, deformation, weight and cost comparison the High strength steel material was chosen over the Aluminium T6. In static structural analysis it was found out the equivalent von misses stresses, maximum principal stresses and maximum shear stresses are less than the allowable yield strength of the material. Then the modal analysis was carried out to extract natural frequencies and mode shapes and mass participation factor of chassis in all 3 directions and it was found that the modes and frequencies in Y direction was critical as it has higher deformation in Y direction under undamped free vibration condition. Then the harmonic analysis was performed in all 3 directions and critical frequency results was found in Y direction as the critical frequencies are having higher mass participation factor in Y direction under dynamic loading conditions and results were analysed for critical frequencies and it was observed that alternating stresses induced in harmonic analysis were below the endurance strength of the material. Fatigue life assessment by using Goodman approach was carried out and it was observed that it will meet the infinite life. Hence it was concluded that no fatigue failure of chassis expected.

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