

Evaluation of Rollover Stability & Dynamic Load Response of Truck Chassis System

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Abstract—The present study focuses on the design, analysis, and optimization of a truck chassis to improve its structural efficiency and dynamic performance under realistic operating conditions. A detailed three-dimensional CAD model of the chassis was developed and analyzed using Finite Element Analysis (FEA). Appropriate boundary conditions and loading scenarios were applied to simulate actual service conditions, including uniformly distributed load (UDL), cornering forces, and bump loads. Two critical load cases, namely UDL with cornering and UDL with cornering combined with bump load, were considered to evaluate the structural response of the chassis under normal as well as severe operating conditions. In addition, modal analysis was carried out to study the dynamic behavior and natural frequency characteristics of the chassis. An iterative optimization approach was adopted to reduce the weight of the chassis while ensuring structural safety. The results indicate that the chassis weight was successfully reduced by approximately 10.5%, from 604.14 kg to 541 kg, with only a marginal increase in stress and deformation. Under both load cases, the von Mises stress and deformation remained within acceptable limits, confirming the structural integrity of the optimized design. Modal analysis results showed only slight variations in natural frequencies, indicating that the optimized chassis does not pose any resonance risk. Overall, the study demonstrates that the application of FEA-based optimization techniques can significantly enhance the performance of a truck chassis by achieving weight reduction without compromising strength, stiffness, and dynamic stability. The optimized chassis design is therefore suitable for practical automotive applications, contributing to improved efficiency and performance of heavy vehicles.

Keywords: Dynamic Loads, Truck Chassis, Finite Element Analysis (FEA), Multi-Body Dynamics (MBD) etc.

I. INTRODUCTION

The chassis is the fundamental load-bearing framework of any commercial vehicle, especially trucks, which carry heavy and varying loads under diverse operating conditions. It serves as the main structural backbone that supports the engine, transmission, suspension, cabin, cargo body, and other major subsystems. As trucks are subjected to fluctuating loads—static, impact, vibrational, and bending—under real-world driving environments, the structural integrity, strength, stiffness, and fatigue life of the chassis become critical determinants of vehicle safety, performance, and reliability.

With the increasing demand for higher payload capacity, improved fuel efficiency, stricter emission norms, and better ride comfort, the automotive industry is compelled to optimize truck chassis design. Traditional chassis designs often rely on conservative safety factors and empirical calculations, which may result in over-designed, heavier structures. Such excess weight reduces fuel economy and limits payload capacity. Conversely, under-designed structures risk fatigue failure, excessive deflection, and poor handling characteristics. Therefore, achieving optimal structural design while ensuring robustness under dynamic loading conditions has become essential for modern truck engineering.

Recent advances in Finite Element Analysis (FEA), material modeling, modal analysis, and optimization techniques enable designers to accurately predict chassis behavior under dynamic loads such as braking, acceleration, cornering, road irregularities, and torsional stress. These tools allow iterative improvement of the chassis geometry, material selection, and cross-sectional dimensions, ensuring that the final design meets all strength and performance requirements while minimizing weight.

The truck chassis is the backbone of any commercial vehicle, serving as the primary load-bearing structure that supports the engine, cab, cargo body, and all auxiliary systems. It must withstand a wide range of static and dynamic loads—including torsion, bending, vibration, and impact—while maintaining structural integrity, safety, and durability. In modern logistics and transportation industries, trucks are expected to operate under increasingly demanding conditions: higher payloads, rough terrains, high-speed highways, and variable loading scenarios. Consequently, the chassis must not only be strong and stiff but also lightweight to improve fuel efficiency and reduce emissions.

II. LITERATURE REVIEW

Harshal V. Pachgade (2021) [1] Focused on improving heavy vehicle chassis using CAE and FEA. Identified stress-prone regions and proposed design modifications to enhance strength, stability, and load capacity. The optimized design reduced stress concentration and improved performance.

Monika S. Agrawal (2018) [2] Studied design and analysis of truck chassis under various load conditions using analytical methods and FEA. Suggested design and material

improvements to enhance strength and durability while maintaining optimal weight.

K. Venkatesh et al. (2014) [3] Worked on optimizing chassis support stiffness to increase natural frequency and avoid resonance. Demonstrated improved vibration resistance and durability through stiffness optimization.

Aadithya R. & Muruganandam D. (2016) [4] Analyzed multiple load cases such as static, dynamic, braking, and cornering loads. Identified critical loading conditions and emphasized comprehensive load analysis for safe chassis design.

Bohuan Ta et al. (2020) [5] Developed a dynamic model considering chassis flexibility for powertrain mounting optimization. Showed that including chassis deformation improves vibration prediction and system performance.

III. FINITE ELEMENT ANALYSIS (FEA)

Finite Element Analysis (FEA) analysis is performed to evaluate the structural behavior of the chassis under various loading conditions that simulate real-world operating scenarios. The primary focus of this chapter is to determine stress distribution, deformation characteristics, natural frequencies, and dynamic response of the chassis frame. The developed CAD model is imported into FEA software and subjected to appropriate meshing, boundary conditions, and load applications as defined in the methodology chapter. Different types of analyses such as static structural analysis, modal analysis, and dynamic (transient) analysis are carried out to understand the performance of the chassis under both steady and time-varying loads.

The results obtained from these analyses are used to identify critical regions of stress concentration and deformation, which may lead to structural failure if not properly addressed. Furthermore, the behavior of the chassis under vibration and dynamic loading is studied to ensure stability, safety, and durability. This chapter also forms the basis for comparison between the existing and optimized chassis designs. The outcomes of the analysis are further utilized in the subsequent chapter to evaluate improvements achieved through design optimization.

3.1 Numerical Calculation

➤ Vehicle Data

- **Gross Vehicle Weight (GVW)** = 8120 kg
- **Wheelbase (L)** = 3820 mm ≈ 3.8 m
- **CG from front axle (b)** = 2654 mm
- **CG from rear axle (c)** = 1096 mm
- **CG height (h)** = 1755 mm = 1.755 m
- **Maximum Power** = 95 PS ≈ 93.7 HP
- **Maximum Speed** = 92–94 km/h

➤ Case 1 – Gross Vehicle Weight as UDL

Total load on chassis:

$$W = m \times g$$

$$W = 8120 \times 9.81$$

$$W = 79,657.2 \text{ N}$$

So,
UDL = 79,657 N

If treated as uniformly distributed load along wheelbase:
 UDL per meter = $79657 / 3.82$
 = 20853 N/m

➤ CASE 2- BUMP FORCE

- $W = 79,657 \text{ N}$
- $c = 1096 \text{ mm} = 1.096 \text{ m}$
- $m = 8050 \text{ kg}$
- $h = 1755 \text{ mm} = 1.755 \text{ m}$
- $L = 3820 \text{ mm} = 3.82 \text{ m}$
- $g = 32.2 \text{ ft/s}$
- $V = 83.85 \text{ ft/s}$
- $HP = 93.7$

Step 1 – Convert Mass to Pounds

$$m = 8050 \times 2.20462$$

$$m = 17,747.2 \text{ lb}$$

Step 2 – Calculate Acceleration (a)

Formula:

$$a = 550 \times (g/V) \times (HP/m)$$

$$a = 550 \times (32.2/83.85) \times (93.7/17747.2)$$

$$a = 1.115 \text{ ft/s}^2$$

Convert to m/s^2 :

$$a = 1.115 \times 0.3048$$

$$a = 0.339 \text{ m/s}^2$$

$$FB = 3/2(Wc/L + mah/L)$$

$$FB = 36163 \text{ N}$$

Load Distribution

- **Force per front wheel**
 $= 36163 / 2$
 $= 18,082 \text{ N}$
- **Force per leaf spring mount**
 $= 18082 / 2$
 $= 9,041 \text{ N}$

taking moment about point A

Z is the reaction force which is $W/4$
 $= 78970/4$
 $= 19742 \text{ N}$

mg at CG is 78970.5 N

Fy is centrifugal force which is $1/2 \times m \times r^2$
 $h = 1755 \text{ mm}$
 $b/2 = 2005/2 = 1002.5 \text{ mm}$

putting in equilibrium

$$mg \times b/2 + Fy \times x \cdot h = Z \times x \cdot b$$

$$79657.2 \times 1.0025 + Fy \times 1.755 = 19914.3 \times 2.005$$

$$Fy = 22756 \text{ N}$$

3.2 Finite Element Analysis

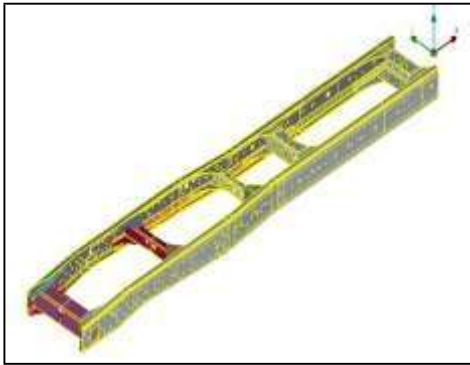


Figure 1-CAD Model of Chasis

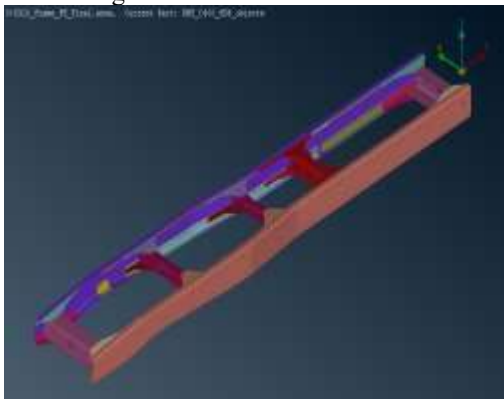


Figure 2-Finite Element Meshing

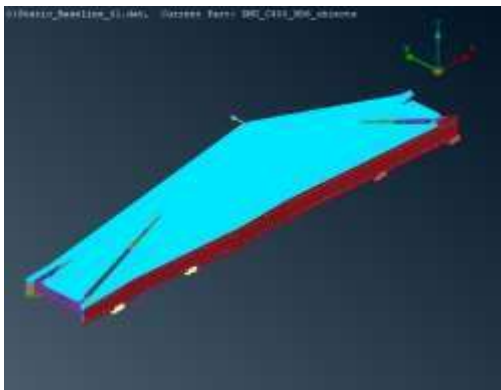


Figure 3-UDL + Cornering - Boundary Conditions

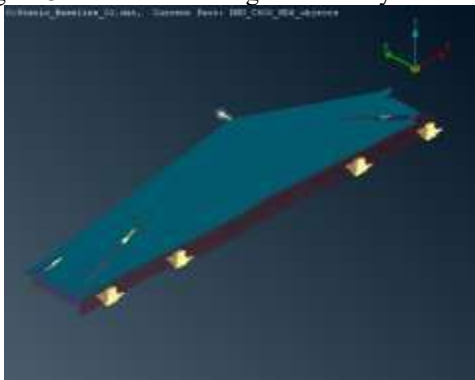


Figure 4-Force applied

➤ Solution Setup



Figure 5-1st Torsion Mode Analysis (10.24Hz)



Figure 6-Lateral Bending Mode Analysis (26.83Hz)



Figure 7- Vertical Bending Analysis (40.80Hz)



Figure 8-2nd Order Torsion Analysis (51.22Hz)

IV. RESULT & DISCUSSION

Table 1-Result of load case 1 FEA Analysis ((UDL+ Cornering)

Load case 1	UDL+ Cornering		
	Von mises stress in Mpa	Deformation in mm	Weight in kg
Baseline	25.47	0.078	604.14
iteration 01	26.14	0.0789	592
iteration 02	28.57	0.086	565
iteration 03	30.5	0.088	541

The table presents the results of Load Case 1 (UDL + Cornering) from a Finite Element Analysis (FEA), tracking how design iterations affect von Mises stress, deformation, and weight. The Baseline design shows a stress of 25.47 MPa, deformation of 0.078 mm, and weight of 604.14 kg. As iterations progress from 01 to 03, stress increases incrementally to 26.14, 28.57, and 30.5 MPa, while deformation rises slightly to 0.0789, 0.086, and 0.088 mm, indicating a stiffer but more loaded structure. Conversely, weight reduces significantly across iterations—from 592 kg to 565 kg, then to 541 kg—demonstrating successful mass reduction. However, this weight optimization comes at the cost of higher stress and deformation, suggesting a trade-off between lightweighting and structural performance under combined uniform distributed load and cornering forces.

Table 2- Result of load case 2 FEA Analysis (UDL+ Cornering + Bump Load)

Load case 2	UDL+ Cornering+ Bump Load		
	Von mises stress in Mpa	Deformation in mm	Weight in kg
Baseline	175.4	0.83	604.14
iteration 01	176.4	0.84	592
iteration 02	176.6	0.84	565
iteration 03	176.9	0.842	541

The table for Load Case 2 (UDL + Cornering + Bump Load) shows how increased loading conditions affect stress, deformation, and weight across design iterations. The Baseline design experiences a von Mises stress of 175.4 MPa and deformation of 0.83 mm at a weight of 604.14 kg. As iterations progress from 01 to 03, stress rises modestly from 176.4 MPa to 176.9 MPa, while deformation increases slightly from 0.84 mm to 0.842 mm, indicating that structural stiffness remains relatively stable despite the added bump load. Meanwhile, weight reduces significantly from 592 kg to 541 kg, similar to the trend in Load Case 1. However, unlike the first load case, the stress and deformation values here are substantially higher—nearly seven times greater in stress—

reflecting the severe demands of the bump load. This suggests that while mass reduction is achieved, the structure operates much closer to its limits under combined UDL, cornering, and bump loading conditions.

Table 5.3-Results of analysis

	Baseline Model (Hz)	Final Iteration (003) Model (Hz)	Difference (Δ Hz<1Hz)
1 st Order Torsion Mode	10.24	10.18	0.06
Lateral bending Mode	26.83	26.10	0.73
Vertical Bending	40.80	40.35	0.45
2 nd Order Torsion Mode	51.22	50.40	0.82

The table compares the natural frequencies of the Baseline Model and the Final Iteration (003) Model across four vibration modes, with all differences recorded as less than 1 Hz. For the 1st Order Torsion Mode, the frequency drops marginally from 10.24 Hz to 10.18 Hz (a difference of just 0.06 Hz), indicating that the fundamental torsional stiffness remains almost unchanged despite weight reduction. The Lateral Bending Mode shows a decrease from 26.83 Hz to 26.10 Hz (difference of 0.73 Hz), suggesting a slightly greater but still modest reduction in sideways stiffness. The Vertical Bending Mode decreases from 40.80 Hz to 40.35 Hz (difference of 0.45 Hz), reflecting a small loss in vertical rigidity. The 2nd Order Torsion Mode exhibits the largest difference, dropping from 51.22 Hz to 50.40 Hz (0.82 Hz), which implies that higher-order torsional behavior is somewhat more sensitive to the design changes introduced in the final iteration. Overall, the table demonstrates that the optimized lightweight model (541 kg) successfully preserves the dynamic characteristics of the baseline (604 kg), with no significant frequency shifts that would risk resonance or compromise structural performance under operating conditions.

V. CONCLUSION

This thesis presents the design, analysis, and optimization of a truck chassis using Finite Element Analysis (FEA) under realistic loading conditions. A detailed CAD model was developed and analyzed by applying boundary conditions and two major load cases (UDL + Cornering and UDL + Cornering + Bump Load) along with modal analysis to assess both structural and dynamic performance. The study achieved effective optimization through iterative design modifications, resulting in a significant weight reduction of approximately 10.5% (from 604.14 kg to 541 kg) while maintaining structural safety. Under Load Case 1, stress and deformation showed controlled increases, whereas under Load Case 2, stress remained nearly constant (~175–177 MPa) with negligible deformation variation, confirming stability under severe conditions. Modal analysis indicated only minor reductions in natural frequencies (all below 1 Hz), ensuring no

risk of resonance. Overall, the optimized chassis demonstrates improved efficiency, reduced weight, and reliable performance, validating the effectiveness of the FEA-based design approach for practical automotive applications.

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