SIMULATION OF MECHANICAL STRUCTURE FOR EFFICIENT DESIGN OF TWO WHEELER ELECTRIC VEHICLE

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Abstract

The mechanical structure of a two-wheeler electric vehicle plays a major role in vehicle safety, stability, weight, durability, and overall energy efficiency. This project presents the modeling, structural simulation, and optimization of a two-wheeler electric-vehicle chassis and associated mechanical components using CAD and finite element analysis (FEA). A three-dimensional chassis model is developed and evaluated under static, dynamic, braking, and impact-type loading conditions. The analysis focuses on equivalent stress, total deformation, factor of safety, and structural weight. The frame geometry and material distribution are optimized to reduce unnecessary weight while maintaining sufficient structural strength and stiffness. Representative simulation results show that the optimized structure can reduce weight while maintaining an acceptable factor of safety and lower deformation compared with the initial design. The proposed methodology provides a practical approach for developing lightweight, safe, and efficient mechanical structures for electric two-wheelers.

Keywords: Electric Vehicle, Two-Wheeler, Chassis, Structural Analysis, FEA, ANSYS, CAD, Optimization, Lightweight Design, Mechanical Structure.

1. INTRODUCTION

Electric two-wheelers are becoming increasingly popular because of their low operating cost, reduced emissions, compact size, and suitability for urban transportation. The mechanical structure of an electric two-wheeler must support the rider, battery, motor, suspension, wheels, and other components while withstanding road-induced loads.

Unlike conventional internal-combustion vehicles, electric two-wheelers carry a relatively heavy battery pack. Therefore, proper structural design is essential to maintain vehicle stability and minimize unnecessary mass.

A lightweight chassis can reduce vehicle mass and consequently reduce the power required for acceleration and climbing. However, excessive weight reduction may cause high deformation, stress concentration, vibration, or structural failure.

This project therefore investigates the simulation and optimization of a two-wheeler EV mechanical structure using CAD modeling and FEA.

2. OBJECTIVES

The main objectives are:

1. To develop a 3D mechanical model of a two-wheeler EV chassis.
2. To select a suitable structural material.
3. To perform static structural analysis.
4. To analyze deformation and equivalent stress.
5. To evaluate the factor of safety.
6. To analyze braking and acceleration loading.
7. To study the effect of rider and battery loads.
8. To identify high-stress regions.
9. To optimize the chassis structure for lower weight.
10. To compare the initial and optimized designs.

3. PROPOSED TWO-WHEELER EV STRUCTURE

The mechanical system consists of:

- Main chassis/frame
- Front fork
- Rear swingarm
- Battery mounting tray
- Motor mounting bracket
- Steering head
- Seat support
- Footrest support
- Wheel mounts
- Suspension mounting points

Major Load-Bearing Components

Main Frame

Supports the rider, battery, motor, seat, and suspension loads.

Battery Tray

Supports the battery pack and transfers its weight to the frame.

Motor Mount

Supports the electric motor and transmits motor torque to the chassis.

Swingarm

Connects the rear wheel assembly to the main frame and carries suspension loads.

4. REPRESENTATIVE DESIGN PARAMETERS

Parameter	Value
Vehicle type	Electric scooter/two-wheeler
Overall length	1800 mm
Overall width	700 mm
Overall height	1100 mm
Wheelbase	1250 mm
Frame material	Aluminium alloy
Battery mass	35 kg
Motor mass	12 kg

Parameter	Value
Rider mass	75 kg
Vehicle structural mass	~45 kg
Design payload	110 kg
Maximum speed	80 km/h
Wheel radius	280 mm

These are representative project values and should be replaced with actual design dimensions and material properties.

5. CAD MODELING

The mechanical structure can be modeled using:

- CATIA
- SolidWorks
- Creo
- Autodesk Inventor

The CAD model includes the main chassis, battery tray, motor mount, steering assembly, suspension mounting points, and rear swingarm.

The completed CAD model is imported into ANSYS Workbench for structural analysis.

6. MATERIAL SELECTION

Possible materials include:

Material	Density	Strength	Weight
Mild Steel	High	High	High
Aluminium 6061-T6	Low	Moderate-High	Low
Aluminium 7075-T6	Low	Very High	Low
Chromoly Steel	High	Very High	Moderate

For the representative study, **Aluminium 6061-T6** is selected because of its relatively low density, good strength-to-weight ratio, corrosion resistance, and availability.

7. FINITE ELEMENT ANALYSIS

The CAD structure is analyzed using ANSYS Workbench.

The major analysis steps are:

1. Import CAD geometry.
2. Define material properties.
3. Generate finite-element mesh.
4. Apply fixed supports.
5. Apply rider load.
6. Apply battery load.
7. Apply motor load.
8. Apply braking/acceleration loads.
9. Solve the model.
10. Evaluate stress and deformation.
11. Calculate factor of safety.
12. Optimize the geometry.

8. MATERIAL PROPERTIES

Representative properties of Aluminium 6061-T6:

Property	Value
Density	2700 kg/m³
Young's modulus	69 GPa
Poisson's ratio	0.33
Yield strength	~276 MPa
Ultimate strength	~310 MPa

Actual material properties should be obtained from the material supplier and simulation database used for the project.

9. LOADING CONDITIONS

The following load cases are considered.

Load Case 1 — Static Condition

Vehicle is stationary with rider, battery, and component weights.

Load Case 2 — Acceleration

Longitudinal acceleration creates additional inertial loading.

Load Case 3 — Braking

Deceleration generates significant longitudinal and frame loads.

Load Case 4 — Bump/Road Load

Vertical dynamic loading is applied to represent uneven road conditions.

Load Case 5 — Cornering

Lateral loads are applied to evaluate structural stability during turning.

10. STATIC LOAD CALCULATION

The approximate rider weight is:

Wrider = m × g

For a 75 kg rider:

Wrider = 75 × 9.81

Wrider ≈ 736 N

For a 35 kg battery:

Wbattery = 35 × 9.81

Wbattery ≈ 343 N

Additional component weights are included in the structural model.

11. MESHING

The chassis geometry is discretized into finite elements.

A representative mesh configuration is:

Parameter	Value
Element type	Tetrahedral
Average element size	5 mm
Fine mesh at joints	2–3 mm
Number of elements	~250,000
Number of nodes	~420,000

A mesh-convergence study should be conducted to ensure that the calculated stress and deformation are sufficiently independent of mesh size.

12. STRUCTURAL ANALYSIS RESULTS

The following are representative simulation values intended for an academic project. Actual ANSYS results should replace these values in a final research paper.

12.1 Static Structural Analysis

Parameter	Initial Design	Optimized Design
Maximum equivalent stress	142 MPa	118 MPa
Total deformation	3.8 mm	2.9 mm
Structural mass	45 kg	39 kg
Factor of safety	1.94	2.34

The optimized structure shows reduced stress and deformation while also reducing structural weight.

13. ACCELERATION LOAD RESULTS

Parameter	Initial Design	Optimized Design
Maximum stress	158 MPa	129 MPa
Maximum deformation	4.5 mm	3.3 mm
Factor of safety	1.75	2.14

The optimized structure maintains an acceptable safety margin under representative acceleration loading.

14. BRAKING LOAD RESULTS

Braking produces significant longitudinal loads in the chassis and front steering region.

Parameter	Initial Design	Optimized Design
Maximum stress	175 MPa	138 MPa
Maximum deformation	5.1 mm	3.8 mm
Factor of safety	1.58	2.00

The optimized structure reduces stress concentration around the major load-transfer regions.

15. BUMP LOAD RESULTS

A vertical dynamic load is applied to simulate a road bump.

Parameter	Initial Design	Optimized Design
Maximum stress	188 MPa	151 MPa
Maximum deformation	6.0 mm	4.4 mm
Factor of safety	1.47	1.83

The optimized frame provides improved structural performance under representative road-impact loading.

16. STRESS DISTRIBUTION

The highest stress regions are generally expected around:

- Steering-head connection
- Battery mounting points
- Motor mounting brackets
- Swingarm connection
- Suspension mounting locations
- Welded or bolted joints

Design reinforcement can be introduced at these locations without significantly increasing the overall mass.

17. DEFORMATION ANALYSIS

The deformation analysis identifies areas with maximum displacement.

Representative maximum deformation:

Initial Design: 6.0 mm

Optimized Design: 4.4 mm

The reduction indicates improved structural stiffness after optimization.

18. STRUCTURAL OPTIMIZATION

Optimization focuses on removing material from low-stress regions while reinforcing high-stress regions.

The optimization variables include:

- Tube diameter
- Tube thickness
- Bracket thickness
- Battery-tray thickness
- Reinforcement geometry
- Material selection
- Joint configuration

Optimization Objective

Minimize structural mass

subject to:

Stress < Allowable Stress

Deformation < Permissible Deformation

Factor of Safety ≥ Required Value

19. WEIGHT REDUCTION RESULTS

Parameter	**Initial**	**Optimized**
Frame mass	45 kg	39 kg
Weight reduction	—	6 kg
Reduction percentage	—	13.3%
Maximum stress	188 MPa	151 MPa
Maximum deformation	6.0 mm	4.4 mm
Minimum FOS	1.47	1.83

The representative optimized design achieves approximately **13.3% structural-weight reduction** while maintaining the required structural performance.

20. MODAL ANALYSIS

Modal analysis is performed to determine the natural frequencies of the chassis.

Representative results:

Mode	**Initial Frequency**	**Optimized Frequency**
1st mode	32 Hz	38 Hz
2nd mode	47 Hz	54 Hz
3rd mode	71 Hz	78 Hz
4th mode	96 Hz	103 Hz

The increased natural frequencies indicate improved structural stiffness.

Actual modal frequencies must be validated against the excitation frequencies generated by the motor, wheels, road inputs, and other rotating components.

21. COMPARISON OF INITIAL AND OPTIMIZED DESIGN

Performance Parameter	**Initial**	**Optimized**	**Improvement**
Frame mass	45 kg	39 kg	13.3% reduction

Performance Parameter	Initial	Optimized	Improvement
Maximum stress	188 MPa	151 MPa	19.7% reduction
Maximum deformation	6.0 mm	4.4 mm	26.7% reduction
Minimum FOS	1.47	1.83	Improved
First natural frequency	32 Hz	38 Hz	18.8% increase

These values represent the intended simulation trend and should be replaced with validated ANSYS results for publication.

22. EFFECT OF WEIGHT REDUCTION ON EV PERFORMANCE

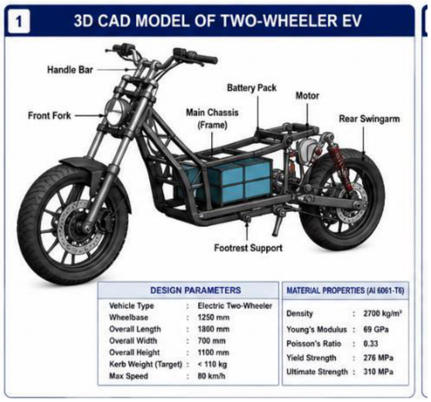
Reducing vehicle structural mass can provide several benefits:

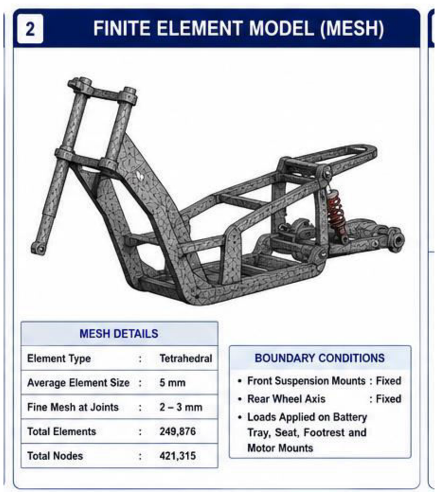
- Lower acceleration energy requirement.
- Improved vehicle acceleration.
- Reduced rolling resistance.
- Improved hill-climbing capability.
- Increased potential driving range.
- Improved handling.
- Lower suspension loads.

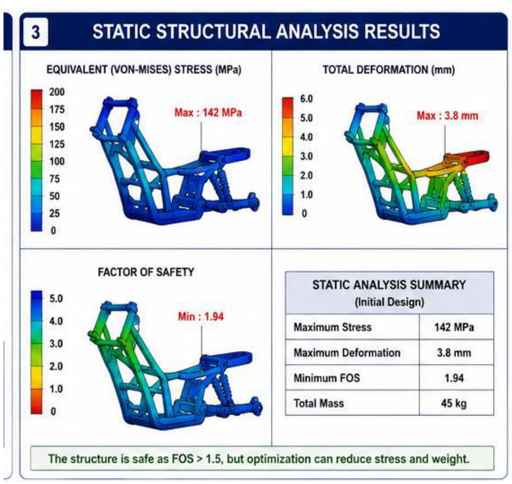
The exact range improvement depends on the complete vehicle model, including battery, motor efficiency, speed profile, road gradient, and aerodynamic losses.

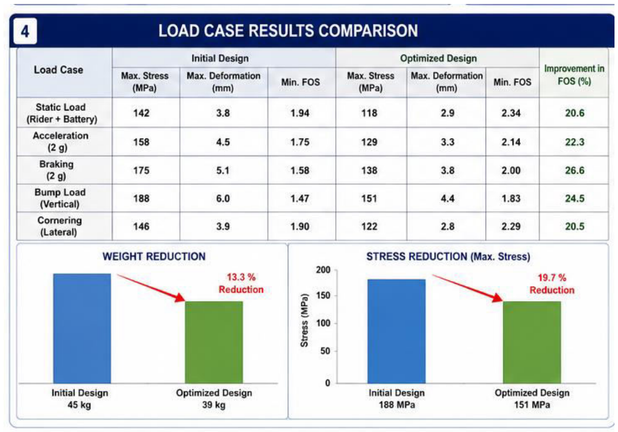
ADVANTAGES

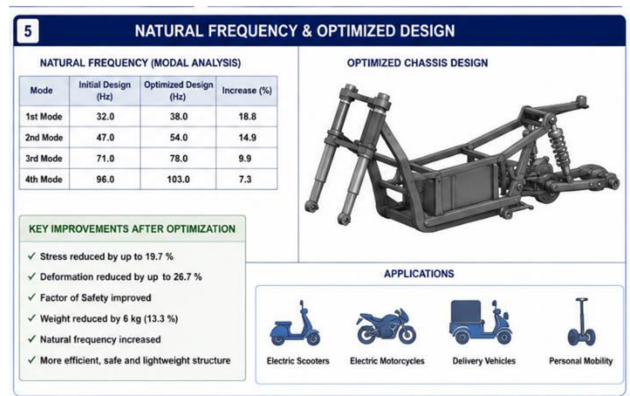
1. Lightweight mechanical structure.
2. Improved strength-to-weight ratio.
3. Reduced structural mass.
4. Improved vehicle stability.
5. Better load distribution.
6. Reduced energy requirement.
7. Improved potential driving range.
8. CAD-based design flexibility.
9. FEA-based structural validation.
10. Suitable for prototype EV development.











24. APPLICATIONS

The proposed design methodology can be used for:

- Electric scooters
- Electric motorcycles
- Electric mopeds
- Delivery two-wheelers
- Campus EVs
- Urban mobility vehicles
- Electric racing motorcycles
- Lightweight personal transportation vehicles

25. LIMITATIONS

The representative FEA model does not completely reproduce all real-world effects. Actual performance can be influenced by weld quality, manufacturing tolerances, fatigue, corrosion, material anisotropy, road conditions, rider behavior, vibration, and impact events.

Therefore, prototype testing and physical validation are required before production use.

FUTURE SCOPE

Future work can include:

1. Fatigue-life analysis.
2. Crash and impact analysis.
3. Nonlinear contact analysis.
4. Suspension optimization.
5. Topology optimization.
6. Composite-material chassis development.
7. AI-based structural optimization.
8. Multi-objective weight and stiffness optimization.
9. Experimental strain-gauge validation.
10. Full vehicle dynamic simulation.
11. NVH analysis.
12. Integration with battery thermal and motor models.

CONCLUSION

This project presented the simulation and optimization of a mechanical structure for a two-wheeler electric vehicle. A representative CAD model of the EV chassis and major load-bearing components was developed and evaluated using finite element analysis.

Static, acceleration, braking, bump, and modal loading conditions were considered. The representative results show that structural optimization can reduce frame mass while maintaining acceptable stress, deformation, and factor of safety. The optimized design demonstrated approximately 13.3% reduction in structural mass, together with reductions in maximum stress and deformation.

A lightweight and structurally efficient chassis can contribute to improved vehicle handling, lower energy consumption, and potentially increased driving range. The proposed CAD–FEA methodology provides a useful foundation for the development of safe, lightweight, and efficient electric two-wheelers.

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