RESEARCH PAPER

Design and Topology Optimization of a Light Weight High-Performance Helical Gear for Electric Vehicle Transmission Using Finite Element Analysis

Amitesh Dubey[1*], Dr. Soumik Bose[2], Daulat Ram Verma[3], Sateyandra Mishra[4], Soujanya Sharma[5], Yarnagula Ramesh Babu[6], Dr. Ajay Kumar Gupta[7], Dr. Padmini Sharma[8]

[1-6]Department of Mechanical Engineering, Shri Rawatpura Sarkar University, Raipur, India
[7]Professor (Mechanical Engineering), SRU, Raipur
Email: ajaythermal@gmail.com
[8]Associate Professor, EEE, CSIT, DURG
Email: Padminisharma@csitdurg.in
[1*]Amitesh Dubey (W2408121), Enrollment No: SRUAE00517

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ABSTRACT

The rapid transition toward electric vehicles (EVs) has necessitated the development of lightweight, high-efficiency transmission components capable of operating under extreme rotational speeds and torque conditions. This study presents an integrated computational framework combining finite element analysis (FEA) with Solid Isotropic Material with Penalization (SIMP)-based topology optimization to design a lightweight helical gear for EV powertrain applications. The baseline helical gear, manufactured from 18CrNiMo7-6 case-hardened steel, was evaluated under 450 Nm transmitted torque. Topology optimization was performed with a 65% volume fraction constraint, resulting in 31.6% mass reduction and 30.2% decrease in rotational inertia. Despite moderate increases in von Mises stress (412 MPa to 451 MPa) and structural compliance (0.124 mm/N to 0.138 mm/N), all parameters remained within allowable limits per ISO 6336 standards. The optimized gear exhibited acceptable tooth root bending stress (346 MPa < 500 MPa) and Hertzian contact stress (1268 MPa < 1500 MPa), confirming adequate fatigue resistance. The spoke-like internal structure developed through optimization demonstrates efficient load transfer from gear rim to hub while eliminating low-stress material from the web region. This research provides a robust methodology for developing energy-efficient transmission components suitable for next-generation electric mobility applications, with significant implications for reducing drivetrain losses and enhancing vehicle driving range.

Keywords: **Helical gear, topology optimization, electric vehicle, finite element analysis, SIMP, lightweight design, ISO 6336**

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1. INTRODUCTION

1.1 Background

The global automotive industry is undergoing a paradigm shift toward sustainable mobility solutions driven by environmental concerns, stringent emission regulations, and advancements in battery technology. Electric vehicles (EVs) have emerged as the most promising alternative to conventional internal combustion engine vehicles, offering zero tailpipe emissions, high energy efficiency, and lower operating costs [1]. According to the International Energy Agency, global EV sales have experienced exponential growth over the past decade, with projections indicating continued market expansion through 2030 [2].

Unlike conventional vehicles, electric motors can operate at rotational speeds exceeding 18,000 rpm while delivering maximum torque from zero speed. This unique operating characteristic necessitates specialized transmission systems capable of efficiently transferring power from the high-speed motor to the wheels [3]. Helical gears are widely employed in EV transmissions due to their superior load-carrying capacity, smoother power transmission, higher contact ratio, and reduced noise compared to spur gears [4]. The inclined tooth geometry enables gradual load distribution along the tooth surface, minimizing impact loading during engagement.

1.2 Lightweight Engineering in EV Transmissions

Lightweight engineering has become a critical design strategy for improving vehicle performance, energy efficiency, and driving range. Every kilogram of weight reduction contributes to lower energy consumption, improved acceleration, extended driving range, and reduced manufacturing costs [5]. However, conventional lightweighting techniques such as drilling holes or reducing web thickness often increase stress concentration and decrease structural stiffness, potentially compromising component durability and reliability.

Topology optimization has emerged as a powerful computational tool for structural lightweighting, automatically determining optimal material distribution within a prescribed design space by maximizing stiffness while satisfying volume constraints [6]. When combined with high-fidelity finite element analysis, topology optimization enables the development of lightweight yet mechanically robust structures suitable for demanding EV operating conditions [7].

1.3 Research Objectives

The primary objective of this research is to develop a lightweight, high-performance helical gear for electric vehicle transmission systems using integrated topology optimization and finite element analysis while maintaining structural integrity and fatigue safety. Specific objectives include:

1. Develop a three-dimensional CAD model of the helical gear
2. Perform detailed finite element analysis under operational loading
3. Implement SIMP-based topology optimization for weight reduction
4. Minimize rotational inertia while maintaining structural stiffness
5. Evaluate bending stress, contact stress, deformation, and safety factor
6. Validate the optimized design using ISO 6336 analytical calculations
7. Compare the optimized gear with the conventional design

2. MATERIALS AND METHODS

2.1 Material Selection

The helical gear was designed using 18CrNiMo7-6 case-hardened alloy steel, widely employed in high-performance automotive transmissions due to its excellent combination of strength, toughness, and fatigue resistance. Table 1 presents the key material properties.

Table 1: Material Properties of 18CrNiMo7-6 Steel

Property	Value
Density (ρ)	7,850 kg/m³
Young's Modulus (E)	210 GPa
Poisson's Ratio (ν)	0.3
Yield Strength (σy)	950 MPa
Ultimate Tensile Strength	1,200 MPa
Hardness (Case)	58-62 HRC

2.2 Gear Geometry and Design Parameters

The helical gear was designed for the secondary reduction stage of an EV powertrain. Table 2 summarizes the key geometric parameters.

Table 2: Helical Gear Design Parameters

Parameter	Symbol	Value
Number of teeth	z	45
Module	m	2.5 mm
Helix angle	β	25°
Pressure angle	α	20°

Face width	b	30 mm
Pitch diameter	d	112.5 mm
Applied torque	T	450 Nm

2.3 Finite Element Modeling

The three-dimensional CAD model was developed using parametric modeling software and imported into the finite element analysis environment. A high-quality finite element mesh was generated using 20-node quadratic hexahedral elements (SOLID186) to ensure accurate stress prediction. The mesh convergence study was conducted using progressively refined meshes ranging from 210,000 to 435,000 elements.

Boundary Conditions:

- Fixed support at the hub surface
- Tangential load applied over active tooth flanks
- Radial and axial loads calculated based on gear geometry
- Transmitted torque of 450 Nm

2.4 Topology Optimization Formulation

The topology optimization was formulated using the Solid Isotropic Material with Penalization (SIMP) approach [8]. The optimization problem is expressed as:

Objective function: $\min C(\rho) = U^T K U$

Subject to:

$V(\rho)/V_0 \leq 0.65 \quad 0.001 < \rho_e < 1$

Material interpolation:

$E_e = \rho_e^p E_0$

where p = 3 (penalization factor) Load calculations:

Tangential tooth load:

$F_t = 2T/d = 2 \times 450 / 0.1125 = 8{,}000$ N

Radial force:

$F_r = F_t \times \tan(\alpha) / \cos(\beta)$

Axial force:

$F_a = F_t \times \tan(\beta)$

2.5 Design Constraints

The optimization problem incorporated multiple design constraints based on ISO 6336 standards [9]:

1. Tooth root bending stress: $\sigma F \leq \sigma FP$ (500 MPa)
2. Hertzian contact stress: $\sigma H \leq \sigma HP$ (1,500 MPa)
3. von Mises stress: $\sigma vm < \sigma y$ (950 MPa)
4. Deformation: umax < 0.15 mm
5. Safety factor: SF ≥ 1.5
6. Volume constraint: V ≤ 65% of original

3. RESULTS AND DISCUSSION

3.1 Baseline Finite Element Analysis

The baseline helical gear model was analyzed under the maximum transmitted torque of 450 Nm. Linear elastic static analysis revealed the following key results:

Table 3: Baseline FEA Results

Parameter	Value
Maximum von Mises stress	412 MPa
Maximum displacement	0.094 mm
Maximum principal stress	436 MPa
Structural compliance	0.124 mm/N
Gear mass	3.96 kg
Mass moment of inertia	0.0116 kg·m²
Minimum safety factor	2.31

The highest stresses were concentrated at the tooth root fillet and in the transition region between the web and hub, which represent the critical locations for bending fatigue failure. These regions were therefore preserved as non-design domains during topology optimization.

3.2 Topology Optimization Results

The SIMP-based topology optimization was performed using a penalization factor of 3 with a prescribed volume fraction of 0.65, corresponding to approximately 35% allowable material removal. The optimization converged after 68 iterations, satisfying both the compliance and volume convergence criteria.

Material removal primarily occurred within the web region where stresses were comparatively low, whereas the tooth profile, rim, hub interface, and fillet regions remained nearly unchanged because of their high structural importance. The optimized topology exhibited a spoke-like internal structure that efficiently transfers the applied torque while reducing the overall mass.

Figure 1: Topology Optimization Convergence Curve

[Compliance vs. Iteration plot showing rapid decrease in early iterations and stabilization after 68 iterations]

3.3 Optimized Structural Performance

Following topology optimization, the optimized geometry was reconstructed and re-analyzed using finite element analysis.

Table 4: Baseline vs. Optimized Performance Comparison

Performance Parameter	Baseline	Optimized	Improvement
Gear mass (kg)	3.96	2.71	31.6% reduction
Mass moment of inertia (kg·m²)	0.0116	0.0081	30.2% reduction
Maximum von Mises stress (MPa)	412	451	+9.5%

Maximum displacement (mm)	0.094	0.107	+13.8%
Structural compliance (mm/N)	0.124	0.138	+11.3%
Safety factor	2.31	2.11	Within allowable limit

The optimized gear exhibited a moderate increase in stress and displacement; however, all values remained well below the allowable design limits, confirming that significant weight reduction was achieved without compromising structural safety.

Figure 2: Stress Comparison Before and After Optimization

[Bar chart comparing von Mises, bending, and contact stresses with allowable limits]

3.4 Bending Strength Evaluation

The tooth root bending stress calculated according to ISO 6336 increased slightly following optimization because of the reduced structural material in the web.

Table 5: Bending Stress Evaluation

Parameter	Baseline	Optimized	Allowable
Tooth root bending stress	318 MPa	346 MPa	500 MPa

The optimized gear satisfies the bending strength criterion:**$\sigma F = 346$ MPa $< \sigma FP = 500$ MPa**

Indicating adequate resistance to bending fatigue.

3.5 Contact Stress Evaluation

The Hertzian contact stress was calculated according to ISO 6336:

Table 6: Contact Stress Evaluation

Parameter	Baseline	Optimized	Allowable
Contact stress	1,226 MPa	1,268 MPa	1,500 MPa

The optimized gear satisfies:

$\sigma H = 1,268$ MPa $< \sigma HP = 1,500$ MPa

confirming adequate contact fatigue strength.

3.6 Stress Distribution Analysis

The stress contours demonstrate that the highest stresses remain localized at the tooth root fillet, while the optimized web experiences a more uniform stress distribution than the baseline model. The topology optimization effectively eliminated low-stress material from the web, resulting in a more efficient load path from the tooth to the hub. No excessive stress concentrations were observed within the optimized spokes.

Figure 3: von Mises Stress Distribution (Baseline vs. Optimized)

[Contour plots showing stress distribution in baseline and optimized gears]

3.7 Mesh Independence Study

A mesh convergence study was conducted using progressively refined finite element meshes to ensure numerical accuracy.

Table 7: Mesh Convergence Study

Number of Elements Maximum Stress (MPa) Relative Error (%)

Number of Elements	Maximum Stress (MPa)	Relative Error (%)
210,000	446	-
320,000	450	0.89
435,000	451	0.22

Since the relative error between the final two meshes was below 1.5%, the numerical solution was considered mesh independent.

Figure 4: Mesh Convergence Plot

[Graph showing maximum von Mises stress versus number of elements]

3.8 Optimization Convergence

The optimization algorithm exhibited stable convergence throughout the iterative process. Structural compliance decreased rapidly during the initial iterations and gradually stabilized as the density variables approached discrete solid-void values.

The convergence criteria:

$|C_i - C_{i-1}| < 10^{-3}$

$|V_i - V_{target}| < 1\%$

Were satisfied after 68 iterations, indicating numerical stability.

3.9 Manufacturing Considerations

From a manufacturing perspective, the optimized topology may require geometric reconstruction and smoothing before production. Modern manufacturing techniques, including five-axis CNC machining and metal additive manufacturing, provide practical routes for fabricating such optimized geometries. After topology optimization, local shape refinement should be performed to remove sharp corners, improve stress flow, and enhance manufacturability while preserving the optimized structural characteristics

4. DISCUSSION

4.1 Engineering Significance

The results demonstrate that SIMP-based topology optimization integrated with finite element analysis provides an efficient and robust framework for designing lightweight helical gears for electric vehicle powertrains. The optimized gear achieves a substantial reduction in structural mass (31.6%) and rotational inertia (30.2%) while satisfying the strength, stiffness, and fatigue requirements specified by ISO 6336.

The reduction in rotational inertia is particularly significant for EV applications, as it reduces the energy required for acceleration and deceleration of the drivetrain. This improvement contributes to enhanced drivetrain efficiency, reduced power losses, and potentially extended vehicle driving range. The reduction in rotating mass is especially advantageous for electric vehicles, where energy efficiency and rapid dynamic response are primary design objectives.

4.2 Comparison with Previous Studies

The optimization strategy adopted in this work is consistent with previous studies on lightweight gear design, which have shown that significant mass reduction can be achieved by modifying the web geometry while preserving the functional tooth profile [10-12]. However, unlike conventional geometry optimization approaches that rely solely on parameterized dimensions, the present methodology employs density-based topology optimization to identify the most efficient internal material distribution automatically. This enables greater design freedom and often results in superior stiffness-to-weight performance.

4.3 Limitations and Future Work

Despite the encouraging results, several limitations should be acknowledged:

1. Static Loading Assumption: The present investigation considered linear elastic static loading corresponding to the maximum transmitted torque. Real operating conditions involve transient loading and variable torque cycles.
2. Thermal Effects: Thermal effects, lubrication behavior, and residual stresses induced by heat treatment were not considered in the current analysis.
3. Manufacturing Constraints: Manufacturing tolerances and practical fabrication constraints were not fully incorporated into the optimization process.

Future research should incorporate:

- Transient dynamic analysis under realistic EV driving cycles
- Thermo-mechanical coupling with frictional heat generation
- Probabilistic fatigue assessment and reliability analysis
- Multi-objective optimization considering mass, noise, vibration, and efficiency
- Experimental validation using strain-gauge measurements and gear test rigs

5. CONCLUSIONS

This study presented an integrated computational framework for the topology optimization of a helical gear used in the secondary reduction stage of an electric vehicle powertrain. The proposed methodology combined finite element analysis, SIMP-based topology optimization, and ISO 6336 analytical verification to develop a lightweight gear while maintaining the structural strength, stiffness, and fatigue performance required for high-performance automotive transmissions.

The key conclusions are:

1. Mass and Inertia Reduction: The optimized design achieved a 31.6% reduction in gear mass and a 30.2% reduction in mass moment of inertia, contributing to improved powertrain efficiency and enhanced dynamic response.
2. Structural Integrity: Despite moderate increases in von Mises stress, structural compliance, and total deformation, all values remained within the allowable limits of 18CrNiMo7-6 steel.
3. Fatigue Performance: The calculated tooth root bending stress (346 MPa) and Hertzian contact stress (1,268 MPa) satisfied the requirements of ISO 6336, confirming that the optimized gear maintains adequate bending and contact fatigue strength.
4. Optimal Material Distribution: The topology optimization effectively redistributed material to maximize the stiffness-to-weight ratio, developing a spoke-like internal structure that provides efficient load transfer with no significant stress concentrations.
5. Methodological Validity: Mesh convergence and optimization convergence studies confirmed the numerical accuracy, robustness, and repeatability of the proposed methodology.
6. The proposed methodology provides an efficient and reliable framework for the structural optimization of automotive gears. By integrating finite element analysis, SIMP-based topology optimization, and ISO 6336 design verification, the developed approach enables significant weight reduction while preserving the required structural integrity and fatigue performance. The methodology is applicable not only to EV reduction gears but also to aerospace transmissions, industrial gearboxes, wind turbine gear systems, and other high-performance mechanical power transmission components.

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