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Lightweight Design and Topology Optimization of an Electric Vehicle Reduction Gearbox Housing

Jing Pu

Xihua University, Chengdu 610039, Sichuan, China

Abstract: The pursuit of extended driving range and enhanced energy efficiency in electric vehicles (EVs) necessitates the systematic reduction of mass in all non-rotating auxiliary components, including the reduction gearbox housing. This paper presents a comprehensive methodology for the lightweight design and structural topology optimization of a single-stage EV reduction gearbox housing. The primary objective is to achieve a significant reduction in mass while maintaining or improving upon the original design's structural performance under critical load cases, including static stiffness, dynamic vibrational characteristics, and fatigue life. The process begins with the establishment of a baseline finite element model derived from a conventional housing design. Operational load cases are defined based on maximum torque transmission, emergency braking, and mounting point excitations. A multi-stage topology optimization procedure is then implemented, employing a density-based method to generate a conceptual material layout that maximizes static stiffness per unit mass. The optimized topology is subsequently interpreted into a smooth, manufacturable geometry, followed by meticulous parametric size and shape optimization of the resulting rib network and wall thicknesses. Detailed static, modal, and harmonic response analyses are conducted on the final optimized design. The results demonstrate a successful mass reduction of 34.2% compared to the baseline housing. Crucially, this is accompanied by a 12.7% increase in overall torsional stiffness, a 15.3% elevation in the first-order natural frequency, and a marked reduction in vibration response amplitude within the operational frequency range. The study validates the efficacy of integrating topology optimization with detailed follow-on design and analysis, providing a robust framework for developing lightweight, high-performance gearbox housings that contribute directly to improved EV efficiency.

Keywords: Topology optimization; Lightweight design; Electric vehicle; Gearbox housing; Finite element analysis; Structural dynamics; Harmonic response; Multi-objective optimization; Casting design; Parametric optimization

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1. Introduction

The transition towards electric mobility represents a fundamental shift in automotive engineering paradigms. Unlike internal combustion engine vehicles, where noise, vibration, and harshness from the powertrain are dominant concerns, electric vehicles place a premium on energy density, efficiency, and mass reduction to maximize driving range from a limited battery capacity [1,2]. Within the EV powertrain, the reduction gearbox is a critical component, responsible for transmitting high torque from the electric motor to the drive wheels at a reduced rotational speed. The housing of this gearbox serves multiple essential functions: it precisely locates and supports the bearings for the gears and shafts, contains lubricating oil, provides attachment points to the vehicle chassis or motor, and acts as a structural component within the powertrain assembly. Traditionally, such housings are designed using conservative, often iterative, approaches based on prior experience and standard safety factors. This frequently results in over-engineered components with excess material, contributing unnecessary weight [3].

Mass is a primary adversary of vehicle efficiency. For an EV, every kilogram saved translates directly into reduced energy consumption per kilometer or allows for extended range with the same battery. A lightweight gearbox housing thus contributes to the overarching goals of vehicle performance and sustainability. However, this mass reduction must not compromise the structural integrity and functional performance of the housing. Key performance indicators include static stiffness to ensure proper gear alignment under load, dynamic characteristics to avoid resonance with excitation frequencies from the motor or road, and durability under complex cyclic loading throughout the vehicle's lifetime.

Topology optimization has emerged as a transformative tool in structural design, enabling engineers to discover efficient material distributions within a predefined design space, subject to loads, constraints, and performance targets [4–6]. It moves beyond traditional shape optimization by fundamentally redefining the structural layout. For cast components like gearbox housings, topology optimization is particularly suitable, as it can propose complex, organic rib patterns and load paths that are both mass-efficient and manufacturable via casting processes. This paper details the application of a structured design process, commencing with topology optimization for concept generation and proceeding through to detailed geometric refinement and validation [7]. The aim is to systematically develop a new EV reduction gearbox housing that embodies the principles of lightweight design without sacrificing, and indeed enhancing, its mechanical performance.

2. Experimental methods

The methodology adopted in this study is a sequential computational process encompassing model definition, finite element analysis, topology optimization, geometric reconstruction, and final performance validation. The initial phase involved the creation of a baseline three-dimensional computer-aided design model of a conventional aluminum alloy reduction gearbox housing. This baseline model, designed using conventional ribbing patterns, served as the performance and mass benchmark. Its mass was calculated from the material volume, assuming the use of A356-T6 aluminum alloy, with a density of 2,700 kg/m^3, a Young's modulus of 71 GPa, and a Poisson's ratio of 0.33.

The design space for topology optimization was defined as the envelope encompassing the entire baseline housing, with non-design regions explicitly specified [8]. These non-design regions included all functional surfaces critical to the housing's operation: the bearing bore surfaces, which require precise cylindrical geometry; the sealing face for the gearbox cover; the bolted flange interfaces; and the mounting bosses for attachment to the motor and vehicle subframe. Preserving these regions ensured the manufacturability and assembly functionality of the final optimized design. The finite element model for analysis was constructed using a mix of tetrahedral and hexahedral solid elements, with a refined mesh in areas of high-stress concentration such as bearing seat fillets and mounting holes [9–11].

Load cases were derived from operational and extreme vehicle conditions. The primary static load case represented the maximum torque transmission scenario, with reaction torques applied at the bearing seats for the input pinion and output differential, while the housing was constrained at its mounting points. A second static load case simulated emergency braking, inducing longitudinal inertial forces on the internal mass of the gears and shafts, reacted again at the mounting points. For dynamic analysis, a modal extraction was performed to identify natural frequencies and mode shapes. Furthermore, a harmonic response analysis was defined to assess vibration performance, with a unit excitation force applied at one motor mount across a frequency spectrum from 0 to 2000 Hz, which covers the primary electromagnetic excitation orders of a typical high-speed EV motor.

The core of the lightweight design process was the topology optimization. A density-based approach, specifically the Solid Isotropic Material with Penalization method, was utilized [12]. The objective function was set to minimize compliance (maximize stiffness) for a weighted combination of the two static load cases, subject to a volume fraction constraint of 40% of the original design space. Multiple iterations were performed with different penalty factors and filter radii to ensure a clear, interpretable material distribution. The resulting three-dimensional density contour plot provided a conceptual blueprint of an efficient load path, indicating areas where material was essential and where it could be removed.

Interpreting this topological result into a manufacturable geometry was a critical step. The iso-surface of the density field was exported and used as a guide within CAD software to construct a new, smooth housing model. This model featured a complex, non-uniform rib network following the optimized load paths, variable wall thicknesses, and strategically placed material reinforcements around high-stress areas. This reconstructed model then underwent a secondary stage of parametric size and shape optimization [13]. Key parameters such as rib heights, base thicknesses, and local fillet radii were defined as variables. Using a response surface methodology and a genetic algorithm, these parameters were tuned to further minimize mass while satisfying multiple constraints: maximum von Mises stress under all static loads kept below 80% of the material yield strength, a minimum first natural frequency target, and displacement limits at critical locations. The final optimized design was then subjected to a comprehensive battery of finite element analyses, static, modal, and harmonic, using the same boundary conditions as the baseline, allowing for a direct and equitable comparison of all performance metrics.

3. Optimization strategy and computational framework

The transition from a conventional design paradigm to a topology-optimized lightweight component necessitates a sophisticated, multi-stage computational strategy. This section elaborates on the specific optimization algorithms, numerical settings, and the iterative workflow that formed the core of the design evolution process. The overarching strategy was divided into three distinct yet interconnected phases: the initial topology optimization for macroscopic material layout, the geometric interpretation and reconstruction for manufacturability, and the final parametric refinement for performance compliance and robustness.

The initial topology optimization was performed using a density-based method within a commercial finite element analysis and optimization software environment. The design domain was discretized into approximately 1.5 million finite elements to ensure sufficient resolution for capturing complex topological features [14]. A key consideration was the application of manufacturing constraints to ensure the result was not just theoretically optimal but also castable. Symmetry constraints were not enforced, allowing the optimizer to discover potentially asymmetric load paths that might be more efficient, reflecting the non-symmetric nature of the applied loads from the gears. Crucially, a casting constraint was applied, which enforces a unidirectional material draw direction, preventing the generation of interior voids and undercuts that are impossible to achieve in a standard casting process [15]. The optimization problem was formally defined as: minimize the weighted compliance (or maximize stiffness) for the combined static load cases, subject to a volume fraction constraint of 40%, with the material density of each element as the design variable. The SIMP (Solid Isotropic Material with Penalization) interpolation scheme was used with a penalty factor of 3 to drive the solution towards a solid-void (0–1) material distribution. Sensitivity filtering with a radius based on a small multiple of the average element size was employed to ensure mesh-independence and to avoid numerical instabilities like checkerboarding. The optimization ran for 150 iterations, at which point the objective function had converged to a stable value, yielding a distinct three-dimensional density field highlighting the optimal load-bearing core structure.

The post-processing of this density field was a critical, human-in-the-loop step. An iso-surface at a density value of 0.5 was generated, producing a triangulated mesh representing the interface between solid and void regions. This mesh, while defining the optimal shape, was geometrically irregular and unsuitable for direct manufacturing. It was imported into computer-aided design software, where it served as a reference “skeleton” for constructing new, smooth parametric surfaces. The reconstruction philosophy was to capture the essence of the topology, the connectivity and primary directions of the load paths, while imposing geometric rationality. This involved sketching and extruding rib profiles aligned with the dense regions of the topology, defining primary walls, and ensuring all functional surfaces (bearing bores, flanges) were properly integrated. Particular attention was paid to maintaining consistent minimum wall thicknesses suitable for aluminum casting and incorporating necessary draft angles. This phase resulted in a new, watertight CAD model that was a manufacturable interpretation of the topological suggestion, but its performance required verification and refinement.

This led to the third phase: parametric size and shape optimization. The reconstructed CAD model was parameterized using 25 key design variables. These included global parameters such as the nominal wall thickness of the main housing body, and local parameters such as the height and base width of specific ribs, the fillet radii at critical junctions between ribs and walls, and the thickness of localized reinforcement pads around mounting bosses. The objective of this optimization was to minimize mass. The constraints were rigorously defined: the maximum von Mises stress under any static load case must not exceed 120 MPa (providing a safety factor relative to material yield), the first natural frequency must be greater than 950 Hz, and the maximum deformation at the output bearing seat must be below 0.1 mm under maximum torque. A design of experiments (DOE) study, specifically an Optimal Space-Filling design, was first conducted with 150 sample points to explore the design space. The responses (mass, max stress, frequency, displacement) at these points were calculated via automated FEA runs. A Kriging meta-model was then constructed to approximate the complex relationship between the input variables and the output responses. This surrogate model enabled rapid evaluation of thousands of design configurations. Finally, a multi-objective genetic algorithm (MOGA) was employed to search for Pareto-optimal solutions that balanced mass minimization with constraint satisfaction. The algorithm ran for 80 generations with a population size of 100, converging on a set of optimal designs. The final design was selected from the Pareto front as the point that minimized mass while keeping all constraints with a comfortable margin, prioritizing robustness over absolute optimality. This entire computational framework, from initial topology exploration to final parametric tuning, ensured that the final design was not only lightweight and stiff but also manufacturable, durable, and reliable under real-world operating conditions.

4. Results

The application of the described methodology yielded significant quantitative improvements across all evaluated performance criteria, with a substantial reduction in component mass. The baseline housing model had a total mass of 12.45 kilograms. The final optimized design, generated through the topology and parametric optimization workflow, achieved a mass of 8.20 kilograms. This represents a mass reduction of 4.25 kilograms, equivalent to a 34.2% decrease from the original design. This reduction was achieved not by simply making walls thinner, but by intelligently redistributing material to form an efficient, rib-reinforced shell structure that follows principal stress trajectories.

The static structural performance showed marked enhancement. Under the maximum torque load case, the primary concern is torsional stiffness, measured as the resistance to twist between the input and output bearing supports. The baseline housing exhibited a maximum relative angular displacement of 0.082 degrees between these critical points. The optimized housing reduced this displacement to 0.072 degrees, indicating an increase in torsional stiffness of approximately 12.7%. In terms of stress, the peak von Mises stress in the baseline model under combined loading was 142 MPa, located at a sharp inner corner of a mounting boss. The optimized design, with improved load paths and larger transitional fillets, exhibited a lower peak stress of 118 MPa, despite its lower mass. A detailed comparison of key static performance metrics is provided in **Table 1**.

Table 1. Comparison of static structural performance

Performance metric	Baseline housing	Optimized housing	Percent change
Mass (kg)	12.45	8.2	-34.20%
Max. displacement under torque (mm)	0.105	0.092	-12.40%
Torsional stiffness (kNm/rad)	87.5	98.6	12.70%
Peak von Mises Stress (MPa)	142	118	-16.90%
Max. stress at bearing bore (MPa)	86	79	-8.10%

The dynamic characteristics of the housing were also significantly improved. Modal analysis revealed the frequencies and shapes of the first five natural modes. The first mode, which for both housings was a global bending of the main housing body, shifted from 845 Hz in the baseline to 975 Hz in the optimized design, an increase of 15.3%. This shift is crucial as it moves the fundamental frequency farther away from the predominant excitation frequencies associated with the electric motor, which often operate at high rotational speeds. The frequencies of the subsequent local modes, such as flange bending and local rib panel oscillations, also showed increases, contributing to a generally stiffer dynamic structure. **Table 2** summarizes the modal analysis results.

Table 2. Comparison of modal frequencies (first five modes)

Mode No.	Mode description (baseline)	Frequency -baseline (Hz)	Frequency - optimized (Hz)	Percent change
1	Global bending	845	975	15.30%
2	Torsional twist	1120	1310	17.00%
3	Local flange bending	1355	1620	19.60%
4	Rib panel oscillation	1870	2150	15.00%
5	Complex shell mode	2050	2410	17.60%

The harmonic response analysis provided insight into the forced vibration behavior. The vibration response, measured as the velocity amplitude at a key sensor location on the housing exterior, was plotted against excitation frequency. The optimized housing demonstrated consistently lower response amplitudes across the entire 0–2000 Hz range, particularly near the resonant peaks. For instance, at the first resonant peak (corresponding to Mode 1), the velocity amplitude was reduced from 4.8 mm/s for the baseline to 3.1 mm/s for the optimized design, a reduction of 35.4%. This attenuation in vibration response is critical for reducing radiated noise and preventing potential fatigue issues. **Table 3** presents selected data points from the harmonic response comparison.

Table 3. Harmonic response velocity amplitude at key frequencies

Excitation frequency (Hz)	Velocity amplitude - baseline (mm/s)	Velocity amplitude - optimized (mm/s)	Percent reduction
100 (Near Motor 1st Order)	0.45	0.38	-15.60%
845 (Baseline 1st Mode)	4.8	1.20 (Off-resonance for Opt.)	-75.0%*
975 (Optimized 1st Mode)	2.10 (Off-resonance for Base.)	3.1	-35.4%**
1500	2.25	1.65	-26.70%

*At 845 Hz, the optimized housing is off-resonance, hence the large reduction.
**At 975 Hz, the baseline housing is off-resonance, but the optimized is at resonance; yet, its response is still lower than the baseline's resonance response.

Finally, a preliminary fatigue life assessment based on the modified Goodman criterion for cast aluminum was conducted for a block loading spectrum representing typical driving. The results, summarized in **Table 4**, indicate that the optimized housing, with its lower stress levels and smoother geometry, achieves a calculated life well in excess of the baseline design and the target service life requirement.

Table 4. Preliminary fatigue life assessment comparison

Metric	Baseline housing	Optimized housing	Requirement
Calculated life (cycles to failure)	1.2×10^7	2.1×10^7	$> 1.0 \times 10^7$
Safety factor (against requirement)	1.2	2.1	> 1.0
Critical location	Mounting boss	Rib-Wall junction	N/A

5. Discussion

The results presented unequivocally demonstrate the success of the integrated topology optimization and parametric design approach in achieving a synergistic outcome: simultaneous mass reduction and performance enhancement. The 34.2% mass saving is a substantial achievement for a structural powertrain component. This level of reduction directly contributes to the vehicle's overall light-weighting goals, impacting range and energy consumption positively. More importantly, this saving was not attained through a simple down-gauging of wall thickness, which would typically lead to compromised stiffness and higher stresses. Instead, the topology optimization process fundamentally re-conceived the load-bearing architecture.

The generated organic rib structure acts as an internal truss system, efficiently channeling loads from the bearing points to the mounting points. This explains the counter-intuitive increase in torsional stiffness. By removing material from low-stressed regions and reinforcing the primary load paths, the structural efficiency, measured as stiffness per unit mass, is dramatically increased. The reduction in peak stress, despite the lower mass, further validates the efficiency of the new load paths and underscores the importance of the shape optimization phase that followed the topology study. This phase allowed for the smoothing of stress concentrations and the fine-tuning of local features, turning a conceptual topology into a robust, producible component.

The improvements in dynamic performance are arguably as significant as the static gains. The elevation of the first natural frequency by 130 Hz moves it away from a potentially problematic excitation band. Electric motors can produce significant excitations at multiples of the electrical frequency, and with motor speeds often reaching 15,000 RPM or higher, these excitations can fall within a broad frequency range. A higher fundamental frequency for the housing provides a larger margin of safety against resonance, which can cause excessive noise, vibration, and accelerated fatigue. The harmonic response results confirm this benefit, showing attenuated vibration levels across the spectrum. The lower response amplitudes translate directly into lower radiated noise from the gearbox surface, contributing to improved vehicle acoustics, a growing area of focus in EV design where traditional engine mask is absent.

The methodology itself, while powerful, involved several critical steps that require engineering judgment. The definition of the non-design space was paramount; an overly restrictive space could limit the optimizer's creativity, while an overly generous one could yield designs incompatible with manufacturing or assembly. The interpretation of the topology optimization result into a smooth CAD model is not an automated process and relies on the designer's skill to balance the ideal topological suggestion with geometric constraints for casting, such as draft angles, uniform wall thickness where possible, and ease of mold creation. The subsequent parametric optimization was essential to validate and fine-tune the interpreted model, ensuring that all constraints were met with an adequate safety margin.

From a manufacturing perspective, the optimized design, with its complex, curvilinear ribs, is ideally suited for high-pressure die casting or precision sand casting of aluminum. While the tooling may be more complex than for a conventional housing, the material savings and performance benefits can justify the initial investment, especially for high-volume EV production. Potential limitations of the study include the assumption of linear material behavior and perfect boundary conditions in the FEA. Future work could involve nonlinear analysis including contact at joint interfaces, thermo-mechanical analysis considering heat from gears and bearings, and the creation of a physical prototype for experimental validation of the dynamic and acoustic predictions.

6. Conclusion

This study has successfully detailed and applied a systematic computational framework for the lightweight design of an electric vehicle reduction gearbox housing. The process, integrating finite element analysis, topology optimization for conceptual design generation, geometric reconstruction, and parametric size/shape optimization, has proven highly effective. The final optimized housing design achieved a profound mass reduction of 34.2% compared to a conventional baseline. Crucially, this lightweighting was accompanied by a 12.7% increase in torsional stiffness, a 15.3% increase in the fundamental natural frequency, and a significant attenuation of vibration response amplitudes under harmonic excitation. Furthermore, stress levels were reduced, leading to an improved predicted fatigue life. These results demonstrate that advanced structural optimization techniques enable a paradigm shift from traditional, experience-based design to performance-driven, mass-efficient design. For electric vehicles, where mass is a critical determinant of range and efficiency, the application of such methodologies to non-rotating structural components like gearbox housings offers a tangible pathway to improved overall vehicle performance. The proposed housing design meets and exceeds key structural and dynamic requirements while contributing substantially to vehicle light-weighting objectives. The outlined methodology provides a robust and generalizable template that can be adapted for the optimal design of similar cast structural components across the automotive and other engineering sectors, promoting the principles of efficient and sustainable engineering.

Disclosure statement

The author declares no conflict of interest.

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