

RESEARCH ON STRUCTURAL OPTIMIZATION DESIGN OF ELECTRIC VEHICLE DRIVE AXLE HOUSING BASED ON ORTHOGONAL TEST METHOD

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Abstract - To address the problem of excessive weight of electric truck drive axle housing that reduces vehicle range, a multi-objective lightweight optimization based on orthogonal experiments is carried out on a three-section stamped and welded electric drive axle housing. SolidWorks is adopted for geometric modeling, and ANSYS is used to conduct multi-condition static, modal and fatigue simulations. The strength, stiffness, natural frequency and fatigue life of the original axle housing all meet the requirements specified in QC/T 533—2020. Seven structural dimensions are selected as design variables to establish an $L_{27}(3^7)$ orthogonal test scheme. Taking axle housing mass, maximum equivalent stress and first-order natural frequency as evaluation indicators, the optimal parameter combination is solved via equally weighted least squares regression. After optimization, the mass of the axle housing is reduced by 5.03%, and the first-order natural frequency increases by 4.05%. Although the maximum stress rises slightly and the fatigue life decreases marginally, all performance indicators remain within the safe range. The proposed method can realize lightweight design while ensuring structural reliability, and provides a reference for the design of drive axles of electric commercial vehicles.

Keywords: Electric Truck; Drive Axle Housing; Finite Element Method; Orthogonal Test; Lightweight Design

1. Introduction

Vehicle curb weight constitutes one of the critical factors determining the driving range of electric vehicles, and the drive axle housing is a key component contributing to overall vehicle weight. Reasonable configuration of structural parameters for the drive axle housing can guarantee its structural reliability while complying with national energy conservation and emission reduction requirements for electric vehicles. As a primary load-bearing component of automobiles, the drive axle housing bears vertical, longitudinal and lateral forces transmitted between the road surface and vehicle frame/body, as well as braking torque and reaction forces. Together with other assemblies, it supports and protects the final drive, differential and half shafts.

In terms of structural design and optimization of drive axle housings, topological optimization and topology design are widely adopted by domestic and international researchers. Guo et al. [1] optimized the wall thickness dimensions of drive axle housings via optimization algorithms, achieving

lightweight design without compromising structural strength. TOPAÇ et al. [2] carried out lightweight design for drive axle housings by constraining housing volume and combining ADAMS simulation with physical tests. Liu et al. realized a 12% weight reduction of the drive axle housing through finite element-based structural optimization. Numerous studies have verified that the combination of finite element method and topological optimization bears significant engineering application value for lightweight design of drive axle housings [3,4]. Material substitution represents another vital engineering approach to simultaneously realize lightweight design and strength improvement of drive axle housings, yet it usually leads to higher manufacturing costs [5,6].

In recent years, emerging design methodologies have brought innovative solutions to structural design and lightweight research of drive axle housings. Amoruso et al. [7] and Su et al. [8] conducted systematic lightweight analysis on drive axle assemblies using multi-objective optimization algorithms and obtained favorable design outcomes. Long et al. [9] reduced the workload of lightweight

design for drive axles by adopting central composite design, which accelerated structural optimization of axle housings. Deep learning and artificial intelligence techniques have yielded preliminary achievements in structural lightweight design, but their applications are mostly limited to high-end engineering fields. Given the well-established conventional optimization approaches for drive axle housings, mature research outcomes based on AI for axle housing design remain scarce [10].

To secure the comprehensive performance of drive axle housings, experimental design or numerical simulation of improved structures is indispensable. Yang et al. [11] performed structural optimization and numerical simulation of drive axle housings based on harmonic response analysis to guarantee structural strength and service life. Tan et al. [12] adopted fatigue life analysis combined with bench tests to further validate the mechanical performance of drive axle housings.

At present, finite element analysis and structural strength verification remain the mainstream technical routes for drive axle optimization [13–15]. Extensive research has been conducted on strength evaluation of drive axles, and experimental design paired with strength checking is still the dominant

strategy for drive axle structural optimization [16–18]. Clear design specifications for drive axle housings have been issued in China as early as 2000 [19]; accordingly, performance improvement of drive axle housings should be implemented on the premise of satisfying these standards.

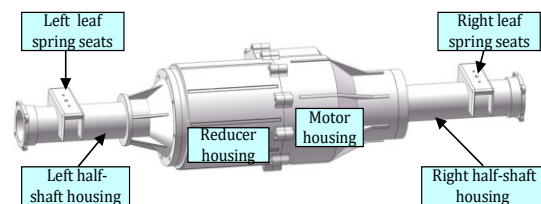
2. Drive Axle Modeling

2.1 Three-Dimensional Modeling of the Drive Axle

Based on physical photos of the drive axle housing (Figure 1a) and engineering drawings provided by the manufacturer, the main structure of the drive axle is constructed using SolidWorks, as illustrated in Figure 1b. This electric drive axle housing adopts a modular flange connection scheme combining stamped-welded axle housings and cast motor housings. The axle housing segments are formed by stamping and splicing high-strength steel plates, while the central motor-reducer housing is cast from aluminum alloy. The two components are rigidly fastened via high-strength bolts, balancing lightweight design and motor heat dissipation performance.



(a) Physical prototype of the drive axle



(b) 3D model of the drive axle

Figure 1: 3D model and assembly of the drive axle

The drive axle adopts a typical three-segment layout, mainly consisting of axle housings, reducer housing and motor housing. The vehicle wheelbase is 3380 mm, and its full-load mass is 3390 kg. The axle housing is made of 40Cr steel, whereas the motor and reducer housings are manufactured from Q235B steel. The intrinsic material properties are listed in Table 1.

Table 1. Intrinsic material parameters

Material	Density(kg/m ³)	Elastic Modulus (GPa)	Poisson's Ratio	Yield Strength (MPa)
Q235B	7800	210	0.27	235
40Cr	7880	211	0.28	787

2.2 Mesh Generation

High-quality mesh generation is critical to guarantee the accuracy of finite element simulation

results for the drive axle housing. An intelligent optimized meshing algorithm with automatic multi-region partitioning is adopted, which delivers precise calculation results while improving computational efficiency. The meshed model is presented in Figure 2.



Figure 2: Meshed finite element model

The global mesh size is set to 4 mm, generating a total of 1,353,484 nodes and 815,856 mesh elements. Element quality evaluation yields an average element quality of 0.82, which meets the basic requirements for finite element simulation.

2.3 Load Calculation

Loads and torques acting on the drive axle housing under four typical operating conditions are calculated, and the corresponding load configurations are shown in Figures 3–6.

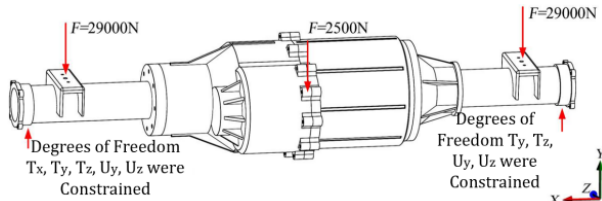


Figure 3: Boundary conditions under full-load bending condition

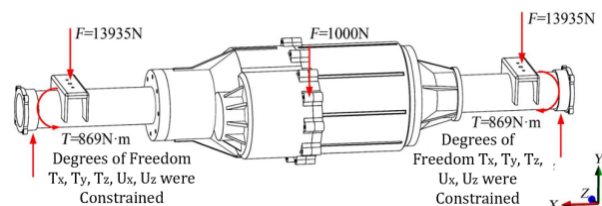


Figure 4: Boundary conditions under maximum driving force condition

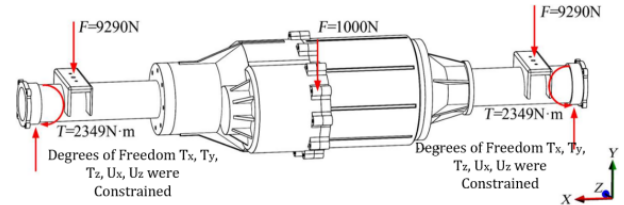


Figure 5: Boundary conditions under maximum braking force condition

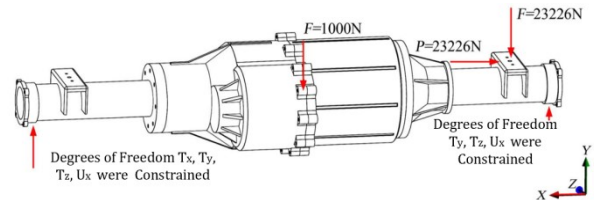


Figure 6: Boundary conditions under maximum lateral force condition

The load and constraint schemes of the drive axle under the four typical working conditions are clarified via force calculation, laying a foundation for subsequent mechanical analysis of the axle housing. Detailed constraint degrees of freedom for each condition are summarized in Table 3.

Table 3. Constrained degrees of freedom under different working conditions

Working Condition	Full-load bending condition	Maximum driving force condition	Maximum braking force condition	Maximum lateral force condition
Constrained DOFs	Left side: Tx, Ty, Tz, Uy, Uz; Right side: Ty, Tz, Uy, Uz	Both left and Right sides: Tx, Ty, Tz, Ux, Uz	Both left and Right sides: Tx, Ty, Tz, Ux, Uz	Left side: Tx, Ty, Tz, Ux; Right side: Ty, Tz, Ux

Note: T denotes translational degrees of freedom, and U denotes rotational degrees of freedom

3. Static Structural Analysis of Drive Axle under Various Working Conditions

3.1 Static Analysis under Full-Load Bending Condition

The contours of stress and displacement distribution of the drive axle housing under this operating condition are displayed in the following figures.

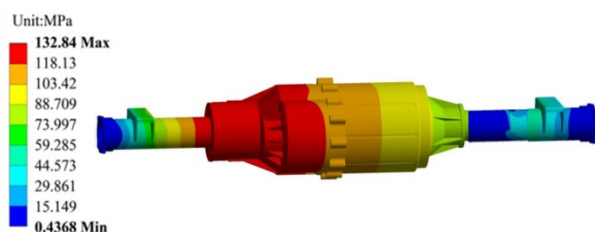


Figure 7: Equivalent stress contour of the axle housing

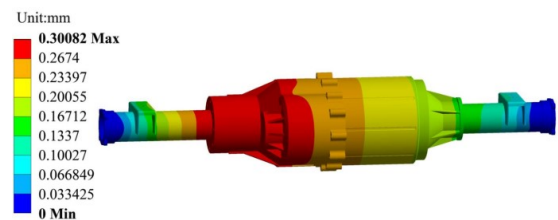


Figure 8: Displacement contour of the axle housing

The simulation results indicate that the maximum equivalent stress of 132.84 MPa appears on the axle housing, which is far lower than the yield strength of the corresponding material and delivers a high safety factor. The maximum deformation reaches 0.30082 mm, and the wheelbase deformation per meter is 0.195 mm/m, much smaller than the limit value of 1.4 mm/m specified in the national standard QC/T 533—2020 *Commercial Vehicle Drive Axle Assembly*.

3.2 Static Analysis under Maximum Driving Force Condition

The stress and displacement contours of the drive axle housing under this condition are shown in Figures 9 and 10.

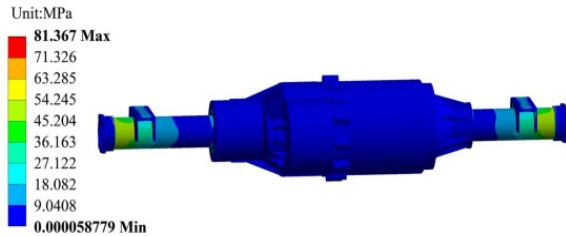


Figure 9: Equivalent stress contour of the axle housing

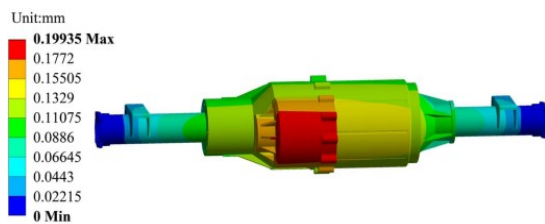


Figure 10: Displacement contour of the axle housing

As illustrated in the figures, the maximum stress of 81.367 MPa occurs on the axle housing, while the maximum displacement of 0.199 mm emerges on the reducer housing. The wheelbase deformation per meter is approximately 0.129 mm/m, complying with the national standard.

3.3 Static Analysis under Maximum Braking Force Condition

The mechanical results of the axle housing under this working condition are presented in Figures 11–12.

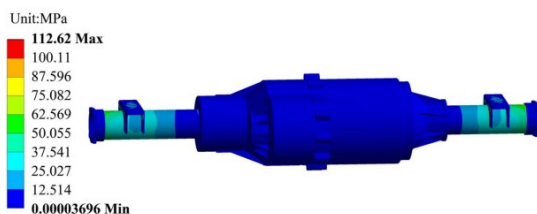


Figure 11: Equivalent stress contour of the axle housing

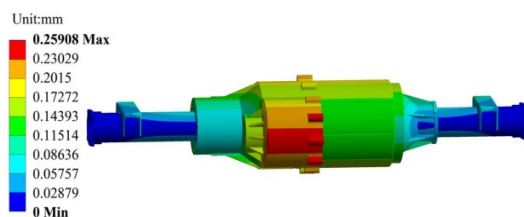


Figure 12: Displacement contour of the axle housing

It can be seen that the maximum stress of 112.62 MPa is located on the axle housing, and the maximum displacement of 0.259 mm occurs at the reducer housing. The wheelbase deformation per meter is 0.168 mm/m, which meets the standard requirements.

3.4 Static Analysis under Maximum Lateral Force Condition

The structural response results of the axle housing under this condition are given in Figures 13–14.

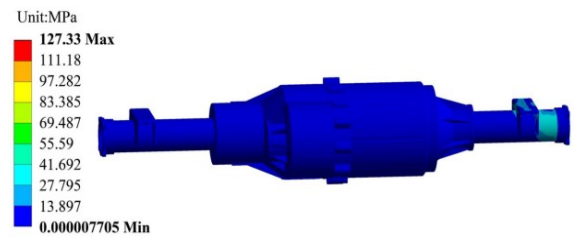


Figure 13: Equivalent stress contour of the axle housing

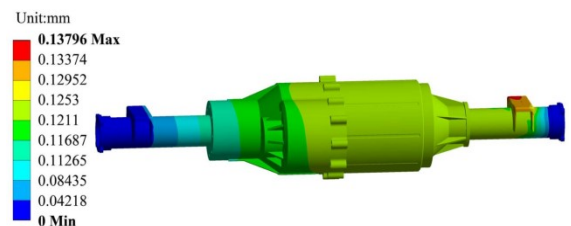


Figure 14: Displacement contour of the axle housing

The contours reveal that the maximum stress of 127.33 MPa concentrates on the axle housing, and the maximum displacement of 0.1379 mm is found on the reducer housing. The wheelbase deformation per meter is about 0.089 mm/m, satisfying the national standard.

Comprehensive analysis demonstrates that the stress and deformation of the drive axle housing under all four working conditions satisfy the relevant specifications. Among all operating conditions, the full-load bending condition brings the most severe deformation of 0.195 mm/m, yet this value still meets the design criteria.

4. Modal Analysis and Fatigue Life Analysis of Drive Axle Housing

4.1 Modal Analysis of the Drive Axle

During vehicle operation, the drive axle is subjected to periodic excitation loads transmitted from the road surface. Such cyclic loads may trigger resonance and further induce structural damage to the axle, which necessitates modal analysis.

Modal analysis is performed within ANSYS Workbench, and the first six orders of natural frequencies and corresponding mode shapes of the drive axle are presented in Figure 15.

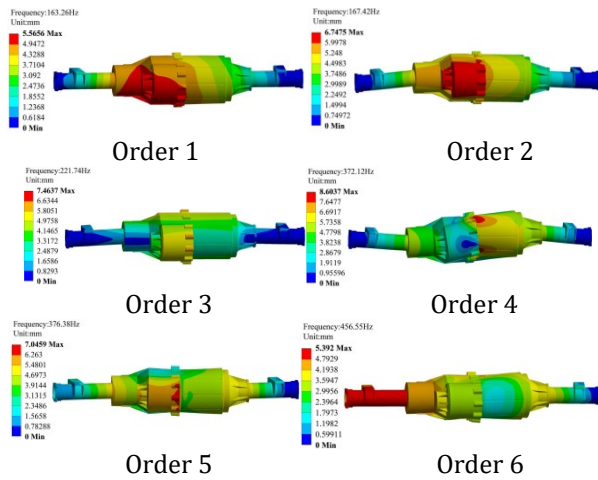


Figure 15: The first six modal shapes of the drive axle housing

In general, the frequency of road excitation loads is below 50 Hz. The first-order natural frequency of the drive axle is calculated as 163.26 Hz, which is substantially higher than the 0–50 Hz low-frequency range. Accordingly, structural damage caused by resonance can be effectively avoided.

4.2 Fatigue Life Analysis of the Drive Axle

Under long-term cyclic service loads, local fatigue cracks may emerge on the drive axle housing, so fatigue life evaluation is essential. The foregoing static structural analysis confirms that the full-load bending condition imposes the most severe loading on the axle. Hence, the maximum cyclic load of 29000 N is adopted for fatigue simulation, and the applied load spectrum is shown in Figure 16.

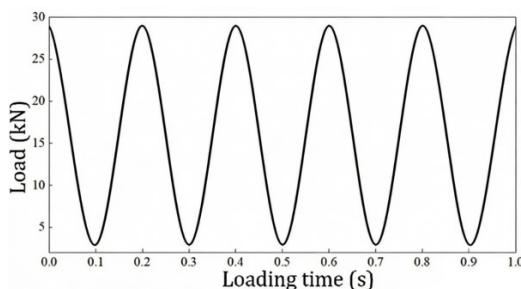


Figure 16: Fatigue load spectrum of the drive axle

Following the fatigue assessment criteria for electric vehicle drive axles specified in national standard QC/T 533—2020, the load spectrum is applied to the axle housing for fatigue life simulation. Combined with the S-N curves of 40Cr and Q235B steel, the fatigue life distribution of the drive axle housing is obtained as illustrated in Figure 17.

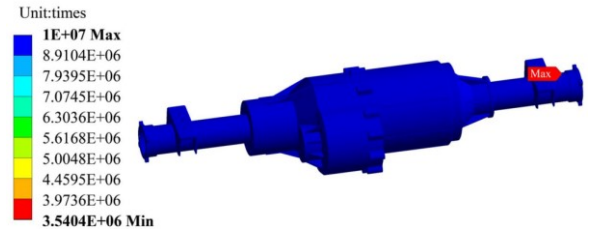


Figure 17: Fatigue life contour of the drive axle

As shown in Figure 17, the minimum fatigue life of the axle housing reaches 3.54×10^6 cycles, far exceeding the threshold of 800,000 cycles required by the national standard, which verifies that the original structure meets the service life demand theoretically. Nevertheless, the flange region is identified as the critical weak zone for fatigue failure, where damage is most likely to occur during operation; structural optimization is therefore required for this position.

5. Orthogonal Experimental Design and Optimization Scheme

5.1 Selection of Design Variables

Based on the stress, displacement and modal contours obtained from static and dynamic analyses, seven structural parameters are selected as design variables for the lightweight optimization of the drive axle housing: left axle housing thickness X_1 , right axle housing thickness X_2 , motor housing thickness X_3 , front reducer housing thickness X_4 , rear reducer housing thickness X_5 , distance between left leaf spring seat and flange X_6 , and distance between right leaf spring seat and flange X_7 . The variation range of each parameter is determined according to practical manufacturing experience, as listed in Table 4.

Table 4. Variation ranges of design parameters

Structural Component	Parameter	Lower limit / mm	Upper limit / mm
Left axle housing thickness	X_1	4	5
Right axle housing thickness	X_2	4	5
Motor housing thickness	X_3	4	5
Front reducer housing thickness	X_4	4	5
Rear reducer housing thickness	X_5	4	6
Distance, left spring seat to flange	X_6	85	100
Distance, right spring seat to flange	X_7	85	100

5.2 Orthogonal Experimental Design and Test Results

For each design variable, three levels are defined: lower limit (1), median value (2) and upper limit (3). An $L_{27}(3^7)$ orthogonal array with 7 factors and 3 levels is established, covering a total of 27 groups of tests.

Table 5. Groups of the $L_{27}(3^7)$ orthogonal experiment with 7 factors and 3 levels

No.	X1	X2	X3	X4	X5	X6	X7
1	1	1	1	1	1	1	1
2	1	1	1	2	2	2	2
3	1	1	1	3	3	3	3
4	1	2	2	1	1	1	1
5	1	2	2	2	2	2	2
6	1	2	2	3	3	3	3
7	1	3	3	1	1	1	1
8	1	3	3	2	2	2	2
9	1	3	3	3	3	3	3
10	2	1	2	1	2	1	3
11	2	1	2	2	3	2	1
12	2	1	2	3	1	3	2
13	2	2	3	1	3	1	2
14	2	2	3	2	1	2	3
15	2	2	3	3	2	3	1
16	2	3	1	1	1	1	3
17	2	3	1	2	2	2	1
18	2	3	1	3	3	3	2
19	3	1	3	1	3	2	3
20	3	1	3	2	1	3	1
21	3	1	3	3	2	1	2
22	3	2	1	1	2	2	1
23	3	2	1	2	3	3	2
24	3	2	1	3	1	1	3
25	3	3	2	1	1	2	2
26	3	3	2	2	2	3	1
27	3	3	2	3	3	1	3

To evaluate the lightweight level, structural strength and anti-resonance performance of the drive axle housing, three evaluation indicators are adopted: axle housing mass M , maximum equivalent stress $\sigma_{\max}$ and the first-order natural frequency $F1$. Parametric finite element analysis is carried out for all 27 groups of parameter combinations, and the calculated results are summarized in Table 6.

Table 6. Orthogonal experimental results

No.	M/kg	$\sigma_{\max}$ /MPa	F1/Hz
1	58.65	164.51	158.14
2	61.14	156.64	156.71
3	62.88	163.37	157.22
4	59.12	150.76	159.22
5	62.25	159.47	155.95
6	62.28	141.22	158.53
7	61.1	159.26	162.39
8	63.57	168.72	157.25
9	64.43	175.38	156.47
10	61.87	149.87	165.22
11	63.45	158.42	162.75
12	64.81	169.33	160.18
13	63.3	155.68	164.37
14	65.12	164.75	166.84
15	65.33	172.14	163.09
16	62.71	148.52	167.51
17	64.94	157.09	165.83
18	65.47	163.27	164.92
19	64.29	156.43	169.76
20	66.01	165.88	168.04
21	66.36	173.21	167.29
22	64.52	154.26	170.15
23	66.78	162.99	169.43
24	67.04	170.84	168.71
25	65.92	159.73	171.28
26	68.13	167.42	170.56
27	68.69	174.55	169.88

5.3 Multi-Objective Optimization Based on Least-Squares Regression

The simulation results cannot simultaneously achieve the minimum mass, minimum stress and minimum first-order natural frequency. Therefore, a mathematical solution is required to obtain the optimal parameter combination. The multi-objective least-squares method is used to analyze the data in Table 6, and the optimization objective function is established as Formula (1):

$$K = \omega_1 \left(\frac{M - M_{\min}}{M_{\max} - M_{\min}} \right)^2 + \omega_2 \left(\frac{\sigma - \sigma_{\min}}{\sigma_{\max} - \sigma_{\min}} \right)^2 + \omega_3 \left(\frac{F - F_{\min}}{F_{\max} - F_{\min}} \right)^2 \quad (1)$$

Where K represents the objective function, ω denotes the weight coefficient, and $\omega_1 = \omega_2 = \omega_3 = 1/3$, for equal weights are assigned to balance lightweight, strength and dynamic performance without subjective preference; M , σ and F are the values extracted from simulation outputs.

All original data are normalized to eliminate dimensional effects, and the final objective function values of each test group are plotted in Figure 18.

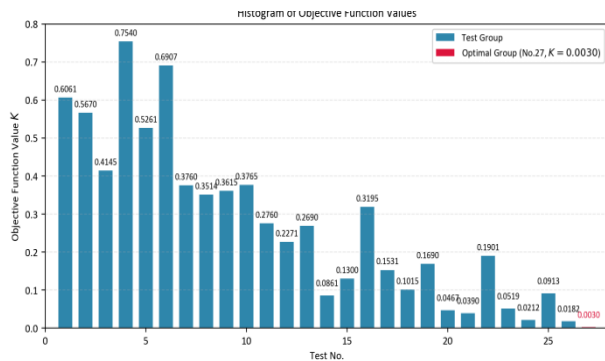


Figure 18: Objective function values of each test group

Least-squares regression analysis reveals that the 27th test group yields the optimal parameter combination. The corresponding structural dimensions are specified as follows: the thickness of both left and right axle housings is 5 mm, the motor housing thickness is 4.5 mm, the front and rear reducer housing thicknesses are X4=5 mm, X5=6 mm,

and the distances between the left/right leaf spring seats are flanges and X6=85mm、X7=100 mm, respectively.

Fatigue simulation is conducted on the optimized drive axle housing, and its fatigue life distribution is presented in Figure 19. The optimized structure still retains a favorable fatigue service life margin.

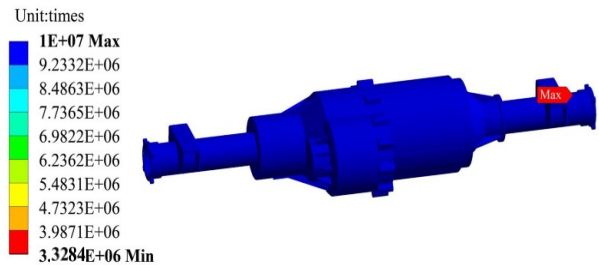


Figure 19: Fatigue life distribution of the optimized axle housing

Table 7 summarizes the performance indicators of the drive axle housing before and after optimization, as well as their corresponding change rates.

Table 7. Change rates of performance indicators after optimization

Target Value	Before Optimization	After Optimization	Change Rate
M/kg	72.33	68.69	Reduced by 5.03%
$\sigma_{\max}$	132.84	174.55	Increased by 11.07%
F1/Hz	163.26	169.88	Increased by 4.05%
Fatigue Life (10^6 cycles)	3.54	3.32	Reduced by 6.21%

The optimized performance indicators obtained from orthogonal optimization are listed in Table 7. The analysis demonstrates that the 27th test group achieves a satisfactory overall optimization effect. Although partial structural performances slightly degrade after optimization, a mass reduction of 5.03% is realized, fulfilling the lightweight target. In addition, the simulation results of other orthogonal parameter combinations all fall within acceptable performance ranges.

6. Conclusions

This work develops a systematic finite element analysis to characterize the mechanical performance of drive axle housings for electric trucks, encompassing geometric reconstruction, mesh discretization, load quantification and multi-condition numerical simulation. A parametric solid model of the target axle housing is constructed in SolidWorks, followed by high-fidelity meshing implemented via an intelligent refinement algorithm.

The discretized model contains 1,353,484 nodes and 815,856 elements, yielding an average element quality of 0.82 that satisfies the precision thresholds for FEA computation. Boundary constraints and external loads corresponding to four representative operating scenarios are imposed to precisely resolve the structural response of the axle assembly. The outcomes validate the robust adaptability of the FEA paradigm for axle structural evaluation, laying a solid numerical basis for follow-up structural optimization.

Static structural and modal analyses constitute indispensable tools for guaranteeing the structural integrity and dynamic stability of drive axle housings. Static simulation reveals that the peak equivalent stress under full-load bending reaches 132.84 MPa, well below the yield limit of the constituent materials, while the wheelbase deflection per meter measures 0.195 mm/m, conforming to the specifications stipulated in national standard QC/T 533—2020. Modal computation identifies the first-order natural frequency of the axle housing as 163.26 Hz, which substantially surpasses the typical 0–50 Hz bandwidth of road-induced excitations and

thereby eliminates the risk of structural deterioration triggered by resonance. Collectively, these numerical findings corroborate that the initial axle prototype fulfills fundamental performance criteria.

An $L_{27}(3^7)$ orthogonal array is adopted to conduct parametric optimization over seven critical structural dimensions, with axle housing mass M maximum equivalent stress $\sigma_{\max}$ and the first-order natural frequency F_1 defined as multi-objective evaluation metrics. Multi-objective least-squares regression is executed to screen the optimal parameter configuration, corresponding to the 27th experimental trial. Post-optimization results demonstrate a 5.03% reduction in structural weight; meanwhile, the first-order natural frequency rises to 169.88 Hz, and the minimum fatigue life is preserved at 10^6 cycles. Despite an 11.07% elevation in maximum stress, the resultant stress magnitude stays within the allowable performance envelope of the raw materials. The proposed optimization strategy thus attains a trade-off among lightweight design, mechanical strength and anti-resonance capacity.

Acknowledgement

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