

Article

Fatigue Performance of a 30CrMoA Gear Shaft Strengthened by Laser Quenching Combined with Ultrasonic Surface Rolling: A Theoretical S–N Curve Approach

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Abstract

Marine gear shafts strengthened by multi-track overlapping laser quenching–ultrasonic rolling composite treatment exhibit complex torsional fatigue damage evolution due to the multi-layer gradient structure of the meshing interface, making quantitative fatigue life evaluation under mean shear stress challenging. Taking 30CrMoA steel as the research object, this study establishes a torsional fatigue S–N model modified by mean shear stress and a layered finite element model comprising the matrix, hardened layer, and tempered zone. Symmetric ($R = -1$) and asymmetric ($R = 0.14435$) cyclic torsional fatigue simulations were performed to clarify load effects on fatigue life distribution and attenuation. A second-order response surface model based on the Box–Behnken design was employed to reveal the interactive effects of hardened layer depth, tempered zone width, and surface residual compressive stress on fatigue life, yielding the sensitivity ranking: residual compressive stress > hardened layer depth > tempered zone width. With maximum fatigue life as the objective, the optimal parameters were determined (0.85 mm hardened layer, 1.60 mm tempered zone, -650 MPa residual stress). Torsional fatigue tests and fracture observations validated the simulations, with a relative error below 15%, providing a theoretical basis for anti-torsional-fatigue design of composite-strengthened shafts.

Keywords: multi-track overlapping laser quenching ultrasonic rolling; composite strengthening; torsional fatigue; mean shear stress; layered finite element model; response surface methodology



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1. Introduction

Gear transmission shafts are core components of the Warship Power Rear Transmission System (WPRTS). The wear resistance, scuffing load capacity, and service reliability of their meshing interfaces directly determine the transmission efficiency and service life of the WPRTS. Subjected to long-term alternating torsional loads combined with complex service conditions, including impact loading and marine corrosion, gear transmission shafts are prone to torsional fatigue failure, which is one of their dominant failure modes. The torsional fatigue resistance of these components directly governs the operational safety and service life of the equipment [1–3]. Under alternating torsional loading, fatigue cracks mostly initiate at the shaft surface, where surface defects, microstructurally softened zones, and residual stress states exert significant influences on crack initiation and propagation. Therefore, surface strengthening technologies, which modify the surface microstructure

and stress state, represent a vital technical route for improving the torsional fatigue life of gear transmission shafts [4–6].

Laser quenching–ultrasonic rolling composite strengthening integrates the advantages of laser phase-transformation hardening and ultrasonic rolling-induced plastic strengthening. This process not only produces a high-hardness hardened layer through rapid phase transformation but also introduces beneficial compressive residual stresses into the surface layer, thereby suppressing fatigue crack nucleation; it therefore shows promising application prospects for heavy-duty shaft and gear components [7–9]. Nevertheless, multi-pass overlapping laser quenching induces tempered softening zones, resulting in axial gradients in microstructure and mechanical properties. The fatigue life of the components is jointly governed by the coupling effects of the tempered zones, hardened layers, and residual stresses [10–12]. Most existing studies have analyzed fatigue performance only under a single strengthening process [13–15], whereas investigations on composite strengthening have mainly focused on the characterization of hardness and wear resistance [16]. Research on the torsional fatigue behavior of the modified layer remains insufficient. In particular, a theoretical S–N curve framework suitable for torsional loading and accounting for the mean shear stress effect is still lacking, making it difficult to quantitatively evaluate the fatigue damage evolution of composite-strengthened components under both symmetric and asymmetric cyclic loading. In engineering practice, gear transmission shafts are predominantly subjected to asymmetric cyclic torsional loading. Mean shear stress alters the fatigue damage accumulation process and accelerates crack propagation; consequently, fatigue properties obtained under symmetric cycling cannot be directly applied to life assessment [17–19]. Conventional fatigue testing is costly and time-consuming. Finite element (FE) simulation combined with theoretical S–N curves enables efficient fatigue life prediction across multiple working conditions and surface layer parameters. However, composite-strengthened components possess a multilayer gradient microstructure comprising the matrix, hardened layer, and tempered zone, with substantial differences in mechanical properties among regions.

Conventional fatigue models formulated for homogeneous materials fail to capture the heterogeneous fatigue response induced by the modified layer [20–22]. Meanwhile, key surface features, such as surface microstructure and surface residual stress, exhibit complex coupling interactions [23]. Quantitative identification of the sensitivity of each parameter with respect to fatigue life maximization remains to be further investigated. To address these issues, this study investigates marine-grade 30CrMoA steel. A torsional fatigue S–N model incorporating mean shear stress correction is developed, and a layered FE model comprising the matrix, hardened layer, and tempered zone is established. Torsional fatigue simulations are performed under symmetric and asymmetric cyclic conditions to analyze the influence of load level on the fatigue life distribution. A response surface model based on the Box–Behnken design is constructed to evaluate the effects and sensitivities of individual surface feature parameters on both the global fatigue life and the fatigue life within the tempered zones, and the optimal combination of surface feature parameters is determined. Torsional fatigue tests combined with fracture morphology observation are conducted to clarify the torsional fatigue failure mechanisms under different stress ratios, and the simulation results are validated against experimental data.

This study establishes a complete technical closed loop comprising theoretical S–N curve fitting, a standardized simulation workflow, and torsional fatigue test validation. It addresses the problems of poorly matched material parameters and the limited adaptability of fatigue damage models in conventional torsional fatigue life prediction methods for laser quenching–ultrasonic rolling-modified gear transmission shafts. The results demonstrate that the overall life prediction error of this co-simulation method is below 15%, providing

a reliable and reusable technical route for the accurate torsional fatigue life prediction of composite-strengthened shaft components.

2. Numerical Simulation of Fatigue Life Under Surface Laser Quenching-Ultrasonic Rolling Composite Strengthening

2.1. Basic Principle of the Theoretical S-N Curve

Based on abundant basic S-N test data, the material S-N curve can be constructed as a piecewise continuous curve consisting of four linear segments in the double-logarithmic coordinate system, as shown in Figure 1. The first inclined straight line corresponds to the low-cycle fatigue (LCF) region, the remaining inclined segments belong to the high-cycle fatigue (HCF) region, and the horizontal asymptote represents the fatigue limit of the material [24,25].

$$b = \frac{\lg S_{E3} - \lg S_{E6}}{\lg 10^3 - \lg 10^6} = -\frac{1}{3} \lg \left(\frac{S_{E3}}{S_{E6}} \right) \quad (1)$$

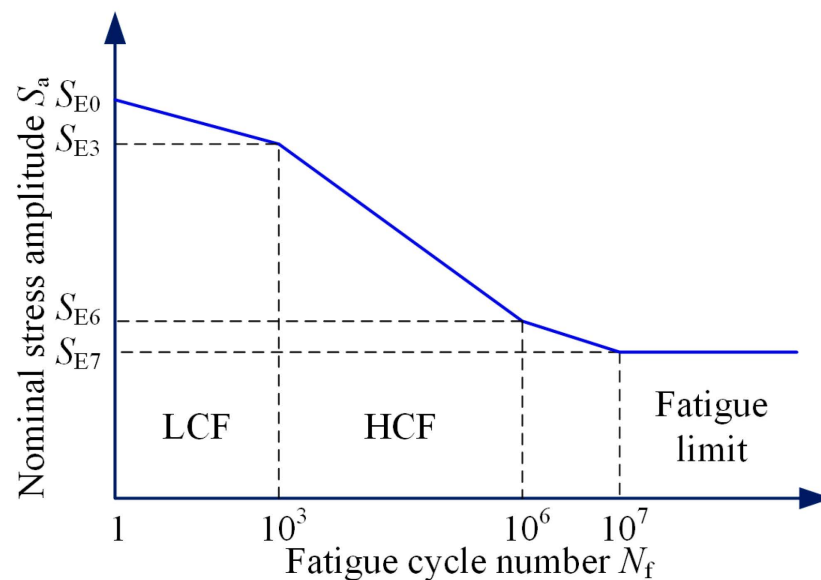


Figure 1. Theoretical S-N Curve.

The inverse slope (slope coefficient) of the S-N curve is denoted by k :

$$k = -\frac{1}{b} \quad (2)$$

For the extended high-cycle fatigue regime of 10^3 – 10^6 cycles, the Haibach method is employed to determine the slope coefficient k' of this curve segment, as expressed by the following equation:

$$k' = 2k - 1 \quad (3)$$

In general, when crack propagation dominates the fatigue process, the S-N curve exhibits a steeper slope; conversely, when the fatigue life is governed by crack initiation, the S-N curve becomes relatively shallow.

2.2. Theoretical Calculation and Modification of the Baseline S-N Curve

When constructing the baseline S-N curve of a component, the ultimate tensile strength σ_b of the material is first obtained to estimate the fatigue strengths at various numbers of cycles. Subsequently, the fatigue strength is modified by comprehensively accounting

for influencing factors such as the loading type, surface roughness, specimen size, and reliability level.

According to the definition of the log–log S–N curve, the fatigue strength S_{E0} of a standard specimen at a cycle number of 1 represents the theoretical initial strength. Under torsional loading, the initial strength S_{E0} of steel can be expressed as:

$$S_{E0} = 0.8\sigma_b \quad (4)$$

The fatigue strength S_{E3} at a cycle number of 10^3 depends on the reliability level and the loading type, and it can be calculated as follows:

$$S_{E3} = S'_{E3}C_R \quad (5)$$

where S'_{E3} denotes the fatigue strength of the standard specimen at 10^3 cycles under different loading conditions. For torsional loading, $S'_{E3} = 0.72\sigma_b$; C_R is the correction factor corresponding to the specified reliability level.

Fatigue test data inherently exhibit statistical scatter. Therefore, a reliability correction factor must be introduced to modify the fatigue strength of components derived from median baseline S–N data. When sufficient fatigue test specimens are unavailable, a detailed statistical analysis of the stochastic characteristics of fatigue life cannot be performed. In such cases, the reliability correction factors C_R corresponding to different reliability levels listed in Table 1 can be directly adopted.

Table 1. Reliability coefficients under different reliability levels.

Reliability	C_R	Reliability	C_R
0.50	1.000	0.999	0.753
0.90	0.897	0.9999	0.702
0.95	0.868	0.99999	0.659
0.99	0.814	0.999999	0.620

A reliability level of 0.90 is uniformly adopted for calculations in this paper, with the corresponding reliability correction factor $C_R = 0.897$.

The fatigue limit is defined as the constant stress amplitude that a component can withstand for an infinite number of cycles without fatigue failure. By introducing the load factor C_L , surface quality factor C_s , size factor C_D and reliability factor C_R to modify the bending fatigue strength S_{be} , the fatigue limit S_{E6} can be estimated. The calculation formula is as follows:

$$S_{E6} = S_{be}C_LC_sC_DC_R \quad (6)$$

The bending fatigue strength S_{be} of the material after quenching and tempering treatment can be estimated by the following equation:

$$S'_{E6} = 0.55\sigma_b \quad (7)$$

According to the von Mises yield criterion, the load factor C_L of the component under torsional loading is obtained as $C_L = 0.58$. Based on the surface roughness of the specimen, the surface quality factor C_s can be determined from Figure 2.

Kuguel's critical volume theory is employed to account for the effect of specimen size on fatigue strength, and the size factor C_D can be expressed as:

$$C_D = 1.189d^{-0.097} \quad (8)$$

Based on the fatigue strength S_{E6} at 10^6 cycles and the slope coefficient k' of the high-cycle fatigue curve within the interval of 10^6 – 10^7 cycles, the fatigue strength S_{E7} at 10^7 cycles can be determined.

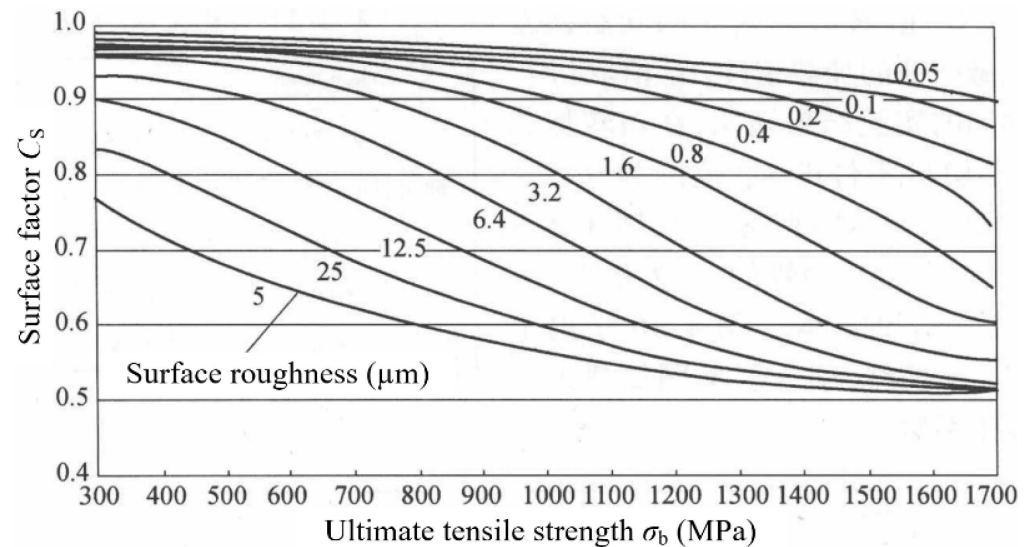


Figure 2. Empirical chart of surface quality factor.

2.3. Fatigue Strength Correction Considering Mean Shear Stress

Shaft components in engineering practice are mostly subjected to asymmetric cyclic loading, in which the mean stress effect cannot be neglected. This effect is particularly prominent in the high-cycle fatigue regime. Mean normal stress mainly governs the opening–closing behavior of microcracks, whereas mean shear stress tends to induce material slip and promote microcrack initiation. When cracks are open, the crack growth rate is further accelerated.

To characterize the influence of mean shear stress on torsional fatigue strength and obtain theoretical S–N curves under different stress ratios, the Goodman mean-stress correction criterion, adapted for torsional loading, is applied to the baseline torsional S–N curve for symmetric cycling (stress ratio $R = -1$). Torsional S–N curves corresponding to different stress ratios R are then derived, providing a theoretical basis for the torsional fatigue life evaluation and reliability design of shaft components. The Goodman criterion employs a linear approximation to describe the degradation of the fatigue limit with increasing mean stress, and its core correction formula can be expressed as:

$$\frac{\tau_a}{\tau_{-1}} + \frac{\tau_m}{\tau_b} = 1 \quad (9)$$

where $\tau_a = (\tau_{\max} - \tau_{\min})/2$ is the stress amplitude; $\tau_m = (\tau_{\max} + \tau_{\min})/2$ is the mean shear stress; τ_{-1} denotes the torsional fatigue limit under symmetric cycles; and τ_b is the torsional ultimate strength of the material.

By deriving the torsional form of the Goodman mean-stress correction equation, the allowable shear stress amplitude under asymmetric cyclic loading can be obtained as:

$$\tau_a = \tau_{-1} \left(1 - \frac{\tau_m}{\tau_b} \right) \quad (10)$$

By introducing the relationship between the stress ratio and the shear stress components, variable substitution yields the following expression:

$$\frac{1-R}{2}\tau_{\max} = \tau_{-1}(N) \left(1 - \frac{1+R}{2} \cdot \frac{\tau_{\max}}{\tau_b} \right) \quad (11)$$

After rearrangement and simplification, the maximum cyclic torsional stress under arbitrary stress ratio R and arbitrary fatigue life N can be obtained as:

$$\tau_{\max}(R, N) = \frac{2\tau_{-1}(N) \cdot \tau_b}{(1-R)\tau_b + (1+R)\tau_{-1}(N)} \quad (12)$$

Further derivation yields the fatigue strength corresponding to fatigue life N under arbitrary stress ratio R :

$$\tau_a(R, N) = \frac{\tau_{-1}(N)}{1 + \frac{\tau_{-1}(N)(1+R)}{\tau_b(1-R)}} \quad (13)$$

2.4. Establishment of the Finite Element Model

The finite element fatigue analysis procedure comprises two main stages: finite element (FE) solution and fatigue calculation. First, a static FE simulation of the target component is performed to accurately capture the stress distribution characteristics in the critical region and to generate result files containing the stress-strain field. These result files are then imported into the nCode fatigue analysis module, where the data configuration, material parameter definition, and loading setup are completed. An appropriate fatigue analysis method is selected, and the corresponding solving task is submitted to calculate the fatigue life of the component. Finally, the fatigue simulation results are analyzed and evaluated. The complete workflow of the finite element fatigue analysis is shown in Figure 3. The procedure can be summarized as follows:

- (1) Conduct a static or dynamic finite element simulation to obtain the corresponding stress result files.
- (2) Import the finite element result files into the fatigue analysis software.
- (3) Configure the data files, define the material properties, and complete the relevant loading configurations.
- (4) Select an appropriate fatigue analysis criterion to calculate the fatigue life of the target component.
- (5) Examine and analyze the predicted fatigue results of the component.

To investigate the effect of combined laser surface quenching and ultrasonic rolling strengthening on the fatigue life of quenched-and-tempered 30CrMoA steel specimens, a three-dimensional cylindrical geometric model (diameter: 6 mm; length: 30 mm) was established using the ANSYS finite element platform. Given that the combined strengthening treatment introduces a gradient microstructure along the radial direction of the specimen, the model was partitioned into three distinct regions: the matrix, the hardened layer, and the tempered zone, with corresponding material properties assigned to each region. The geometric model of the fatigue specimen subjected to combined strengthening is illustrated in Figure 4.

In the numerical calculation, structured hexahedral meshes were adopted for full-domain discretization of the established three-dimensional layered geometric model. The total number of mesh elements of the model is 188,800, and the finite element mesh model is presented in Figure 5.

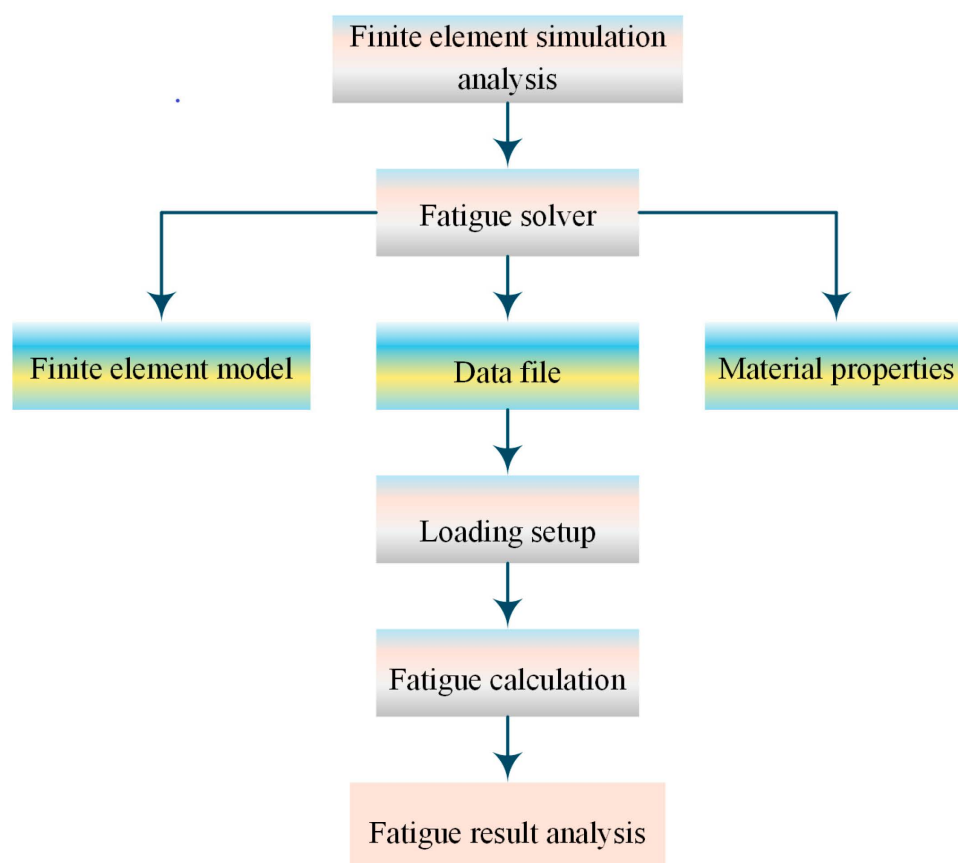


Figure 3. Workflow of finite element fatigue analysis.

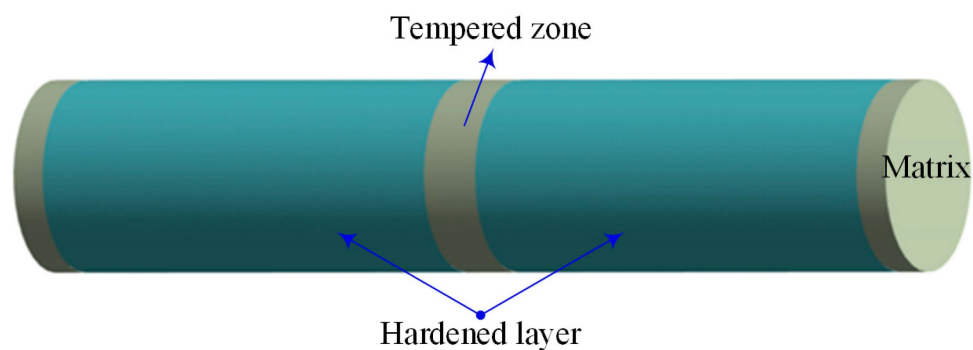


Figure 4. Geometric model of fatigue specimen treated by laser quenching-ultrasonic rolling composite strengthening.

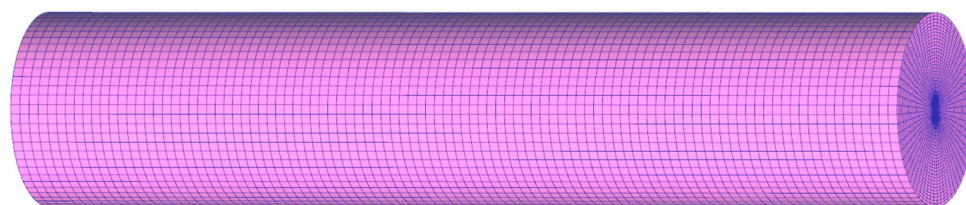


Figure 5. Finite element mesh model of fatigue specimen treated by laser quenching-ultrasonic rolling composite strengthening.

The chemical composition of 30CrMoA steel used in this study is listed in Table 2.

Table 2. Chemical composition of 30CrMoA steel.

Element	C	Si	Mn	S	P	Cr	Ni	Cu	Mo	Fe
Content	0.30	0.27	0.55	0.02	0.02	0.95	0.03	0.02	0.20	Balance

Using JMatPro software and the CALPHAD method, the material properties of 30CrMoA steel under different states were calculated. The obtained parameters were imported into the finite element model to improve the accuracy of numerical simulation. After laser quenching–ultrasonic rolling composite strengthening, the specimen is divided into the matrix, hardened zone and tempered zone. The basic mechanical property parameters of each region are listed in Table 3.

Table 3. Basic mechanical property parameters of different regions of 30CrMoA steel.

Parameter	Matrix	Hardened Zone	Tempered Zone
Elastic modulus (GPa)	210	210	210
Poisson's ratio	0.3	0.3	0.3
Density ($\text{kg}\cdot\text{m}^{-3}$)	7850	7850	7850
Yield strength (MPa)	760	1600	680
Ultimate tensile strength (MPa)	960	1800	860

During finite element simulation, the corresponding mechanical parameters and corrected theoretical S–N curves are assigned to different subregions of the model, so as to realize the numerical calculation of fatigue life considering the surface characteristics of composite strengthening.

A three-region finite element model of the solid cylindrical torsional fatigue specimen, surface-hardened layer (0.85 mm baseline), tempered zone (1.60 mm), and base material, each with independent elastoplastic and S–N parameters, was constructed. One end face was fully constrained (machine clamping); the other was kinematically coupled to an on-axis reference point, where the torque was applied as a concentrated moment, enforcing uniform angular displacement without point-loading artifacts. Mesh M3, 188,800 eight-node reduced-integration hexahedra with enhanced hourglass control (0.20 mm strengthened zone, 0.5 mm core), met the convergence criteria ($<2\%$ in maximum surface shear stress and $<5\%$ in predicted life between successive refinements) and was used throughout. The theoretical S–N curves of the base material, hardened layer, and tempered zone are shown in Figure 6. The ultimate torsional strengths and all empirical correction factors are summarized in Tables 4 and 5.

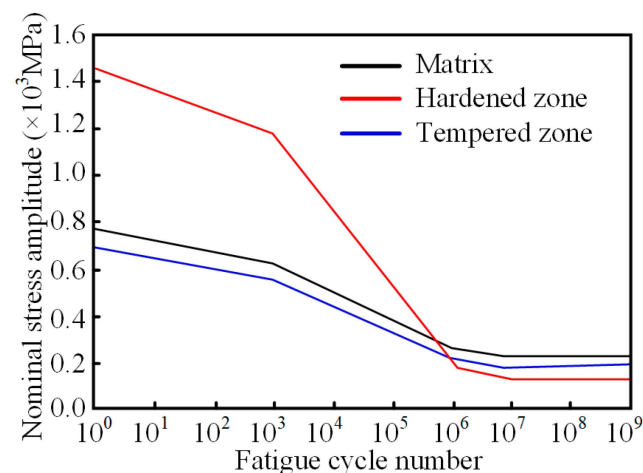
**Figure 6.** Theoretical S–N curves for different zones of 30CrMoA specimen.

Table 4. Mechanical properties for different zones of 30CrMoA specimen.

Parameter	Matrix	Hardened Zone	Tempered Zone
Torsional ultimate strength (MPa)	770	1450	690

Table 5. Empirical correction factors.

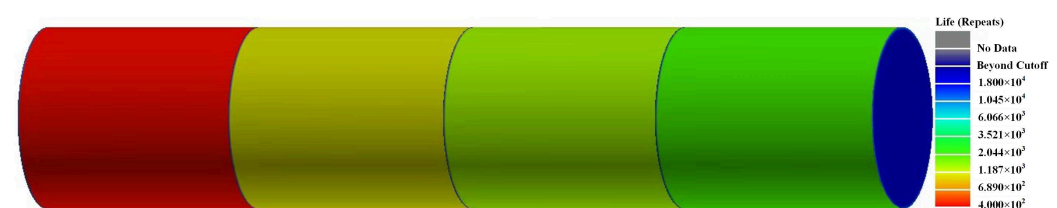
Parameter	Matrix	Hardened Zone	Tempered Zone
C_L	0.58	0.58	0.58
C_S	0.93	0.73	0.94
C_D	0.99931	0.99931	0.99931
C_R	0.897	0.897	0.897

A sinusoidal torque history (90–120 N·m; 10^4 – 10^6 cycles) was applied via an amplitude curve; gauge-section stress non-uniformity remained below 2%, confirming negligible Saint-Venant end effects. Depth-dependent compressive residual stresses measured by XRD ($\sin^2\psi$ method) were prescribed as an initial stress field (*INITIAL CONDITIONS, TYPE=STRESS) and converted via the von Mises relation into an equivalent mean-shear-stress shift for the mean-stress-corrected S–N model. Quasi-static analyses (Abaqus/Standard; combined isotropic–kinematic hardening in all layers) yielded stabilized peak-load stress/strain fields, which were transferred to the fatigue post-processor for life prediction. The base-material ultimate torsional strength (620 MPa) was measured in static torsion tests (GB/T 10128, three-specimen mean); those of the hardened layer (1100 MPa) and tempered zone (540 MPa) were derived from the hardness–strength correlation. Laser-quenching–ultrasonic-rolling strengthening is thus captured solely through the layered material properties and residual stress field, with no additional factor, hence no double counting.

3. Numerical Prediction of the Fatigue Life of Laser Surface Quenching–Ultrasonic Rolling Composite-Strengthened Specimens

3.1. Fatigue Life Analysis Under Fully Reversed Cyclic Loading

Symmetric cyclic torsion is one of the most typical fatigue loading modes for shaft components. Because the mean stress is zero under this condition, the fatigue response is free from the influence of superimposed mean-stress effects and thus directly reflects the intrinsic fatigue resistance of the material; it is therefore commonly adopted as the baseline condition for evaluating the fundamental fatigue performance of materials. In this section, numerical simulations of the torsional fatigue life of 30CrMoA steel shaft specimens are performed under symmetric cyclic loading (stress ratio $R = -1$). A series of torque amplitudes are applied to investigate the influence of the load level on the spatial distribution and degradation behavior of the fatigue life, thereby providing a baseline for subsequent studies on asymmetric stress-ratio conditions and on the fatigue performance of composite-strengthened surface layers. The fatigue life contours obtained under the various loading conditions are presented in Figure 7.

**Figure 7.** Fatigue life nephograms under symmetric cycles for different torque amplitudes (a) 27 N·m; (b) 26 N·m; (c) 25 N·m; (d) 24 N·m.

It can be observed from Figure 6 that the fatigue-critical regions of the shaft are consistently located at the outer surface under all applied load levels, which is consistent with the characteristic distribution of torsional shear stress maximum at the surface and zero at the core along the radial cross-section. Fatigue cracks therefore initiate at the surface layer of the shaft. As the torque amplitude increases stepwise from 24 N·m to 27 N·m, the minimum fatigue life at the shaft surface decreases to 2029, 1476, 1088, and 418 cycles at 24, 25, 26, and 27 N·m, respectively, indicating an overall decreasing trend with increasing load. Within the torque range of 24–26 N·m, the material response is dominated by elastic deformation, and the alternating shear stress remains below the torsional yield strength; this corresponds to a typical high-cycle fatigue (HCF) regime, in which the fatigue life decreases gradually with increasing load. When the torque amplitude reaches 27 N·m, the surface shear stress approaches the torsional yield strength of the material and localized cyclic plastic deformation occurs. Consequently, the initiation and propagation rates of fatigue cracks increase markedly, leading to a sharp reduction in fatigue life and a significantly enhanced load sensitivity. This behavior conforms to the Basquin power-law fatigue degradation model [26], providing a baseline reference for subsequent fatigue performance analyses under non-symmetric stress ratios and for composite-strengthened surface layers.

3.2. Torsional Fatigue Life Analysis Under a Typical Stress Ratio

In practical service, shaft-type transmission components are generally subjected to asymmetric cyclic torsional loads. The mean stress exerts a significant influence on the fatigue damage evolution of materials, resulting in fatigue behavior that differs markedly from that under symmetric cyclic loading. In this work, a stress ratio of $R = 0.144435$ is selected as a typical asymmetric loading condition, and numerical simulations of torsional fatigue life are performed under a series of stepwise torque amplitudes. The distribution characteristics and degradation law of the fatigue life of the shaft under the combined action of mean stress and alternating load are analyzed, providing a basis for fatigue life evaluation under practical engineering conditions. The contour plots of the fatigue life distribution of the shaft under asymmetric cyclic loading with different torque amplitudes are shown in Figure 8.

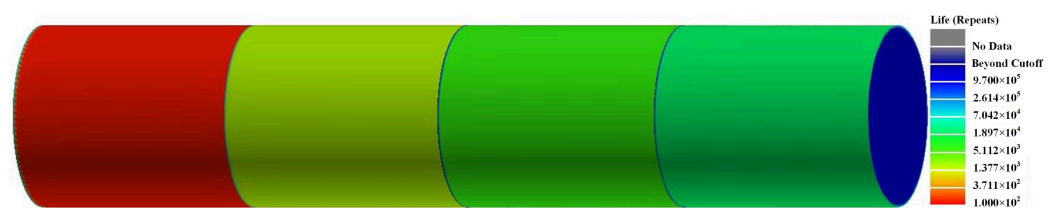


Figure 8. Fatigue life cloud diagrams under asymmetric cycles at different torque amplitudes: (a) 13 N·m, (b) 12 N·m, (c) 11 N·m, (d) 10 N·m.

The numerical results indicate that the fatigue-critical regions of the shaft are consistently localized at the outer surface under all applied load levels. As the torque amplitude increases from 10 N·m to 13 N·m, the surface fatigue life of the shaft decreases monotonically, with the minimum fatigue lives being 30,170, 8,175, 2,262, and 145 cycles, respectively. Under asymmetric cyclic loading, the fatigue life exhibits a pronounced nonlinear inverse dependence on the torque amplitude, following the power-law decay characteristics of the Basquin relationship. Specifically, within the torque amplitude range of 10–12 N·m, the fatigue life decreases in an approximately exponential manner with increasing torque amplitude.

From the perspective of the damage mechanism, the alternating shear stress amplitude, which is governed by the torque amplitude, serves as the dominant factor controlling the

fatigue life. A higher stress amplitude accelerates the accumulation of fatigue damage per cycle, thereby shortening the fatigue life. Meanwhile, a positive mean shear stress is present under this loading condition. According to the Goodman mean-stress correction theory, the superimposed mean shear stress reduces the allowable alternating stress of the material, promotes fatigue crack initiation, and further degrades the fatigue performance. In particular, under the highest torque amplitude of 13 N·m, the maximum surface shear stress approaches the torsional yield strength of the material, and local cyclic plastic deformation is significantly intensified, such that the mean-stress-induced damage effect is further amplified, resulting in a sharp reduction in fatigue life. Overall, the fatigue life evolution observed under this typical stress ratio is consistent with the fundamental characteristics of asymmetric torsional fatigue of metallic materials. The present simulation results provide a loading baseline for the subsequent evaluation of the life improvement achieved by composite strengthening processes.

3.3. Numerical Simulation Study on the Fatigue Life of Composite-Strengthened Shafts

To investigate the influence of different surface characteristic states on the torsional fatigue performance of shafts subjected to laser quenching–ultrasonic rolling composite strengthening, three key surface characteristic parameters, hardened layer depth, tempered zone width, and surface residual stress, are selected as the research variables. The corresponding three-level values of each parameter are summarized in Table 6. The simulation results reveal that the fatigue life of the composite-strengthened shaft exhibits distinct axial zoning characteristics, which correspond closely to the spatial distribution of the surface modified layer. The fatigue life distribution of the shaft subjected to the composite strengthening treatment is presented in Figure 9.

Table 6. Surface characteristic parameters of composite strengthening.

Surface Characteristic Parameters	Parameter Values
Hardened layer depth, H/mm	0.65/0.75/0.85
Tempered zone width, W/mm	1.60/1.75/1.90
Average surface residual stress, σ /MPa	−650/−500/−350

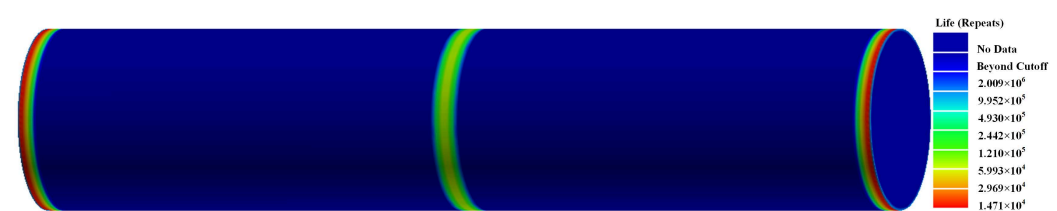


Figure 9. Fatigue life distribution of the shaft treated by composite strengthening.

Three characteristic regions can be identified: the end heat-affected zones, the middle-hardened zone, and the central overlapping zone. The end heat-affected zones are the weakest regions in terms of fatigue resistance, with a minimum surface fatigue life of 14,710 cycles. Because these regions are subjected only to cyclic laser thermal loading and do not undergo sufficient martensitic transformation, their surface hardness and torsional yield strength are lower than those of the hardened zone; consequently, they exhibit the poorest fatigue resistance and serve as preferential sites for fatigue crack initiation. The middle-hardened zone possesses a fatigue life exceeding 10^7 cycles, as complete laser phase-transformation hardening combined with ultrasonic rolling plastic strengthening substantially increases its surface hardness and yield strength, thereby achieving the optimal strengthening effect. The central overlapping zone exhibits a slightly lower fatigue

life than the middle-hardened zone but remains far superior to the end heat-affected zones. In this zone, secondary cyclic laser thermal loading induces surface temper softening, which moderately weakens the hardening effect and marginally reduces the fatigue life. Nevertheless, the hardened microstructure and residual compressive stresses are retained, so this zone does not constitute a new weak link for fatigue failure.

The double-pass overlapping composite strengthening process does not alter the fundamental pattern of surface-initiated cracking under torsional fatigue; however, it significantly modifies the axial distribution of fatigue-weak sites along the shaft. In the unstrengthened state, fatigue-weak sites are distributed almost uniformly over the entire surface of the shaft segment. After composite strengthening, the primary fatigue-weak sites shift to the heat-affected zones at both ends, whereas the fatigue performance of the hardened and overlapping zones is substantially improved. Even in the heat-affected zones, which exhibit the poorest performance after strengthening, the minimum fatigue life is increased to 14.7 times that of the unstrengthened specimen, demonstrating the remarkable fatigue-life enhancement achieved by this composite strengthening process.

The combined laser-quenching and ultrasonic surface rolling treatment induces a near-equibiaxial compressive residual stress at the shaft surface, dominated by the axial and circumferential (hoop) components. The stress field, measured by X-ray diffraction ($\sin^2\psi$ method) with successive electrolytic layer removal, exhibits the typical strengthened-layer profile: approximately -650 MPa at the surface, a peak at 0.05 – 0.15 mm depth, and a near-exponential decay vanishing at the hardened-layer boundary (≈ 0.85 mm); the component on the critical shear plane was extracted and superimposed on the applied torsional shear stress field. An exponential fit of the measured profile was mapped onto the integration points of the strengthened-layer elements in the layered FE model and introduced as a self-equilibrating initial stress field (via *INITIAL CONDITIONS, TYPE=STRESS or the SIGINI subroutine in Abaqus), followed by a zero-load equilibrium step before cyclic loading. In the fatigue life prediction, the residual stresses enter the mean-shear-stress-corrected S–N model through the local mean stress, and the node-by-node superimposed stress field is transferred directly to the fatigue solver, fully preserving the depth-wise gradient. The residual stresses are assumed stable during cyclic loading on three grounds: the maximum superimposed stress remains below the cyclic yield strength, limiting macroscopic relaxation; deep work-hardened layers (high dislocation density and refined grains) reportedly retain most of their compressive residual stress at comparable stress levels; and the predicted fatigue lives agree with torsional fatigue tests within 15%.

3.4. Response Surface Modeling and Parameter Sensitivity Analysis of the Fatigue Life of Composite-Strengthened Shafts

A fatigue life prediction model based on the response surface methodology (RSM) is developed to quantify the mapping relationship between the surface characteristic parameters of composite-strengthened shafts and their fatigue life. The hardened layer depth H , the tempered zone width W , and the average surface residual stress σ are selected as the input variables. A total of 17 numerical simulation runs is arranged according to the Box–Behnken design to obtain the global fatigue life N_g and the tempered-zone fatigue life N_t . The design matrix and the corresponding simulation results are summarized in Table 7. The comparison between the predicted and simulated results is summarized in Table 8.

Based on the Box–Behnken design simulation data, a second-order response surface model is established for each response indicator. The results of the analysis of variance (ANOVA) and significance tests indicate that the tempered zone width W has no statistically significant effect on the global fatigue life; therefore, this term is removed from the corresponding model to improve its predictive accuracy. The final response surface models are expressed as follows:

$$N_g = 5.5882 \times 10^5 H^2 + 0.79982 \sigma^2 - 1415.3 H \sigma - 1.3492 \times 10^6 H + 1642.0 \sigma + 8.0291 \times 10^5 \quad (14)$$

$$N_t = 7.8021 \times 10^6 H^2 + 4.8280 \times 10^6 W^2 + 4.2789 \sigma^2 - 1.2822 \times 10^7 H W - 11493 H \sigma + 9036.3 W \sigma + 7.0570 \times 10^6 H - 4.2851 \times 10^6 W - 4499.3 \sigma + 2.1391 \times 10^5 \quad (15)$$

The ANOVA and significance tests are performed on the established regression models, and the results are summarized in Table 9. The p -values of all response models are less than 0.0001, indicating that the overall models are highly statistically significant. Moreover, the mean square of the regression is much larger than that of the lack-of-fit term for each model, demonstrating that the models fit the simulation data well and can accurately characterize the mapping relationship between the response indicators and the surface characteristic parameters.

Table 7. Box–Behnken design scheme and simulation results.

No.	Surface Characteristic Parameters			Fatigue Life	
	H (mm)	W (mm)	σ (MPa)	N_g	N_t
1	0.75	1.60	−650	64,410	964,900
2	0.75	1.75	−500	14,710	76,210
3	0.65	1.75	−350	493	4402
4	0.65	1.75	−650	27,560	108,000
5	0.65	1.90	−500	5850	8677
6	0.85	1.90	−500	36,126	62,830
7	0.75	1.75	−500	14,710	76,210
8	0.75	1.60	−350	2372	55,970
9	0.75	1.90	−350	2373	3963
10	0.75	1.75	−500	14,710	76,210
11	0.75	1.75	−500	14,710	76,210
12	0.85	1.60	−500	36,120	901,700
13	0.75	1.90	−650	64,420	99,630
14	0.75	1.75	−500	14,710	76,210
15	0.65	1.60	−500	5848	78,240
16	0.85	1.75	−350	6418	48,220
17	0.85	1.75	−650	118,400	841,400

Table 8. Comparison between predicted and simulated results.

Surface Characteristic Parameters			Predicted Values		Simulated Values	
H (mm)	W (mm)	σ (MPa)	N_g	N_t	N_g	N_t
0.65	1.60	−650	3.0599×10^4	4.56762×10^5	2.756×10^4	4.15500×10^5
0.75	1.75	−650	6.5903×10^4	4.10157×10^5	6.441×10^4	3.85765×10^5
0.85	1.60	−350	4.143×10^3	3.73148×10^5	5.418×10^3	4.67814×10^5
0.85	1.60	−650	$112,384 \times 10^5$	1.595534×10^6	1.484×10^5	2.0795×10^6

The parameter sensitivity analysis based on the F -values and p -values obtained from ANOVA indicates that the average surface stress σ and the hardened layer depth H are the primary sensitive factors affecting the global fatigue life. For the tempered-zone fatigue life, all three parameters (σ , H , and W) exert highly significant effects. The average surface stress σ is the primary common sensitive factor for both fatigue-life responses, followed by the hardened layer depth H and the tempered zone width W in descending order of sensitivity. In the global fatigue life model, the highly significant terms include the $H\sigma$

interaction term and the quadratic terms H^2 and σ^2 , whereas all interaction and quadratic terms in the tempered-zone fatigue life model are highly significant. According to the comprehensive sensitivity results, the priority order for parameter regulation of the fatigue life of composite-strengthened components is $\sigma > H > W$.

Table 9. Analysis of variance for regression models of response values.

Response	Source	Sum of Squares	df	Mean Square	F-Value	p-Value	Significance
Global fatigue life N_g	Model	1.510×10^{10}	5	3.020×10^9	171.11	<0.0001	***
	H	3.093×10^9	1	3.093×10^9	175.25	<0.0001	***
	σ	8.655×10^9	1	8.655×10^9	490.32	<0.0001	***
	$H\sigma$	1.803×10^9	1	1.803×10^9	102.12	<0.0001	***
	H^2	1.319×10^8	1	1.319×10^8	7.47	0.0195	***
	σ^2	1.367×10^9	1	1.367×10^9	77.46	<0.0001	***
	Residual	1.942×10^8	11	1.765×10^7			
	Lack of Fit	1.942×10^8	7	2.774×10^7			
	Pure Error	0.0000	4	0.0000			
	Total	1.530×10^{10}	16				
Tempered-zone fatigue life N_t	Model	1.771×10^{12}	9	1.967×10^{11}	505.76	<0.0001	***
	H	3.423×10^{11}	1	3.423×10^{11}	880.04	<0.0001	***
	W	4.167×10^{11}	1	4.167×10^{11}	1071.17	<0.0001	***
	σ	4.519×10^{11}	1	4.519×10^{11}	1161.80	<0.0001	***
	HW	1.480×10^{11}	1	1.480×10^{11}	380.39	<0.0001	***
	$H\sigma$	1.189×10^{11}	1	1.189×10^{11}	305.63	<0.0001	***
	$W\sigma$	1.653×10^{11}	1	1.653×10^{11}	425.10	<0.0001	***
	H^2	2.563×10^{10}	1	2.563×10^{10}	65.89	<0.0001	***
	W^2	4.969×10^{10}	1	4.969×10^{10}	127.74	<0.0001	***
	σ^2	3.903×10^{10}	1	3.903×10^{10}	100.33	<0.0001	***
	Residual	2.723×10^9	7	3.890×10^8			
	Lack of Fit	2.723×10^9	3	9.076×10^8			
	Pure Error	0.0000	4	0.0000			
	Total	1.773×10^{12}	16				

Note: *** highly significant, ** significant, * insignificant.

Table 10 summarizes the goodness-of-fit and prediction accuracy metrics of each response surface model. The R^2 values of all models exceed 0.95, with the adjusted R^2 values above 0.95 and the predicted R^2 values remaining above 0.90. These results demonstrate that the established second-order response surface models achieve excellent fitting performance and possess good generalization and predictive capability.

Table 10. Error statistics of response surface equations.

Evaluation Index	N_g	N_{t5}
R^2	0.9873	0.9985
Adjusted R^2	0.9815	0.9965
Predicted R^2	0.9233	0.9754

3.5. Influence of Surface Characteristic Parameters on the Fatigue Life of Composite-Strengthened Shafts and Multi-Objective Optimization

Response surface plots provide an intuitive visualization of the interactions among the input factors and their effects on the response variables. The response surface plots for the global fatigue life and the tempered-zone fatigue life are presented in Figure 10.

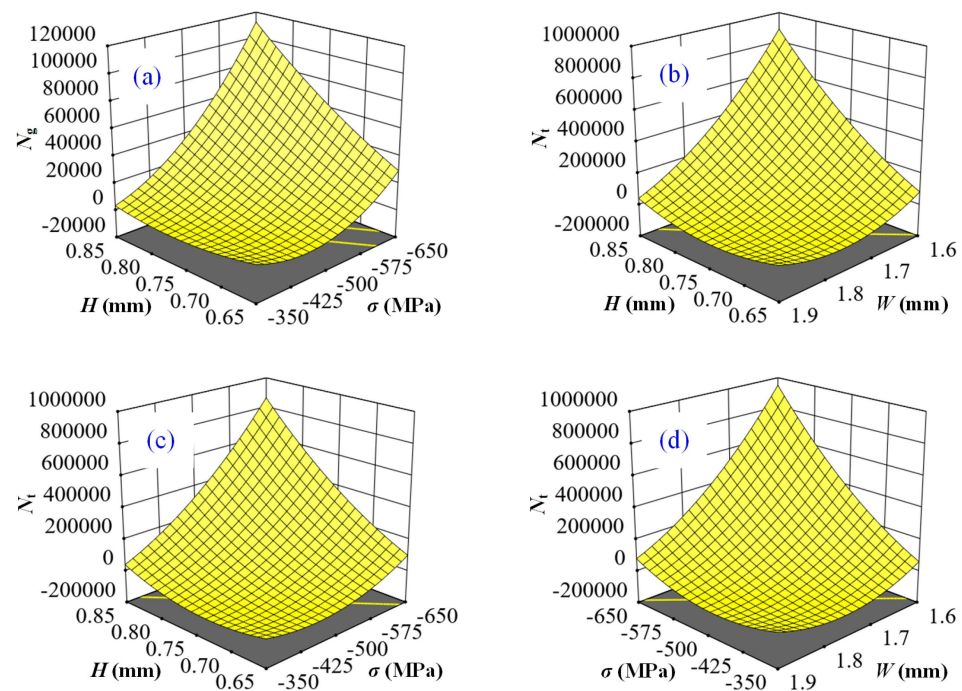


Figure 10. Response surfaces of fatigue life for composite strengthening (a) Response surface of N_g as a function of H and σ ; (b) Response surface of N_t as a function of H and W ; (c) Response surface of N_t as a function of H and σ ; (d) Response surface of N_t as a function of W and σ .

Figure 10a shows the response surface plot of the global fatigue life, while Figure 10b–d present the response surface plots of the tempered-zone fatigue life. The global fatigue life is governed by the coupling effect of the hardened layer depth H and the average surface residual stress σ . The average surface residual stress σ exerts the strongest influence: increasing the magnitude of the compressive residual stress markedly improves the global fatigue life, and a greater hardened layer depth H is also beneficial to the fatigue performance. The steeper slope of the response surface along the σ direction indicates that σ carries a greater weight than H , whereas the pronounced curvature of the surface demonstrates a significant interaction between the two parameters. The tempered-zone fatigue life is jointly governed by H , W , and σ : reducing the average surface residual stress σ , increasing the hardened layer depth H , and decreasing the tempered zone width W all contribute to an improved tempered-zone fatigue life. The pronounced curvature of the H – W , H – σ , and W – σ response surfaces reveals significant interactions among these parameters. Taken together with the surface characteristic analysis, the average surface residual stress is identified as the core parameter governing the fatigue life of the component, followed by the hardened layer depth, whereas the tempered zone width exerts a notable influence only on the tempered-zone fatigue life.

The steepest ascent method is first employed to preliminarily locate the optimal parameter combination. In conjunction with the established quadratic regression models, local optimization algorithms such as gradient descent are then applied to progressively reduce the iteration step size and refine the search space. After multiple iterations, the optimal parameter combination is identified. Within the parameter ranges considered in this study, the optimization objectives are to maximize both the global fatigue life and the tempered-zone fatigue life. The optimization model and the associated parameter constraints are summarized in Table 11.

Table 11. Constraints for optimal parameters.

Find	H, W, σ
Range	$0.65 \text{ mm} \leq H \leq 0.85 \text{ mm}; 1.60 \text{ mm} \leq W \leq 1.90 \text{ mm}; -650 \text{ MPa} \leq \sigma \leq -350 \text{ MPa}$
Maximize	N_g, N_t

Under this optimization model, the optimal surface characteristic parameters for composite strengthening are determined as follows: a hardened layer depth $H = 0.85 \text{ mm}$, a tempered zone width $W = 1.60 \text{ mm}$, and an average surface residual stress $\sigma = -650 \text{ MPa}$. All three parameters are located at the boundaries of their respective variation ranges, which is consistent with the variation trends of the fatigue life revealed by the response surface analysis. With this optimal parameter combination, the predicted global fatigue life reaches 112,384 cycles, while the predicted tempered-zone fatigue life reaches 1,595,534 cycles, demonstrating a remarkable improvement in the fatigue performance of the component.

4. Torsional Fatigue Experiments of Shaft Steel

4.1. Torsional Fatigue Experimental Design for Shaft Steel

The chemical composition of the 30CrMoA steel used in this study is summarized in Table 1. The initial microstructure of the material is primarily composed of 86% pearlite and 14% ferrite, with an initial hardness of approximately 245 HV. After forging, the specimens were subjected to quenching and tempering treatments: they were heated to 880°C and oil-quenched to room temperature, followed by tempering at 540°C and water cooling to room temperature. The hardness after quenching and tempering is approximately 260 HV. Standard torsional fatigue specimens were fabricated by wire electrical discharge machining followed by grinding, and their geometry and dimensions are shown in Figure 11. The torsional fatigue tests were conducted on an MTS 215.32 torsional fatigue testing machine (Figure 12), and its main technical parameters are summarized in Table 12.

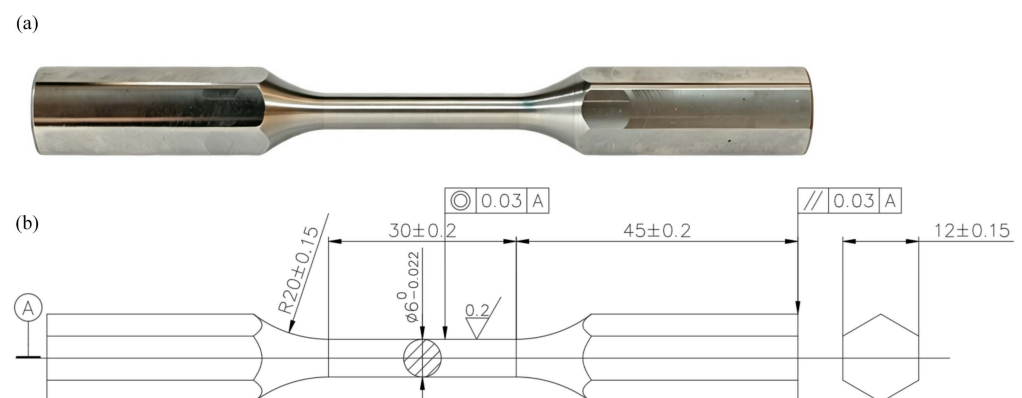


Figure 11. Schematic diagram of dimensions of torsional fatigue specimen (a) Photograph of the torsional fatigue specimen; (b) Dimensioned engineering drawing of the torsional fatigue specimen.

The tester operates on the principle of servo closed-loop control. During testing, both ends of the specimen are rigidly clamped, and the servo actuator system applies an alternating torsional load to the specimen. During loading, the torque and angular displacement sensors measure the torque and torsion angle in real time and feed the measured signals back to the closed-loop control system for load correction.

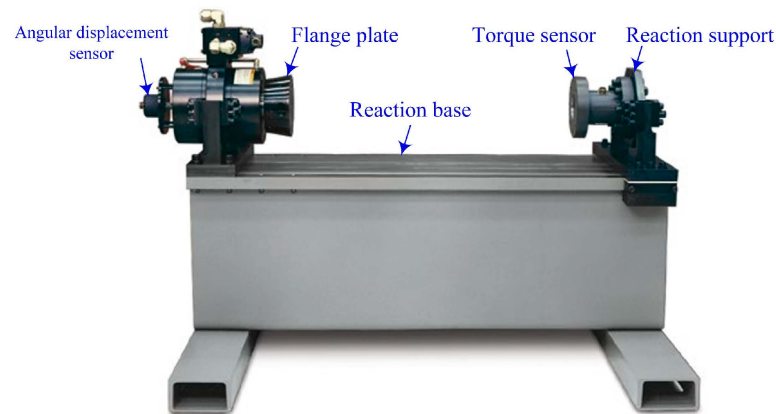


Figure 12. Torsional fatigue test platform.

Table 12. Technical parameters of MTS215.32 torsional fatigue testing machine.

Parameter	Value	Parameter	Value
Maximum test torque	$\geq 200 \text{ N}\cdot\text{m}$	Data acquisition frequency	$\geq 120 \text{ kHz}$
Torsion angle range	$-135^\circ \sim 135^\circ$	Data acquisition resolution	$\geq 24 \text{ Bit}$
Alternating load frequency	$\geq 50 \text{ Hz}$	Extensometer gauge length	225 mm
Closed-loop control frequency	$\geq 6000 \text{ Hz}$	Torque strain	$\pm 5^\circ$

In this study, macroscopic fracture of the specimen is defined as the failure criterion. Three replicate tests are performed at each load level to ensure the repeatability of the results, and the detailed test matrix is summarized in Table 13.

Table 13. Torsional fatigue test scheme.

No.	Torque Amplitude (N·m)	Stress Ratio
1	24	−1
2	25	−1
3	26	−1
4	27	−1
5	10	0.14435
6	11	0.14435
7	12	0.14435
8	13	0.14435

The depth of the hardened layer is governed by the laser power and the scanning speed, which together define the linear energy density. The higher the energy density, the greater the austenitizing depth and, correspondingly, the deeper the martensitic layer after self-quenching. The process parameters of the multi-pass overlapping laser quenching are summarized in Table 14. Within the linear energy density range adopted in this study, the theoretical austenitizing depth is approximately 0.8–0.9 mm, which is in good agreement with the measured hardened layer depth of 0.85 mm. The tempered zone is an inherent consequence of multi-pass overlapping: the thermal influence of the adjacent pass induces temper softening in the overlapping region of the preceding pass. For overlap rates of 30–50%, the width of the temper-softened band is approximately 1.5–1.7 mm, consistent with the measured value of 1.60 mm, thereby validating the physical basis for the layered modeling of the hardened layer coupled with the tempered zone.

Table 14. Process parameters for multi-pass laser hardening.

Parameter	Value	Remarks
Laser type	Fiber/diode laser	Continuous-wave (CW) output
Laser power, P	1.2–1.8 kW	Adjusted according to the target hardening depth
Spot size	3 mm × 3 mm (rectangular homogenized spot)	Ensures a uniform temperature field
Scanning speed, v	8–12 mm/s	—
Multi-pass overlap ratio	30–50%	Forms a tempered zone
Shielding gas	Ar, flow rate 15 L/min	Prevents surface oxidation
Surface pretreatment	Blackening/light-absorbing coating	Enhances laser energy absorptivity

For laser quenching alone, the surface layer exhibits residual tensile stress or low-level compressive stress, arising from the competition between transformation-induced expansion and thermal contraction. Ultrasonic surface rolling (USRP) induces plastic deformation of the surface layer through high-frequency impact; the process parameters are summarized in Table 15. This treatment introduces residual compressive stresses of substantial depth and high magnitude. The level of -600 to -700 MPa is consistent with reported values for laser-quenched 30CrMo-series steels in the literature. Residual compressive stresses of this magnitude can substantially offset the externally applied mean shear stress, which is mutually corroborated by the response surface analysis in this study, wherein the residual compressive stress is identified as the most sensitive factor governing fatigue life.

Table 15. Process parameters for ultrasonic surface rolling.

Parameter	Value	Remarks
Ultrasonic vibration frequency	20 kHz	—
Amplitude	6–10 μm	—
Static force	600–1000 N	—
Carbide ball diameter	$\phi 6$ mm	WC ball
Feed rate	0.05–0.10 mm/rev	—
Spindle speed	50–100 r/min	—
Number of rolling passes	2–3	—

The measured mean values (0.85 mm/1.60 mm/ -650 MPa) are consistent with the optimal surface-layer parameters obtained from response surface optimization, and the measured surface characteristics of the gear shaft are presented in Table 16. This demonstrates that the optimization results are achievable and reproducible within the existing process window, thereby strengthening the chain of evidence for the reliability of parameter control and directly corroborating the experimental validation conclusion that the fatigue life prediction error remains below 15%.

The hardened layer depth was determined in accordance with GB/T 5617 Determination of the effective hardened layer depth of steel after induction hardening or flame hardening, using the hardness gradient method with a limiting hardness of 550 HV. The residual stress was measured by X-ray diffraction (XRD) using the $\sin^2\psi$ technique, with multiple measurements taken at different positions along both the axial and circumferential directions and averaged. Microhardness was tested along the cross-sectional depth

direction at a step spacing of 0.05 mm to ensure adequate resolution of the hardness gradient curve.

Table 16. Measured surface-layer characteristics of the gear shaft.

Test Item	Test Method	Measured Result
Hardened layer depth	Metallographic method + microhardness gradient method (GB/T 5617)	0.82–0.88 mm (mean $\approx$ 0.85 mm)
Tempered zone width	Metallographic method (softened band in the overlap region)	1.55–1.65 mm (mean $\approx$ 1.60 mm)
Surface microhardness	Micro-Vickers hardness tester, load of 0.5 kgf	Hardened layer: 650–720 HV _{0.5} ; tempered zone: 480–520 HV _{0.5} ; matrix: $\sim$ 300 HV _{0.5}
Surface residual stress	X-ray diffraction $\sin^2\psi$ method (Cr K α radiation, {211} lattice planes)	After combined treatment: -610 to -690 MPa (mean $\approx$ -650 MPa)

4.2. Fatigue Test Analysis Under Symmetric Cyclic Loading

Symmetric cyclic loading corresponds to a stress ratio of $R = -1$, under which the forward and reverse torque amplitudes of the applied alternating torsional load are equal. This loading condition serves as the baseline case for constructing the theoretical S–N curve. The fatigue test and simulation results obtained under different torque amplitudes are presented in Figure 13.

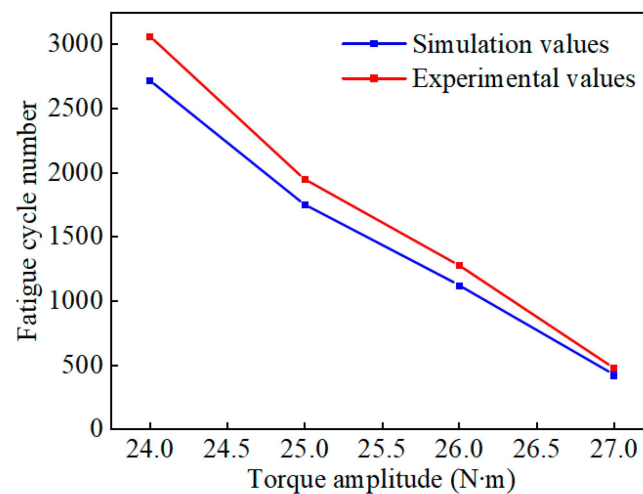


Figure 13. Comparison of experimental and simulated torsional fatigue life under symmetric cyclic loading.

Under symmetric cyclic conditions, the mean shear stress is zero, and the fatigue damage is entirely attributed to the alternating shear stress amplitude; accordingly, crack initiation and propagation are dominated by the shear mechanism. As indicated by the trends of the curves in Figure 13, both the experimental and simulation results reveal that the fatigue life decreases with increasing torque amplitude. An increase in the torque amplitude elevates the alternating shear stress on the critical cross-section of the specimen, thereby promoting repeated dislocation slip within the material and aggravating local plastic deformation. Consequently, cracks preferentially initiate at surface defects and microstructural inhomogeneities, and the crack propagation rate increases accordingly. The simulated fatigue lives are generally lower than the experimentally measured values, indicating that the model provides conservative predictions, whereas the two curves exhibit

highly consistent slopes. As further confirmed by the quantitative data in Table 10, the relative errors between the simulated and experimental results at all load points under this loading condition remain within 12.5%. These findings demonstrate that the established fatigue damage model can accurately characterize the evolution of the fatigue life with the torque amplitude and achieves good predictive accuracy.

4.3. Fatigue Test Analysis Under Typical Stress Ratio Conditions

Torsional fatigue tests were conducted under an asymmetric cyclic loading condition with a stress ratio of $R = 0.14435$, so as to simulate the engineering scenario in which an alternating torsional load is superimposed on a static preload, as typically encountered by transmission shafts in service. The experimental and simulation results are presented in Figure 14. Under asymmetric cyclic loading, both the experimental and simulation results show that a higher torque amplitude leads to a shorter fatigue life. Compared with the symmetric cyclic loading at $R = -1$, the introduction of the mean shear stress induces additional static damage and accelerates fatigue crack nucleation. The quantitative results in Table 10 indicate that the simulated fatigue lives under this condition are also slightly lower than the experimental values, with the relative errors at all load points remaining within 15%. The errors increase slightly under low loads, mainly because the effects of random experimental factors, such as the stochastic nature of the material microstructure and specimen surface machining defects, are amplified in the long-life regime. A sharp drop in the fatigue life occurs in the medium-load range, demonstrating that a transition in the crack propagation mechanism occurs once the load exceeds the critical threshold.

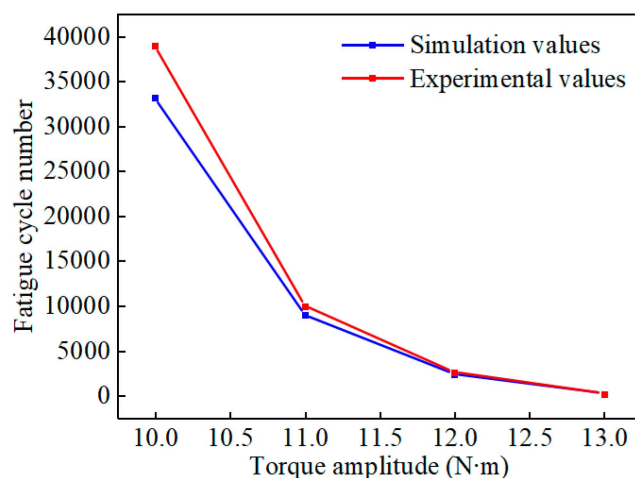


Figure 14. Fatigue life curves at different torque amplitudes under typical stress ratio condition.

In asymmetric cycles, the coupling effect of the alternating shear stress and the mean stress leads to a markedly different crack propagation driving force compared with that under the pure-shear symmetric cycle. The comparison between the experimental and simulation results of this loading group provides data support for the fatigue life prediction of shafts subjected to a mean torque and further verifies the applicability of the proposed model under asymmetric stress ratio conditions.

4.4. Fracture Morphology and Failure Characteristics of Base-Material Specimens

The fracture morphology of the specimens fractured under the symmetric cyclic and asymmetric stress ratio conditions was observed using a VHX-100 digital microscope with a large depth of field (Keyence Corporation, Osaka, Japan). The representative fracture images are presented in Figure 15.

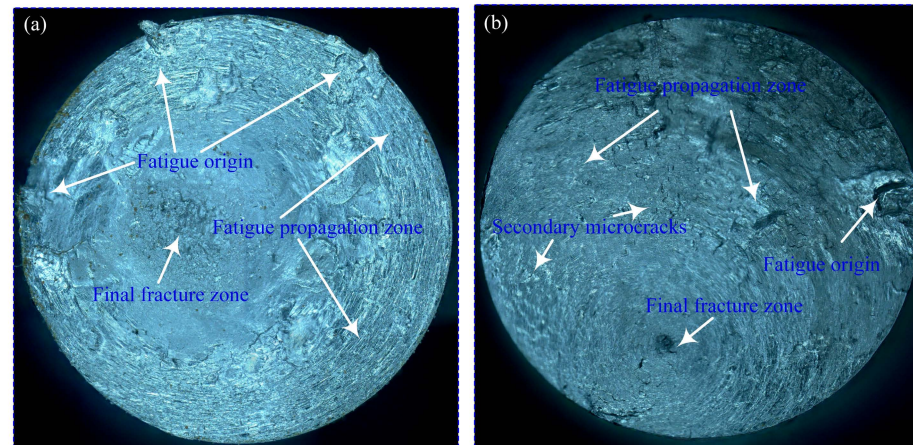


Figure 15. Fracture morphologies under different loading conditions: (a) symmetric cyclic loading; (b) asymmetric cyclic loading.

Figure 15a presents the fracture morphology under symmetric cyclic loading. Clear annular fatigue beach marks are observed on the fracture surface, and the fracture exhibits multi-origin crack initiation characteristics, with all fatigue origins nucleating at the outer surface of the specimen. Under torsional loading, the shear stress reaches its maximum at the outer surface of the specimen; therefore, surface machining defects and microstructural inhomogeneities preferentially serve as crack initiation sites, from which the cracks propagate inward toward the specimen core. The fracture surface is covered with layered and grooved features formed by shear sliding, which are typical characteristics of shear-dominated torsional fatigue fracture. The central region of the fracture surface corresponds to the final rupture zone and exhibits a rough morphology.

Figure 15b shows the fracture morphology under the asymmetric stress ratio condition. The fracture is also of the surface-initiated fatigue type, in which the fatigue cracks extend inward from the specimen surface and form distinct fatigue propagation zones. Compared with the fracture under symmetric cyclic loading, a larger number of secondary microcracks are observed on this fracture surface. This is because the mean shear stress contributes to damage accumulation and promotes the continuous branching of the main crack during propagation, thereby generating numerous secondary microcracks.

Torsional fatigue failures under both loading conditions are characterized by surface crack initiation. Symmetric cyclic loading is dominated by alternating shear damage accompanied by multi-origin crack initiation, whereas asymmetric cyclic loading involves alternating shear damage superimposed on the static damage induced by the mean stress, leading to a markedly increased number of secondary cracks.

5. Conclusions

To address the complex torsional fatigue damage evolution and the difficulty of quantitative fatigue life evaluation arising from the multilayer gradient structure formed on the shaft surface by multi-track overlapping laser quenching–ultrasonic rolling composite strengthening, this study systematically investigates the torsional fatigue performance of marine 30CrMoA steel shafts under mean shear stress, combining theoretical modeling, numerical simulation, and experimental verification. A torsional fatigue S–N model modified by the mean shear stress and a layered finite element model comprising the matrix, hardened layer, and tempered zone were established, and symmetric and asymmetric cyclic torsional fatigue simulations were performed to explore the effects of the applied load on the fatigue life distribution and degradation. The response surface methodology was adopted to analyze the interactive effects of the hardened layer depth, tempered zone width,

and surface residual compressive stress on the global and tempered-zone fatigue lives, thereby optimizing the surface-layer characteristics for maximum fatigue life. Torsional fatigue tests combined with fracture morphology observations were conducted to clarify the failure mechanisms under different stress ratios and to verify the simulation accuracy and the reliability of parameter regulation. The main conclusions are drawn as follows:

- (1) A theoretical torsional fatigue S–N model incorporating the mean shear stress correction and a layered finite element model consisting of the matrix, hardened layer, and tempered zone were constructed for laser quenching–ultrasonic rolling composite-strengthened shafts, enabling the numerical simulation of their torsional fatigue behavior.
- (2) Torsional fatigue simulations of 30CrMoA steel were performed under a symmetric cycle ($R = -1$) and an asymmetric cycle ($R = 0.14435$). The fatigue-critical regions under both loading conditions are located at the outer surface of the component. With increasing torque amplitude, the fatigue life decays following the Basquin power law. The positive mean shear stress aggravates fatigue damage, and the local cyclic plastic deformation induced by high loads leads to a sharp reduction in the fatigue life.
- (3) A series of torsional fatigue simulations were performed on the laser quenching–ultrasonic rolling composite-strengthened 30CrMoA specimens, and a second-order response surface model for the fatigue life was established via the Box–Behnken design, revealing the influences of the individual surface-layer characteristic parameters on the fatigue life. The parameter sensitivity is ranked in the following order: surface residual compressive stress > hardened layer depth > tempered zone width.
- (4) With the maximization of both the global and tempered-zone fatigue lives as the optimization objective, the optimal surface-layer characteristic parameters were determined as follows: a hardened layer depth of 0.85 mm, a tempered zone width of 1.60 mm, and a surface residual compressive stress of -650 MPa. The corresponding predicted global and tempered-zone fatigue lives reach 112,384 and 1,595,534 cycles, respectively, indicating that the torsional fatigue performance of the shaft component can be significantly improved.
- (5) Torsional fatigue tests on standard 30CrMoA specimens were conducted to clarify the failure mechanisms under different loading conditions. Symmetric cyclic loading is characterized by alternating shear damage and multi-origin crack initiation, whereas asymmetric cyclic loading additionally introduces static damage induced by the mean shear stress, which promotes the proliferation of secondary microcracks. The simulation results agree well with the experimental measurements, with the overall relative error in the predicted fatigue life being less than 15%.

These findings provide a theoretical basis and technical guidance for the design of composite-strengthened shaft components with enhanced torsional fatigue resistance.

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