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Optimization of Topological Modification Parameters for Heavy-Duty Helical Gears

YING WANG¹, HUILIANG WANG^{1,2,3}, XING LIU², AND YILIN FU²

¹School of Mechanical and Electrical Engineering, Zhengzhou University of Industrial Technology, Zhengzhou, Henan 450064, China

²School of Mechatronics Engineering, Henan University of Science and Technology, Luoyang, Henan 471003, China

³Collaboration Innovation Center of Machinery Equipment Advanced Manufacturing of Henan Province, Henan University of Science and Technology, Luoyang, Henan 471003, China

Corresponding author: Huiliang Wang (whl-ly@163.com)


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ABSTRACT To improve the meshing state of gear pairs, the abrasion of tooth surfaces must be reduced and on the other hand, the functional requirement of transmission stability for different conditions of heavy-duty helical gears must be increased. To do this requires a topological modification of the tooth profile and tooth alignment. The topological modification parameter optimization model is established based on Hertz theory, which takes the functional requirement as a goal, and the tooth surface topological modification parameters as the design variables. The constraint condition equations in the tooth contact analysis process are derived, and the improved genetic algorithm is used to solve the numerical conditions of the tooth surface microgeometry parameter optimization model. Then, the modification parameters optimized by different modification schemes are obtained. The loaded transmission errors, the mesh misalignment movements, and the distribution characteristics of the tooth surface contact stress are studied. The parameter optimization model of the tooth surface topological modification for the functional requirements is established. With different modification parameters heavy-duty helical gear pairs under various loads, the research results provide a theoretical basis for tooth surface micro design of heavy-duty helical gears in practical application and in determining the reasonable modification parameters.

INDEX TERMS Heavy-duty helical gears, topological modification, parameter optimization, loaded transmission error, mesh misalignment.

I. INTRODUCTION

Heavy-duty helical gears are an important transmission component of the automobile, wind power generation, metallurgy, mining, oil and cement, and other large industries involving the transmission of mechanical and electrical equipment parts. At high-speed operation of such equipment, the helical gears bear enormous loads, which cause unfavorable displacements in tooth contact resulting in edge contact, highly concentrated stress, shock, and vibration [1], [2]. In these devices during times of significant loads, the helical gears bearing will produce elastic deformation and base pitch error. The influence of processing error, assembly error, and cyclical changes in tooth stiffness leads to the actual meshing point deviating from the theoretical meshing point,

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which changes the transmission characteristics of the gear producing partial loading and causing vibration and noise. This requires the tooth manufacturing factory to actively adapt to the requirements of the equipment to improve its service life and reliability. With the development of science and technology, the demand for gear transmission systems is always increasing because the characteristics of the meshing gears directly impact the life of the gears. Therefore, new design ideas are needed which allow the fast and convenient design and production of the tooth surface for adapting to different specifications to meet the functional requirements of gear transmission systems in different applications.

Tooth surface modification can improve the gear pair meshing state to effectively prevent installation error, machining error, and bearing deformation factors that lead to the edge of the contact, such as reducing vibrations and

noise [3]–[5]. Jiang [6] proposed a design method of modified cylindrical gears with a controllable higher-order polynomial function of the transmission error to reduce vibration and noise. Litvin [7], [8] proposed a novel topology with involute and crowned zones for modified helical gear tooth surfaces. If misalignment does not occur, an involute zone is supplied in the middle portion of the gear tooth surfaces to facilitate line contact. When misalignments develop, zones on the top, bottom, front, and back sides of tooth surfaces are crowned, allowing the bearing contact to be localized. Tran and Wu [9] proposed a numerical approach for the closed-loop topology modification on cylindrical gears that have double-crowned and anti-twist tooth flanks for a CNC internal-meshing gear honing machine. To overcome the problem of flank twist induced by lead modifications of gear flanks, a method for topological modification of gear flanks in continuous generating grinding based on contact trace evaluated genetic algorithm (CTEGA) has been presented [10]. Assembly errors of gears significantly influence the surface contact stress and root bending stress. There has been an early focus on modification technology as an effective means to improve the performance of gears. These technologies can improve meshing conditions through tooth surface modification, effectively preventing edge contact and reducing vibration and noise [11]–[15]. Zhang and Fang [16] established a model for the analysis of tooth contact and transmission errors of helical gears with modified tooth surfaces and found that modified gears can provide gear meshing smoothness and lower the magnitude of transmission errors under a certain load. Of all the factors affecting the performance of a gear drive, the load distribution is considered to be one of the most important. Conry *et al.* [17], [18] presented a mathematical programming method for the evaluation of load distribution and the optimal modifications for gear systems. Optimal modification can achieve optimum load distribution and prolong the life of gears. The load distribution is non-uniform along the contact lines of engaged teeth of two gears in a mesh. This is generally attributed to manufacturing and assembling errors and to the deformation of gears, shaft, bearings, and housing by the loading. Tooth profile modifications and tooth crowning are imperative to improve the resistance against failures from gear teeth rubbing against surfaces. Simon [19] proposed a method to determine load and stress distributions along the contact lines of the instantaneously engaged teeth of helical gears and for the optimization of topological modifications for helical gears. The method of optimization of the load contact pattern is used in hypoid gears by automatic topography modification [20]. This method can be adapted to helical gear drives by its systematic and versatile nature. Kahraman and Bajpai [21], [22] examined a surface wear model for helical gears to determine the influence of tooth modifications on helical gear wear and durability. When properly selected, topological modifications allow significant stress reductions to improve durability and wear. Tavakoli and Houser [23] developed a method for computing static transmission errors

and tooth load sharing to minimize the transmission error of heavily loaded gears. Through the above works, it is clear that the modification of tooth surfaces can improve gear meshing characteristics.

Several works have been published on gear modification using simulation technologies such as TCA, LTCA, and finite elements to solve for modification parameters to improve gear transmission characteristics and obtained good application effects in engineering practice [24], [25]. Kim *et al.* [26] optimized the macro geometry of a helical gear pair for low weight, high efficiency, and low noise. Younes *et al.* [27] implemented multi-objective optimization of a gear unit to minimize the power loss and the vibrational excitation generated by the meshing, via a multi-scale approach that extends from gear contact to the complete transmission. A novel multi-objective optimization of a two-stage helical gearbox is carried out with a comprehensive range of constraints. Two objective functions are formulated, the first minimizes the volume and the second minimizes the total power loss in the gearbox [28]–[30]. Weight reduction with a simultaneous increase in transmission efficiency, by optimization method, is very much essential for gear applications that transmit very high mechanical power, with a significant speed reduction [31]–[33]. In the process of actual transmission, due to the involved stresses, gears will experience an elastic deformation. Along with the influence of manufacturing error, assembly error, and periodic changes in the tooth stiffness, the actual meshing point will inevitably deviate from the theoretical meshing point, thus changing the transmission characteristics of the gear and causing vibration and noise. This paper aims to meet various functional requirements based on the loaded contact analysis and a microgeometry parameter optimization analysis under various loads to choose the ideal gear modification parameters. As a result, the edge contact due to installation error was effectively prevented, and the load can be evenly distributed on the teeth surface.

II. TOPOLOGICAL MODIFICATION OF TOOTH SURFACE

According to different processing requirements including the loading capacity and dynamic characteristics, the tooth surface of helical gears can be designed using the method of changing the modification parameters. From a microscopic perspective, the tooth profile and tooth alignment modification were completed. Tooth surface modification can reduce impact while meshing in and out reduces vibration and noise. These modifications can reduce partial loading, allow the load to be more evenly distributed on the teeth, increase the loaded capacity, improve the lubrication state of the tooth surface when meshing, enhance the anti-adhesion ability of the gears, and reduce the tooth surface relative sliding rate and transmission error. Currently, in order to avoid marginal contact on the tooth surface due to misalignment, we try to achieve this by contouring and longitudinal crowning. This crowning we call topology modification, where the tooth surface is divided into involute and crowning zones.

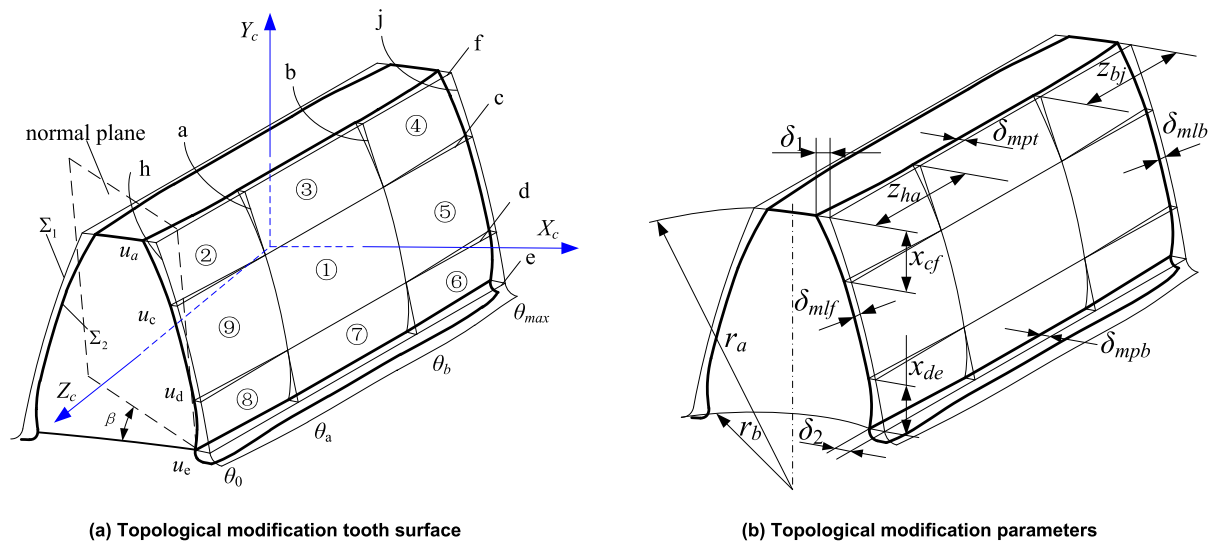


FIGURE 1. Schematic diagram of topological modification of helical gear tooth surface.

A modification-free zone is provided in the central area of the gear tooth face, which will allow line contact if there are no mounting errors. Other areas including the top, bottom, front and rear of the tooth face are crowned to improve bearing contact when a mounting error occurs. The authors propose a grinding wheel finishing procedure that permits topological alteration of the tooth face to avoid significant contact stress areas, reduce transmission errors, and reduce alignment error sensitivity. The essential concept of topological tooth face alteration by grinding form is to use a changed grinding wheel and grinding path.

The topological modification requires a tooth profile and alignment modification at the same time, as shown in Fig. 1 (a). Σ_1 is the theoretical involute tooth surface and Σ_2 is the topological modification tooth surface. Region ① is the involute tooth surface; the corresponding grinding wheel profile is solved according to the involute tooth profile, while the grinding wheel has no radial feeding motion. Regions ②, ④, ⑥, and ⑧ are the tooth profile and tooth alignment bidirectional modifications. Regions ⑤ and ⑨ are tooth lead unidirectional modifications and Regions ③ and ⑦ are tooth profile unidirectional modifications. As shown in the figure, the boundary lines between different regions are $a, b, c, d, e, f, h, j, u_e, u_d, u_c,$ and u_a and are the respective expansion angles of the involute at the tooth profile modification boundary lines. The variables $e, d, c, f, \theta_0, \theta_a, \theta_b,$ and θ_{max} are the respective rotation angles of the gears at the tooth alignment modification boundary lines, $h, a, b,$ and j . By controlling the corresponding involute expansion angles at the tooth profile and alignment modification boundary lines and the size of the gear rotation angle, the division of the tooth surface topological region boundary can be realized. The topological modification parameter distribution diagram is shown in Fig. 1 (b), where, r_a is the tip radius, r_b is the base circle radius, δ_1 and δ_2 are the

maximum amount of modification at the tip and root of gear, respectively.

The modification parameters can be expressed as:

$$f(MP) = f(\delta_{mpt}, \delta_{mpb}, \delta_{mlf}, \delta_{mlb}, z_{ha}, z_{bj}, x_{cf}, x_{de}) \quad (1)$$

Here, δ_{mpt} and δ_{mpb} are the amount of modification at the tooth tip and root, respectively, δ_{mlf} and δ_{mlb} are the amount of modification at the tooth alignment top and bottom, respectively, z_{ha} and z_{bj} are the length of the tooth lead top and bottom modification, respectively, and x_{cf} and x_{de} are the length of tooth tip and root modification, respectively.

The axial profile of the wheel spirals along the axis direction and does parabolic movement in the radial direction of the work gear at the same time. The formed track is thought of as the topological modification tooth surface, so the cross-line equation of the wheel axial profile and middle section of the work gear can be used to obtain the tooth flank equation as described by Eq. (2), shown at the bottom of the next page, in Ref. [34], [35]. where, (u_1, θ_1) are surface parameters, r_b is the base radius, σ_1 is tooth space half-angle, ΔL is profile crowning value, a_x is longitudinal motion value, p_1 is helix parameters, respectively.

Topological crowning includes profile modification and longitudinal modification. The former uses wheel crowning to accomplish the process; the latter changes the relative moving trajectory of the wheel and the gear. The applied profile crowning equation and wheel feeding motion are shown in Table 1. The modified topology of the gear tooth surface shown in Figure 1 can be obtained by application of Eq. (2). Crowning coefficients in profile and longitudinal are applied as follows:

(1) On the top side, regions ②, ③, and ④ will be provided by profile crowning with a coefficient $a_{mp(cf)}$ starting at u_c .

TABLE 1. Applied relations of motion at zones during gear form grinding.

zone	Profile crowning equation	Wheel feed equation
1	$\Delta L = 0$	$a_x = 0$
2	$\Delta L = a_{mp(cf)} r_b^2 (u_1 - u_c)^2$	$a_x = a_{ml(ha)} P_1^2 (\theta_a - \theta_1)^2$
3	$\Delta L = a_{mp(cf)} r_b^2 (u_1 - u_c)^2$	$a_x = 0$
4	$\Delta L = a_{mp(cf)} r_b^2 (u_1 - u_c)^2$	$a_x = a_{ml(bj)} P_1^2 (\theta_1 - \theta_b)^2$
5	$\Delta L = 0$	$a_x = a_{ml(bj)} P_1^2 (\theta_1 - \theta_b)^2$
6	$\Delta L = a_{mp(de)} r_b^2 (u_d - u_1)^2$	$a_x = a_{ml(bj)} P_1^2 (\theta_1 - \theta_b)^2$
7	$\Delta L = a_{mp(de)} r_b^2 (u_d - u_1)^2$	$a_x = 0$
8	$\Delta L = a_{mp(de)} r_b^2 (u_d - u_1)^2$	$a_x = a_{ml(ha)} P_1^2 (\theta_a - \theta_1)^2$
9	$\Delta L = 0$	$a_x = a_{ml(ha)} P_1^2 (\theta_a - \theta_1)^2$

(2) On the bottom side, regions ⑥, ⑦, and ⑧ will be provided by profile crowning with a coefficient $a_{mp(de)}$ ending at u_d .

(3) On the front side, regions ④, ⑤, and ⑥ will be provided by longitudinal crowning with a coefficient $a_{ml(bj)}$ starting at θ_b .

(4) On the backside, regions ②, ③, and ⑧ will be provided by longitudinal crowning with a coefficient $a_{ml(ha)}$ ending at θ_a .

III. PARAMETER OPTIMIZATION OF THE TOOTH SURFACE TOPOLOGICAL MODIFICATION

A. TOOTH SURFACE CONTACT MODEL BASED ON HERTZ THEORY

In practical applications, the gear pair cannot reach the ideal meshing state due to errors and deformation. The partial load appeared in the meshing process and the load distribution along the tooth width is usually non-uniform, leading to a decrease in the meshing quality, which produces vibration and noise. Under loading, the gear teeth and other parts that transmit the load in the transmission system will most likely experience deformation. Therefore, there was an error between the actual tooth form and the theoretical tooth form. The negative effect of this error on the loaded capacity and noise performance can be reduced by tooth profile and alignment modification. Fig. 2 is the elastic deformation diagram of the gear pair contacting the tooth surfaces. The tooth surface affected by the external load produced an elastic

deformation. Assuming the larger wheel is fixed, the small wheel moves along the normal direction under the load P. The two contact points corresponding to the two tooth surfaces were divided into (1-1'), (2-2'), ..., (k-k'), ..., (n-n'), where F_k is the load on any two contact points (k-k'), ε_k represent the clearance between two contact points (k-k') before contact, δ_0 is the initial minimum clearance between the two contact surfaces and Z represent the normal displacement of the inner gear teeth in one meshing period.

As known from the geometric relationship in Fig. 2 (b)

$$\omega_k + \omega_{k'} + \varepsilon_k - Z \geq 0 \tag{3}$$

where ω_k and $\omega_{k'}$ are the elastic deformations produced by the two contact points (k-k').

At any point, k on the left and side of Eq. (3) is a non-negative number. Therefore, defining a new variable $Y_k \geq 0$, Eq. (3) can be transformed into

$$\omega_k + \omega_{k'} + \varepsilon_k - Z - Y_k = 0 \tag{4}$$

As known from the Hertz contact theory, Y_k meets the following conditions:

$$\begin{cases} Y_k = 0, & \text{if } F_k \geq 0 \\ Y_k \geq 0, & \text{if } F_k = 0 \end{cases} \tag{5}$$

where either Y_k or F_k is zero.

When analyzing the contact between the two tooth surfaces, only the elastic deformation was considered and the influence of other factors was neglected. The elastic deformation at any pair of contact points (k-k') is the sum of the deformation of all forces at this point, as

$$\begin{cases} \omega_k = \sum_{j=1}^n a_{kj} F_j \\ \omega_{k'} = \sum_{j'=1}^n a_{k'j'} F_{j'} \end{cases} \tag{6}$$

where a_{kj} and $a_{k'j'}$ are the respective unit deformation coefficients of the applied force F_j and $F_{j'}$ at discrete points (k-k') and are found using the finite element method [36].

Substituting Eq. (6) into (4) obtains,

$$\sum_{j=1}^n (a_{kj} + a_{k'j'}) F_j + \varepsilon_k - Z - Y_k = 0 \tag{7}$$

Considering the elastic deformation of other discrete points, Eq. (7) is expressed in matrix form as

$$-SF + Ze + IY = \varepsilon \tag{8}$$

$$\mathbf{r}_1 = \begin{bmatrix} r_b \cos(\sigma_1 + u_1 - \theta_1) + (r_b u_1 - \Delta L) \sin(\sigma_1 + u_1 - \theta_1) - a_x \cos \theta_1 \\ r_b \sin(\sigma_1 + u_1 - \theta_1) - (r_b u_1 - \Delta L) \cos(\sigma_1 + u_1 - \theta_1) + a_x \sin \theta_1 \\ p_1 \theta_1 \\ 1 \end{bmatrix} \tag{2}$$

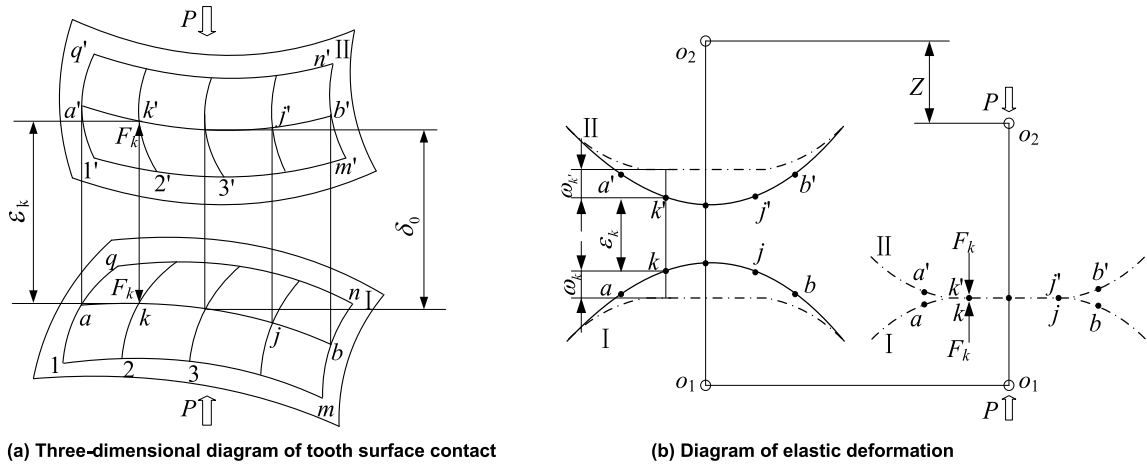


FIGURE 2. The model of two tooth surface contact.

where

$$\begin{aligned}
 S &= [S_{kj}]_{n \times n} \\
 &= [a_{kj} + a_{k'j'}]_{n \times n}, \quad k = 1, 2, \dots, n; j = 1, 2, \dots, n \\
 F &= [F_1 \quad F_2 \quad \dots \quad F_K \quad \dots \quad F_n]^T \\
 Y &= [Y_1 \quad Y_2 \quad \dots \quad Y_K \quad \dots \quad Y_n]^T \\
 \epsilon &= [\epsilon_1 \quad \epsilon_2 \quad \dots \quad \epsilon_K \quad \dots \quad \epsilon_n]^T \\
 e &= [1 \quad 1 \quad \dots \quad 1 \quad \dots \quad 1]^T
 \end{aligned}$$

Assuming that the forces between all the contact points are consistent with the direction of the load P and considering the small instantaneous contact area, the external load P can be thought of as a sum of the forces of all the contact points. The equation is expressed by

$$\sum_{j=1}^n F_j = P \tag{9}$$

where F_j is the loaded capacity for each contact point.

The constraint conditions formed by the above equations are shown as

$$\begin{cases} -SF + Ze + IY = \epsilon \\ e^T F = P \\ F_k = 0 \text{ or } Y_k = 0 \\ F_k \geq 0, \quad Y_k \geq 0, \quad Z \geq 0 \end{cases} \tag{10}$$

To add an artificial variable X_j for Eq. (10) and using the improved simplex method to solve, Eq. (10) can be converted into

$$\begin{aligned}
 &\min(\sum_{j=1}^{n+1} X_j) \\
 &\begin{cases} -SF + Ze + IY + I\bar{X} = \epsilon \\ e^T F + X_{n+1} = P \\ F_k = 0 \text{ or } Y_k = 0 \\ F_k \geq 0, \quad Y_k \geq 0, \quad Z \geq 0, \quad X_j \geq 0 \end{cases} \tag{11}
 \end{aligned}$$

Considering the above iterative method to solve Eq. (11) where load P is known, a_{kj} is the deformation coefficient under unit load obtained by using the finite element method and ϵ_k represents the initial clearance of the tooth from the contact point k , obtained by TCA. Therefore, parameters, S , F , and ϵ are known and the unknown parameters include Z , X , and Y . Combined with linear programming, the improved simplex method can be used to solve for these unknown parameters.

Eq. (11) belongs to a linear programming problem, which is solved by a modified simplex method, considering the introduction of artificial variables X_j and relaxation variables Y_j . The tabular form of the simplex method is shown in Table 3-4. The tabular form is one of the most common forms of the simplex method, where the artificial variables are introduced as the optimization objectives at a later stage and the relaxation variables change the inequality constraint into an equation constraint. Let the initial values of variables F_j, Z , and Y are equal to zero, as shown in row 7 of Table 2, and the initial value of the objective function of equation (11) under the initial conditions is also equal to X_0 . Normalize Table 2 by subtracting the sum of the first $n+1$ rows from the $n+2$ row to obtain the row transformation, as shown in Table 3. How to make the value of X_0 smaller, it is necessary to construct the basis matrix and perform several iterations to get the optimal solution of the target equation.

In Table 3, the values of the elements in row $n+2$ are

$$X' = X_0 - P - \sum_{j=1}^n \epsilon_j \tag{12}$$

$$d_k = \begin{cases} -1 + \sum_{j=1}^n S_{jk}; & k = 1, \dots, n \\ -n; & k = n + 1 \\ -1; & k = n + 2, \dots, 2n + 1 \end{cases} \tag{13}$$

Equation (11) is solved by the above iterative method, where the load P is known; a_{kj} is the deformation coefficient

TABLE 2. Initial simplex algorithm table.

F_1	F_2	...	F_{n-1}	F_n	Z	Y_1	Y_2	...	Y_{n-1}	Y_n	X_1	X_2	...	X_n	X_{n+1}	
$-S_{11}$	$-S_{12}$...	$-S_{1n-1}$	$-S_{1n}$	+1	1					1					$= \varepsilon_1$
$-S_{21}$	$-S_{22}$...	$-S_{2n-1}$	$-S_{2n}$	+1		1					1				$= \varepsilon_2$
\vdots	\vdots			\vdots	\vdots			\ddots					\ddots			\vdots
$-S_{n1}$	$-S_{n2}$...	$-S_{nn-1}$	$-S_{nn}$	+1					1				1		$= \varepsilon_n$
1	1	...	1	1											1	$= P$
											1	1	...	1	1	$= X_0$

TABLE 3. Standardized processing.

F_1	F_2	...	F_n	Z	Y_1	Y_2	...	Y_n	
$-S_{11}$	$-S_{12}$...	$-S_{1n}$	+1	1				$= \varepsilon_1$
$-S_{21}$	$-S_{22}$...	$-S_{2n}$	+1		1			$= \varepsilon_2$
\vdots	\vdots		\vdots	\vdots			\ddots		\vdots
$-S_{n1}$	$-S_{n2}$...	$-S_{nn}$	+1				1	$= \varepsilon_n$
+1	+1	...	+1						$= P$
d_1	d_2	...	d_n	d_{n+1}	d_{n+2}	d_{n+3}	...	d_{2n+1}	$= X'$

TABLE 4. Basic parameters of the gear pairs.

Parameters	Driving gear	Driven gear
Number of teeth N	40	228
Module, m_n /mm	22	22
Pressure angle α_n /($^\circ$)	20	20
Spiral angle β /($^\circ$)	7.5(left)	7.5(right)
Tooth width b /mm	450	450
Modification coefficient x_n	0	0
Center distance E /mm	2948.438	

per unit load, which is obtained using the finite element method; ε_k is the initial clearance at the kth contact point of the tooth pair, which is obtained by TCA. Thus the parameters S, F, ε are known, while the unknown parameters are Z, X, Y form a linear programming, which can be solved by the modified simplex method.

B. THE PARAMETER OPTIMIZATION MODEL OF THE TOOTH SURFACE MODIFICATION

The normal displacement of the gear tooth is transformed into the displacement of the meshing line and the load

transmission error is expressed in the form of an angle as

$$\Delta\phi_2 = \frac{3600 \times 180Z}{\pi r_{b2} \cos \beta_{b2}} \tag{14}$$

where r_{b2} and β_{b2} are, respectively, the driven wheelbase circle radius and spiral angle.

The loaded transmission error is expressed as a displacement on the meshing line as

$$TE = \frac{Z}{\cos \beta_{b2}} \tag{15}$$

Changing the modification parameters allows the variation of the transmission error to be controlled within a certain

range. The amplitude of the transmission error is minimized, which is part of the parameter optimization problem and can be solved with the genetic algorithm. The amplitude of the transmission error is given by

$$\Delta TE(MP) = \max(TE) - \min(TE) \quad (16)$$

where MP is the modification parameters.

Assuming that the contact length of point k is Δ_k , the average load density at point k is F_k/Δ_k and the maximum unit load should satisfy

$$DF \leq p_{\max}e \quad (17)$$

where

$$D = \begin{bmatrix} 1/\Delta_1 & 0 & \cdots & 0 \\ 0 & 1/\Delta_2 & \cdots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \cdots & 1/\Delta_n \end{bmatrix} \quad (18)$$

The multi-objective optimization problem of the heavy-duty gear aims to minimize the amplitude of the loaded transmission error and the unit normal load. The concrete expression is given as

$$\begin{cases} \min F(\delta_{mpt}, \delta_{mpb}, \delta_{mlf}, \delta_{mlb}, z_{ha}, z_{bj}, x_{cf}, x_{de}) \\ = \min [w_1 \Delta TE(MP) + w_2 p_{\max}(MP)] \\ C_{a \min} \leq \delta_{mpt} \leq C_{a \max} \\ C_{f \min} \leq \delta_{mpb} \leq C_{f \max} \\ C_{c \min} \leq \delta_{mlf}, \delta_{mlb} \leq C_{c \max} \\ l_{t \min} \leq x_{cf} \leq l_{t \max} \\ l_{b \min} \leq x_{de} \leq l_{b \max} \\ l_{c \min} \leq z_{ha}, z_{bj} \leq l_{c \max} \end{cases} \quad (19)$$

where after the tooth profile modification, $C_{a \min}$ and $C_{a \max}$ are the minimum and maximum modification values of the tip, respectively, $C_{f \min}$ and $C_{f \max}$ are the minimum and maximum modification values of the root, respectively, $C_{c \min}$ and $C_{c \max}$ are the minimum and maximum modification values of the top and bottom, respectively; $l_{t \min}$, $l_{t \max}$, $l_{b \min}$, and $l_{b \max}$ represent the length constraints of the tooth profile modification; $l_{c \min}$ and $l_{c \max}$ represent length constraints of the tooth alignment; and w_1 and w_2 represent the weighting coefficients, which are equal in this instance.

To determine the reasonable modification parameters, the optimization process is conducted by solving the nonlinear contact problem where the tooth surface microscopic geometric transformation is conducted for different modification parameters. Changing the conditions of the tooth surface contact is an iterative process solving TCA and LTCA until the gear pair meets the specific functional requirements. The optimization variables are the gear modification parameters, δ_{mpt} , δ_{mpb} , δ_{mlf} , δ_{mlb} , z_{ha} , z_{bj} , x_{cf} , and x_{de} , and the goal of the optimization is to minimize the amplitude of the loaded transmission error, ΔTE . The variable p_{\max} represents the unit length of the normal load, the constraint conditions C_a , C_f , and C_c represent the amount of modification in

two directions of the tooth profile and tooth alignment, and the modification lengths are l_t , l_b , and l_c . There are no specific analytical expressions between the optimization goal and the optimization variables leading to the difficulty in acquiring an accurate analytical solution. There are multiple locally optimal solutions in the optimization space. As a result, the improved genetic algorithm is used to solve the optimization problem process as shown in Fig. 3. The improved optimization algorithm has global convergence, and after a certain number of iterations for a basic stable solution.

IV. EXAMPLES AND ANALYSIS

The optimization problem determining the amplitude of the loaded transmission error and the maximum value of the unit normal load to reach the minimum goal is an optimization process of the nonlinear contact problem. Changing the modification parameters causes a change in the tooth surface state. Then, the contact conditions are changed to meet certain functional requirements of the gear pair under the specific working condition. This example takes the gear pair shown in Table 4 as the research object, the modification parameters of the gear pair as the optimization variables, and the transmission characteristics of the gear pair as the optimization goal. There are multiple locally optimal solutions in the optimization space so it is more suitable to use the improved genetic algorithm. The improved genetic algorithm is chosen as the optimization algorithm since it can optimize the macroscopic geometric parameters and the topological modification parameters of the gear pair. Assuming the speed of the small gear is 2000 r/min with a weight of $w_1 = 0.3$, the size of the genetic algorithm population is 50, the evolution algebra is 20, the crossover probability is 0.2, and the mutation probability is 0.3. Under a variety of loads (from 1 kNm, 3 kNm, 5 kNm, 10 kNm, 15 kNm, 20 kNm, 25 kNm to 30 kNm), the resulting simulation characteristics of the transmission error and stress distribution of the gear pair with different modification parameters are shown in Table 5. Here C_1 is the non-modification case, C_2 is the topological modification case, and C_3 is the segment modification case, respectively.

A. ANALYSIS OF THE MESH DISLOCATION CHANGE

Because of the transverse bending deformation and torsion deformation of the supporting shaft, the phenomenon of mesh misalignment along the direction of the tooth is produced. According to the definition of the ISO6336 standard, the mesh misalignment is defined by the maximum clearance of the two corresponding points along the tooth width direction. It varies under different working conditions due to the gear pairs being under different loads. Under non-modification, as shown in Fig. 4 (a), the load is 1 kNm and the change range of the mesh misalignment is $0.11\mu\text{m}$. When the load reached more than 5 kNm, the mesh misalignment range was approximately $0.42\mu\text{m}$. The range of the transmission dislocation change increased rapidly as

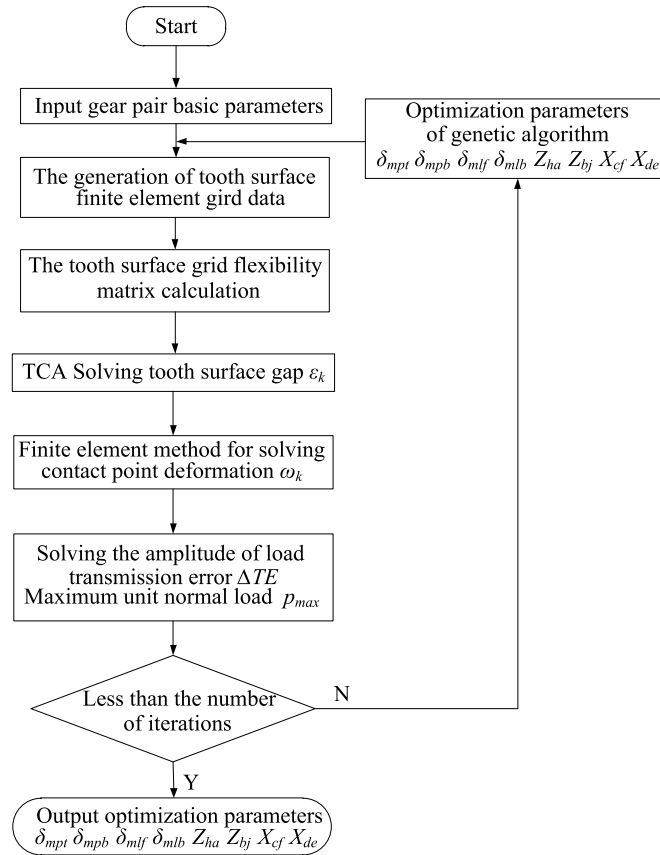


FIGURE 3. The optimization process for the modification parameters.

TABLE 5. Design of different modification parameters under load 30 kNm.

Case	Tooth profile modification parameters				Lead modification parameters			
	$\delta_{mpt}/\mu\text{m}$	$\delta_{mpb}/\mu\text{m}$	x_{cf}/mm	x_{de}/mm	$\delta_{mlf}/\mu\text{m}$	$\delta_{mlb}/\mu\text{m}$	z_{ha}/mm	z_{bj}/mm
C_1	0	0	0	0	0	0	0	0
C_2	20	20	22	27.5	15	15	225	225
C_3	25	25	12.37	7.42	25	25	67.5	382.5

the load increased. When the load reached 30 kNm, the range of the mesh misalignment change reached $1.94\mu\text{m}$. The distribution of the mesh misalignment change of the topological modification as shown in Fig. 4 (b) gives an amplitude that is decreased as compared to the conditions given for Fig. 4 (a). This result shows that the topological modification improves the condition of dislocation change. The mesh misalignment change of the segment topological modification is shown in Fig. 4 (c). When the load is 30 kNm, the minimum value of the mesh misalignment change reached $-2.92\mu\text{m}$. Fig. 4 (d) shows the mesh misalignment change of the different modification methods. When the load is 30 kNm, the amplitude variation range of the mesh dislocation of the topological modification gear pair is minimized, which is beneficial to the stability of the system

transmission. From the above analysis, it can be seen that the topological modification is advantageous for reducing the mesh misalignment change.

B. ANALYSIS OF LOADED TRANSMISSION ERRORS

The simulation results of the transmission errors under multiple loads are shown in Fig. 5. With the load gradually increasing, the amplitude of the transmission error also gradually increases. The smaller the load is, the smaller the amplitude of the transmission error will be. As shown in Fig. 5 (a), under the non-modification condition and when the load is 1 kNm, the maximum amplitude of the transmission error is $0.24\mu\text{m}$ and the range of change is $0.026\mu\text{m}$. When the load is 30 kNm, the maximum amplitude of the transmission error is $6.39\mu\text{m}$ and the range of change

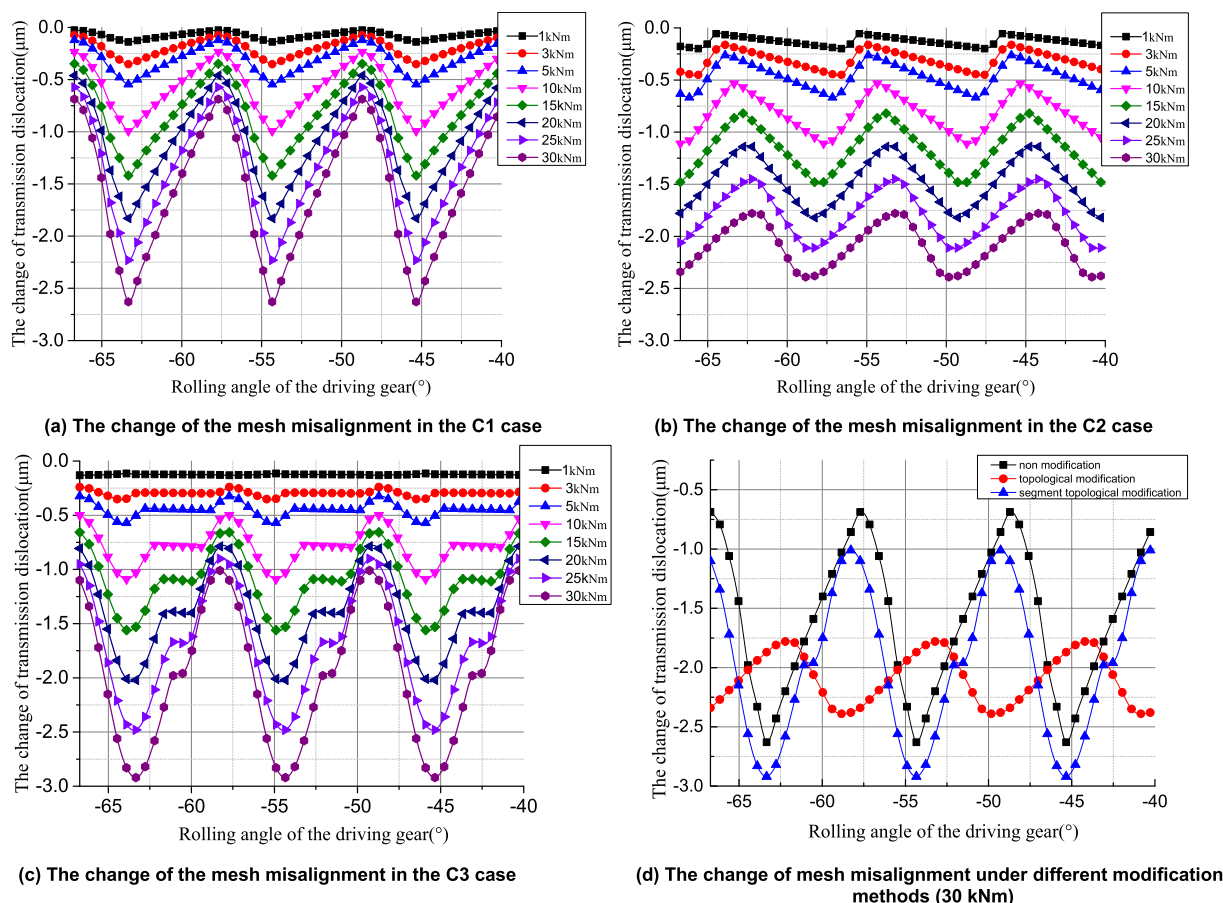


FIGURE 4. The change in the mesh misalignment under different loads.

is $0.64\mu\text{m}$. The transmission error under the maximum load increases by a factor of 24 compared to the minimum load. Therefore, the greater the load, the more obvious the deformation performance of each part of the transmission system, and the more profound the impact on transmission performance.

The distribution of the topological modification transmission error, as shown in Fig. 5 (b), when the load is 1 kNm, gives a maximum amplitude in the transmission error of $10.56\mu\text{m}$ and a varied range of $7.54\mu\text{m}$. When the load is 30 kNm, the maximum amplitude of transmission error is $38.06\mu\text{m}$ and the variation range is $10.56\mu\text{m}$, the transmission error under the maximum load increased by a factor of 3.7 compared to the minimum load. The topological modification gear pair and the non-modification gear pair under a particular load had a significantly reduced fluctuation range of the transmission error. This indicates that the modification affected determining the deformation of the gear teeth. On the other hand, when the load is 30 kNm, the amplitude of the transmission error of the topological modification gear pair increased by 50% compared to the amplitude of the maximum transmission error of the non-modification gear pair. This indicated that the modification could not ensure the transmission error under all loads

can be reduced and depends on the load of the gear teeth.

The distribution of the transmission error of the segment topological modification, as shown in Fig. 5 (c), when the load is 1kNm gave the maximum amplitude of the transmission error of $0.99\mu\text{m}$ and the variation range of the transmission error is $0.04\mu\text{m}$. When the load is 30 kNm, the maximum amplitude of the transmission error is $16.55\mu\text{m}$ and the variation range is $4.38\mu\text{m}$. Fig. 5 (d) shows the transmission error curve of the different modification methods. Under the load of 30kNm, the maximum amplitude of the non-modification transmission error is the lowest and the variation range of the amplitude is $0.64\mu\text{m}$, while the maximum amplitude of the topological modification transmission error is the highest with a varied range of the amplitude of $1.29\mu\text{m}$. The variation range of the amplitude of the segment topological modification transmission error is $4.38\mu\text{m}$. As a result, the modification can reduce transmission error under certain conditions. However, when under the different loads, the transmission error can be increased and reduced. The transmission performance can be improved under a particular load. In summary, under certain working conditions, the topological modification can reduce transmission error and improve transmission performance.

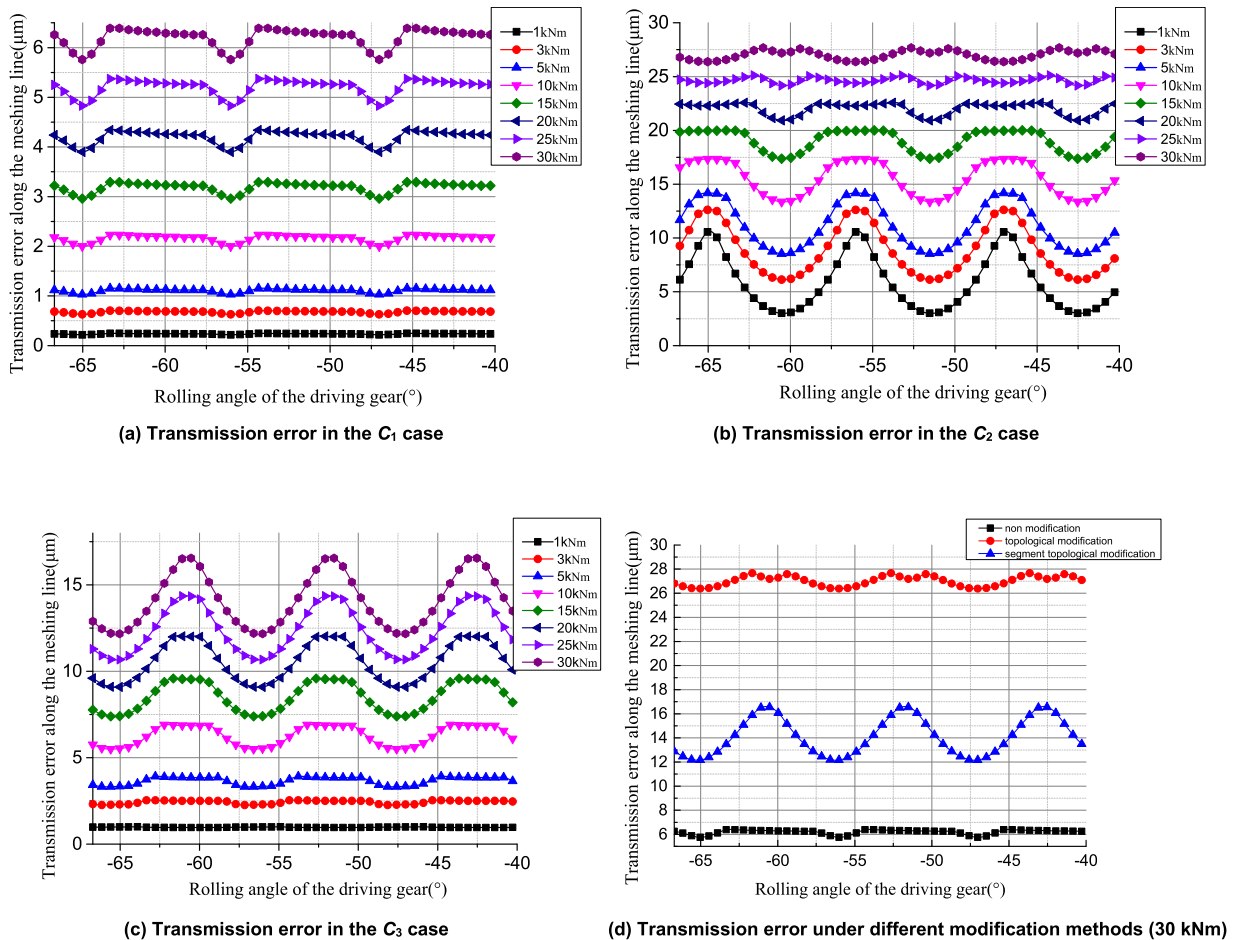


FIGURE 5. Simulation results of the transmission error under different load.

C. ANALYSIS OF TOOTH CONTACT STRESS

To test the loaded capacity of the helical gear pair with different modification parameters, need an analysis of the distribution characteristics of contact stress and bending stress along the direction of tooth profile and tooth alignment is needed. To provide a reference basis for the modification parameter optimization, only an analysis of the unit length contact load and maximum contact stress of tooth surface is needed when the load is 15 kNm. Fig. 6 shows the distribution characteristics of the unit length contact load and contact stress of the tooth surface. Under the non-modification conditions, the maximum value of the unit contact load of the tooth surface modification is 58.981 N/mm, the maximum contact stress of the tooth surface is 149.727 MPa, and the load distribution is non-uniform. Under the topological modification condition, the distributions of the tooth surface load and stress are shown in Fig. 6(c) and Fig. 6(d). From the figure, the maximum unit contact load is 274.434 N/mm and the maximum contact stress of the tooth surface is 288.406 MPa. The stress began to move towards the middle of the tooth profile and tooth alignment and the value of the unit length contact load and contact stress increased. Therefore,

through the tooth profile and tooth alignment modification, edge contact was avoided but the phenomenon of stress concentration occurred. Fig. 6(e) and Fig. 6(f) show the distribution situation of the tooth surface load and stress under the segment topological modification condition. In the figure, the maximum unit contact load is 150.064 N/mm and the maximum contact stress of the tooth surface is 210.658 MPa. The value of the contact stress was lower than that of the topological modification tooth surface, and the unit contact load also decreased.

From the above analysis, it can be seen that the tooth surface modification can improve the distribution of the tooth surface load, but can also produce the condition of stress concentration. To achieve the two goals of optimization, both of the reasonable modification parameters need to be selected. The topological modification caused the tooth surface contact to move towards the middle of the tooth profile along its direction and towards the middle of the tooth width along the direction of the tooth alignment. Choosing reasonable modification parameters can increase the stress distribution region on the tooth surface to avoid excessive stress concentration. The uniform distribution of the tooth

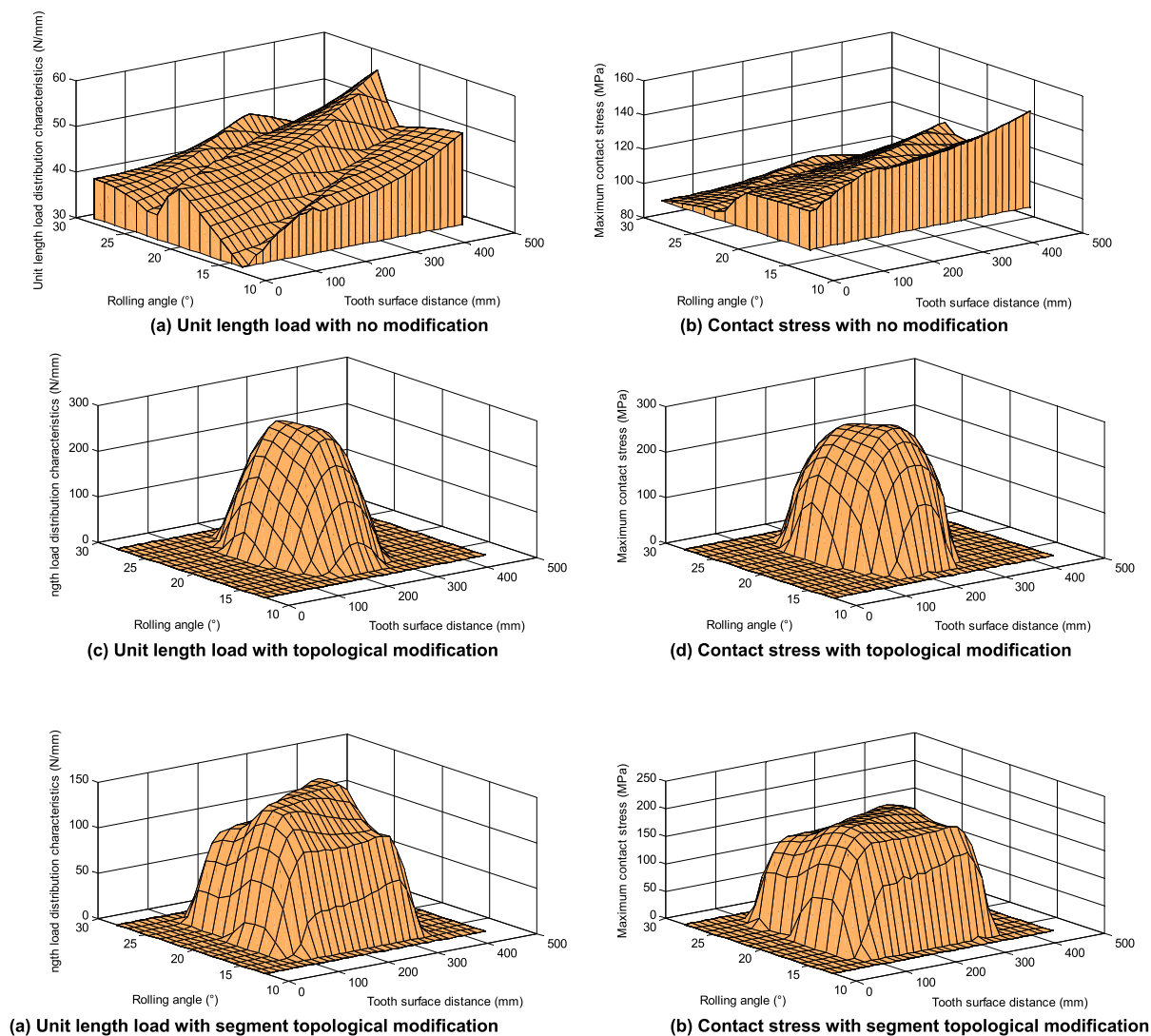


FIGURE 6. The bearing analysis of the segment topological modification tooth surface (15 kNm).

surface load delays the tooth surface failure rate and prolongs the life of the gear.

V. GEAR GRINDING AND MEASUREMENT

To verify the feasibility of the topological modification method on the CNC forming gear grinding machine, the driving wheel of topological modification is ground and measured as shown in Fig.7. After inputting the basic gear parameters, grinding wheel mounting parameters, and other process parameters, the system automatically generates the grinding processing program, which mainly calls the grinding wheel dressing subroutine and parameter subroutine. During the grinding process, the grinding wheel automatically trims the grinding profile after every five teeth are ground to reduce the machining errors caused by grinding wheel wear. After grinding one tooth, the grinding wheel is withdrawn from the grinding position and the rotary table is automatically indexed for grinding the next tooth. The grinding wheel speed is set

to 5000r/min, the workpiece indexing speed is 0.523rad/s, the diamond wheel speed is 3000r/min, the grinding wheel is trimmed once after grinding five teeth at an interval, the single stroke of the grinding wheel is 0.005mm along with the radial feed of the workpiece, the overtravel of the grinding wheel is set to 10mm, and the stroke along the Z-direction is set to 460mm.

The measurement results are shown in Fig .8. It can be seen that the normal error of the tooth face is similar to a parabolic shape, with a high middle and low sides. The normal error is the largest at both ends of the tooth surface. The dashed line in the topologically shaped tooth surface region ②, region ④, region ⑥ and region ⑧ clearly shows a parabolic-like shape in both the tooth profile and the tooth upward, and the maximum value of the measured value (tooth surface deviation and shaping amount) is divided into 56.4 μm at the top of the tooth. The measured value in the right tooth surface center region ① is close to $-2.7 \mu\text{m}$. It can be seen that the measured

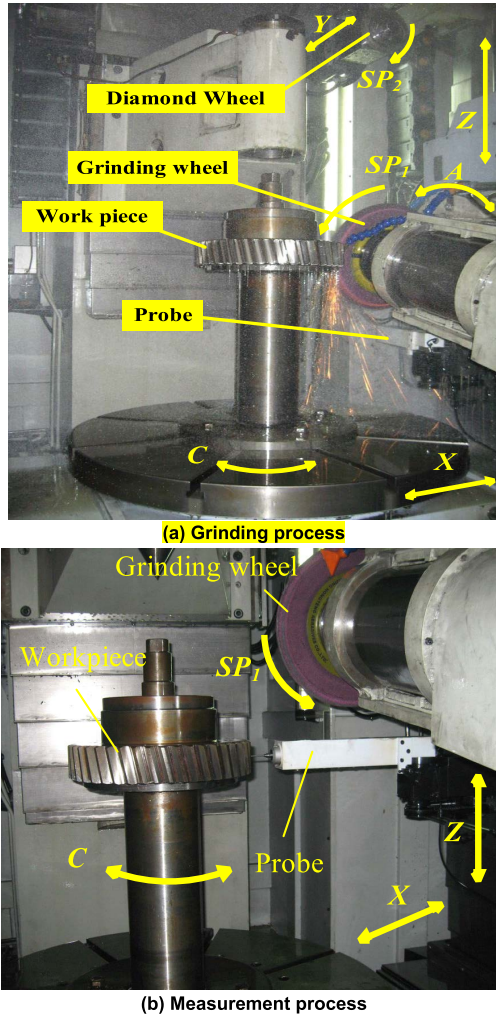


FIGURE 7. Gear machining and measurement experimental scenarios.

value is the smallest in the middle of the tooth profile and the middle of the tooth upward, and gradually increases toward the respective ends, showing a trend of higher in the middle and lower in the sides. The measured values in the middle of the tooth profile and the middle of the tooth orientation are the smallest and increase gradually towards the respective ends, showing a trend of high in the middle and low in the sides.

The loading test bench consists of a gearbox, magnetic powder brake, speed control motor, speed and torque sensor, sound level meter, and other devices, and the structure sketch as shown in Fig.9.

As it can be seen from Fig.9, during the test, the motor speed is changed by adjusting the output frequency of the inverter; the loading load is controlled by the current controller 6 connected with the magnetic powder brake 5, which changes the size of the DC input to the magnetic powder brake; the torque-speed sensor 4 measures the torque, speed and power of the output shaft; the acceleration sensor is displayed at five test points to measure the vibration signal

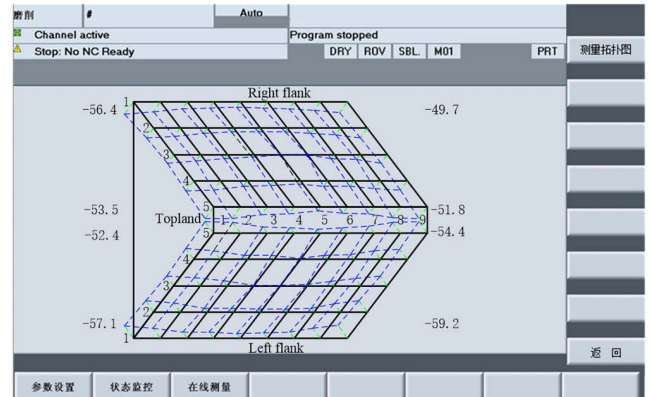


FIGURE 8. Topology measurement results.

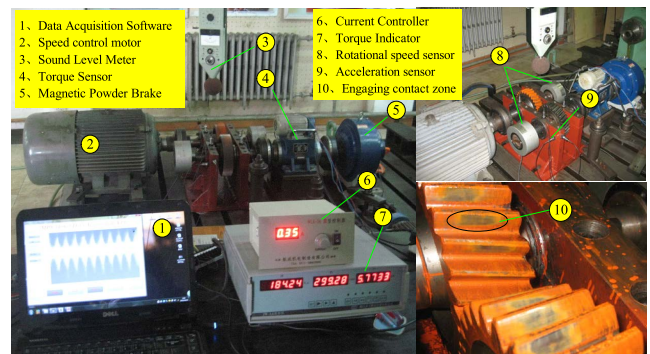


FIGURE 9. Gear loading test stand.

at different positions; two rotational speed sensors 8 measure the speed of the input shaft and the output shaft; the sound level meter 3 measures the field noise near the gear sub. The acceleration sensor is distributed in five test points to measure the vibration signal at different locations. The vibration signal and the pulse signal of the angled grating are collected in the field and input to the computer, and the data acquisition system 1 realizes data acquisition, processing, and analysis. The data displayed by the instrument is the signal at a certain moment of the test site, where the current controller 6 outputs 0.35A to the magnetic powder brake 5, and the torque measured by the torque-speed sensor 4 is 184.24Nm, the speed is 299.28r/min, and the power is about 5.77kW, and the speed of the inverter-controlled motor is 300r/min at this time.

Under the load, the gear teeth and all other parts of the transmission device that transmit the load will be more or less deformed, including bending deformation and torsional deformation. The actual tooth shape and the theoretical tooth shape will have errors, resulting in contact along one end of the gear teeth, uneven load distribution, and bias load phenomenon. The contact marks of the topology modified gear pair meshing drive, where the contact marks are in the middle of the tooth width and height, reduce the sensitivity to installation errors.

VI. CONCLUSION

This paper analyzed the principle of topological modifications of the tooth surface formation and divided the topological modification boundaries. Changing the modification boundary allows control over the size of each topological region of the tooth surface. Then, based on Hertz's theory, the tooth surface loaded contact model was established to find the mechanism of the tooth surface loaded deformation. The parameter optimization model of the tooth surface topological modification for the functional requirements was established to study the distribution regulation of the loaded transmission error and contact stress of the tooth surface under different loads. The functional requirements of the gear transmission were the goals of the optimization and the tooth surface topological modification parameters were the design variables. The tooth surface bearing contact analysis based on the loaded transmission error provided the maximum contact stress of the tooth surface to determine the reasonable modification parameters.

REFERENCES

- [1] T. Eritenel and R. G. Parker, "Nonlinear vibration of gears with tooth surface modifications," *J. Vib. Acoust.*, vol. 135, no. 5, Oct. 2013, Art. no. 051005.
- [2] V. V. Simon, "Multi-objective optimization of hypoid gears to improve operating characteristics," *Mechanism Mach. Theory*, vol. 146, Apr. 2020, Art. no. 103727.
- [3] T. Praveenkumar, B. Sabhrish, M. Saimurugan, and K. I. Ramachandran, "Pattern recognition based on-line vibration monitoring system for fault diagnosis of automobile gearbox," *Measurement*, vol. 114, pp. 233–242, Jan. 2018.
- [4] A. Parmar, P. Ramkumar, and K. Shankar, "Macro geometry multi-objective optimization of planetary gearbox considering scuffing constraint," *Mechanism Mach. Theory*, vol. 154, Dec. 2020, Art. no. 104045.
- [5] Z. Li and K. Mao, "The tooth profile modification in gear manufacture," *Appl. Mech. Mater.*, vols. 10–12, pp. 317–321, Dec. 2007.
- [6] J. Jiang and Z. Fang, "Design and analysis of modified cylindrical gears with a higher-order transmission error," *Mechanism Mach. Theory*, vol. 88, pp. 141–152, Jun. 2015.
- [7] F. L. Litvin, I. Gonzalez-Perez, A. Fuentes, K. Hayasaka, and K. Yukishima, "Topology of modified surfaces of involute helical gears with line contact developed for improvement of bearing contact, reduction of transmission errors, and stress analysis," *Math. Comput. Model.*, vol. 42, nos. 9–10, pp. 1063–1078, Nov. 2005.
- [8] F. L. Litvin, A. Fuentes, I. Gonzalez-Perez, L. Carvenali, K. Kawasaki, and R. F. Handschuh, "Modified involute helical gears: Computerized design, simulation of meshing and stress analysis," *Comput. Methods Appl. Mech. Eng.*, vol. 192, nos. 33–34, pp. 3619–3655, Aug. 2003.
- [9] V.-Q. Tran and Y.-R. Wu, "A novel method for closed-loop topology modification of helical gears using internal-meshing gear honing," *Mechanism Mach. Theory*, vol. 145, Mar. 2020, Art. no. 103691.
- [10] X. Tian, D. Li, X. Huang, H. Liu, J. Han, and L. Xia, "A topological flank modification method based on contact trace evaluated genetic algorithm in continuous generating grinding," *Mechanism Mach. Theory*, vol. 172, Jun. 2022, Art. no. 104820.
- [11] Q. Fan, R. S. DaFoe, and J. W. Swanger, "Higher-order tooth flank form error correction for face-milled spiral bevel and hypoid gears," *J. Mech. Design*, vol. 130, no. 7, pp. 0726011–0726017, Jul. 2008.
- [12] J. Argyris, M. De Donno, and F. L. Litvin, "Computer program in visual basic language for simulation of meshing and contact of gear drives and its application for design of worm gear drive," *Comput. Methods Appl. Mech. Eng.*, vol. 189, no. 2, pp. 595–612, Sep. 2000.
- [13] C.-K. Lee, "Manufacturing process for a cylindrical crown gear drive with a controllable fourth order polynomial function of transmission error," *J. Mater. Process. Technol.*, vol. 209, no. 1, pp. 3–13, Jan. 2009.
- [14] H. M. Noh, "Contribution analysis of interior noise and floor vibration in high-speed trains by operational transfer path analysis," *Adv. Mech. Eng.*, vol. 9, no. 8, pp. 6–9, Aug. 2017.
- [15] X. B. Chen, X. G. Zhi, and H. C. Song, "Optimization of transmission system design based on genetic algorithm," *Adv. Mech. Eng.*, vol. 8, no. 5, pp. 117–120, Mar. 2016.
- [16] Y. Zhang and Z. Fang, "Analysis of transmission errors under load of helical gears with modified tooth surfaces," *J. Mech. Des.*, vol. 119, no. 1, pp. 120–126, Mar. 1997.
- [17] T. F. Conry and A. Seireg, "A mathematical programming technique for the evaluation of load distribution and optimal modifications for gear systems," *J. Eng. Ind.*, vol. 95, no. 4, pp. 1115–1122, Nov. 1973.
- [18] T. F. Conry and A. Seireg, "A mathematical programming method for design of elastic bodies in contact," *J. Appl. Mech.*, vol. 38, no. 2, pp. 387–392, Jun. 1971.
- [19] V. Simon, "Optimal tooth modifications for spur and helical gears," *J. Mech., Transmiss., Autom. Des.*, vol. 111, no. 4, pp. 611–615, Dec. 1989.
- [20] A. Artoni, A. Bracci, M. Gabiccini, and M. Guiggiani, "Optimization of the loaded contact pattern in hypoid gears by automatic topography modification," *J. Mech. Des.*, vol. 131, no. 1, Jan. 2009, Art. no. 011008.
- [21] A. Kahraman, P. Bajpai, and N. E. Anderson, "Influence of tooth profile deviations on helical gear wear," *J. Mech. Des.*, vol. 127, no. 4, pp. 656–663, Jul. 2005.
- [22] P. Bajpai and A. Kahraman, "Influence of tooth profile modification on helical gear durability," *J. Mech. Des.*, vol. 124, no. 3, pp. 501–510, Sep. 2002.
- [23] M. S. Tavakoli and D. R. Houser, "Optimum profile modifications for the minimization of static transmission errors of spur gears," *J. Mech., Transmiss., Autom. Des.*, vol. 108, no. 1, pp. 86–94, Mar. 1986.
- [24] P. Wagaj and A. Kahraman, "Influence of tooth profile modification on helical gear durability," *J. Mech. Des.*, vol. 124, no. 3, pp. 501–510, Sep. 2002.
- [25] H. L. Wang, X. Z. Deng, and K. Xu, "Load tooth contact analysis (LTCA) of helical gears using topographic modification and installation errors," *J. Northwestern Polytech. Univ.*, vol. 32, no. 5, pp. 781–786, Oct. 2014.
- [26] S.-C. Kim, S.-G. Moon, J.-H. Sohn, Y.-J. Park, C.-H. Choi, and G.-H. Lee, "Macro geometry optimization of a helical gear pair for mass, efficiency, and transmission error," *Mechanism Mach. Theory*, vol. 144, Feb. 2020, Art. no. 103634.
- [27] E. B. Younes, C. Chagnenet, J. Bruyère, E. Rigaud, and J. Perret-Liaudet, "Multi-objective optimization of gear unit design to improve efficiency and transmission error," *Mechanism Mach. Theory*, vol. 167, Jan. 2022, Art. no. 104499.
- [28] M. Patil, P. Ramkumar, and K. Shankar, "Multi-objective optimization of two stage spur gearbox using NSGA-II," SAE INDIA, Chennai, India, SAE Tech. Paper 2017-28-1939, 2017, doi: [10.4271/2017-28-1939](https://doi.org/10.4271/2017-28-1939).
- [29] M. Patil, P. Ramkumar, and K. Shankar, "Multi-objective optimization of the two-stage helical gearbox with tribological constraints," *Mechanism Mach. Theory*, vol. 138, pp. 38–57, Aug. 2019.
- [30] J. A. Korta and D. Mundo, "Multi-objective micro-geometry optimization of gear tooth supported by response surface methodology," *Mechanism Mach. Theory*, vol. 109, pp. 278–295, Mar. 2017.
- [31] G. Chandrasekaran, V. S. Sreebalaji, R. Saravanan, and J. Maniraj, "Multiobjective optimisation of bevel gear pair design using NSGA-II," *Mater. Today, Proc.*, vol. 16, pp. 351–360, May 2019.
- [32] N. Li, W. Huang, W. Guo, G. Gao, and Z. Zhu, "Multiple enhanced sparse decomposition for gearbox compound fault diagnosis," *IEEE Trans. Instrum. Meas.*, vol. 69, no. 3, pp. 771–781, Mar. 2020.
- [33] Q. Li and S. Y. Liang, "Weak crack detection for gearbox using sparse denoising and decomposition method," *IEEE Sensors J.*, vol. 19, no. 6, pp. 22243–22253, Mar. 2019.
- [34] H. Wang, J. Li, Y. Gao, and J. Yang, "Closed-loop feedback flank errors correction of topographic modification of helical gears based on form grinding," *Math. Problems Eng.*, vol. 2015, Jan. 2015, Art. no. 635156.
- [35] H. Wang, X. Deng, J. Han, J. Li, and J. Yang, "Mathematical model of helical gear topography measurements and tooth flank errors separation," *Math. Problems Eng.*, vol. 2015, Jan. 2015, Art. no. 176237.
- [36] F. Yang, Z. Zhang, and X. Long, "Finite element analysis of helical gear's contact stress based on precise model," *Mech. Sci. Tech.*, vol. 22, no. 2, pp. 206–208, Mar. 2003.



YING WANG received the M.S. degree in electrical engineering major from the Nanjing University of Science and Technology, Nanjing, Jiangsu, China, in 2016. She is currently a Senior Experimenter with the Zhengzhou University of Industrial Technology. Her research interest includes mechatronics technology.



XING LIU received the bachelor's degree in mechanical engineering from the Yenching Institute of Technology, in 2015. He is currently a Graduate Student at the Henan University of Science and Technology, Luoyang, China. His research interests include mechanical engineering and the digital design of gears.



HUILIANG WANG received the Ph.D. degree in mechanical design and theory from Northwestern Polytechnical University, Xi'an, Shanxi, China, in 2016. He is currently an Associate Professor with the School of Mechatronics Engineering, Henan University of Science and Technology, Luoyang, China. His research interests include industrial engineering, mechanical product digital design and performance analysis, and computer integrated manufacturing systems.



YILIN FU received the bachelor's degree in industrial engineering from the Henan Institute of Technology, in 2015. He is currently a Graduate Student at the Henan University of Science and Technology, Luoyang, China. His research interests include industrial engineering and reliability.

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