

Article

Research on Calculation and Optimization Methods for Tooth Flash Temperature and Meshing Power Loss of the Gear System in Drum Shearer

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Abstract: The operating conditions of the drum shearer are very complex, and its ranging arm gear system often suffers from gear scuffing and wear. Gear scuffing is caused by the adhesive wear, which is due to the instantaneous friction and flash temperature of the tooth surface, and the gear meshing power loss is also caused by tooth surface friction. In order to resist tooth scuffing and improve meshing efficiency of the transmission system, an improved semi-analytical tooth surface flash temperature calculation method was used. The tooth flash temperature status under various working conditions were analyzed in detail. Based on the mechanical model of the shearer drum picks, the load condition of the drum was analyzed. Under these load and boundary conditions, the misalignments of each gear pair in the ranging arm were calculated. The tooth surface load distribution was calculated under the gear misalignments, and then the theoretical tooth surface flash temperature and meshing power loss were determined. Next, the tooth micro-geometry was modified to reduce flash temperature and meshing power loss. The flash temperature distribution pattern of the optimized tooth surface was studied under various working conditions, and the meshing power loss was also obtained. Finally, experiments were conducted to verify the effects of the optimized tooth surface on the friction temperature rise and the effectiveness of the modification method. Tooth surface optimization aimed at reducing tooth surface flash temperature can also effectively reduce meshing power loss, which has a significant effect on gear anti-scuffing and energy saving.

Keywords: gear system; tooth flash temperature; meshing power loss; tooth modification; drum shearer; gear scuffing



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1. Introduction

Drum shearers have the advantages of high mining efficiency and a wide application range and are widely used in coal mines. Due to the diverse structures of coal seams, shearers need to face various complex working conditions, and the cutting resistance changes frequently, which has a very obvious impact on the gear transmission system [1,2]. Under such extremely poor working conditions, tooth surface wear, scuffing, fatigue cracks, and tooth fracture are prone to occur [3], transmission system failures are very common in shearer failures [4,5]. It is difficult to maintain the gears under coal mine conditions, which seriously affects work efficiency.

There is rich research in the design and dynamic performance of drum shearer transmission systems, Boiko et al. [6] discussed a method which aimed at calculating optimum transmission ratio of the reduction gears for the optimization of the revolution rate of shearers, considering the effects of the following factors on shearer operation: resistance of coal to cutting, shearer dimensions, properties of the drive system, and reduction gear system. Yang et al. [7] designed a new kind of high reliability short-range cutting transmission system that can realize buffering and damping with the load mutations, and the roller speed can be controlled with the variation of coal seam. Zhang et al. [8] used Matlab software to

conduct a reliable and robust optimization design of the shearer ranging arm transmission system, proving that this design is feasible, practical, and effective. Ge et al. [9] proposed active torque control for gear dynamic load suppression in a drum shearer transmission system. Chen et al. [10] studied the change in tooth backlash caused by the tooth bulk temperature, which in turn affects the dynamic performance of gear meshing in the shearer. Yuan et al. [11] revealed that tooth surface contact temperature and dynamic wear will cause changes in the contact position on the line of action, thereby changing the dynamic characteristics of the gear system. Liu et al. [12] developed three drums with different helical angles to investigate improved loading performance and studied the loading performance of the shearer drum. Jia and Qin [13] found that the three-directional forces exerted by the coal seam on the drum are the main factor causing large deformation of the ranging arm, which will adversely affect the dynamic meshing performance. Gear micro-geometry optimization can improve some problems that occur in gear meshing, thereby avoiding failure of the gear system and improving the reliability of the machine. Bahk et al. [14] proposed a tooth profile model for modified gears and investigated the impact of tooth profile modification on spur planetary gear vibration. Huangfu et al. [15] investigated optimal parameters of tooth modifications in order to reduce the wear of tooth surface. Karpat et al. [16] studied that tip relief modification has a significant effect on reducing wear on the tooth surface. Chen and Shao [17] studied the solution of gear mesh stiffness considering tooth surface modification, error and misalignment, and proposed a tooth surface optimization method based on this stiffness model.

Higher flash temperature of tooth surface will also have an adverse effect on the meshing characteristics of the transmission system. Gou et al. [18] established an equation for tooth surface flash temperature and contact temperature based on Blok flash temperature theory and calculated the tooth surface deformation caused by contact temperature. Pan et al. [19] found that the tooth surface contact temperature will change the time-varying mesh stiffness and lead to changes in the periodic motion characteristics of the gear-rotor system. Liu et al. [20] also studied the time-varying meshing stiffness of the planetary gear train due to thermal deformation of the gear teeth caused by the thermal equilibrium temperature of the system, which increased system vibration. Li et al. [21] found that the spur gear tooth will become a non-involute shape due to the increase in tooth surface contact temperature. Mao et al. [22] proved that tooth modification can improve the temperature distribution and reduce the temperature during meshing. Yu et al. [23] based on the thermal shock model and Hertzian contact theory, derived the calculation method of the instantaneous contact temperature and bulk temperature of the gear tooth, analyzed the influence of the profile shift coefficient and tooth surface modification on the contact temperature, and proved that reasonable selection of profile shift coefficient and modification parameters can improve the ability of anti-scuffing. Wei et al. [24] studied the mechanism of tooth surface pitting caused by factors such as surface morphology, rolling and sliding contact stress, magnitude and depth of subsurface shear stress of spur gears under EHL conditions.

Gear tooth scuffing is one of the failure forms of the gear transmission system. It is caused by the instantaneous friction flash temperature of the tooth surface being too high, causing the contact temperature to exceed the scuffing temperature of the lubrication oil, and causing the oil film to break and the rough peaks to be in direct contact. The gear meshing power loss is also caused by tooth surface friction. Caused by this, in order to avoid scuffing failure and obtain higher meshing efficiency when designing the gear system, it is necessary to accurately calculate the tooth surface flash temperature distribution. At present, the common calculation method of flash temperature is the standard equation method, such as ISO 6336-20:2022 [25] and AGMA 925-A03 [26], they are both evolved based on Blok flash temperature theory [27–29], and the load in equations is the average unit load regardless of the detail load distribution in the tooth lead direction. Although the tooth face load factor $K_{\beta\beta}$ is introduced, it is still impossible to determine the coordinate position where the maximum flash temperature occurs. Therefore, it is impossible to determine

the specific parameters for using tooth modification to reduce the flash temperature. The flash temperature is closely related to the working conditions. With different loads and boundary conditions, the maximum load position on the tooth surface will be different, which will lead to different flash temperature.

In short, there are many research results on the tooth flash temperature and meshing power loss of the drum shearer gear system, but these studies usually calculate the flash temperature and efficiency according to international standard with average unit load, the influence of three-directional forces from coal seam on tooth surface load distribution is not considered, which makes it impossible to accurately calculate the flash temperature distribution and detailed meshing power loss of the gearset. Therefore, in order to study the method of reducing gears' flash temperature and meshing power loss under operation condition, this article proposes a tooth surface optimization method using micro-geometry modification and an improved semi-analytical tooth surface flash temperature calculation method. Finally, experiments were used to verify the effects of the optimized tooth surface on the friction temperature rise, and effectiveness of modification method.

2. Model, Loads, and Boundary Conditions of Gear System in Drum Shearer

2.1. Drum Shearer Transmission System Model

A drum shearer is mainly composed of a shearer body, left and right ranging arms, and a cutting drum. The structural model is shown in Figure 1. When the shearer is working, the motor provides driving power, which decelerates and increases torque through the ranging arm gear system, and finally drives the cutting drum to rotate to cut the coal seam. The drum is the working execution mechanism to break coal seam, which includes picks, end disks, and spiral blades. Most of the shearer's power is consumed in the cutting drum.

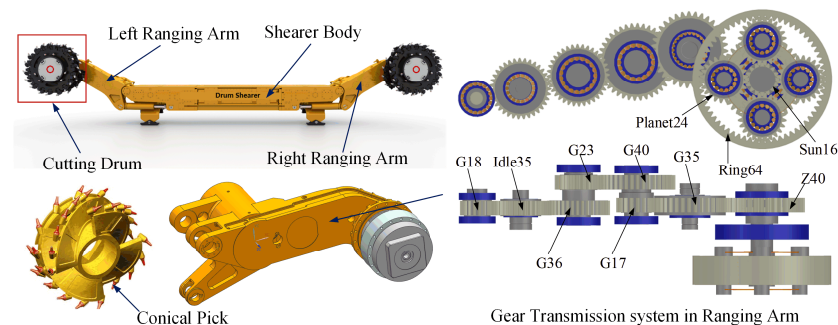


Figure 1. A drum shearer and its components.

The ranging arm transmission system includes parallel gear train and planetary gear train. The parallel gear train consists of 8 spur gears to form a three-stage fixed-axis gear train, of which two idler gears are used to increase the length of the arm. The planetary gear train is a single-stage NGW, which includes a sun gear, an internal ring gear, and four planetary gears. The parameters and codes of gears are shown in Table 1. It can be easily calculated that the total transmission ratio of the system is 40.92.

Table 1. Basic parameters of gearsets.

Gear ID	G18	Idle35	G36	G23	G40	G17	G35	Z40	S16	P24	R64
Tooth number/ z	18	35	36	23	40	17	35	40	16	24	64
Module/ m_n		8 mm		9 mm			10 mm			9 mm	
Pressure angle/ α_n						20°					
Face width/ b				70 mm					140	135	140
Steel density						7850 kg/m ³					
Young's modulus/ E						206 GPa					
Poisson's ratio						0.3					
Specific heat						450 J/(kg·K)					
Thermal conductivity						43 W/(m·K)					

2.2. Gear System Working Loads and Boundary Conditions

The load condition of the ranging arm gear system is determined by the load state of the drum during cutting. The load state of the drum is complex, which mainly includes the resistance force and torque generated by the drum cutting the coal seam. In order to obtain the accurate load conditions of the gears, it is necessary to conduct a force analysis on the drum. When the drum cuts the coal seam, the factors that affect the force on each pick and the resultant force on the drum can be mainly divided into three parts. The first part is the design parameters of the drum, which mainly include the geometric parameters of the drum, the arrangement of the picks, the working width of the picks, the influence coefficient of the picks angle, etc.; the second part is the coal seam parameters, which mainly includes the impedance of the coal seam and the brittleness coefficient of the coal rock, etc.; the third part is the speed parameters, including the walking speed of the shearer and the rotation speed of the drum.

When the drum cuts the coal seam, a single pick will be affected by cutting resistance, feeding resistance and the lateral force exerted on it by the coal seam, as shown in Figure 2. According to the theory of cutting and breaking coal [30], during the cutting process, the lateral force, cutting resistance, and feeding resistance on a single pick can be expressed as follows:

$$X_0 = Z_0 \left(\frac{C_1}{C_2 + h} + C_3 \right) \frac{h}{t} \quad (1)$$

$$Y_0 = k_q \cdot Z_0 \quad (2)$$

$$Z_0 = 10A_z \cdot h \cdot t \cdot k_y \cdot k_m \cdot k_a \cdot k_f \cdot k_p \frac{0.3 + 0.35b_p}{(b_p + Bh^{0.5})k_\phi} \cdot \frac{1}{\cos \beta} \quad (3)$$

The forces on the drum at a certain position are shown in Figure 3. The forces on all the picks can be converted to three directions: A, B, and C. Direction A is the traction resistance of the drum, Direction B is the lateral force of the drum, and Direction C is the vertical resistance of the drum. It can be concluded that the expression of loads on the drum can be defined as follows:

$$F_A = -\sum_{i=1}^{n_j} (Y_i \sin \varphi_i \cos \beta_i + Z_i \cos \varphi_i) \quad (4)$$

$$F_B = \sum_{i=1}^{n_j} X_i \quad (5)$$

$$F_C = \sum_{i=1}^{n_j} (Z_i \sin \varphi_i - Y_i \cos \varphi_i \cos \beta_i) \quad (6)$$

$$M_R = \frac{D_C}{2} \sum_{i=1}^{n_j} Z_i \quad (7)$$

In these equations, M_R denotes the cutting resistance torque of the drum, in Nm; φ_i is the position angle of the i th pick, in rad; X_i denotes the lateral force of the i th pick at position φ_i , in newtons; Y_i denotes the feeding resistance of the i -th pick at the position φ_i ; Z_i denotes the cutting resistance of the i th pick at the φ_i position; D_C is the cutting diameter of the drum, in meters; n_j is the number of picks that instantaneously participate in cutting. Other various parameters and influence coefficients are not the focus of this article, and their detailed meanings can be found in Ref. [30].

Since the parameters of various coal seams and the cutting thickness are different, the load of the shearer changes randomly when working. Putting the parameters of this type of drum shearer when working in a typical coal seam into Equations (1)–(7), it can be calculated that $F_A = -37.393$ kN, $F_B = -13.532$ kN, $F_C = 42.28$ kN, $M_R = 81.85$ kNm.

According to the gear parameters, a ranging arm model is established in the transmission system software Romax R20, and the traction resistance, lateral force, vertical resistance and cutting resistance torque of the drum under different working conditions are all applied to the square head of drum. The boundary conditions of the ranging arm include constraints on two hinges. The upper hinge hole constrains axial displacement and three degrees of freedom for rotation around its own x-axis and y-axis. The lower hinge

hole constrains all degrees of freedom except rotation around its own axis, as shown in Figure 4.

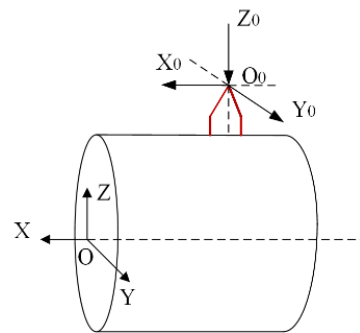


Figure 2. Force diagram for a cutting tooth (red line) of a cutter drum.

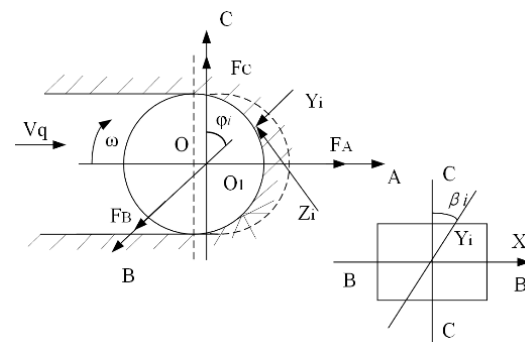


Figure 3. Force diagram of a cutter drum.

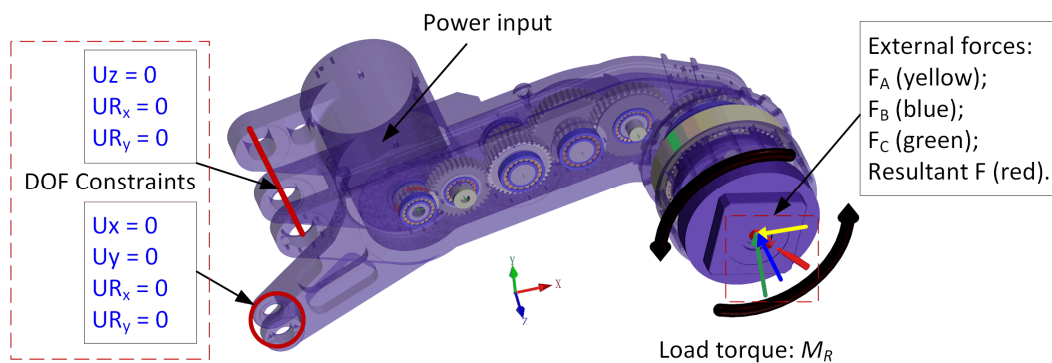


Figure 4. The external load (forces and torque) and boundary condition on the ranging arm.

2.3. Gear Pairs Misalignments in Ranging Arm

Since the mechanical structure is elastic, all parts of the ranging arm will elastically deform after being loaded. At this time, the gears will be misaligned meshing due to the deformation of the shafts and housing. The gear misalignment refers to the component of the axis non-parallelism of a gear pair in the plane of action, and a misaligned gear pair cannot meet the ideal meshing state with full tooth surface contact, but will suffer from unbalanced force and stress concentration.

Through static analysis, the misalignment results of all gear pairs in the ranging arm under the maximum load condition (input torque 2000 Nm) can be obtained, as shown in Figure 5, where the red curve represents the three-direction forces exerted by the coal seam on the drum, and the blue curve does not include the effect of external forces. It can be seen that the impact of external forces on the gear misalignment is relatively significant, with some misalignment increasing and some decreasing. The maximum absolute change in the misalignment is 12.1 μm , and the maximum relative change is about 50%.

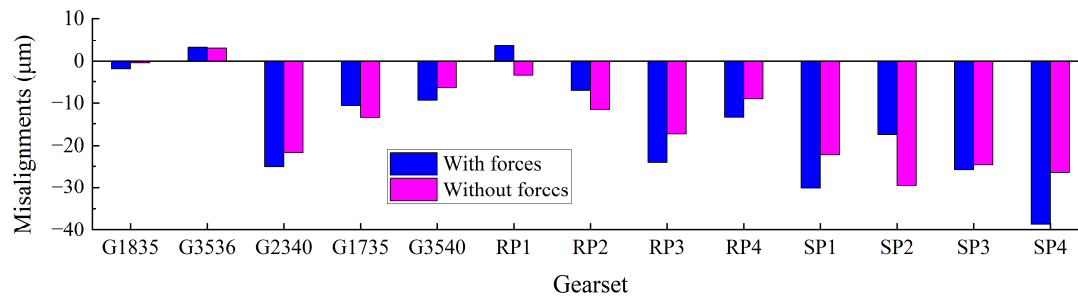


Figure 5. The misalignments of all gear pairs.

Different gear misalignments have completely different tooth surface load distribution. On this basis, the tooth surface flash temperature and friction power loss will be different. Therefore, accurate load conditions are the prerequisite for calculating tooth surface flash temperature distribution and friction power loss.

3. Analysis and Optimization of Friction Flash Temperature of Gears Tooth Surface

3.1. Semi-Analytical Method for Solving Tooth Surface Friction Flash Temperature

Due to the existence of friction between tooth surfaces, flash temperature will be generated during gear meshing. When the flash temperature is higher than the allowable temperature, tooth surface scuffing will occur. Scuffing is different from fatigue in that it occurs instantly. There are usually two methods for calculating tooth surface flash temperature: the Blok flash temperature method and the integrated temperature method. The integrated temperature is the average of the flash temperatures and adds the influence factor to the load sharing factor, the resulting value is close to the maximum contact temperature, so the scuffing risk assessment is approximately the same as the flash temperature method. Situations in which there are local temperature peaks, such as gear pairs; low contact ratios; tooth tip and root edge contact; or contact at other geometrically sensitive locations, the integrated temperature method has low sensitivity and cannot reflect the tooth surface flash temperature distribution. In order to study the flash temperature distribution of the tooth surface when the gear pair undergoes unbalance load after elastic deformation, it is more appropriate to use the improved flash temperature method for analysis.

The flash temperature at the instant meshing position of the tooth surface is described as follows [25]:

$$\Theta_{fl} = 1.11 \cdot \frac{\mu_m \cdot X_\Gamma \cdot X_J \cdot w_n}{\sqrt{2} \cdot b_H} \cdot \frac{abs(v_{g1} - v_{g2})}{B_{M1} \cdot \sqrt{v_{g1}} + B_{M2} \cdot \sqrt{v_{g2}}} \quad (8)$$

For involute cylindrical gears, the tooth surface tangential velocity components v_{g1} and v_{g2} are related to the local radius of curvature of the tooth surface and the rotation speed, so the Equation (8) can also be equivalently replaced by Equation (9).

$$\Theta_{fl} = 2.52 \cdot \mu_m \cdot \frac{X_M}{50} \cdot X_J \cdot \sqrt[4]{(X_\Gamma \cdot w_n)^3} \cdot \sqrt{n_1/60} \cdot \frac{abs(\sqrt{\rho_{y1}} - \sqrt{\rho_{y2}/u})}{\sqrt[4]{\rho_{rely}}} \quad (9)$$

The friction coefficient for calculating the tooth surface flash temperature, which is obtained through a large number of experimental fittings based on the EHL theory, is defined as follows in Equation (10).

$$\mu_m = 0.06 \cdot \left(\frac{w_t}{v_s \cdot \rho_y} \right)^{0.2} \cdot X_L \cdot X_R \quad (10)$$

In the above equations, w_n and w_t represent the normal and transverse unit loads, respectively. Since the transverse unit load calculated using ISO 6336-20:2022 is the ratio

of the tangential force to the tooth width, it is an average value and cannot show the load distribution on instantaneous contact line. Due to the existence of meshing misalignment and tooth surface modification, the load on the contact line must be unevenly distributed. In order to investigate the real load distribution on the tooth surface, this paper uses gear loaded tooth contact analysis (LTCA) based on the finite element method to calculate the w_n distribution. If a meshing cycle (an angular pitch, $2\pi/z$) is equally divided into N instantaneous positions, and M grid nodes along the tooth width direction, then after finite element calculation converges, the normal load on the current contact line at the i th instant meshing position can be obtained as in Equation (11), where the superscript “ n ” represents the normal direction.

$$w_i^n = (w_{i1}^n, w_{i2}^n, w_{i3}^n, \dots, w_{iM}^n) \quad (11)$$

After solving all N meshing positions, the normal load matrix can be assembled in the form of Equation (12).

$$w^n = \begin{bmatrix} w_{11}^n & w_{12}^n & w_{13}^n & \dots & w_{1M}^n \\ w_{21}^n & w_{22}^n & w_{23}^n & \dots & w_{2M}^n \\ w_{31}^n & w_{32}^n & w_{33}^n & \dots & w_{3M}^n \\ \dots & \dots & \dots & w_{ij}^n & \dots \\ w_{N1}^n & w_{N2}^n & w_{N3}^n & \dots & w_{NM}^n \end{bmatrix} \quad (12)$$

The detailed calculation process can be found in Ref. [31], which will not be repeated here. Then Equation (13) is used to calculate w_t . The local relative radius of curvature of the tooth surface is calculated by Equations (14)–(17).

$$w_t = w_n \cdot \cos \alpha_{wn} \cdot \cos \beta_w \quad (13)$$

$$\rho_y = \frac{1}{1/\rho_{y1} + 1/\rho_{y2}} = \frac{\rho_{y1} \cdot \rho_{y2}}{\rho_{y1} + \rho_{y2}} \quad (14)$$

$$\rho_{y1} = \frac{1 + \Gamma_y}{1 + u} \cdot a \cdot \sin \alpha_{wt} \quad (15)$$

$$\rho_{y2} = \frac{u - \Gamma_y}{1 + u} \cdot a \cdot \sin \alpha_{wt} \quad (16)$$

$$v_s(i) = \text{abs}(v_{g1} - v_{g2}) = \text{abs}(\omega_1 \cdot \rho_{1y} - \omega_2 \cdot \rho_{2y}) / 1000 \quad (17)$$

X_M represents the thermoelastic coefficient, X_J denotes the meshing-in coefficient, X_Γ is the load sharing coefficient, Γ_y is the contact position parameter along the line of action, X_L denotes the lubricating factor, and X_R is the roughness factor. ω_1 and ω_2 are the rotational speed of the driving pinion and the driven wheel, respectively, in rad. The specific calculation of these factors can be found in Ref. [25].

3.2. Micro-Geometry Modification Method to Reduce Tooth Surface Flash Temperature

Since the mechanical system will produce elastic deformation after being loaded, the gear will not mesh at the theoretical contact position, but will cause eccentric load and stress concentration, as shown in Figure 6. This will lead to failure modes such as early pitting, scuffing, and tooth fracture, among others, and can also cause NVH problems such as meshing impact and gear whine in the transmission system [31].

The emergence of gear modification technology is to solve the above problems. Gear modification refers to the small micron-level modification of the tooth surface. The modified tooth surface deviates from the theoretical tooth surface, becoming no longer the theoretical involute helical surface. For open helical surfaces, gear modification includes profile modification, lead modification, triangular modification, twist modification, and topological modification. Tooth surface modification can improve the transmission accuracy and stability of gears and eliminate phenomena such as eccentric load and stress concentration, thereby increasing the service life and improving the load capacity of gears.

For continuous operation of high-power heavy machinery such as drum shearers, gear modification can also reduce the contact temperature of the meshing tooth, which can effectively improve the anti-scuffing ability of the gear and can also reduce meshing power loss, improving the efficiency of the machine.

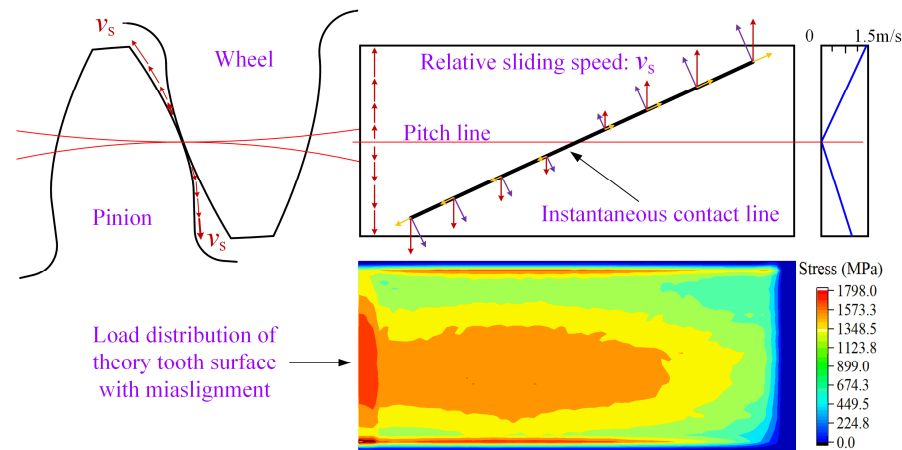


Figure 6. Relative sliding speed and load distribution on the tooth surface of involute cylindrical gears.

As shown in Figure 6, a pair of meshing tooth surfaces undergoes both rolling and sliding during the meshing process. The pitch position is pure rolling, and the other positions have relative sliding speeds. The farther away from the pitch, the greater the relative sliding speed v_s . The maximum value area of v_s is close to the tooth tip or root, so the friction power loss in this area is larger. The theory tooth surface often causes edge contact due to elastic deformation near the tooth tip, root, or both ends of the tooth face. If the contact load in these areas is greater, the friction loss will be greater, which will produce a higher flash temperature and cause scuffing failure. The principle of tooth modification with the purpose of reducing flash temperature and improving efficiency is to use a variety of modification forms to remove some trace amounts of material from the tooth surface, thereby avoiding heavy load contact in the area around the tooth tip and root during the meshing process. This will reduce the friction power loss in areas with the higher relative sliding speeds, reduce the tooth surface flash temperature, and avoid tooth surface scuffing failure.

Based on the meshing misalignment results analyzed in Section 2.3, comprehensive modifications to the tooth surface of each gear are carried out, including tooth tip and root relief, tooth profile crowing and slop, tooth lead crowing, slop, and end relief. Through these means, the modified gears can meet the requirement that the contact pattern is located in the middle of the tooth surface under small and medium loads and the tooth surface full contact under full load. The tooth surface flash temperature is positively related to the local load and relative sliding speed v_s , and v_s is related to the gear macro geometry, so it cannot be changed through micro-modification. However, the contact load at the position where the relative sliding speed is higher can be minimized through micro-modification, so that the flash temperature of the tooth surface can be reduced. Taking gear G17 as an example, the micro-geometry parameters of comprehensive modification are shown in Table 2, and the optimization topological surface of the tooth is shown in Figure 7.

First, apply the LTCA method to calculate the tooth surface load distribution and stress distribution after tooth modification. Second, iteratively calculate and evaluate whether the modification parameters are optimal according to the load distribution state. Third, take the unit load of tooth surface at current contact position into Equations (9)–(17), and the tooth surface flash temperature at the current meshing position can be obtained. Fourth, after calculating N meshing positions in one meshing cycle, store the maximum flash temperature value of each node of each meshing position. Finally, the maximum flash

temperature of each node is integrated into a developed plane of tooth surface, and the flash temperature distribution of the entire tooth surface can be obtained.

Table 2. Micro-geometry parameters of gearset-G1735 modification.

	Item	Symbol	Pinion (G17)	Wheel (G35)
Tooth profile	Barrelling/ μm	C_α	10	0
	Slope/ μm	$F_{H\alpha}$	0	0
	Tip relief start/mm	u_T	43	70
	Tip parabolic relief/ μm	F_{Ko}	28	32
Tooth lead	Lead crown/ μm	C_β	9	0
	Lead slope/ μm	$F_{H\beta}$	−11	0
	Bottom end relief length/mm	L_B	10	10
	Bottom parabolic end relief/ μm	δ_{BP}	−22	0
	Top end relief length/mm	L_T	10	10
	Top parabolic end relief/ μm	δ_{TP}	−20	0

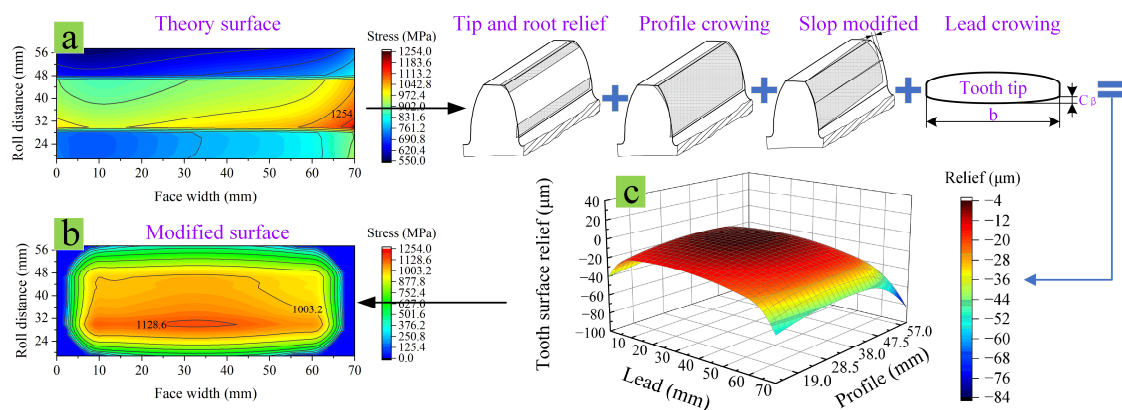


Figure 7. The scheme and evaluation of tooth surface micro-geometry modification. (a) Theory tooth surface contact stress. (b) Modified tooth surface contact stress. (c) Tooth surface modification.

The changes of tooth surface flash temperature distribution with speed and torque before and after modification of G1735 gear pair are shown in Figure 8. It can be seen that the flash temperature distribution of the theory tooth surface is biased, and the maximum flash temperature appears at the tip and root of tooth right end, the maximum value at full load is 110.6 °C. After tooth modification, the maximum flash temperature is significantly reduced, and is mainly distributed in the middle of the double-tooth contact area symmetrical with the pitch line. The maximum flash temperature is 73.5 °C. When the speed is constant, as the torque increases, the maximum contact flash temperature position gradually transitions from the left to the middle of the tooth face, and the distribution area also expands to the entire tooth surface except the surface edge. When the torque is constant, as the rotational speed increases, the flash temperature distribution area almost does not expand, while the maximum flash temperature increases from 39 °C to 73.5 °C. It can be seen that because the relative sliding speed of the tooth surfaces near the pitch line is relatively low, the flash temperature on both sides near the pitch line is generally very low—only a few degrees Celsius.

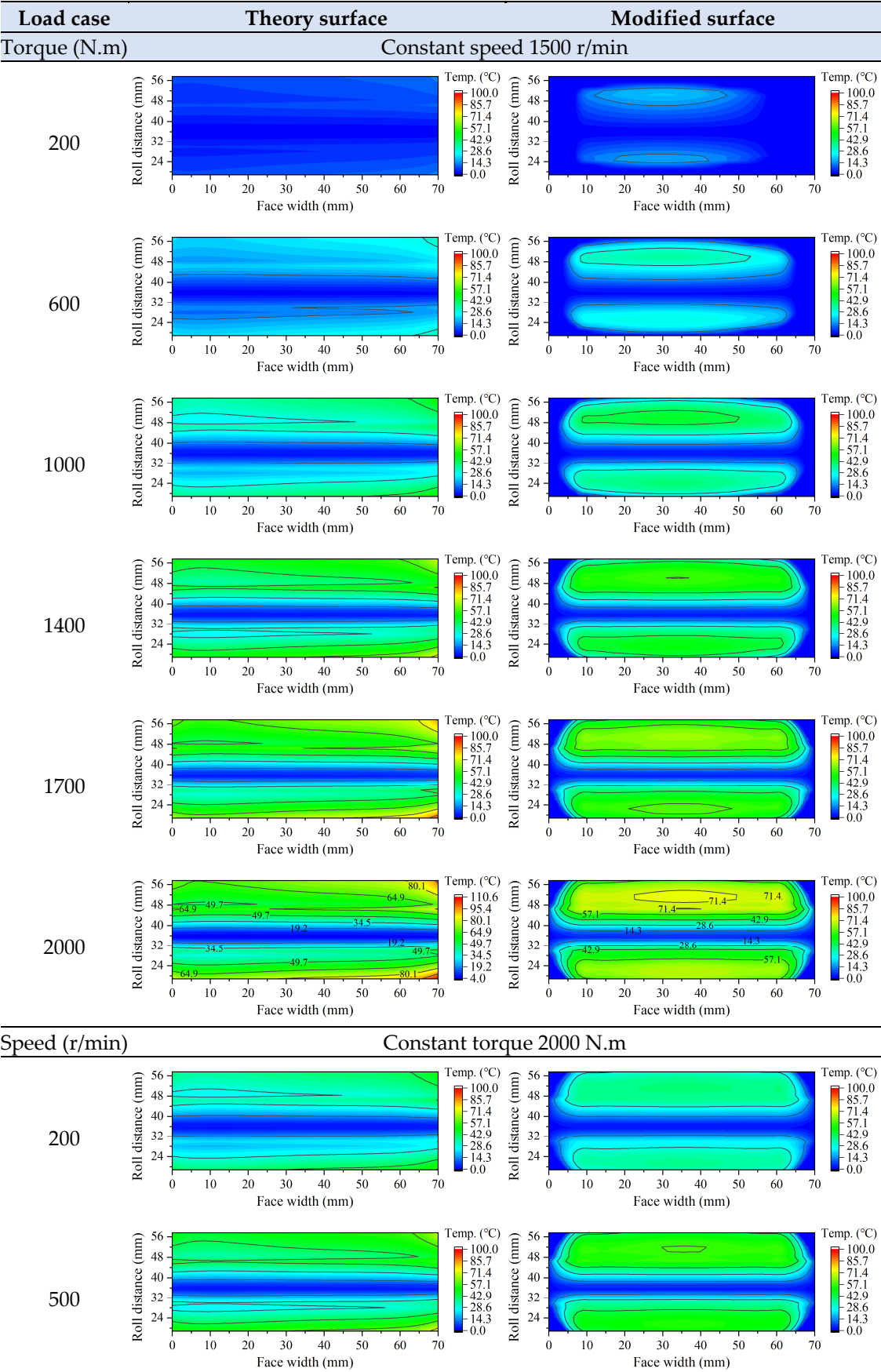


Figure 8. Cont.

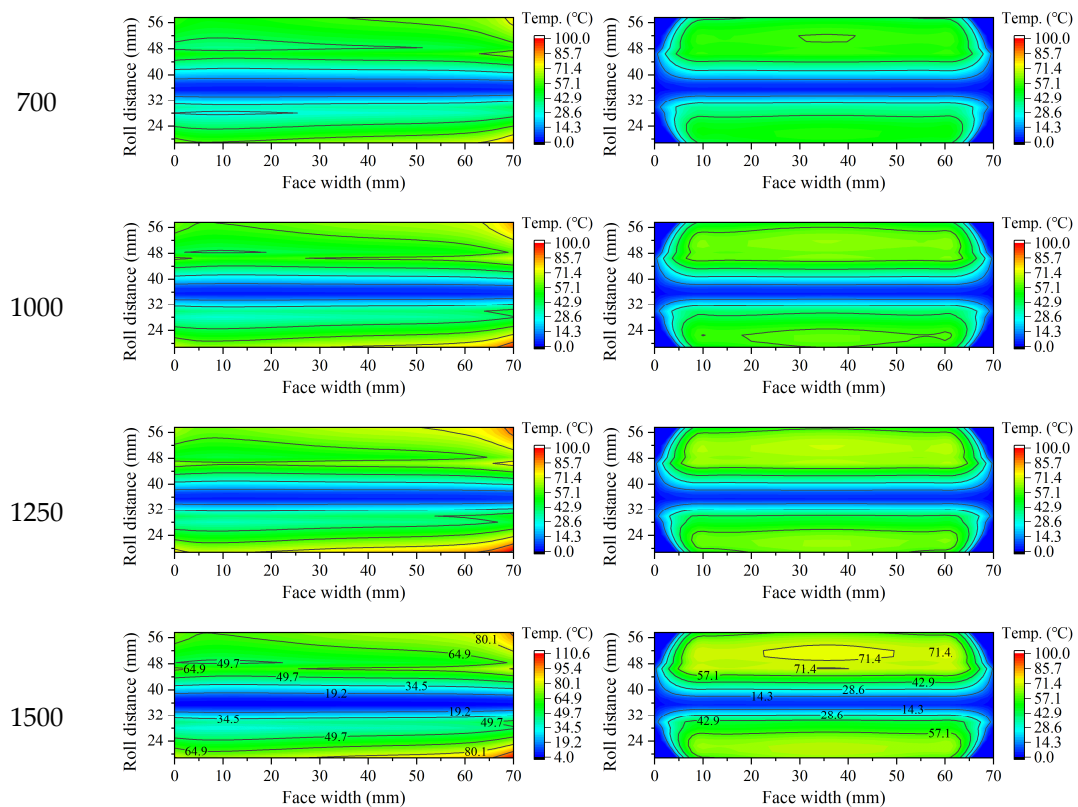


Figure 8. Colormap of tooth surface flash temperature distribution of gearset-G1735.

The influence of tooth surface modification on the flash temperature of the gear pair is significant. It can be seen from Figure 9, the full-load maximum flash temperature of the 13 groups gear pairs after modification is lower than before, and the range of reduction is 7.9–37.1 °C, the reduction rate is 17.1–42.8%. This result shows that properly designed tooth surface modification can greatly improve the gear anti-scuffing capacity.

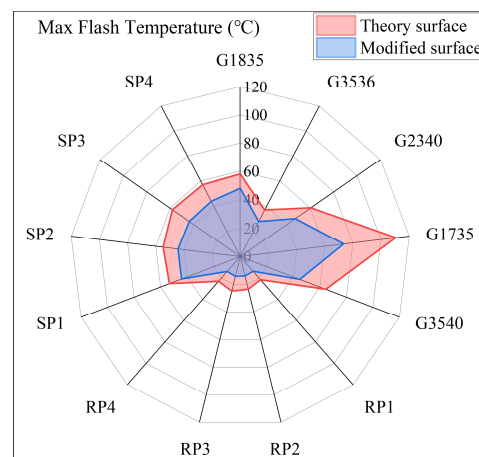


Figure 9. Maximum flash temperature of gearsets before and after tooth modification under full load conditions.

Taking the G1735 gear pair as the research object, under the load spectrum, the maximum flash temperature distribution pattern before and after tooth modification is shown in Figure 10. It can be seen that the highest flash temperature point is located at the maximum power point (1500 r/min, 2000 Nm), and taking this load point as the center, it

spreads and decreases in a quarter-circle shape outward toward the peripheral load case. The minimum power point has the lowest flash temperature. Due to the contribution of tooth surface modification, the maximum flash temperature is reduced from 110.6 °C to 73.5 °C, and compared to the theoretical tooth surface, the reduction rate is 33.5%.

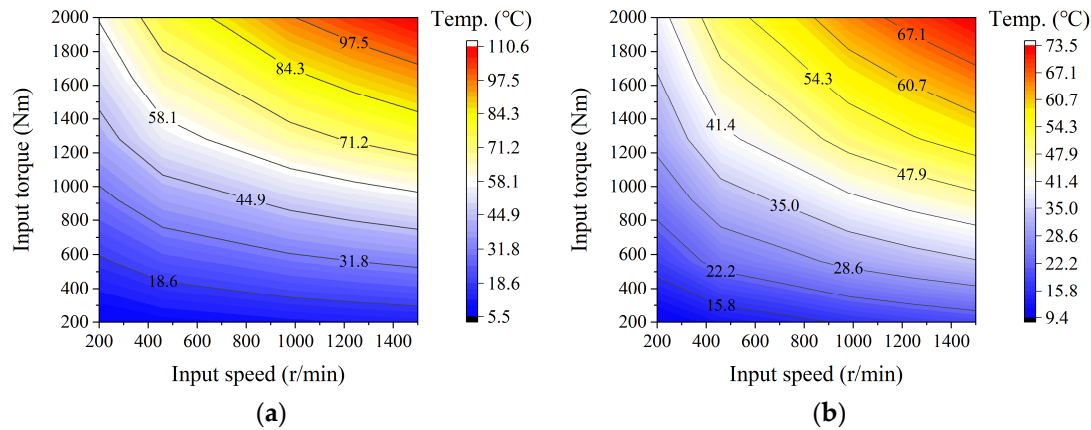


Figure 10. Tooth surface flash temperature of G1735 under load spectrum. (a) Non-modified tooth surface; (b) modified tooth surface.

3.3. Analysis of Tooth Surface Flash Temperature under Various Load Conditions

In order to study the relationship between the maximum flash temperature and the working conditions, the working conditions can be divided into two categories:

1. Constant torque and speed-up: in this condition, the input torque is maintained at 2000 Nm, while the input speed increases from 200 r/min to 1500 r/min.
2. Constant speed and torque-up: here, the input speed is kept at 1500 r/min, and the input torque increases from 200 Nm to 2000 Nm.

The analysis results of the maximum flash temperature of all gear pairs under each working condition are shown in Figure 11.

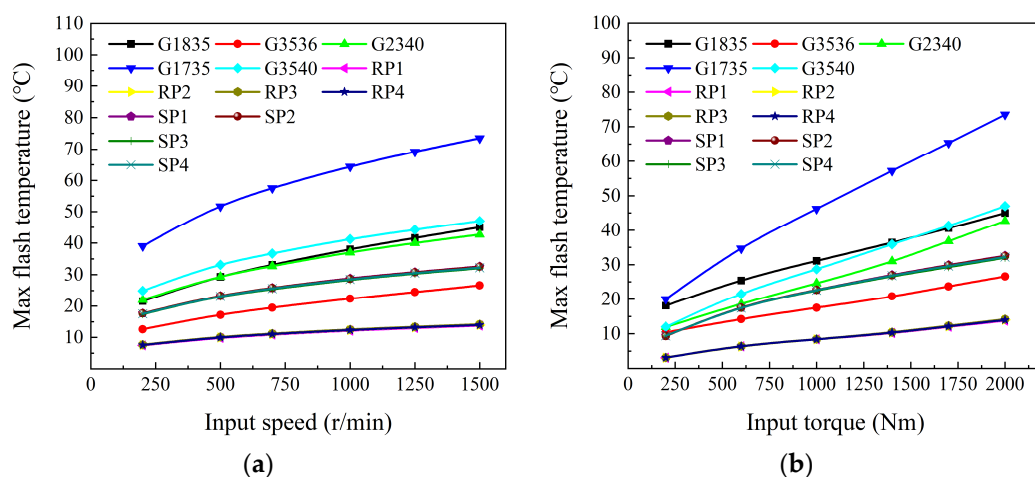


Figure 11. Tooth surface flash temperature under various operation conditions. (a) Flash temperature curves with speed variation; (b) flash temperature curves with torque variation.

The variation curve of the maximum flash temperature with the rotation speed of each gear pair is shown in Figure 11a. It can be seen that the maximum flash temperature increases with non-linear deceleration as the speed increases. Among all gear pairs, G1735 consistently exhibits the highest flash temperature. Following G1735 are the three fixed-axis gear train sets (G3540, G1835, and G2340). The meshing between the sun gear and the four

planetary gears in the planetary gear train have nearly identical flash temperatures. Gear pair G3536 falls next, and finally, the four meshing pairs between the planetary gears and the internal ring gear exhibit the lowest flash temperatures.

The variation curve of the maximum flash temperature with torque of each gear pair is shown in Figure 11b. It can be seen that the maximum flash temperature increases approximately linearly with the increase of torque, and the order of the maximum flash temperature values of each gear pair is almost the same as that under speed-up conditions.

4. Investigation of Reduction of Gear Meshing Power Loss

4.1. Gear Meshing Power Loss Solution

Meshing efficiency, a key performance indicator for high-performance gear transmissions, is affected by tooth surface friction. This friction, influenced by both topography and load conditions, leads to a portion of the input power being lost. Gear meshing power loss quantifies the energy consumed by tooth surface friction. This instantaneous meshing power loss is defined as follows:

$$P_z(i) = \mu_{mz}(i) \cdot F_n(i) \cdot v_s(i) \quad (18)$$

The meshing power loss is related to three key parameters: the instantaneous friction coefficient μ_{mz} of the meshing surfaces, the total normal force F_n , and the relative sliding speed v_s at the instant meshing position. These parameters can be determined using Equations (19)–(21), respectively.

$$\mu_{mz}(i) = 0.048 \cdot \left(\frac{w_i^n}{v_e(i) \cdot \rho_t(i)} \right)^{0.2} \cdot Ra^{0.25} \cdot \eta_{oil}^{-0.05} \cdot X_L \quad (19)$$

$$F_n(i) = \sum_j^M w_{ij}^n \quad (20)$$

$$v_e(i) = v_{g1}(i) + v_{g2}(i) \quad (21)$$

$$\rho_t = \rho / \cos \beta_b \quad (22)$$

Among these equations, Ra demotes the average surface roughness of a gear pair, v_e is the entrainment speed, ρ_t is the transverse relative curvature radius, and the instantaneous relative sliding speed v_s is calculated by Equation (17). If a meshing cycle (an angular pitch, $2\pi/z$) is divided into N equal parts, the rotation angle of each part is θ , unit in rad.

$$\theta = 2\pi / (z_1 \cdot N) \quad (23)$$

The meshing time of each equal part of the angle is t , in seconds.

$$t = \theta / \omega_1 \quad (24)$$

The meshing loss energy of one meshing cycle is Q_z , in joules.

$$Q_z = \sum_i^N P_z(i) \cdot t_i \quad (25)$$

Since the energy lost in each meshing cycle is equal, the average meshing power loss of the gearset is P_{mz} , in watts.

$$P_{mz} = Q_z / (N \cdot t) \quad (26)$$

The instantaneous meshing efficiency and average meshing efficiency of a gear pair are Equations (27) and (28), respectively.

$$\eta_z(i) = \frac{T_1\omega_1 - P_z(i)}{T_1\omega_1} \quad (27)$$

$$\eta_{mz} = \frac{T_1\omega_1 - P_{mz}}{T_1\omega_1} \quad (28)$$

The results below present the average meshing power loss and average efficiency for each gearset under a single load condition.

4.2. Verification of the Effectiveness of Modification Methods Aimed at Reducing Flash Temperature in Improving Meshing Efficiency

This section will verify whether the previous modification scheme optimized for tooth surface flash temperature can also reduce meshing power loss. First, the gear LTCA based on FE is used to calculate the instantaneous load distribution on the tooth surface, and the normal force distribution w_n is obtained. Take the geometric parameters of gears into the Equations (14)–(17) to obtain local curvature radius and relative sliding speed of the meshing tooth surface. The average meshing power loss and average meshing efficiency can then be obtained by Equations (18)–(28).

Taking the analysis results of gear pair G1735 as an example, Figure 12a is the theoretical tooth surface meshing power loss colormap; the maximum power loss is 4279 W, which occurs under full load conditions; and the average meshing power loss under the load spectrum is 1266.9 W. Figure 12b is the colormap of the meshing power loss for optimized tooth, the maximum loss also occurs under full load conditions, which is 3176 W, and the average power loss is 924.1 W. Compared to the theoretical tooth surface, the maximum meshing power loss of the optimized tooth surface is reduced by 1103 W, a reduction rate of 25.8%, and the average meshing power loss is reduced by 342.8 W, a reduction rate of 27.1%.

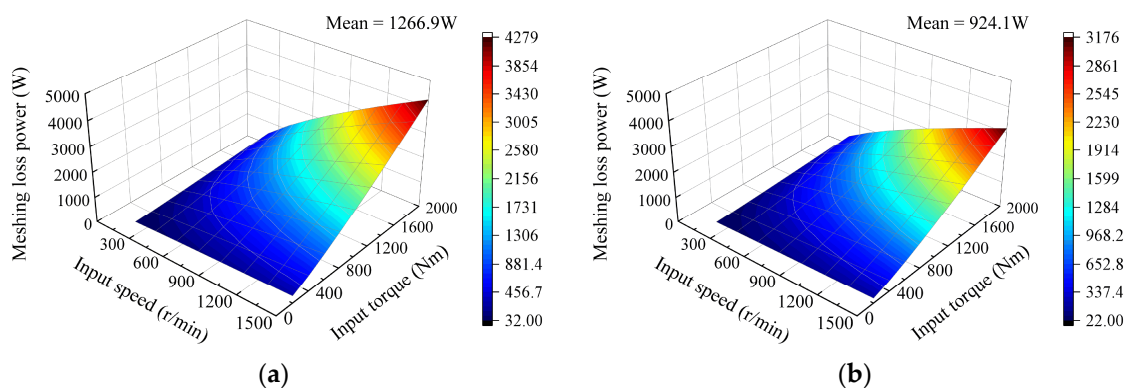


Figure 12. Gear meshing power loss under load spectrum. (a) Theory tooth surface of G1735; (b) improved tooth surface of G1735.

Figure 13a is a colormap of meshing efficiency of the theoretical tooth surface of gear pair G1735. Its maximum meshing efficiency is 99.11%, which occurs under the highest speed and light load level conditions, and the average meshing efficiency under the load spectrum is 98.57%. Figure 13b is a colormap of meshing efficiency of the optimized tooth surface, the maximum efficiency—which is 99.36%—also occurs under the highest speed and light load level conditions, and the average meshing efficiency is 98.97%. Compared to the theoretical tooth surface, the maximum meshing efficiency of the optimized tooth surface is increased by 0.28%, and the average meshing efficiency is increased by 0.4%.

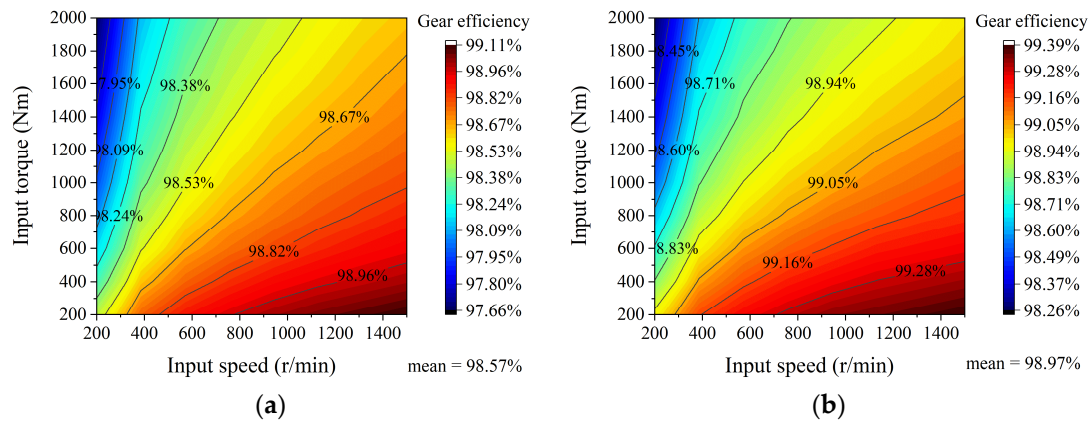


Figure 13. Gear meshing efficiency under load spectrum. (a) Theory tooth surface of G1735; (b) optimized tooth surface of G1735.

It can be seen from the above analysis that the proposed modification method to reduce meshing power loss and improve meshing efficiency is effective. Through the micro-geometry design of the tooth surface, the gear meshing friction loss can be greatly reduced, and an efficient transmission system can be obtained. Since tooth surface modification does not change the macro-parameters of the gear system, it is the most economical scheme to save energy for the machines.

4.3. Analysis of Meshing Power Loss under Various Load Conditions

To study the relationship between meshing power loss/efficiency and working conditions after gear tooth modification, the simulated working conditions were divided into two categories: One is constant torque and speed-up. In these conditions, the maximum torque is maintained at 2000 Nm, while the input speed increases from 200 r/min to 1500 r/min. The other is constant speed and torque-up conditions. Here, the input speed is held constant at 1500 r/min, and the input torque increases from 200 Nm to 2000 Nm. The analysis results for the accumulated power loss of the gear pairs under each working condition are shown in Figure 14.

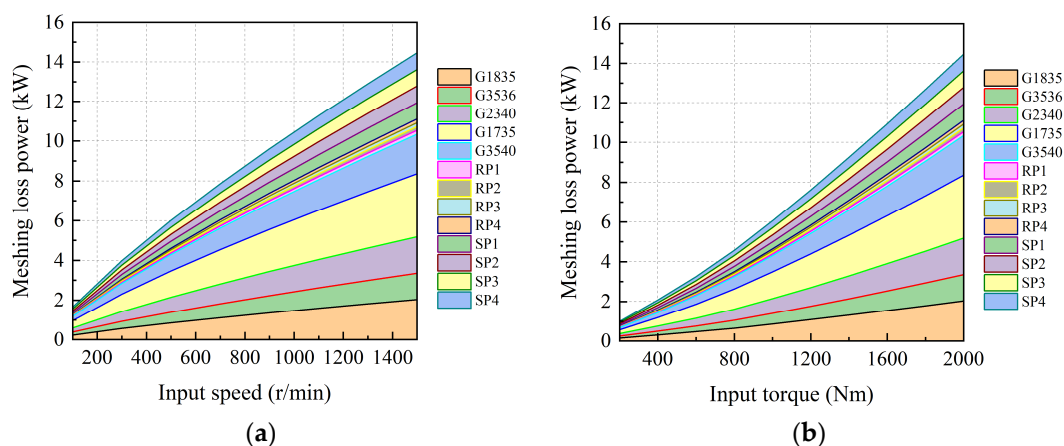


Figure 14. Gears meshing accumulated power loss under various operation conditions. (a) Accumulated power loss with speed variation; (b) accumulated power loss with torque variation.

The cumulative meshing power loss of 13 gear pairs changes with the rotational speed as shown in Figure 14a. It can be seen that the meshing loss increases approximately linearly with the increase of rotation speed. The five gear pairs of the fixed-axis gear train have high power loss in all working conditions. They are arranged in descending order as G1735, G1835, G2340, G3540, and G3536. The power losses of the sun and four planetary

gears meshing in the epicyclic gear train are basically equal; the least power loss is the internal ring and the four planetary gears. The cumulative power loss curve of each gear pair changes with the torque as shown in Figure 14b. The meshing loss increases with non-linear acceleration as the torque increases. The order of meshing loss of each gear pair is the same as the speed-up working condition. Under the whole load spectrum, the max meshing power loss of the entire gear system is 14.45 kW and the overall meshing efficiency is 95.4%.

The curves of meshing efficiency-speed of each gear pair as shown in Figure 15a. It can be seen that the meshing efficiency increases with non-linear deceleration as the rotational speed increases. Under low-speed conditions, the meshing efficiency is lower and the change rate is large, but in high-speed conditions, the rate of change decreases and gradually converges. The changing curves of meshing efficiency with torque are shown in Figure 15b. Although the efficiency curves vary, the overall meshing efficiency decreases with the torque-up. The G1735 gear pair has the largest decline slope for meshing efficiency, followed by the four meshing pairs of the sun gear and the planet gear. However, their efficiency is relatively stable in the low torque range and drops rapidly at medium and high torques. The efficiency–torque curves of the other gear pairs are approximately linear decline.

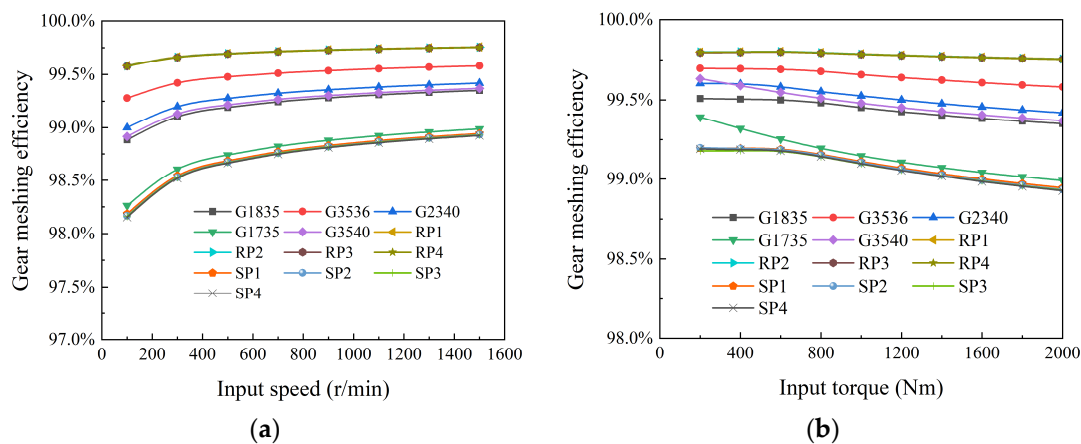


Figure 15. Gear meshing efficiency under various operation conditions. (a) Meshing efficiency vs. speed; (b) meshing efficiency vs. torque.

5. Experiments

5.1. Experimental Procedure

In order to verify that the proposed modification scheme has the effect of reducing the meshing friction loss of the gear system, it is first necessary to manufacture two groups of gears, the one is the gears with theoretical involute tooth, and the other is the gears with proposed modified tooth surface. A formed grinding was used in the finishing process of the gears (Figure 16a). After the gears ground, it was found through measurement that the tooth profile and lead curves have met the designed modification intention (Figure 16b). The gears' accuracy grade is ISO-1328-level 7, and the tooth surface roughness is $0.8 \mu\text{m}$. The two finished groups of gears were assembled into the two ranging arms, respectively, and ISO-VG220 synthetic lubricating oil was added to the gear box. The lubricating oil parameters are shown in Table 3.

The experiment was carried out on the ranging arm test bench. Since the ranging arm is a large and heavy piece of equipment with an output torque of about 80 kNm. Due to the limitations of the test equipment, it has not been equipped with a high-precision torque sensor that meets this large range. Therefore, the temperature rise test was only conducted this time. The installation position of the temperature sensor is shown in Figure 17. Since the precise measurement of tooth surface flash temperature of closed gearbox has not been reported and cannot be measured directly, some flash temperature measurements require

the test gear pair to be made of different conductive materials [32]. Hence, this test is indirectly measured by measuring the internal structure temperature near the gear meshing point. Due to the conduction, convection, and radiation effects of heat, the measured value is generally lower than the tooth surface flash temperature, but the effectiveness of the modified gear can be verified through comparative tests.

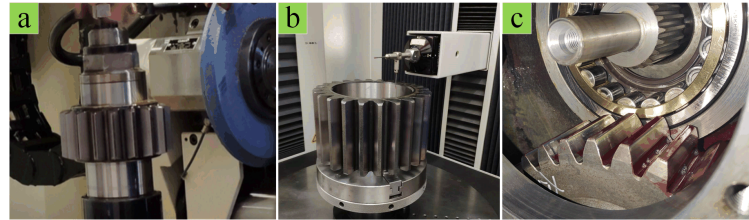


Figure 16. Modified tooth surface of gears were (a) ground, (b) measured, and (c) tested for load contact pattern.

Table 3. The parameters of ISO-VG220 synthetic lubrication oil.

Name	Kinematic Viscosity @40 °C	Kinematic Viscosity @100 °C	Density @15 °C	FZG Scuffing Test Grade
Value	220 mm ² /s	39 mm ² /s	1026 kg/m ³	12

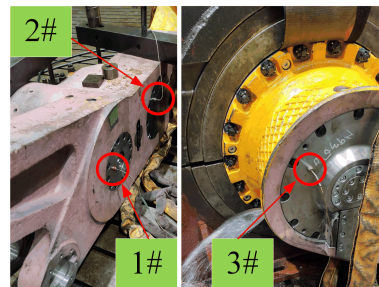


Figure 17. Layout position of temperature sensors.

The 1#, 2#, and 3# sensors measure the flash temperature of the G1835, G1735, and the sun gear, respectively. First, a no-load running-in test was conducted for 120 min. Then, under the same working conditions and boundary conditions as the simulations, temperature rise tests were conducted on the two group ranging arms. When the ambient temperature was 20 °C and the input speed was 1500 r/min, three stages of sequential loading conditions are performed. Running at an input torque of 1000 Nm for 60 min, 1500 Nm for 60 min, and 2000 Nm for 30 min, and the temperature was measured and recorded every 30 min during the process.

5.2. Experimental Results

During the loading process, multiple measurements are taken every half an hour, the measured values of the temperature sensor were recorded and averaged. The results are shown in Figure 18. By comparing the two groups' temperature rise test results, it can be found that the temperature curve near G1835 is recorded by the 1# sensor; the maximum temperature of the unmodified gear is 71.2 °C; the modified gear is 67.1 °C, a decrease of 4.1 °C; and the reduction rate is 5.8%. The temperature curve near the G1735 recorded by the 2# sensor is the highest; the maximum temperature of the unmodified gear is 74.2 °C; and after modification, it is 65.1 °C, which is a decrease of 9.1 °C and a reduction rate of 12.2%. The temperature curve near the sun gear recorded by the 3# sensor shows that the maximum temperature of the unmodified gear is 57.6 °C and that after modification, it is 54 °C, a decrease of 3.6 °C and a reduction rate of 6.3%.

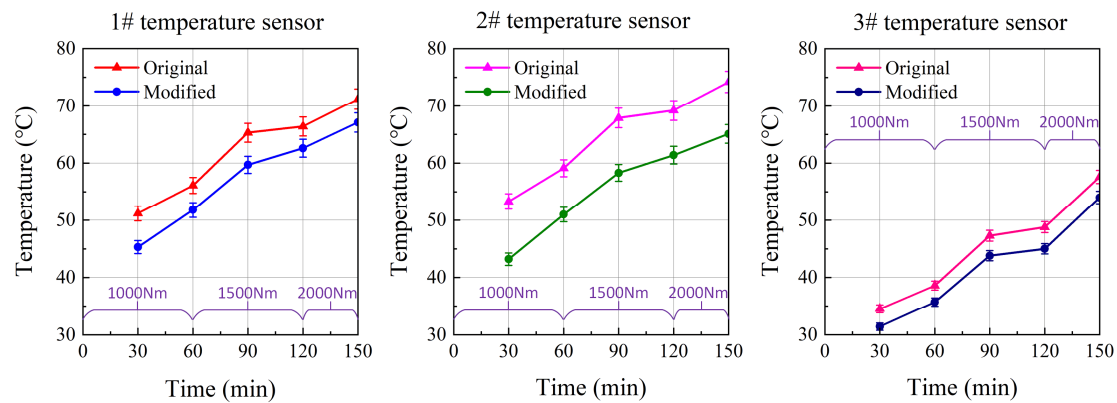


Figure 18. Measurement results of temperature sensors.

It can be seen that as the input power and the time increases, the flash temperature of the gears after optimization are overall lower than unmodified gears, with a maximum temperature reduction of 9.1 °C and the maximum reduction rate of 12.2%. Since the flash temperature value is indirectly measured, there are other energy dissipation factors in the system. Although there are some discrepancies between the tests and the simulation for flash temperature and reduction rate, the overall flash temperature reduction trend is consistent, which can prove that the proposed gear modification method for reducing flash temperature and decreasing meshing power loss are effective.

6. Conclusions

In order to study the methods of reducing tooth surface flash temperature and meshing power loss during gears operation, a targeted tooth surface modification optimization scheme was proposed, and a semi-analytical simulation was conducted for the optimized scheme. Finally, experiments were used to verify the effectiveness of the modification scheme, and the following conclusions can be drawn.

1. A tooth surface modification method with the goal of reducing tooth surface flash temperature and meshing power loss is proposed. After simulation analysis and experimental verification, this scheme can significantly reduce tooth surface flash temperature and meshing power loss.
2. A semi-analytic tooth surface flash temperature calculation method is proposed, which makes up for the inability to calculate the real tooth surface load distribution in the ISO standard method, and it is also faster than the pure finite element method of fluid-structure-thermal coupling to calculate friction flash temperature.
3. For theory teeth, the maximum flash temperature occurs near the tooth tip or root end area, where there is a higher relative sliding speed. For modified gear teeth, it is mainly distributed in the middle of the double-tooth contact area symmetrical with the pitch line. The maximum flash temperature increases with non-linear deceleration as the speed-up, and with approximately linearly as the torque-up. The meshing efficiency increases with non-linear deceleration as the rotational speed increases, and overall meshing efficiency decreases with the torque increase. Under low-speed conditions, the meshing efficiency is lower and the change rate is large, but in high-speed conditions, the rate of change decreases and gradually converges.
4. Through the comparison of the two gear groups' temperature rise test, it can be seen that as the input power and the time increases, the tooth temperatures of optimized gears are overall lower than unmodified gears. Since the flash temperature value is indirectly measured, there are other energy dissipation factors in the system, so there are some discrepancies between the tests and the simulation for flash temperature and reduction rate, but the overall flash temperature reduction trend is consistent, which can prove that the proposed gear modification method for reducing flash temperature and decreasing meshing power loss are effective.

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