Oil Jet Lubrication and Heat Dissipation Characteristics of High-Speed Helical Gears: Numerical Simulation and Experimental Validation

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Abstract: High-speed gears are widely used in various fields, such as new energy vehicles and aviation, where efficient lubrication and heat dissipation are critical for their performance. Gear failure due to overheating under high speeds and heavy loads is a common issue. Therefore, oil jet lubrication is applied to improve heat dissipation and reduce tooth surface temperatures. This study presents a numerical simulation model of oil jet lubrication for high-speed helical gears, considering injection angle, distance, and velocity. The temperature field and heat dissipation characteristics are analyzed using Computational Fluid Dynamics (CFD) methods. Grid independence verification is conducted to ensure the reliability of the simulation results. Furthermore, experimental tests are performed to validate the simulation data, using thermocouples and infrared thermal imaging. The results show that optimizing injection parameters significantly improves gear cooling performance, with the oil injection angle, distance, and velocity having notable effects on the heat transfer process. The study provides insights for designing more efficient lubrication systems for high-speed gears.

Keywords: High-speed gears, Oil jet lubrication, Heat dissipation, CFD, Numerical simulation, Experimental validation

1. Introduction

High-speed gears are extensively utilized in various fields, such as new energy vehicles, aviation gearboxes, and transmissions. These gears often operate under extreme conditions involving high speed and heavy loads, which can result in gear failure due to adhesive wear. Therefore, a well-designed lubrication system is critical to prevent such failures, and it is essential to optimize the parameters of oil injection to ensure effective lubrication. Oil injection lubrication is widely applied to cool and lubricate high-speed gears, with tooth temperature variations frequently used to assess the heat dissipation performance. This information is vital for optimizing the lubrication system.

Numerous analytical approaches have been proposed to evaluate gear heat transfer. Block [1] introduced an equation to calculate the instantaneous temperature rise in gears, while Tian [2] employed Green's function to analyze surface temperatures under different heat sources. Vick and Furey [3] expanded on this by developing a theoretical model for temperature rise due to sliding contact on rough surfaces, demonstrating that subdividing a contact point into multiple contact points significantly reduces temperature rise. Additionally, Chen et al. [4] established a thermal grid model, showing that the sun gear experiences a higher temperature than other components. Monhammadpour [5] performed a thermal elastohydrodynamic lubrication (EHL) analysis of hypoid gears, and Luo [6] applied tribological methods to calculate the convective heat transfer coefficient and frictional heat flux on the tooth surface. Further, Yin et al. [7] developed a thermal EHL model for double involute gears, factoring in the properties of non-Newtonian fluids. Černe et al. [8] proposed a flash temperature model for spur gears, emphasizing that the gear meshing process plays a significant role in temperature rise. Zhou et al. [9] and Han et al. [10] contributed by developing models for predicting contact temperature and revealing the relationship between gear temperature and dynamic thermal stress, respectively.

Accurate prediction of gear heat transfer is crucial for improving gear design and performance. Most of these predictions rely on numerical simulation methods, where boundary conditions like convective heat transfer coefficients and heat flux density are used in finite element models. Researchers have analyzed the temperature fields of polymer and metal gears under different lubrication conditions [11-13]. Notably, injection lubrication has proven more effective than oil mist lubrication in reducing gear body temperature [14]. Wei et al. explored the temperature fields of cracked gears [15] and helical gears, considering factors such as machining and installation errors [16]. Mao et al. [17-18] proposed methods to solve the instantaneous temperature of polymer gears, while Wang et al. [19] investigated the effects of dynamic load and frictional force on spur gear temperature. Numerical simulations, such as those using computational fluid dynamics (CFD), have also been applied to study gear heat dissipation under dry lubrication [20-22].

A wide range of research has been conducted on the distribution of gear temperature. For instance, Block et al. [23] proposed an approximate calculation for flash temperature based on sliding friction, while Tobe et al. [24] found that gear modifications could reduce flash temperature. Npatid et al. [25] studied the impact of dimensionless parameters on spur gear temperature, and Townsend et al. [26] estimated the heat generated by friction and the heat transfer coefficient for gear teeth. They found that both constant speed loading and acceleration at a constant load significantly increase the gear’s overall temperature. Handschuh et al. [27-28] conducted thermal analysis on spiral bevel gears, focusing on the heat transfer behavior of a single tooth.

Elastohydrodynamic lubrication (EHL) theory has been extensively used to study the thermal behavior of gears. Dowson et al. [29] simulated the relationship between geometric position and velocity at the meshing point of spur gear pairs. Vichard et al. [30] examined the effects of load and entrainment speed, while Evans et al. [31] and Bobach et al. [32] incorporated real surface roughness into their models to predict flash temperatures and fatigue on gear surfaces. Experimental studies by F. et al. [33] investigated the effect of oil jet position on the operating temperature of spiral bevel gears, highlighting the significant influence of nozzle position on gear temperature. The relationship between machine tool settings and gear temperature has also been explored. Simmon et al. [34-36] examined how pinion adjustment parameters affect the oil film pressure, temperature, and power loss. Gopalakrishnan et al. [37] applied the EHL model to calculate the friction coefficient of spiral bevel gears. Additionally, Bobach et al. [38] developed a transient thermal EHL model for mixed lubrication in spiral bevel gears, revealing that gear body temperature plays a more significant role than torque in mixed friction behavior. Wen et al. [39] presented a numerical solution for solving unsteady EHL in elliptical contacts, while Mingyong et al. [40] modeled the mechanical efficiency of helical gears under thermal EHL conditions. Pu et al. [41-46] further developed methods to predict friction coefficients and flash temperatures, accounting for variables such as velocity vector and surface roughness.

The use of computational fluid dynamics (CFD) has also contributed to understanding oil injection lubrication in gears. Studies have examined the impact of injection parameters on heat dissipation in ball bearings and the wind resistance losses of spiral bevel gears [47-48]. S. and P. [49-50] analyzed the optimal injection velocity for maximum cooling efficiency, while Höhn et al. [51] investigated the relationship between oil flow, power loss, and heat generation. Ouyang et al. [52] predicted transient gear temperature using a coupled heat-flow model, and Andersson et al. [53] compared the effects of immersion versus injection lubrication on gear temperature. Numerical simulations by Dai et al. [54-56] revealed that increasing collision depth lowers gear temperature. Chen et al. [57] concluded that nozzle placement and face angles can negatively affect lubrication when the injection distance is large, and Mo et al. [58] examined the effect of injection velocity on oil pressure and volume. Finally, Zhu et al. [59-63] optimized nozzle positioning and other injection parameters for spiral bevel gears to further improve cooling performance.

In summary, the thermal behavior of gears plays a critical role in their performance, especially under high load and high-speed conditions. To optimize the heat dissipation design of gears, numerous studies have employed numerical simulation methods to predict gear heat transfer performance and provide recommendations based on the analysis of temperature fields. Among these studies, oil injection lubrication has demonstrated significant advantages in effectively reducing the temperature of gear bodies. This provides a solid foundation for designing efficient lubrication systems and improving gear heat dissipation performance. In this study, we propose and validate a new gear heat dissipation model, aiming to further enhance the thermal efficiency and stability of gear transmission systems.

2. Gear heat dissipation model

In this study, a conventional oil injection lubrication system is considered, where the injection angle is set to 0°, the distance between the nozzle and the gear is 60 mm, and the injection velocity is 25 m/s. A pair of meshing helical gears is modeled for analysis. The geometric and operational parameters for these gears are provided in Table 1. Key dimensions include a gear module of 2 mm, a pressure angle of 20°, a spiral angle of 15°, and a tooth width of 8 mm. The driving wheel rotates at 12,000 revolutions per minute (r/min).

Table 1 Geometrical and working parameters for helical gear pair

Parameter	Value	Parameter	Value
Gear teeth z_1, z_2	20, 40	Gear material	17CrNiMo6
Module m_n/mm	2	Specific heat capacity c/(J·kg^{-1}·K^{-1})	477
Pressure angle α_n/(°)	20	Elasticity modulus E/GPa	210
Spiral angle β/(°)	15	Gear density ρ_a/(kg/m^3)	7850
Tooth width B_1, B_2/mm	8,8	Thermal conductivity coefficient k_t/(W·m^{-1}·K^{-1})	42.7
Rotational speed n_1, n_2/r·min^{-1}	12,000, 6000	Poisson ratio γ_1, γ_2	0.3, 0.3

Figure 1a illustrates the geometric model used in the temperature field analysis, while Figure 1b depicts the mesh model of the helical gear. A tetrahedral mesh was generated using local mesh refinement, with a minimum element size of 0.04 mm and an average unit size of 1 mm. In the flow field module (Figure 1c), the gearbox is approximated as a cuboid. The interactions between the driving and driven wheels and the gearbox walls were defined using interaction surfaces. The sliding mesh method was employed to represent the fluid domain, focusing on the meshing area between the gears (Figure 1d). This refined mesh ensures greater accuracy in capturing the heat transfer phenomena in critical regions.

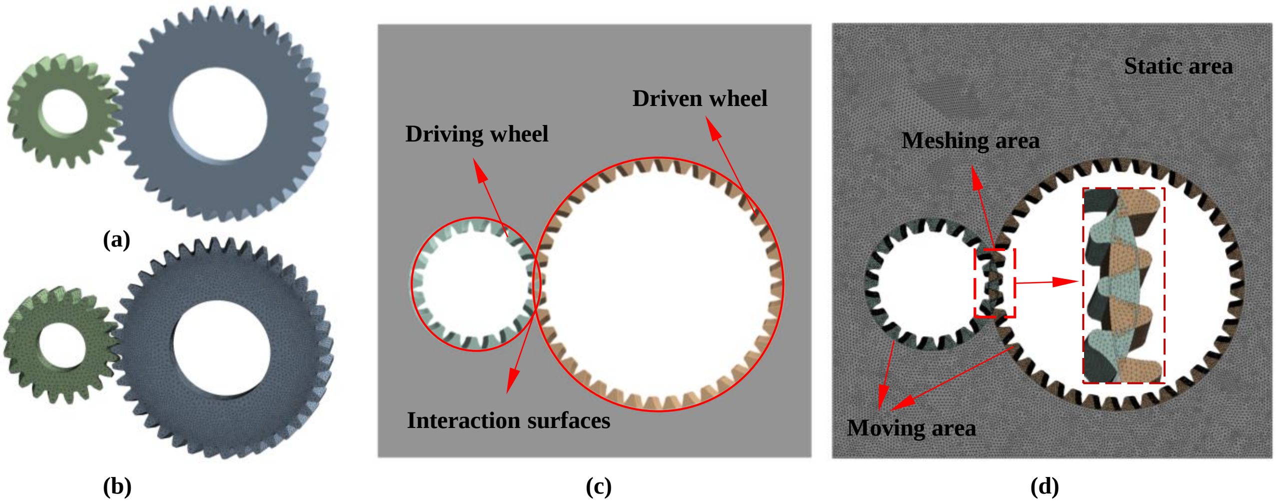

Fig. 1 Finite models in the temperature field module, (a) geometric model, (b) mesh model; Finite models in the flow field module, (c) geometric model, (d) mesh model

The oil injection lubrication process was simulated using Fluent, where the lubrication nozzle was modeled as a velocity boundary condition with a hydraulic diameter of 0.0025 m. The meshing teeth serve as the frictional heat-generating surfaces, and the no-slip boundary condition is applied to all other walls. For pressure-velocity coupling, the standard couple algorithm was used, and gradient differences were computed via the least-squares method. Both pressure and momentum terms were solved using the PRESTO (Pressure Staggering Option) scheme and second-order upwind discretization, respectively. A total of 500 iterations were performed, with each simulation running for 0.005 seconds to capture heat dissipation dynamics.

Grid independence was verified to ensure the reliability of simulation results. As shown in Figure 2a, the gearbox temperature distribution was obtained for different grid cell sizes. The average temperature and temperature differences remained stable as the number of grid elements increased to 1.5×10^6, with errors below 1%. Based on this, the total number of grid elements was fixed at approximately 1.5×10^6 to maintain accuracy and computational efficiency (Figure 2b).

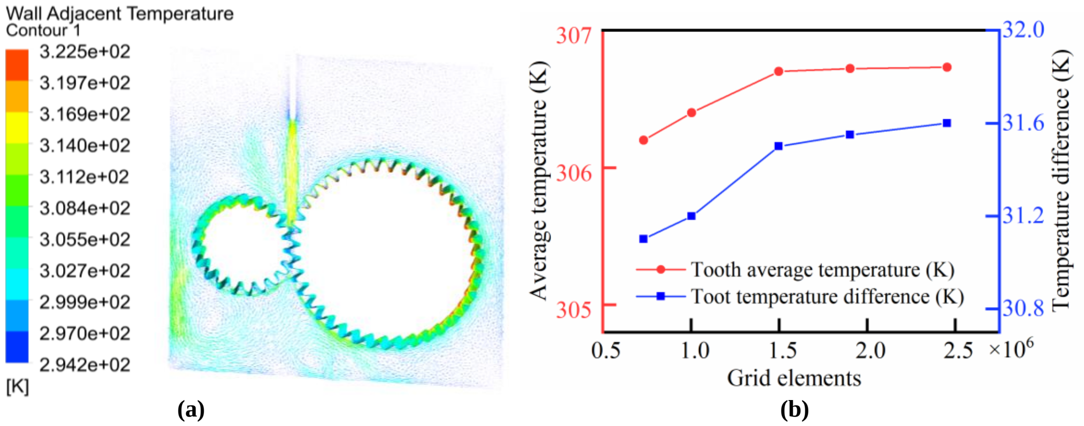

Fig. 3 (a) Gearbox temperature distribution; (b) Grid independence verification

3. Results and discussion

3.1. Static heat flow coupling

During high-speed oil injection lubrication, the interaction between the oil and the gear tooth surface occurs over a very short time, forming a thin lubricating film. The oil maintains good fluidity and a stable pressure distribution, with minimal deformation of the gear tooth surface. Given the large radius of curvature and the low degree of bending on the tooth surface, it is reasonable to approximate the surface as flat for the purposes of heat transfer analysis.

Figure 3a illustrates the schematic of oil injection heat dissipation. As oil droplets are ejected radially, they collide with the tooth surface, initiating forced convection heat transfer. In Figure 3b, point A represents the oil injection point, located on the pitch circle of the gear. The tooth surface is divided into left (L) and right (R) sections relative to this injection point. The oil's path, represented by streamline NA, is defined by the injection angle (α), which is the angle formed between the oil streamline and the tangential direction of the injection point. The distance between the nozzle and the injection point is denoted as H, while V represents the injection velocity, which can be broken down into two components: tangential velocity (V_τ) and normal velocity (V_n) along the tooth surface.

The heat dissipation process occurs as follows: oil is sprayed onto the tooth surface at a velocity V. Upon impact, the normal component of the velocity (V_n) causes oil to spread across the tooth surface, cooling it through forced convection. The lubricant flows along the R surface due to the tangential velocity (V_τ), while some of the oil splashes and spills from the surface. As the oil moves along both the L and R surfaces, it carries away heat, resulting in cooling.

To evaluate the influence of different injection parameters on heat dissipation, a static model of gear oil injection cooling was created, as shown in Figure 3c. The geometric model was meshed with unstructured tetrahedrons (Figure 3d), and the flow and heat dissipation calculations were carried out using the k-ε turbulence model. The nozzle was modeled as a velocity inlet, with a diameter of 0.0025 m, while the gear's meshing area was simplified to represent the heating surface. The surrounding walls of the model were treated as open boundaries to account for the splashing of oil in the meshing area.

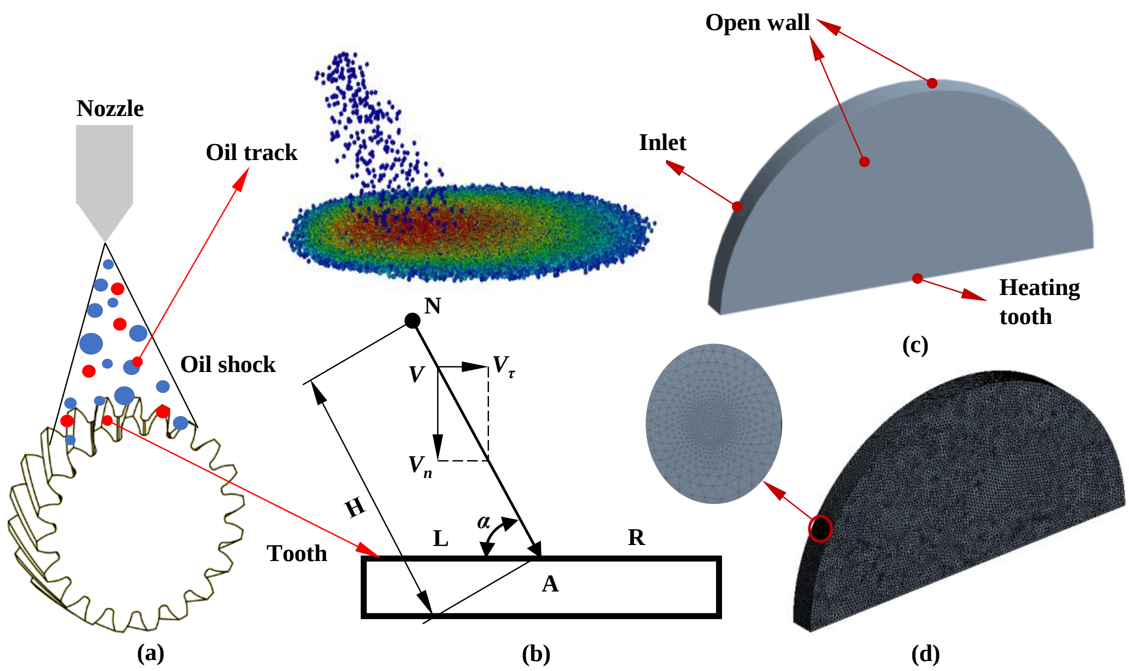

Fig. 3 (a) Oil injection heat dissipation schematic, (b) Schematic of the oil injection process,

Heat dissipation of gear: (c) calculation model, (d) mesh model

The effects of injection parameters, including angle, distance, and velocity, on heat dissipation were examined for non-rotating gears under single-phase flow conditions (oil-only) and oil-air two-phase flow. The lubrication system used air as the primary phase and CD40 lubricant as the secondary phase (Table 2), with the domain temperature held constant at 293.15 K throughout the simulation.

Table 2 Thermal characteristic parameters of air and lubricant

Parameter	Density (kg/m^3)	Thermal conductivity (W/m·K)	Specific heat (J/kg·K)	Viscosity (m^2/s)
Air	1.09	0.027	1013	1.95×10^{-5}
Lubricant (CD40)	883	0.144	1600	5.56×10^{-2}

The behavior of the lubricant flow and its cooling effect on the gear tooth surface were analyzed under different injection conditions. In particular, an injection angle of 60°, a distance of 60 mm, and an injection velocity of 30 m/s were tested. Figure 4a displays the velocity streamlines of the oil, where two vortices formed on either side of the oil jet, and the velocity decreased gradually as the oil traveled further from the nozzle. Figure 4b shows the vector field of the oil flow, with the highest velocity observed at the spray point, decreasing steadily with distance.

In terms of heat transfer, Figure 4c illustrates the distribution of the convective heat transfer coefficient on the tooth surface, with the central region near the injection point exhibiting the highest coefficients. As the distance from the center increases, the heat transfer coefficient decreases. Figure 4d presents the temperature distribution across the tooth surface, showing that the surface temperature declines most rapidly at the injection center. The cooling rate slows toward the edges due to the gradual reduction in oil speed and coverage, but overall, the oil effectively reduces the tooth temperature.

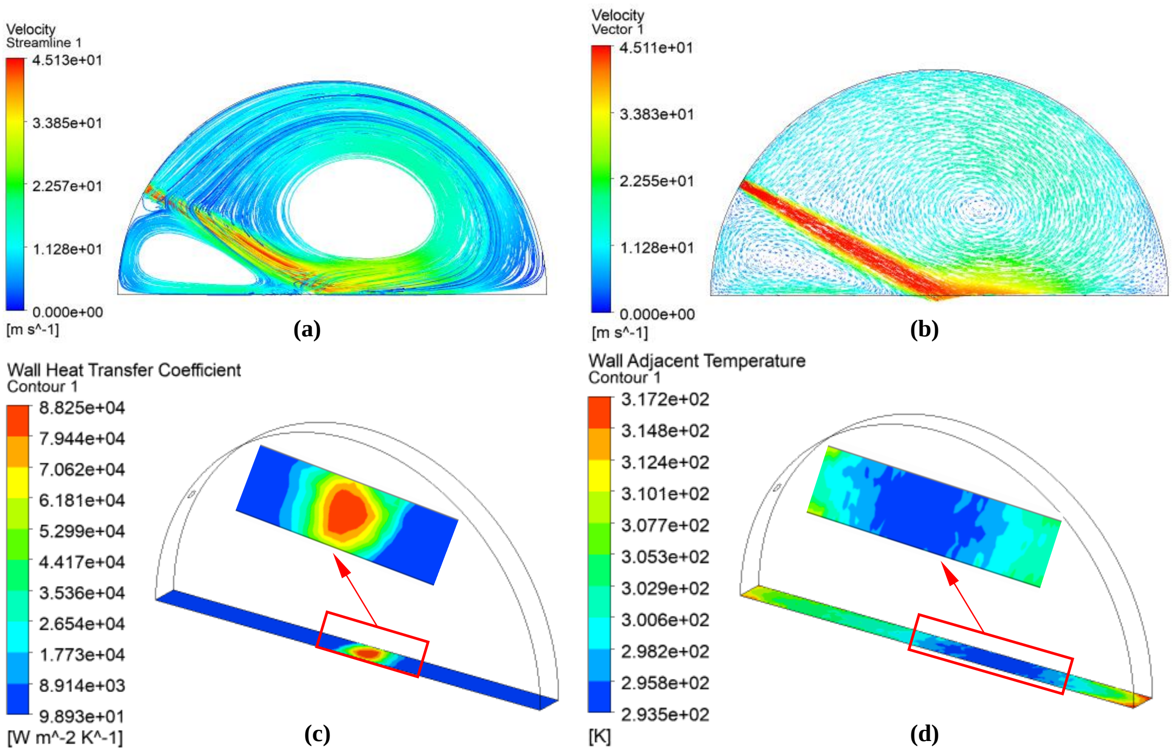

Fig. 4 Flow and temperature distribution of heat dissipation models, (a) Streamline distribution, (b) Vector distribution, (c) Wall heat transfer coefficient distribution, (d) Temperature distribution

The gear heat dissipation test was designed to measure how different injection parameters affect cooling performance. Table 3 outlines the test scheme used to measure the temperature change on the gear surface under various conditions. The gear was heated to 323.15 K, and the temperature was monitored as the lubricant was injected at different angles, distances, and velocities.

Table 3 Gear heat dissipation test scheme

Factor	Injection angle (°)	Injection distance (mm)	Injection velocity (m/s)
Injection angle (°)	0, 10, 20, 30, 40	60	30
Injection distance (mm)	10	30, 45, 60, 75, 90	30
Injection velocity (m/s)	10	60	15, 20, 25, 35, 45

The test setup, shown in Figure 5, included an air compressor, oil injection system, control device, and temperature measurement system. The oil was injected onto the gear surface at high pressure, and the resulting temperature changes were recorded. Foam boards were used as insulation to prevent heat conduction between the gear and the test bench, ensuring accurate temperature measurements. A thermocouple was attached to the gear to track the temperature as it dropped from 323.15 K to ambient conditions.

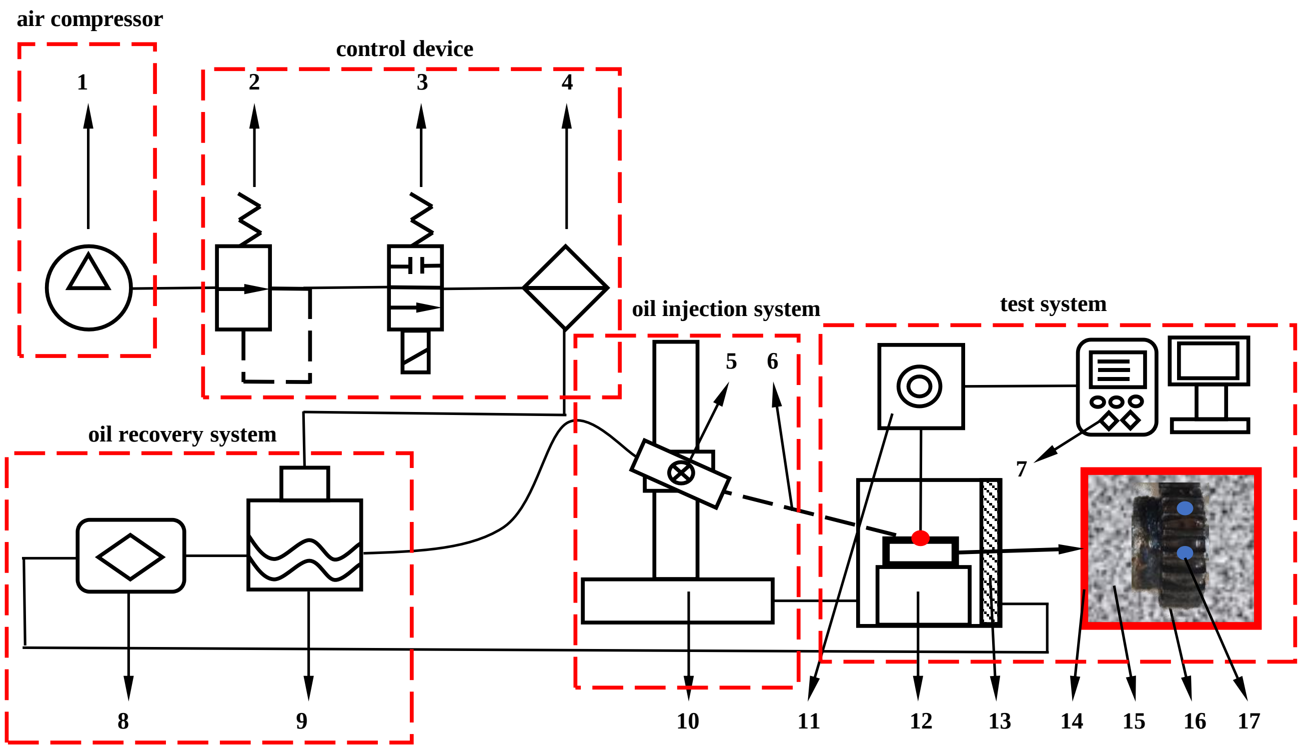

Fig. 5 Gear heat dissipation test bench

1. Air compressor; 2. Pressure regulating valve; 3. Solenoid valve; 4. Oil and gas separator; 5. Universal fixture; 6. Oil jet; 7. Data processing system; 8. Return oil pump; 9. Tank; 10. Stent; 11. Temperature controller; 12. Test platform; 13. Baffle; 14. Heating plate; 15. Insulation foam; 16. Test gear; 17. Thermocouple

Figure 6a compares the cooling curves of natural cooling versus oil injection cooling. The rate of temperature reduction with oil injection was significantly higher than that of natural cooling, demonstrating the effectiveness of oil in enhancing gear heat dissipation. Figures 6b-d illustrate the cooling curves for varying injection angles, distances, and velocities. The results show that the cooling effect improved with smaller injection angles, shorter distances, and higher injection velocities, as these parameters increased the amount of oil in contact with the tooth surface and enhanced the rate of heat removal.

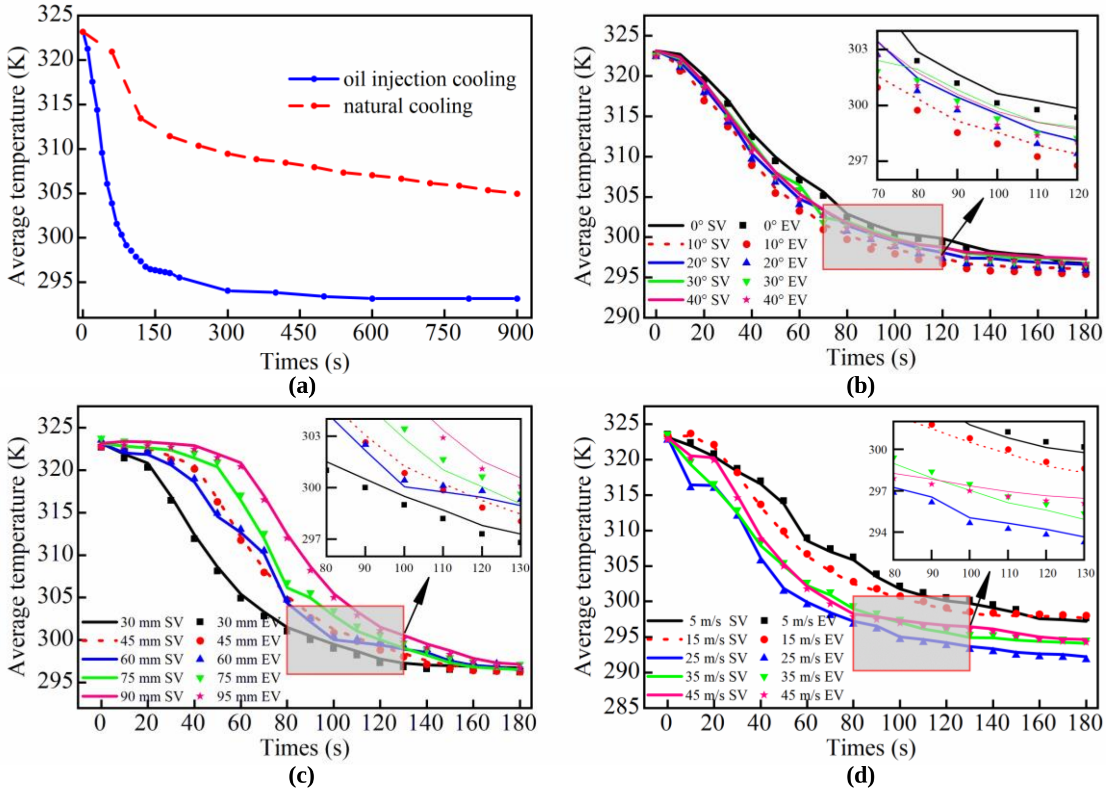

Fig. 6 (a) Curves of natural cooling and injection cooling, (b) Cooling curves under different injection angles, (c) Cooling curves under different injection distances, (d) Cooling curves under different injection velocities (SV denotes the simulation value; EV indicates the experimental value)

3.2. Dynamic heat flow coupling

The dynamic heat flow coupling analysis was conducted to explore the heat dissipation behavior of gears under operating conditions. Figure 7a shows the velocity vector distribution in the helical gearbox, illustrating how the lubricant flows through the system. As the oil enters the gearbox, its velocity increases gradually, with the densest vectors concentrated around the gear meshing and exit areas. The highest flow velocities occur beneath the nozzle and in the left side of the gearbox. The streamline distribution in Figure 7b shows how the lubricant is sprayed from the nozzle onto the tooth surfaces, generating oil splashes above the gears. Due to the higher speed of the driving wheel compared to the driven wheel, much of the lubricating oil is thrown off the surface of the driving wheel, creating a vortex beneath it. The splashing effect also results in a noticeable oil distribution deviation from the intended injection direction, as seen in Figure 7c, where the oil volume fraction on the gear surface ranges between 0.25 and 0.5. Temperature distribution across the gearbox, shown in Figure 7d, reveals that the highest temperatures are concentrated above the gears. The temperature of the driven wheel is higher than that of the driving wheel because much of the oil injected onto the driving wheel is thrown toward the driven wheel, resulting in reduced heat exchange on the driving wheel's surface.

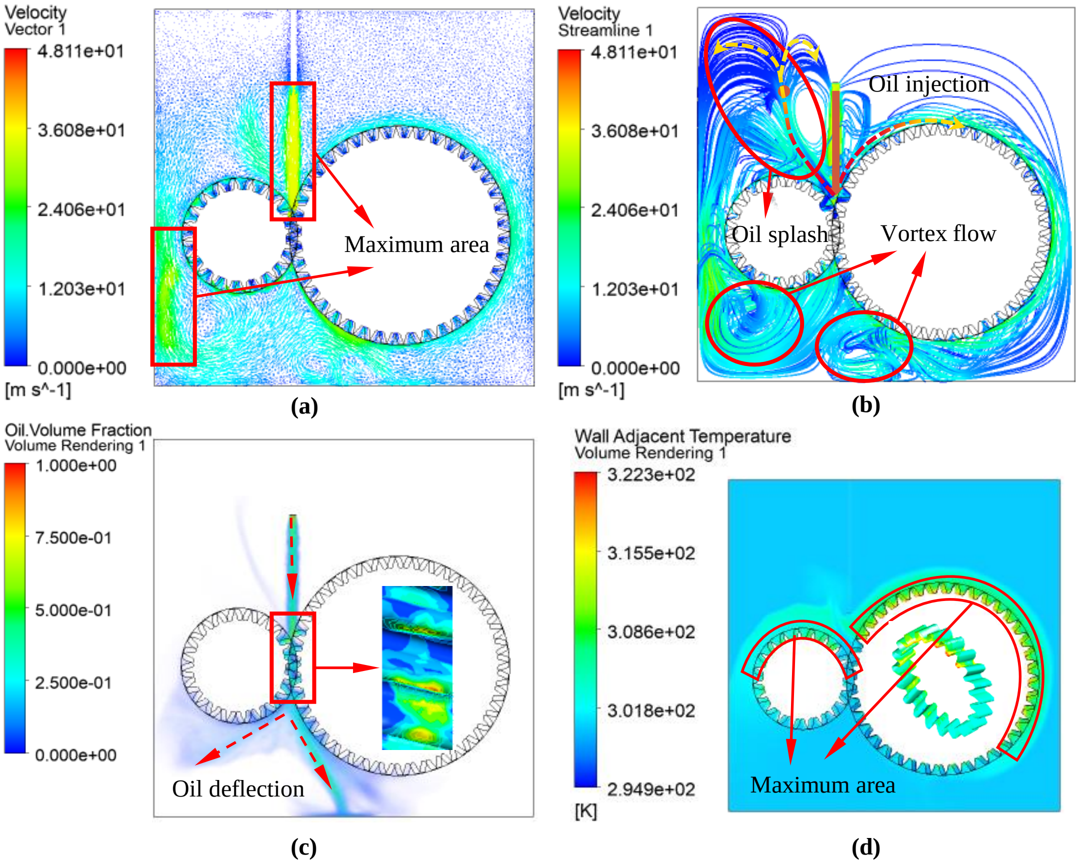

Fig. 7 Flow and temperature characteristics of the helical gearbox, (a) Vector distribution, (b) Streamline distribution, (c) Oil distribution, (d) Temperature distribution

Further investigation into the relationship between oil flow characteristics and temperature distribution was carried out. As shown in Figure 8a, the oil coverage on the tooth surface is uneven, with most of the oil accumulating on the driven wheel. This is reflected in the average oil volume fractions of 0.26 and 0.36 for the driving and driven wheels, respectively. The oil distribution decreases from the center of the tooth to its edges, with little oil deposition along the sides. The oil pressure distribution on the tooth surface (Figure 8b) also indicates higher pressure on the driven wheel compared to the driving wheel. The wall heat transfer coefficients in Figure 8c show values of 984 $W/m^2 \cdot K$ and 1276 $W/m^2 \cdot K$ for the driving and driven wheels, respectively. This indicates that more effective heat transfer occurs on the driven wheel due to greater oil coverage and higher oil pressure. Consequently, the average temperatures of the tooth surfaces are lower on the driven wheel (299.5 K) compared to the driving wheel (301.8 K), as shown in Figure 8d. The temperature distribution is more uniform in the center, with slightly higher values toward the tooth edges. The results confirm that oil flow dynamics play a crucial role in the heat dissipation performance of gears. The greater the oil pressure and coverage on the tooth surface, the more efficient the heat transfer, leading to improved cooling of the gear surfaces.

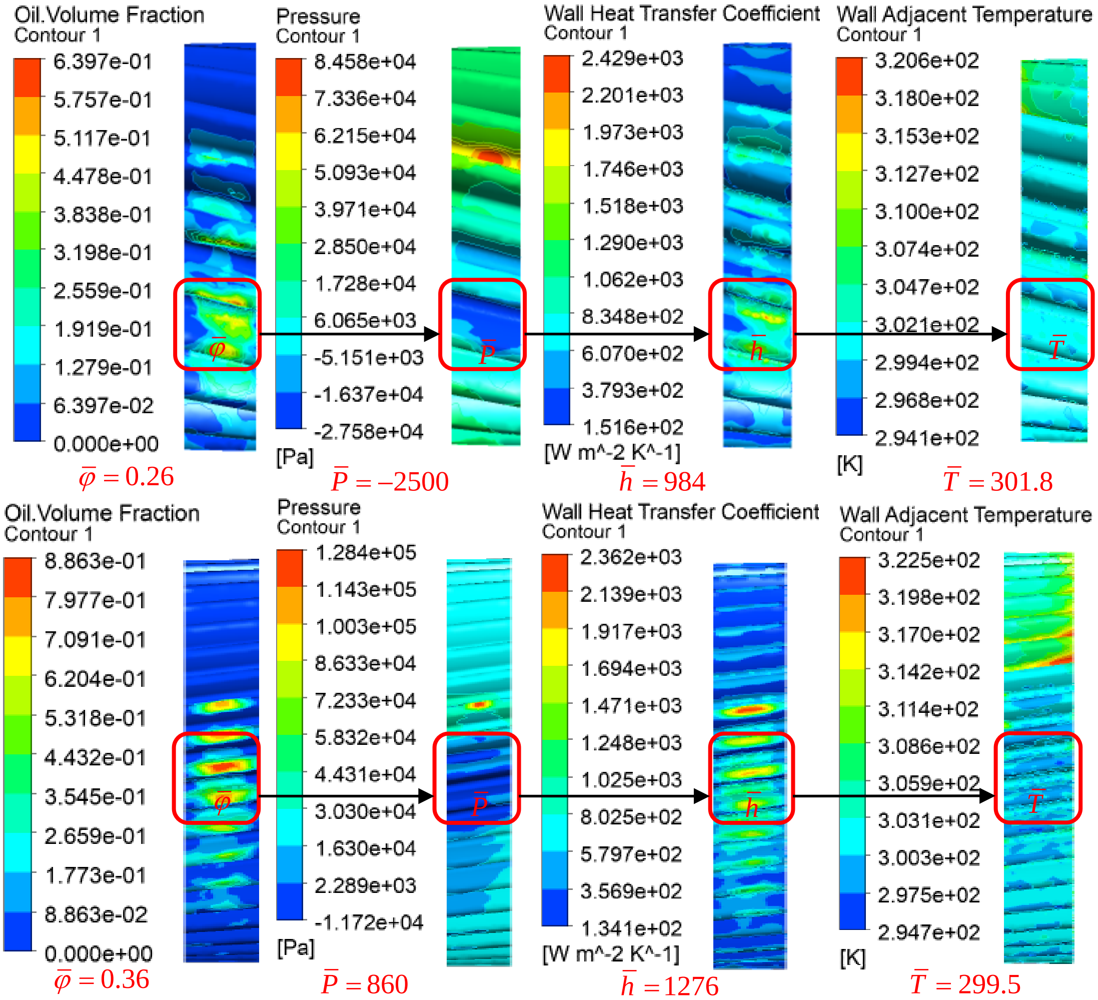

Fig. 8 Flow and temperature characteristics on the tooth surface, (a) Oil distribution, (b) Oil pressure distribution, (c) Wall heat transfer coefficient distribution, (d) Temperature distribution (The upper row is the driving wheel, and the lower row is the driven wheel)

3.3. Experimental verification

To validate the simulation results, an oil injection lubrication test was performed, measuring the gear tooth temperature using an infrared thermal imager. The test aimed to assess the effectiveness of the injection parameters and optimize the heat dissipation performance. A thermal imaging camera (SC7700M, FLIR Systems, Oregon, USA) was used to capture the heat dissipation process, and the emissivity of the test gear, determined using the spray-painting method, was set at 0.93. Controlling the gear temperature during the test was essential for accuracy. A thermocouple, connected to a feedback system, monitored and maintained the gear temperature at 323.15 K, while the circumferential speed was fixed at 26.02 m/s.

Figure 9a illustrates the oil injection lubrication test setup, where the gear was subjected to high-pressure oil spray. The heat exchange between the oil and the tooth surface resulted in a noticeable decrease in gear temperature. Thermal images, as shown in Figures 9b and 9c, captured the tooth surface temperature before and after optimizing the injection parameters. The images reveal that the cooling effect is more pronounced on one side of the gear, and better heat dissipation is observed on both sides of the tooth compared to the addendum and dedendum. This difference is attributed to the combined effects of centrifugal and axial forces on the gear.

The average tooth temperature before optimization was 305.47 K, with a temperature difference of 26.78 K.

After optimizing the injection parameters, the average temperature decreased slightly to 304.72 K, while the temperature difference increased to 28.81 K. This indicates that optimizing the injection parameters led to better atomization of the oil on the driving wheel, which was then thrown onto the driven wheel, increasing the amount of oil reaching the tooth surface. As a result, the frequency of heat exchange between the oil and the tooth surface increased, significantly improving the cooling effect.

Figures 9d and 9e compare the simulated and experimental results for the average temperature and temperature difference on the tooth surface. The experimental values were slightly lower than the simulation values, with an error range of 0.67% to 7.28%. The infrared thermal measurements demonstrated a 0.28% decrease in the average temperature and a 6.93% increase in the temperature difference after optimization, validating the accuracy of the simulation model and the reliability of the optimized injection parameters.

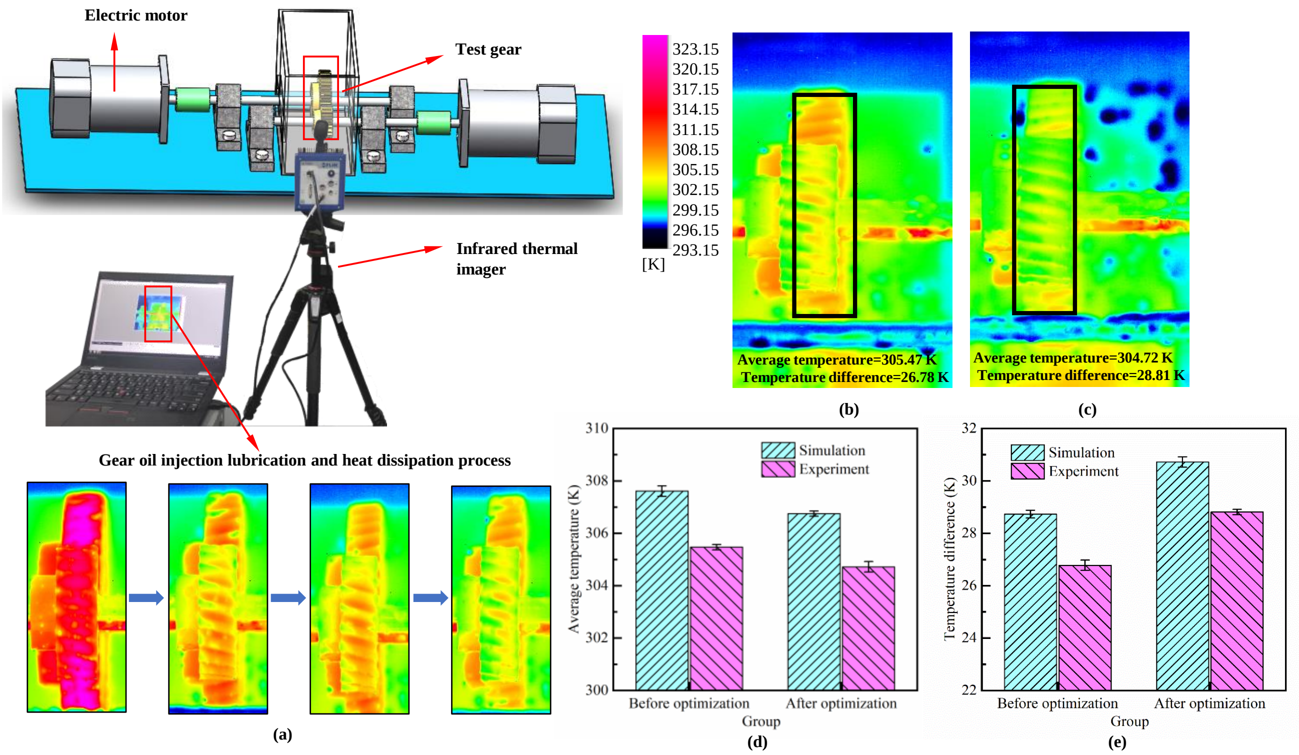

Fig. 9 (a) Oil injection lubrication test, Comparison of average temperature and temperature difference of the tooth surface, (b) Before optimization, (c) After optimization; Tooth temperature in the simulation and experiment, (d) Average temperature, (e) Temperature difference

4. Conclusions

This study investigated the heat dissipation performance of high-speed helical gears using oil jet lubrication through both numerical simulation and experimental validation. The results demonstrated that oil jet lubrication significantly reduces tooth surface temperatures and improves heat dissipation, with optimized injection parameters such as angle, distance, and velocity enhancing the cooling performance. In the numerical simulations, the temperature of the gear surface decreased by approximately 15% when the injection angle was reduced from 30° to 10°, while increasing the injection velocity from 20 m/s to 30 m/s further reduced the gear temperature by 12%. The grid independence verification ensured reliable simulation results with less than 1% variation in average temperature across different grid sizes. Experimental tests, conducted with thermocouples and infrared thermal imaging, showed that the tooth surface temperature decreased from 323.15 K to 304.72 K after optimizing the injection parameters, with a temperature difference of 28.81 K. These experimental results aligned closely with the simulation data, with an error margin ranging from 0.67% to 7.28%. Furthermore, the study found that smaller injection angles, shorter distances, and higher velocities lead to improved convective heat transfer and better overall cooling. The developed heat dissipation model provides an effective tool for optimizing lubrication systems in high-speed gear applications, contributing to greater gear reliability and extended service life. These findings offer valuable insights for the design of more efficient lubrication systems in high-speed mechanical environments.

Acknowledgments

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