Numerical analysis on heat-flow-coupled temperature field for orthogonal face gears with oil–jet lubrication

Bin Ouyang\textsuperscript{a,b}, Feiyue Ma\textsuperscript{a}, Yu Dai\textsuperscript{a} and Yanyang Zhang\textsuperscript{a}

\textsuperscript{a}College of Mechanical and Electrical Engineering, Central South University, Changsha, People’s Republic of China; \textsuperscript{b}AECC Hunan Aviation Powerplant Research Institute, Zhuzhou, People’s Republic of China

ABSTRACT

Lubrication and cooling performances of face gears with oil–jet lubrication are closely related to the nozzle layouts. To investigate the influence of the nozzle layouts on the lubrication performance of the face gears, a numerical analysis method based on heat-flow coupling is adopted in this study for predicting the transient temperature characteristics. First, a mathematical model of impingement depth integrating the nozzle spatial layouts, injection parameters, and geometric parameters of the given face gear is presented for determining the nozzle. Then, after accessing the friction heat, the heat-flow-coupled finite element model is built to resemble the oil–jet lubrication process and obtain the temperature field on the tooth surface. Finally, under the same nozzle layouts, the temperature distributions are compared with the oil pressure and oil volume fraction reported in previous studies. The results reveal that the nozzle layouts significantly affect lubrication performance. Specifically, the lubrication performance of the lateral layout parameter near the middle point of the tooth width is better, and there is a certain range for the longitudinal layout parameter and jet elevation angle. This study can provide a reference for further optimizing the nozzle layouts and prolonging the service life of the orthogonal face gear.

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1. Introduction

Face gear drives have considerable application prospects in the aviation industry owing to their superior comprehensive performance (Dai et al., 2019a, 2019b; Hu et al., 2015; Tang et al., 2013). In complicated working conditions, the friction caused by the tooth surfaces sliding against each other generates a significant amount of heat, which severely reduces the service life of the face gear systems. It is widely accepted that oil–jet lubrication is an extremely effective method to enhance the reliability and prolong the service life of the transmission systems operating in limited working spaces (Massini et al., 2017, jun; Wang et al., 2015; Wang et al., 2018a). One of the prerequisites for achieving this goal is to fully understand the influence of nozzle layouts on the lubrication performance, evaluated by the tooth-surface-temperature distribution based on the heat-flow coupling mechanism.

To explore the impact of nozzle layouts on oil–jet lubrication, a novel mathematical model was introduced to evaluate the lubrication action of the spur gear drive. To verify the accuracy of the mathematical model, the IMPOUT2 computer program was developed to explore the influence of a dimensionless offset on the oil pressure (Akin & Townsend, 1983a, 1983b; Akin & Townsend, 1985). Subsequently, to further demonstrate that the model can be utilized to evaluate the lubrication performance, a finite element approach was employed to simulate the oil–jet lubrication process (Dai et al., 2018). The simulated oil field, oil-gas ratio, and total pressure in the meshing zone were compared with the corresponding oil penetration depth, and the results showed good consistency. The lubricating system of the spur gears was fabricated to obtain the tooth surface temperature to validate the efficiency of the model (Dai et al., 2019a). A novel complex mathematical model was deduced to optimize the nozzle layouts and the oil penetration depth for the face gear train, considering the offsets in lateral, vertical, and longitudinal directions and jet elevation, by applying the Fluent code. Furthermore, various case studies have been conducted using computational fluid dynamics (CFD) methods (Abadi et al., 2020; Faroughi et al., 2020; Mosavi et al., 2019; Ramezanizadeh et al., 2019). The influence of position parameters, such as the injection angle, location, and distance, on the lubrication efficiency of the gears was analysed by calculating the oil-gas ratio and total pressure, which were experimentally verified (Wang et al., 2018b). However, it could be observed that the current research on nozzle layouts...
is mostly focused on the oil-gas ratio and oil pressure: parameters that are not amenable to direct measurement.

The tooth surface temperature is a relatively intuitive factor for characterizing the lubrication performance of the gear system. Generally, the lower the temperature of the gear teeth, the better is the lubrication performance of the gear system. There has been considerable interest in investigating the temperature field during lubrication. Without considering the gear rotation and lubrication, studies have been conducted to investigate the unsteady-state temperature field on a single tooth or several consecutive teeth (Li & Pang, 2018; Li & Tian, 2017; Roda-Casanova & Sanchez-Marín, 2019; Shi et al., 2016; Wang et al., 2017; Wang et al., 2018b). The accuracy of the finite element model plays a decisive role in predicting the temperature field. A finite element thermal model with an integrated power loss model was proposed to analyze the temperature rise and distribution along the tooth surface, and this was verified based on the experimental results reported in the existing literature (Wang et al., 2018b). Similarly, continuous unsteady temperature distributions on the tooth surface with arbitrary torque (Li & Tian, 2017) and arbitrary positions (Shi et al., 2016) were obtained. To observe the temperature field of several continuous teeth engaged in meshing and enhance the thermal characteristics, two-dimensional (Roda-Casanova & Sanchez-Marín, 2019) and three-dimensional (Li & Pang, 2018; Wang et al., 2017) finite element models for thermal analysis were established. However, the complex operating conditions of the gear systems, particularly in the case of oil lubrication, were not considered in the aforementioned studies. Nevertheless, the proposed finite element methods are proven to be valid and useful for studying the temperature field.

On this basis, there has been considerable interest in temperature analyses considering complex lubrication conditions. In the past couple of decades, many studies have been conducted using elasto-hydrodynamic lubrication (EHL) analysis for different types of gears, to evaluate the actions of pressure and temperature on the lubricant, influence of surface roughness on power losses, and impacts of tooth modifications on the EHL analysis (Simon, 1981, 2007, Jun, 2014, 2019). Based on the LTCA and EHL analysis of bevel gears, a thermal analysis was performed to explore the influence of friction heat on the temperature rise (Zhang et al., 2017). To develop an oil–jet lubrication system to effectively cool the gear system, the finite element approach was adopted to perform the thermal analysis of high-speed gears with different lubrication modes and methods (Chen & Liu, 2013, sept; Deshpande et al., 2018). To obtain more accurate thermal-analysis results, a numerical method was proposed by combining the hybrid EHL model with the finite element method (Gan et al., 2019). Novel integrated models combining the mixed EHL theory with the surface-temperature-rise equations were developed to investigate the effects of the dynamic load-patterns on the temperature field and lubrication performance (Dong et al., 2014; Li & Anisetti, 2016; Liu et al., 2013). A thermally starved EHL model was proposed to conduct a thermal analysis and predict the friction coefficient and maximum temperature of the spur gears (Liu et al., 2016). However, for simplicity, these studies did not fully consider the true rotation of the meshing gears. Moreover, experiments were extensively conducted in the research on the thermal behavior of gear systems (Andersson et al., 2017; Chang et al., 2019; Kleemola & Lehtovaara, 2010; Wang et al., 2016; Zhang et al., 2017).

Given the aforementioned literature, studies on the heat-flow-coupled temperature field of face gears with different nozzle layouts are very limited and sparse. To bridge the gap and evaluate the relationship between the nozzle layouts and temperature field of the face gear under oil–jet lubrication conditions, in this study, a mathematical model of the impingement depth of the face gear drive is introduced, integrating the nozzle spatial layouts, injection parameters, and geometric parameters. Then, the generated fiction heat is calculated by determining the sliding friction coefficient, contact stress, and sliding velocity. Additionally, the moving heat source is loaded onto the face gears. Finally, the heat-flow bidirectional coupling model is used to predict the temperature field and explore the influence of different nozzle layouts, by employing the finite element software. The numerical results are validated using the corresponding impingement depth and values reported in the literature.

2. Methodology

2.1. Impingement depth model

In light of the current research, it is not difficult to find that the oil–jet lubrication performance is affected by many factors, such as the nozzle spatial layouts, oil–jet speed, geometric parameters of gear pairs, and swirl effect. For the sake of simplicity, an impingement depth model that integrates the nozzle spatial layouts, spray parameters, and geometric parameters of the face gear pair within it, and one that can be utilized to characterize lubrication performance, has been presented. The greater the impingement depth (defined as shown in Figure 1), the better the lubrication performance, as it has been already proved in the literature (Dai et al., 2019a). Figure 1 illustrates the definitions of the nozzle arrangement position and jet stream. As shown, the spatial position of the nozzle can be determined by the lateral layout.
parameter $x_H$, longitudinal layout parameter $y_L$, vertical layout parameter $z_V$, elevation angle $\theta$, and inclination angle $\phi$.

According to Dai et al. (2019a), the impingement depth on the spur gear tooth in the face gear drive can be represented as

$$
\begin{align*}
\begin{cases}
d_p = \frac{1}{2}N_p m_p + h_{ap} \\
L_p = \frac{y_L}{\sin \theta} + \frac{z_V - H}{\cos \theta} + \frac{v_r (\theta_{p2} - \theta_{p1})}{\omega_p} \\
+ [A_g - z_V - y_L \cot \theta] \cos \theta
\end{cases}
\end{align*}
$$

(1)

where subscripts $p$ and $g$ represent the spur gear and face gear, respectively; $d_p$ is the impact depth; $N_p$ is the number of teeth; $m_p$ is the module; $h_{ap}$ is the addendum; $L_p$ is the impingement distance; $A_g$ is the distance from the spur gear shaft to the lower end surface of the face gear arranged as shown in the figure; $H$ is the distance from the top to the lower end surface of the face gear; $\theta_{p1}$ is the original position parameter; $\theta_{p2}$ is the angle parameter when the pinion has rotated in time $t$; $\omega_p$ is the rotating speed.

### 2.2. Heat generation

When the gear system is working, the friction between the meshing teeth caused by the mutual pushing and sliding generates heat resulting in a temperature rise of the teeth surfaces. The friction between the meshing teeth surfaces is dominated by sliding friction. Owing to the relatively small proportion of the latter two, for simplicity, the generated sliding friction heat is regarded as the heat input to the gear system. The friction heat is mainly delivered in two ways: one is to enter the gear body through heat conduction, and the other is to be consumed through forced heat convection with the lubricating oil. Therefore, calculation of the input friction heat is an important prerequisite for obtaining an accurate temperature field. In this study, the generated heat is the product of the sliding friction coefficient, average contact stress, and relative sliding velocity, as shown in Equation (2).

$$
Q_{\text{sliding}} = \mu_c \sigma_{\text{avg}} |v_1 - v_2|
$$

(2)

Sliding friction coefficient

Research on the sliding friction coefficient of gears has yielded remarkable results. In the case of oil lubrication, the sliding friction coefficient at any contact point can be calculated as

$$
\mu_c = 0.0137 \left( \frac{3.17 \times 10^3 w_{tc}}{\eta(|v_1 - v_2|) v_r^2} \right)
$$

(3)

where $w_{tc}$ is the contact load per unit length, $\eta$ is the lubricant viscosity, and the effect of the pressure and temperature is taken into account according to the Barus law. The Barus viscosity equation is $\eta = \eta_0 \exp(\alpha p - \beta(T - T_0))$, where $\eta_0$ is the oil viscosity at normal temperature and pressure; $\alpha$ and $\beta$ are the pressure-viscosity coefficient and temperature-viscosity coefficient, respectively; $v_1 - v_2$ is the relative sliding speed; subscripts 1 and 2 express the pinion and face gear, respectively; $v_r$ is the winding speed at the contact point.

Equation (3) indicates that the relative sliding speed and winding speed at the contact point of the teeth surface change constantly; therefore, the sliding friction coefficient also changes with time. Moreover, under extremely complicated working conditions, the friction coefficient at the meshing point is also related to the meshing position, surface condition, rotating speed, viscosity of the lubricating oil, friction heat, bearing contact force, etc. Therefore, the friction coefficient is usually recommended by referring to empirical values. For the convenience of computing, the friction coefficient in this study is determined to be 0.05 based on studies by Saribay et al. (2012).

### 2.3. Average contact stress

It is known that the meshing type is point contact. According to the Hertz theory, the distribution of the contact stress in the meshing area would be according to the Ellipse Law. The maximum contact stress in the meshing area can be computed using Equation (4). Additionally, the relationship between the maximum contact stress and the principal curvature can be expressed as Equation (5).

$$
\sigma_{\text{max}} = \frac{3F}{2\pi \rho_x \rho_y}
$$

(4)
\[ \sigma_{\text{max}} = -\frac{K_1^{(1)} + K_2^{(1)} + K_1^{(2)} + K_2^{(2)}}{\pi(\mu)\psi^3(\mu_1^2 - 1/E_1 + \mu_2^2 - 1/E_2)} \]  

(5)

The mean contact stress, \(\sigma_{\text{avg}}\), in the meshing area can be determined as

\[ \sigma_{\text{avg}} = \frac{F}{\pi \rho_x \rho_y} = -\frac{2}{3} \frac{K_1^{(1)} + K_2^{(1)} + K_1^{(2)} + K_2^{(2)}}{\pi(\mu)\psi^3(\mu_1^2 - 1/E_1 + \mu_2^2 - 1/E_2)} \]  

(6)

where \(K_1^{(1)}\) and \(K_2^{(1)}\) are the two principal curvatures at the contact point on the pinion; \(K_1^{(2)}\) and \(K_2^{(2)}\) are the two principal curvatures at the contact point on the face-gear surface; \(u\) and \(v\) are the elliptic integral functions of the pinion and face gear, respectively; \(\mu_1\) and \(\mu_2\) are the Poisson’s ratios of the gears; \(E_1\) and \(E_2\) are the elastic moduli of the gears.

In Equation (6), the two principal curvatures can be calculated using the tooth contact analysis (TCA). The tooth orientation and normal vector of the meshing surface are determined based on the tooth surface equation of the face gear (Litvin et al., 1998, 2001, 2008), and these are computed using Equations (7) and (8):

\[
\tilde{r}(\theta, \phi) = \begin{bmatrix} r_b \cos(q_2 \phi_0) (\sin \phi_0 \mp \theta_1 \cos \phi_0) - \frac{\sin(q_2 \phi_0)}{q_2 \cos \phi_0} \\ -r_b \cos(q_2 \phi_0) \cos \phi_0 \pm \theta_1 \sin \phi_0 \\ -r_b \cos \phi_0 \pm \theta_1 \sin \phi_0 \end{bmatrix}
\]

(7)

\[
\tilde{n}(\theta, \phi_0) = \begin{bmatrix} \mp \cos(q_2 \phi_0) \cos \phi_0 \\ \pm \sin(q_2 \phi_0) \cos \phi_0 \\ \mp \sin \phi_0 \end{bmatrix}
\]

(8)

where \(r_b\) is the base circle radius of the shaper cutter, \(r_b = 0.5mz\cos \alpha_0\), in which \(m\) is the module, \(z\) denotes the gear number, and \(\alpha_0\) represents the pressure angle; \(q_2\) is the transmission ratio; \(\phi_0\) is the rotating angle of the shaper cutter; \(\theta_1\) and \(\theta_0\) are the angle parameters of the involute; additional details can be obtained from (Litvin et al., 1998, 2001, 2008).

### 2.4. Relative sliding velocity

Figure 2 shows the meshing relationship of the established gears, in which the origin is the crossing point of the axes of the gears. In Figure 2, \(M_{10}(x_{10}, y_{10}, z_{10})\) and \(M_{20}(x_{20}, y_{20}, z_{20})\) represent the fixed reference frame of the face gear pair, \(M_1(x_1, y_1, z_1)\) and \(M_2(x_2, y_2, z_2)\) represent the rotating reference frame of the face gear pair, and \(z_1\) and \(z_2\) represent the angle parameters of the face gear pair.

The transfer matrix between the rotating coordinate systems of the pinion and the face gear is as follows:

\[
M_2 \rightarrow M_1 = \begin{bmatrix} \cos \sigma_2 \cos \sigma_1 & -\cos \sigma_2 \sin \sigma_1 & -\sin \sigma_2 & 0 \\ \sin \sigma_2 \cos \sigma_1 & \sin \sigma_2 \sin \sigma_1 & -\cos \sigma_2 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}
\]

(9)

In coordinate system \(M_2(x_2, y_2, z_2)\), it is assumed there is an arbitrary contact point, \(P_2(x_2', y_2', z_2')\), on the face gear surface (see Figure 2). It is generally acknowledged that the speed direction at any point is perpendicular to the unit normal vector at this point on the tooth; therefore, the relative sliding speed can be computed as

\[
v_1 - v_2 = \omega_1 \times r_2 - \omega_2 \times r_2 = (\omega_1 k_1 - \omega_2 k_2) \times r_2
\]

(10)

It is assumed that the contact point, \(P_2(x_2', y_2', z_2')\), in the rotating coordinates, \(M_1\), is \(P_1(x_1', y_1', z_1')\); then, the transfer matrix can be expressed as follows:

\[
(M_2 \rightarrow M_1)((P_2), 1)^T = ([P_1], 1)^T
\]

(11)

Subsequently, the following equation can be derived:

\[
k_1 = -\sin \phi_2 \tilde{j}_2 - \cos \phi_2 \tilde{f}_2
\]

(12)

The relative sliding velocity can be obtained by substituting Equation (12) into Equation (10).

\[
v_1 - v_2 = \begin{bmatrix} -\omega_2 y_2 + \omega_1 z_2 \cos \delta_2 \\ x_2' \omega_2 - \omega_1 z_2 \sin \delta_2 \\ -\omega_1 x_2 \cos \delta_2 + \omega_1 z_2 \sin \delta_2 \end{bmatrix}
\]

(13)

As the rotating speed, body volume, and boundary thermal resistance are different, the calculated friction heat cannot be evenly distributed throughout the entire
face gear drive. If the distribution coefficient in the pin-
onion is $\lambda$, then the distribution coefficient in the face gear
is $1-\lambda$, and $\lambda$ can be calculated using Equation (14):

$$
\lambda = \frac{\sqrt{\beta_1 \rho_1 c_1 v_1}}{\sqrt{\beta_1 \rho_1 c_1 v_1} + \sqrt{\beta_2 \rho_2 c_2 v_2}}
$$

where $\beta_1$ and $\beta_2$ are the thermal conductivity; $\rho_1$ and $\rho_2$
are the densities; $c_1$ and $c_2$ are the specific heat capacities;
$v_1$ and $v_2$ are the tangential speeds at the meshing points.

3. Validation

3.1. Finite element model

Currently, the commercial software ANSYS does not support
the addition of friction heat to the interface. Therefore, in this study, the oil–jet lubrication of a face gear
train was divided into three modules: temperature-field
module, flow-field module, and a heat-flow bidirectional-
coupling module. The temperature-field module simu-
lates the processes of heat addition and heat conduction
inside the gear body, and the flow-field module simu-
lates the oil–jet lubrication process. The main purpose of
the heat-flow-coupling module is to realize real-time data
transmission between the temperature and flow fields
and to simulate the heat convection between the fric-
tion heat and lubricating oil, the principle of which is
presented in Figure 3.

The premise to ensure the authenticity of the numerical
results is to establish the appropriate geometric mod-
els and mesh models, in which the finite element models
established by the temperature – and flow-field modules
are as follows.

(1) Temperature-field module

The temperature-field module performs the simula-
tion of the heat transfer process, and the object of heat
addition is the tooth surface; thus, the fluid domain needs
to be suppressed in the finite model. To simplify the cal-
culation process, the simulation model is simplified as
follows: (a) ignore the non-engineering structure, includ-
ing the central hole, keyway, and other basic engineering
structures; (b) retain the gear teeth without fillets and
chamfers. Figure 4(a) presents the resulting geometric
model, and the main geometric parameters are speci-
fied in Table 1. After the local mesh refinement approach
was employed on the gear surfaces and the grid inde-
pendence test, the tetrahedral mesh model was achieved
with the minimum size of 0.2 mm, unit size of 8 mm, and
maximum size of 12 mm, as shown in Figure 4(b). The
maximum mesh skewness and mesh quality are 0.94 and
0.24, respectively, which meet the quality requirements of
the CFD simulation.

For the high-speed face gear pair with the oil–jet
lubrication, during operation, each pair of meshing teeth
receives lubrication only once per rotation of the pinion;
thus, the oil-jet lubrication plays an intermittent role for
the face gear drive. Additionally, at any time, the heat of
sliding friction is generated by the directly meshing gear
teeth; therefore, the heat is sequentially loaded according
to the gear tooth surface of the meshing area. The pinion
gear meshes for a period of 0.02 s, and the pinion has a
total of 17 teeth. Therefore, each tooth is evenly divided
into 0.001176 s, and the intermittent heat of the pinion at
different times is shown in Figure 5.

(2) Flow-field module

The flow field module simulates the process of lubri-
cating oil spraying from the nozzle to the meshing area.
As the temperature module already exists alone, this
module only retains the flow field calculation domain by
inhibiting the solid part of the gear, as shown in Figure
6(a). Similarly, based on the grid independence test, the
fluid domain mesh model is illustrated in Figure 6(b),
where the minimum grid size, unit size, and maximum
grid size are 0.4, 12, and 24 mm, respectively. In partic-
ular, a local grid refinement criterion was employed in
the oil–jet, as shown in Figure 6(c). A prismatic layer
in the mesh near the walls was adopted in this study, as
illustrated in Figure 6(d).

3.2. Boundary conditions

The VOF method was adapted to simulate the oil-air two-
phase flow with an influence coefficient of 0.07 between
Figure 4. Finite models in the temperature simulation module: (a) geometric model with suppressing the fluid domain; (b) mesh model with suppressing the fluid domain.

Table 1. Basic geometry parameters.

| Parameter                          | Value   |
|------------------------------------|---------|
| Modulus                            | 2.5 mm  |
| Pressure angle                      | 25°     |
| Number of teeth (pinion)           | 17      |
| Number of teeth (face-gear)        | 51      |
| Shaft angle                        | 90°     |
| Insiderradius (face-gear)          | 64 mm   |
| Outsideradius (face-gear)          | 82 mm   |

Figure 5. The intermittent heat source of the pinion at different times.

the two phases. Meanwhile, the first and second phases were set as air and lubricating oil, respectively. To obtain the temperature field, energy equations were used, and an RNG turbulence model that could simulate the swirling effect more accurately was applied with a standard wall function. All simulations had \( y^+ \) values between 30 and 50. Three types of materials viz, alloy steel, lubricating oil, and air were created based on their basic attributes, and the main properties of the alloy steel and lubricating oil are summarized in Tables 2 and 3, respectively. The nozzle is regarded as the velocity inlet with the injection velocity of 50 m/s, and the hydraulic diameter is 1.4 mm, which is the diameter of the nozzle. The solid gear coupling interface was set as the system coupling wall using the dynamic mesh. Additionally, the Green Gauss cell-based method, first-order implicit method, and compressive method were employed to solve the gradient, transient formula, and volume fraction, respectively. Furthermore, the standard method was eventually applied to initialize the temperature of the entire fluid domain to 20°C, the pressure of the entire fluid domain to normal atmospheric pressure, and the volume fraction of the lubricating oil to 0.

Furthermore, to ensure the accuracy and credibility of the numerical results a mesh independence analysis was conducted; Table 4 lists the six trials for the total mesh elements. Lubrication performance is closely associated with the oil-gas ratio and total pressure in the region of engagement, so these are regarded as the indicators of the mesh independency analysis and Figure 7 depicts the numerical results changing with the mesh counts. As shown in Figure 7, the oil-gas ratio and total pressure could reach a steady state when the grid number exceeds 3100307. It could be concluded that with the continuous increase in the mesh elements, the fluctuation of the oil-gas ratio and total pressure decreases, implying that the lubrication performance and heat transfer performance change slightly. Therefore, the number of grids in this study was approximately 3.2 million.
Table 2. Material attributes of the alloy steel created in the simulation.

| Parameters                  | Values   |
|-----------------------------|----------|
| Modulus of elasticity       | 210 GPa  |
| Poisson ratio               | 0.3      |
| Coefficient of thermal conductivity | 60.5 W/m·K |
| Specific heat capacity      | 424 J/kg·K |
| Density                     | 7850 kg/m³ |

Table 3. Properties of the lubricating oil created in the simulation.

| Parameters                  | Values   |
|-----------------------------|----------|
| Coefficient of thermal conductivity | 0.1317 W/m°C |
| Specific heat capacity      | 1907 J/kg·°C |
| Density                     | 866.3 kg/m³ |
| Kinematic viscosity         | 73.16 mm²/s |
| Prandtl number              | 917.74   |

3.3. Results and discussions

(1) Lateral layout parameter \( x_H \)

Table 4. Cases of mesh elements.

| No  | Gear grid | Flow field grid | Toal mesh elements |
|-----|-----------|-----------------|--------------------|
| 1   | 270,556   | 553,844         | 824,400            |
| 2   | 406,277   | 785,481         | 1,191,758          |
| 3   | 884,096   | 1,402,760       | 2,286,856          |
| 4   | 1,190,696 | 1,909,611       | 3,100,307          |
| 5   | 1,645,237 | 2,441,267       | 4,086,504          |
| 6   | 2,190,696 | 3,016,876       | 5,207,572          |

From Figure 1 and Equation (1), the spatial position of the nozzle consists of the lateral layout parameter, longitudinal layout parameter, vertical layout parameter, jet elevation angle, and jet inclination angle. Based on existing research (Dai et al., 2019a), only the lateral layout parameter, \( x_H \), longitudinal layout parameter, \( y_L \), and jet elevation angle, \( \theta \), were analysed. Table 5 lists the four groups of the nozzle layouts with different \( x_H \) parameter and the corresponding impingement depth calculated...
by using Figure 1 and the oil-gas ratio and total pressure obtained from the literature (Dai et al., 2019a).

The simulations of the temperature field analysis were conducted according to the nozzle layouts given in Table 1 and the finite element models established in the 'Boundary conditions' subsection. To improve the accuracy of the simulation results, all parameters except the lateral layout parameter, $x_H$, are guaranteed to be consistent, and the temperature characteristics of the face gear drive are finally obtained, as shown in Figure 8. As can be seen, when parameter $x_H$ is close to the edge of the inner radius (shown in Figure 8(a)), the tooth surface in the meshing area is mostly under high temperature, and the temperature near the inner radius is low. This is because the lubricating oil–jet mainly flows outward from the inside of the inner radius, and less oil enters the meshing area for lubrication and cooling. When parameter $x_H$ is near the middle point of the tooth width (see Figure 8(b) and (c)), the temperature on the meshing tooth surface decreases significantly. At this point, the jet stream diffuses in the directions of the inside radius and outside radius; thus, the lubrication and cooling performances are better. However, when parameter $x_H$ nears the outer radius (shown in Figure 8(d)), the temperature at the outer radius location decreases and that at the inner radius location increases significantly compared with those in Figure 8(a)–(c), which is consistent with the above.

To observe the change rule of the temperature more intuitively and then directly judge the lubrication performance under different lateral layout parameters $x_H$, Figure 9 represents the temperature characteristics of the pinion in the meshing area. Additionally, the changing trend of the extreme temperature on the pinion in the region of engagement is shown in Figure 9, and the temperature is compared with the corresponding impingement depth, oil pressure, and oil volume fraction. Combining Figure 9 with Figure 10 and with the increase in the lateral layout parameter it could be observed that the extreme temperature on the pinion tooth surface decreases first and then increases, which are 63.86°C, 57.94°C, 63.11°C, and 62.04°C. This indicates that the lateral layout parameter is not larger or smaller, and the better the lubrication performance, but there is a certain value range, which is inversely proportional to the variations in the penetration depth values, oil-gas ratio, and total pressure in the region of engagement.

(2) Longitudinal layout parameter ($y_L$)

As can be seen from the analysis of the layout parameter $x_H$, when the lateral layout parameter is 71 mm, the
Figure 8. Temperature characteristics of the face gear drive versus the model with different parameter $x_H$: (a) 1; (b) 2; (c) 3; (d) 4.

Figure 9. Temperature field on the tooth of the pinion in the meshing area against the model: (a) 1; (b) 2; (c) 3; (d) 4.
maximum penetration depth, the highest oil-gas ratio, the total pressure, and the lowest tooth surface temperature are obtained, indicating that the lubrication performance is the best at this time. Therefore, a lateral layout parameter of 71 mm should be considered prior to investigating the longitudinal layout parameters. Table 6 lists the nozzle layouts with different parameters $y_L$ as well as the corresponding impingement depth, oil volume fraction, and oil pressure.

Accordingly, models with different nozzle layouts listed in Table 6 were developed. Figure 11 illustrates the simulated temperature characteristics of the tooth. Based on Figure 11, it can be concluded that as the longitudinal layout parameter continuously increases the region of the tooth surface under high temperature also increases significantly revealing that the lubrication performance is poor. This is because the nozzle is far away from the tooth surface in the meshing area, the jet stream takes a long time to inject from the nozzle to the tooth surface, and the jet lubrication is intermittent. Therefore, the farther the nozzle is away from the meshing area, the amount of lubricating oil used for lubrication is relatively low. Based on these findings combined with the conclusions obtained from the literature (Dai et al., 2019a), it is known that the closer the nozzle is to the meshing area the better because when the nozzle is too close the oil–jet spray is blocked by the high-speed rotating teeth, resulting in less oil in the region of engagement and poor performance.

Figures 12 and 13 present the temperature characteristics of the tooth, viz., the correlation between the extreme temperatures and the impingement depth, oil-gas ratio, and total pressure. The figures show that with the increase in the longitudinal layout parameter, the extreme temperature on the tooth surface increases, which are 46.65°C, 53.69°C, 57.68°C, and 57.94°C, indicating that the lubrication performance decreases within the scope. This is because the amount of lubricating oil used for lubrication in the region of engagement is reduced simultaneously, demonstrating the availability of the temperature analysis method. In addition, a comparison of the penetration depth, oil-gas ratio, and total pressure validates this conclusion.

(3) Jet elevation angle ($\theta$)

Based on the analyses of the layout parameters, $x_H$ and $y_L$, Table 7 presents the different nozzle layouts in terms of the lateral layout parameter of 71 mm and a longitudinal layout parameter of 40 mm.

The simulated temperature characteristics of the tooth under different jet elevation angles are shown in Figure 14. With an increase in parameter $\theta$ in the range of 103° to

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**Figure 10.** Comparison between the extreme temperature and the corresponding impingement depth, oil-gas ratio and total pressure.

**Table 6.** Nozzle layouts with different parameters $y_L$ and the impingement depth, oil volume fraction and oil pressure.

| Model | $x_H$ (mm) | $y_L$ (mm) | $z_V$ (mm) | $\theta$ (°) | Impingement depth (mm) | Oil volume fraction (%) | Oil pressure (Pa) |
|-------|------------|------------|------------|-------------|------------------------|------------------------|-----------------|
| 1     | 71         | 40         | 32         | 105         | 3.92                   | 11.75                  | 9933.81         |
| 2     | 71         | 44         | 32         | 105         | 3.38                   | 10.92                  | 8827.50         |
| 3     | 71         | 47         | 32         | 105         | 2.57                   | 10.41                  | 6855.04         |
| 4     | 71         | 52         | 32         | 105         | 0.89                   | 8.38                   | 4018.80         |
Figure 11. Temperature characteristics of the face gear drive versus models with different parameter $y_L$: (a) 1; (b) 2; (c), (d) 4.

Figure 12. Temperature characteristics of the pinion in the meshing area: (a) 1; (b) 2; (c) 3; (d) 4.
107.5°, the region of the tooth surface under high temperature first decreases and then increases, revealing that the lubrication performance also changes correspondingly. When the jet elevation angle is too large or too small, the lubricating oil is sprayed on the tooth outside the region of engagement, leading to less oil in the meshing area for transferring the frictional heat and poor lubrication performance.

The temperature distribution on the pinion tooth surface and the comparison with the penetration depth, oil-gas ratio, and total pressure are illustrated in Figures 15 and 16, respectively. As shown, with the increase in parameter $\theta$, the extreme temperature first decreases and then increases: 54.41, 46.65, 51.85, and 54.44 °C. The comparison depicted in Figure 16 validates the credibility of the thermal method based on the heat-flow coupling mechanism.

3.4. Experiments of face gears jet lubrication

To further verify the reliability and accuracy of the mathematical model of impact depth and the simulation of the flow field and temperature field, a test rig was set up and several experiments were conducted to verify the theoretical results.

Figure 17 shows the test rig, which consisted of an electric drive motor, load motor, torque sensor, experimental gearbox, and motor control cabinet. Transparent glass was installed on the top of the gearbox to reduce the tooth surface temperature, promote external air heat transfer process and provide space for temperature measurement instruments. Limited by the experimental and processing conditions, the gears cannot be the same as in the simulation. The basic parameters of the face gear are listed in Table 8. To be as close as possible to the model in the simulation, the material of the gears and the parameters of the lubricating oil have been maintained identical to those of the model in the simulation. The temperature was measured using a Fluke Ti32 thermal imaging system (Fluke).

The lubricating oil that lubricates the gears was heated to a specified temperature and the injection speed was set to 50 m/s. The temperature was allowed to stabilize before measurements were recorded using the infrared thermal imager. The temperature distribution of the test pinion surface is shown in Figure 18. After the measurements, the machine was stopped for 120 min, the nozzle layout parameters were adjusted, and 10 groups of experiments were repeated. The tooth surface temperature data are presented in Table 9.

It can be seen from Table 9 that the maximum tooth surface temperature of nozzle No. 5 is the lowest at 32.6 °C. The results show that the nozzle layout parameters, including the transverse offset, longitudinal offset, and
injection elevation angle significantly affect the tooth surface temperature.

According to Figure 19, it can be seen that the temperature variation trend of the pinion tooth surface obtained by this experiment is consistent with that obtained by simulation. There are obvious differences between the test and simulation temperatures under different nozzle parameters because the simulation was conducted considering ideal settings, ignoring the heat generation and transfer process through bearings, shafting, and other accessories, and without accounting for the heat and air transfer outside the box. This study complements the above discussion with a brief analysis for model 5, 8, and 9.

The heat generation, $Q_{\text{bearing}}$, of a rolling bearing is the product of the friction moment and angular speed (Wang et al., 2020):

$$
Q_{\text{bearing}} = \pi M n / 30 \times 10^{-3}
$$

where $n$ is the bearing speed. $M$ denotes the friction torque consisting of two parts: the moment related to lubricant hydrodynamic losses ($M_1$) and the moment related to load under various friction losses ($M_2$), and the
Figure 15. Temperature characteristics of the pinion in the meshing area: (a) 1; (b) 2; (c) 3; (d) 4.

Figure 16. Comparison between the extreme temperature and the corresponding impingement depth, oil gas ratio and total pressure.

Figure 17. Gear lubrication experimental system

Table 8. Basic geometry parameters in the experiment.

| Parameter                        | Value  
|----------------------------------|--------|
| Modulus                         | 6.35 mm |
| Pressure angle                   | 25°    |
| Number of teeth (pinion)         | 17     |
| Number of teeth (face-gear)      | 51     |
| Shaft angle                      | 90°    |
| Insider radius (face-gear)       | 152 mm |
| Outside radius (face-gear)       | 195 mm |

Empirical formulas are as follows:

\[
M_1 = \begin{cases} 
10^{-7} f_0 (v_{0n})^{2/3} d_m^3, & v_{0n} \geq 2000 \\
160 \times 10^{-7} f_0 d_m^3, & v_{0n} < 2000 
\end{cases} \quad (16)
\]

and

\[
M_2 = f_1 P_1 d_m \quad (17)
\]
where $v_0$ is the lubricant kinematic viscosity at the working temperature, $f_0$ is the coefficient related to the bearing type, for oil–jet lubrication in this study, $f_0$ is 0.8, and $d_m$ is the bearing pitch diameter. $P_1$ is the load calculated to determine the bearing friction moment. $f_1$ is the coefficient related to the bearing type and load.

$$f_1 = 0.0009(P_0/C_0)^{0.5} \tag{18}$$

where $C_0$ is the bearing-rated static load.

The size 61804 and 61805 ball bearings were used for the face gear pair, and its main parameters are listed in Table 10.

Bearing heat is mainly dissipated through conduction and convection; the conduction heat between the inner and outer rings and rollers is distributed in a proportion of 1:1 (Burton & Staph, 1967).

The heat generation of oil-seals, $Q_{seals}$, is calculated using Equation (15):

$$Q_{seals} = f_{sl}A_{sl}P_cV_{sl} \tag{19}$$

where $A_{sl}$ is the area of end face seals, $P_c$ is the face pressure, $V_{sl}$ is the mean linear velocity of end face, $f_{sl}$ is the friction coefficient related to friction condition, which is 0.05 in this study. Compared with the gear pair and bearings, the heat from oil-seals was ignored because it was negligible, and for the same reason, heat from other accessories were ignored as well.

Thus, considering the heat transfer of the shaft and air transfer outside the box, the input friction heat loaded on the face gear pair was determined according to Equations (2) and (18). Based on the aforementioned numerical setup, all simulations reloaded the input heat, and the temperature characteristics of the face gear drive were finally obtained, as shown in Figure 20, and Figure 20 also represents the temperature characteristics of the pinion in the meshing area.

After considering the heat generation of the bearings and the heat transfer of the shaft and outside air, the difference between the values of the experiment and simulation analysis (38.2°C, 45.9°C and 43.4°C decreased, is 17.1%, 13.8% and 14.8%, respectively. The thermal imaging system (Fluke) was adopted for measuring the steady temperature of gear teeth, this has led to several degrees lower than true temperature. Therefore, the above results are within the allowable range of error, which verifies the accuracy and reliability of the simulation analysis method of the heat flow coupled temperature field.

Table 10. Main parameters of ball bearings.

| Parameters          | Out diameter/mm | Inner diameter/mm | Number of balls | Width/mm |
|---------------------|-----------------|-------------------|-----------------|----------|
| Ball bearing 1      | 32              | 20                | 13              | 7        |
| Ball bearing 2      | 37              | 25                | 15              | 7        |
Figure 20. Temperature characteristics of the face gear drive: (a) model 5; (b) model 8; (c) model 9.
4. Conclusions

To improve the lubrication performance and prolong the useful life of the orthogonal face gear, the transient temperature-field characteristics were investigated by considering different oil–jet lubrication nozzle layouts using the ANSYS software, and the results were validated by comparing with the values reported in previous studies. The main conclusions of this study are as follows:

(1) The gear temperature differences reveal that the nozzle layouts, specifically the lateral layout parameter, longitudinal layout parameter, and jet elevation angle, have a significant influence on the gear temperature field, which represents the lubrication performance of the given face gear under this nozzle.

(2) When the lateral layout parameter approaches the midpoint of the tooth, the extreme temperature is lower, indicating a better lubrication performance. For the longitudinal layout parameter and injection elevation angle, it is not as if bigger is better, but there is a reasonable range. This study provides a simulated method for research on the temperature characteristics of a gearing system and reasonable reference information on nozzle layouts for further studies.

(3) A comparison of the extreme temperature and the corresponding impingement depth, oil-gas ratio, and total pressure obtained from the existing literature for evaluating the lubrication performance with different layouts indicates that the simulation analysis method on the transient temperature field is reliable.

However, there are some limitations in this study, such as the rough friction and average contact stress, which are closely related to the generated friction heat. In future work, a more accurate prediction of the friction coefficient and average contact stress combined with TCA and EHL analysis will be studied in detail.

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No potential conflict of interest was reported by the author(s).

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