Research Article

Analysis of Flow Characteristics of Straight Conjugate Crescent Gear Pump at Variable Working Conditions

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The working conditions (i.e., gas content, working pressure, and working temperature) of oil in the actual hydraulic system are always in a dynamic process, which will cause the basic properties of oil to change and further generate changes of the flow characteristics in the straight conjugate crescent gear pump (SCCGP). However, no clear explanation has been given for the variation of the flow characteristics of the SCCGP with the working conditions in previous studies. This paper first expounds the gear pair profile equations used to draw the tooth profile curves and the equation applicable to the calculation of the geometric displacement at the given geometric parameters. Subsequently, the mathematical models of the oil’s basic properties are analyzed, and it is explained how the changing working conditions affect the changes in oil properties. After that, the paper designs an orthogonal test scheme and applies it to the study of the flow characteristics of the SCCGP for the first time to optimize its working conditions. Then, the paper performs simulations in commercial CFD numerical software, and the change laws of the internal flow field, flow pulsation, pressure pulsation, volumetric efficiency, and total efficiency of the SCCGP at different working conditions are analyzed in detail. The working conditions for decreasing pulse rate and improving total efficiency are summarized. Finally, for model validation purposes, the paper elucidates the construction process of a high-pressure positive displacement pump test system and carried out experimental research. The simulation and actual measurement results are compared under the same working conditions. The results validate the accuracy of the simulation model and the potential for future analytical studies.

1. Introduction

The oil is used for energy and signal transmission by the hydraulic system. Hydraulic oil is equivalent to the blood of the hydraulic system. It communicates the various components in the system as an organic whole. Its basic properties (i.e., density, absolute viscosity, effective bulk modulus) have a significant impact on the performance of the system. Due to the complexities of the working environment, load change, and itself structure, the gas content, the working pressure, and the working temperature of oil in the actual hydraulic system are always in a dynamic process, which will cause the basic properties of oil to change.

As the power element of the hydraulic system, the working characteristics of positive displacement pumps determine the dynamic behavior of the whole system. Internal gear pumps have the characteristics of high reliability, high energy density, and high pollution resistance, so they are widely used in construction machinery, injection molding machinery, metallurgical machinery, and other fields.

As a new type of internal gear pump, the straight conjugate crescent gear pump (SCCGP) was developed after the involute and cycloid internal gear pumps. Its moving parts consist of an external gear with a straight tooth profile and an internal gear ring with a high-order arc tooth profile [1]. For the research on the crescent gear pumps, Rundo and Corvaglia divided the fluid domain in the pump into several control bodies, and then, the lumped parameter models of the pump were established in the commercial software LMS.
Amesim according to the connection between the control bodies [2]. Later, Rundo [3] published a review that compared two models for simulating outlet flow (i.e., lumped parameter model and distributed parameter model). In addition, Chen et al. [4] established a simplified gear pump Amesim simulation model to study flow pulsation. The above studies did not consider the changes in the basic properties of the oil.

The research on the SCCGP mainly focuses on the following aspects: the tooth profile shape of the gear pair reflects the transmission performance of the gear pump. Wei et al. constructed a general mathematical model of the overall tooth profile with the tooth thickness coefficient as a parameter and proposed a solution to the interference between the dedendum and the addendum [5]. Yang and Bai derived the straight conjugate tooth profile equations based on the meshing angle function [6]. Hu et al. optimized the basic parameters of gear pair with the flow unevenness coefficient as objective function and the change of the gear pair volume, displacement, and oil trapped volume as constraint function [7]. Gu et al. optimized four parameters of modulus, teeth number, tooth profile half angle, and addendum height coefficient based on a genetic algorithm with the flow pulsation rate as an objective function [8].

Flow pulsation is an inherent property of positive displacement hydraulic pumps, which is a deleterious feature. The pressure pulsation caused by it not only damages weak parts but also causes system vibration and various noises [9, 10]. Cui and Qin first used the gear meshing principle to derive the instantaneous flow equation. However, the study did not determine the geometric flow pulsation rate equation [11]. Later, Xu and Song [12], and Wang et al. [13] derived the approximate equation of geometric flow pulsation rate based on the meshing transmission of gear pair and calculated the pulsation rate at given parameters. The theoretical value is much lower than the actual value without considering the internal leakage and the change of oil property. In order to accurately procure the actual value of the flow pulsation rate, the author’s research team analyzed in detail the main factors affecting the instantaneous flow pulsation of the outlet based on the combination of numerical calculation and experimental verification and determined the primary and secondary relationship of each factor [14].

The coincidence degree of gear transmission not only affects the smoothness of gear pair operation but also causes an oil trapped phenomenon. Gao et al. [15] and Duan et al. [16] deduced the calculation formula of coincidence degree according to the gear meshing principle and analyzed the influence of the single basic parameters of gear pair on coincidence degree. Zhang et al. deduced the formula of oil trapped volume change based on the meshing angle function and procured the value range of the geometric parameters of gear pair with the goal of weakening the oil trapped phenomenon [17]. The overlapping interference of the tooth profile affects the normal transmission of gear pair, especially in the internal meshing gear pair with less tooth difference. Zhang et al. determined the limit meshing point of the tooth profile by solving the extreme value of the radius equation of the tooth profile meshing point, thereby deduced the conditions of tooth profile overlap interference [18]. Li and Cui analyzed the three main types of interference in detail and determined the constraint conditions of the noninterference calibration problem [19]. The level of meshing efficiency directly affects the quality of gear transmission. Wang et al. deduced the meshing efficiency function on the basis of instantaneous efficiency and procured the average meshing efficiency formula by integrating it in the meshing interval [20].

From the brief summary of the research status provided above, it can be concluded that the current research on SCCGP is concentrated on the theoretical analysis stage and does not explicate the influence of fluid-particle behavior on the flow characteristics of the SCCGP at variable working conditions. In view of this, the orthogonal test is applied to the study of the flow characteristics of the SCCGP for the first time in this paper to optimize its working conditions. With this goal, first of all, this paper proposes a tooth profile normal method for solving the tooth profile equations and utilizes the volume change method to determine the geometric displacement equation (Section 2). Subsequently, the paper reviews the mathematical models between the basic properties of oil and working conditions and analyzes in detail the change laws of oil density, absolute viscosity, and effective bulk modulus with working conditions. Then, the orthogonal test scheme is designed in the paper. The modeling process of the three-dimensional fluid finite element model is elaborated, and the medium properties of oil at different working conditions are calculated. On the basis of the orthogonal test, the numerical calculations are carried out by commercial CFD software (Section 3). After that, the paper comprehensively analyzes the change laws of internal flow field, flow pulsation, pressure pulsation, volumetric efficiency, and total efficiency of the SCCGP under different working conditions, and the working conditions for decreasing the pulsation rate and improving the total efficiency are summarized (Section 4). Finally, the paper introduces the construction process of a high-pressure positive displacement pump test system and conducts experimental activities. In particular, the experimental and simulation results are compared, and the validation results are shown in Section 5.

2. SCCGP Modeling: Geometric Modeling

2.1. Tooth Profile Equation of External Gear. In the SCCGP, the tooth profile of the external gear is a symmetrical straight line. As shown in Figure 1, the rectangular coordinate system $x, y$ is established based on the external gear center $o_x$ as the coordinate origin. Assuming that the left side tooth profile equation of the external gear is

\[ y = kx + b. \]

According to the geometric relationship in Figure 1, the straight tooth profile equation can be obtained by substituting the two points $(-r_1 \sin \theta, -r_1 \cos \theta)$ and $(0, r_1 \cos \theta + r_1 \sin \theta \cot \beta)$ on the left tooth profile into equation (1).
y_1 = x_1 \cot \beta + r_1 \cos \theta + r_1 \sin \theta \cot \beta. \quad (2)

2.2. Tooth Profile Equation of Internal Gear Ring. The tooth profile curves of the internal gear ring and external gear are a pair of plane meshing conjugate curves. The tooth profile equation of the internal gear ring can be solved by the tooth profile normal method. As shown in Figure 2, o_1 and o_2 are the centers of the external gear and the internal gear ring, respectively, and the rectangular coordinate systems xoy, x_1o_1y_1, and x_2o_2y_2 are, respectively, fixedly connected to the frame, the external gear, and the internal gear ring. g_1 and g_2 represent straight tooth.

Symbol list

\( g_1 \) and \( g_2 \) represent the tooth profiles of straight line and straight-line conjugate, respectively. \( p_1 \) and \( p_2 \) represent the pitch circles of \( g_1 \) and \( g_2 \), respectively.

Assuming that the external gear rotates counterclockwise, according to the basic theorem of tooth profile meshing, it can be known that

\[ \varphi_1 = \frac{\pi}{2} - (\psi_1 + \gamma_1). \] (3)

Assuming that the coordinate of the intersection point \( p' \) between the normal line of any point \( C_1(x_1, y_1) \) on the tooth profile and the instant center line is \( (X_1, Y_1) \), according to the geometric relationship, it can be procured.

\[ \frac{Y_1 - y_1}{X_1 - x_1} = - \cot y_1. \] (4)

\[ r_1 \cos \psi_1 = X_1 \cos y_1 + Y_1 \sin y_1. \]

Thus, the relationship between the contact point \( C_1 \) and the external gear angle \( \varphi_1 \) can be procured.

\[ \begin{aligned}
\cos \psi_1 & = \frac{x_1 \cos y_1 + y_1 \sin y_1}{r_1} \\
\varphi_1 & = \frac{\pi}{2} - (y_1 + \psi_1).
\end{aligned} \] (5)

Equation (5) is the condition that the point \( C_1 \) on the external tooth profile becomes the meshing point. Since \( C_1 \) is arbitrary, equation (5) is equivalent to the meshing line equation. The relationship diagram of the point \( C_1 \) meshing is established, as shown in Figure 3.

In Figure 3, the coordinates \((x_2, y_2)\) of the meshing point \( C_1 \) in coordinate system \( x_2o_2y_2 \) can be procured by coordinate transformation.

\[ \begin{aligned}
x_2 = x_1 \cos (\varphi_1 - \varphi_2) - y_1 \sin (\varphi_1 - \varphi_2) + a \sin \varphi_2, \\
y_2 = x_1 \sin (\varphi_1 - \varphi_2) + y_1 \cos (\varphi_1 - \varphi_2) + a \cos \varphi_2.
\end{aligned} \] (6)

Furthermore, the coordinates \((x_0, y_0)\) of the meshing point \( C_1 \) in the coordinate system \( xoy \) can be procured.
\[ \begin{align*} 
  x_0 &= x_1 \cos \varphi_1 - y_1 \sin \varphi_1, \\
  y_0 &= x_1 \sin \varphi_1 + y_1 \cos \varphi_1 + a. 
\end{align*} \]  

(7)

Equation (7) is the meshing line equation of the conjugate tooth profile procured by using the tooth profile normal method.

2.3. Tooth Profile Curves Drawing of the Gear Pair. The gear pair parameters are listed, as shown in Table 1.

The range of independent variable \( x_1 \) is determined according to Table 1, and the parameters of Table 1 are substituted into equations (2) and (6). The fitting curves of the straight tooth profile and straight conjugate tooth profile are drawn by the MATLAB program, as shown in Figure 4.

Figure 4 shows that the straight conjugate tooth profile is a slightly concave high-order arc curve. Finally, the fitting curve is mirrored and arrayed to procure the tooth profile curves of the gear pair as shown in Figure 5.

2.4. Geometric Displacement Equation. The volume change method is utilized in this part to describe the geometric displacement of the SCCGP. The working principle diagram of the gear pump is established, as shown in Figure 6. The oil in the oil-discharge chamber is compressed by the full tooth profiles \( mn \) and \( m'n' \) to make its volume smaller. In the meantime, due to the continuous rotation of the other meshing tooth profiles \( gk \) and \( g'k \), the volume of oil-discharge chamber is enlarged. The rest of the tooth profiles that are completely surrounded by high-pressure oil does not participate in the work and has no effect on oil-discharge.

The reduced volume of oil-discharge chamber caused by the full tooth profile \( mn \) and \( m'n' \) is

\[ \Delta V_1 = \frac{B}{2} \left[ (r_{a1}^2 - r_{j1}^2) \Delta \varphi_1 + (r_{j2}^2 - r_{a2}^2) \Delta \varphi_2 \right]. \]  

(8)

Meanwhile, the enlarged volume of the oil-discharge chamber caused by the meshing tooth profile \( gk \) and \( g'k \) is

\[ \Delta V_2 = \frac{B}{2} \left[ (\rho_1^2 - r_{j1}^2) \Delta \varphi_1 + (r_{j2}^2 - \rho_2^2) \Delta \varphi_2 \right]. \]  

(9)

The oil-discharge volume of a single pair of teeth can be procured by subtracting equations (8) and (9).

\[ \Delta V = \frac{B}{2} \left[ (r_{a1}^2 - \rho_1^2) \Delta \varphi_1 + (\rho_2^2 - r_{a2}^2) \Delta \varphi_2 \right]. \]  

(10)

The distance from the gear center to the meshing point can be described by the equation of the change of external gear angle. The schematic diagram of the meshing point changing with external gear rotation angle is established, as shown in Figure 7.

From the geometric relationship in Figure 7, the equation of \( \rho_1 \) and \( \rho_2 \) about the external gear rotation angle can be procured.

\[ \begin{aligned} 
  \rho_1^2 &= l^2 + r_1^2 \cos^2 (\beta - \varphi), \\
  \rho_2^2 &= [a \sin (\beta - \varphi) + l]^2 + r_2^2 \cos^2 (\beta - \varphi). 
\end{aligned} \]  

(11)

According to the meshing process of gear pair [21], the starting point of entering meshing is the intersection of the internal gear ring addendum circle and the meshing line, and this corresponds to the initial meshing angle \( \delta_0 \). When the meshing is carried out to the intersection of the external gear addendum circle and the meshing line, the two gears are about to exit meshing, and this corresponds to the end
meshing angle \( \delta_1 \). Based on this, equation (12) can be procured.

\[
\delta_0 = \beta - \arcsin \left( \frac{al + \sqrt{a^2l^2 - (r_{12}r_1 - a^2)(r_{12}^2 - l^2)}}{r_{12}^2 - a^2} \right)
\]

\[
\delta_1 = -\arccos \left( \frac{\sqrt{r_{12}^2 - l^2}}{r_1} \right)
\]

(12)

\[
V = \frac{BZ_1}{8} \left\{ \left( r_1^2 + r_{12}^{-1}a_1^2 - i_{12}r_1^2 \right) \left[ 2(\delta_1 - \delta_0) - \sin 2(\delta_1 - \beta) + \sin 2(\delta_0 - \beta) \right] + 8r_{12}^{-1}al \left[ \cos(\delta_1 - \beta) - \cos(\delta_0 - \beta) \right] + \cdots 
+ 4 \left( r_{a1}^2 + i_{12}r_1^2 + r_{12}^{-1}l^2 - r_{12}^{-1}r_{a2}^2 - r_1^2 - l^2 \right)(\delta_1 - \delta_0) \right\}
\]

(13)

Assuming

\[
K_{v1} = 2(\delta_1 - \delta_0) - \sin 2(\delta_1 - \beta) + \sin 2(\delta_0 - \beta),
\]

\[
K_{v2} = \cos(\delta_1 - \beta) - \cos(\delta_0 - \beta), \quad \text{and} \quad K_{v3} = \delta_1 - \delta_0,
\]

then equation (13) can be expressed as

\[
V = \frac{BZ_1}{8} \left[ K_{v1}(r_1^2 + i_{12}a_1^2 - i_{12}r_1^2) + 8K_{v2}r_{12}^{-1}al 
+ 4K_{v3}(r_{a1}^2 + i_{12}r_1^2 + r_{12}^{-1}l^2 - r_{12}^{-1}r_{a2}^2 - r_1^2 - l^2) \right].
\]

(14)

By using equation (14) and combining the gear pair parameters, the geometric displacement is 51.1 cc/r.

3. SCCGP Modeling: Fluid Dynamics Model

3.1. Mathematical Model of Basic Properties of Oil. The three basic properties affecting the dynamic characteristics of oil are density, effective bulk modulus, and viscosity, respectively. A certain amount of air is inevitably dissolved and mixed in the oil. With the change in working pressure and temperature, the basic properties of the oil will change.

3.1.1. Density of Gas-Containing Oil. Dissolved gas is separated when working pressure \( p \) is lower than air separation pressure. Therefore, there are generally three components in oil: pure oil, air, and oil vapor.

The density \( \rho \) of the oil-gas two-phase mixture can refer to the following equation:

\[
\frac{1}{\rho} = \frac{\alpha_a}{\rho_a} + \frac{\alpha_o}{\rho_o} + \frac{1 - \alpha_a - \alpha_o}{\rho_v}.
\]

(15)

The densities of gas-phase components can be procured according to the gas state equation.
\[\rho_a = \rho_{oil}\left(\frac{P}{P_0}\right)^{1/3}\]

\[\rho_v = \rho_{oil}\left(\frac{P}{P_v}\right)^{1/3}\] (16)

The functional relationship between density, working pressure, and working temperature can be procured according to the Bode state equation [22].

\[\rho = \rho_b (1 - \zeta T)\]

\[\frac{1}{1 - \alpha_p + A}\log\left(1 + p - \frac{\rho_{oos}/B_1 + B_2 T + B_3 T^2 + B_4 T^3}{\rho - \rho_{oos}}\right)\] (17)

where \(\rho\) is the oil density (kg/m\(^3\)); \(\rho_b\) is the density at standard state; \(\zeta\) is the volume expansion coefficient; \(A, B_1, B_2, B_3, B_4\) are parameters related to density.

According to the measured data, the maximum oil temperature of general hydraulic systems at stable working conditions is 60°C, and the maximum gas content of oil under normal conditions is 1% [23]. Combined with the rated pressure (125 bar) of the gear pump, the variation curves of the oil density with working pressure (0–125 bar) and working temperature (0–60°C) at different gas contents (i.e., 0.1%, 0.5%, and 1%) can be procured from equation (19), as shown in Figure 8.

Figure 8 shows that with the increase in working pressure and working temperature, the density first increased from 845.4 kg/m\(^3\) to 849.3 kg/m\(^3\) and then continued to decrease until 842.2 kg/m\(^3\). The gas contents basically have no effect on the oil density.

3.1.2. Viscosity of Gas-Containing Oil. The viscosity \(\mu\) of the oil-gas two-phase mixture can refer to the following equation:

\[\mu = \alpha_o \mu_o + \alpha_g \mu_g + (1 - \alpha_o - \alpha_g)\mu_v\] (18)

The relationship between oil viscosity, working pressure, and working temperature can be procured from the literature [24].

\[\mu = 0.0457 \exp\left\{6.58 \left[1 + 5.1 \times 10^{-9} P\right]^{3.3 \times 10^{-9} \left(T - 138\right)^{1.16}} \right\}.\] (19)

The effective bulk modulus of the oil-gas two-phase mixture is expressed as

\[E = \frac{V_o + V_g + V_v}{dV_o/dp + dV_g/dp + dV_v/dp}\] (21)

According to the definition of derivative, the above equation can be transformed.

\[E = \frac{1}{\alpha_o/\lambda p + \alpha_g/\lambda p + 1 - \alpha_o - \alpha_g / E_o} \] (22)

The functional relationship of effective bulk modulus \(K_{ef}\) of gas-containing oil with working pressure and temperature can be referred to IFAS model. Kim and Murrenhoff [25] verified the model under low pressure. The results show that the calculated effective bulk modulus is in good agreement with the experimental test curves.

\[K_{ef} = \frac{\left[1 - \alpha\right] + m\left(p - p_0\right)/E_{oil}\right]^{-1/m} + \alpha\left(p_0/p\right)^{1/\lambda} + \alpha\lambda p_0\left(p_0/p\right)^{1+1/\lambda}}{\left[1 - \alpha\right] + m\left(p - p_0\right)/E_{oil}\right]^{-1/m} + \alpha\lambda p_0\left(p_0/p\right)^{1+1/\lambda}}\] (23)

The variation curves of effective bulk modulus of oil with working pressure (0–125 bar) and working temperature (0–60°C) at different gas contents (i.e., 0.1%, 0.5%, and 1%) can be procured from equation (23), as shown in Figure 10.
Figure 8: Density variation curves with working pressure and working temperature.

Figure 9: Absolute viscosity variation curves with working pressure and working temperature.
the effective bulk modulus. The lower the gas content is, the higher the corresponding effective bulk modulus is.

3.2. Orthogonal Test Design. In order to procure sufficient and effective data with as few test times as possible and analyze the test results to elicit reliable conclusions, the orthogonal method is used to design the test process in this study.

3.2.1. Design of Factor Level Table. There are three factors studied in this paper, namely gas content (i.e., free gas) A, working pressure B, and working temperature C. According to the maximum value of the factors mentioned above, it is assumed that the level values of gas content A are 0.1%, 0.5%, and 1%, those of working pressure B are 75 bar, 100 bar, and 125 bar, and those of working temperature C are 40°C, 50°C, and 60°C, respectively. In this way, the factors and the levels under consideration are listed in the form shown in Table 2.

3.2.2. Determination of Test Scheme. Table 2 shows that an equal-level orthogonal table with three factors and three levels needs to be designed. According to the properties of the orthogonal table, the orthogonal table of $L_9(3^3)$ is designed. In this way, nine specific test conditions can be procured, and the corresponding test scheme is shown in Table 3.

3.2.3. Determination of Test Objectives. In order to comprehensively analyze the influence of different working conditions on the performance of the SCCGP, the test targets selected in this study are flow pulsation, pressure pulsation, volumetric efficiency, and total efficiency.

3.3. CFD Simulation Model

3.3.1. Three-Dimensional Fluid Finite Element Model. There are four pairs of friction pairs in the SCCGP, namely, the friction pair between the gear end face and the shell, the friction pair between the addendum and the crescent diaphragm, and the friction pair between the outer wall of the internal ring gear and the shell, the friction pair between the meshing tooth surfaces. The sliding surface of the friction pair is separated by a certain thickness of oil film to achieve fluid lubrication to reduce the friction damage between the parts. In order to ensure sealing and effective transfer force, the oil film thickness between each friction pair is determined according to the oil film theory of hydrostatic support [26]. The oil film thickness of gear end face, addendum, and outer wall of internal gear ring are all set to 30 μm, and the oil film thickness between tooth surfaces is set to 4 μm.

The internal gear pair is the moving part of the gear pump, and through the meshing transmission of the gear pair, the oil transportation function is realized. As shown in Figure 11, the three-dimensional fluid model of the gear pump is established and divided into five parts, namely, inlet
and outlet regions 1 and 2, rotor region 3, and inlet and outlet oil supplement regions 4 and 5. Next, the model is imported into CFD simulation software for meshing, and the divided finite element model is shown in Figure 12.

In Figure 12, 1 is the axial oil film, and 2 is the radial oil film. The computational domain is discretized into a combination of Cartesian grids and structured dynamic grids, and three prismatic boundary layers are used to capture the flow behavior along the wall. MGI technology is used to set the interface between the dynamic and static fluid domains. The mesh quality is aimed at no finely broken surfaces, and the mesh in the rotor region is self-adaptively refined. According to the type of gear pump, the main performance parameters of the pump are listed, as shown in Table 4.

3.3.2. Setting of Simulation Parameters. As one of the three most critical stages in flow field simulation, the parameters of the flow channel model directly affect the speed of the simulation process and the accuracy of the simulation results. Therefore, setting reasonable simulation parameters has far-reaching significance.

(1) Boundary Conditions. According to the working condition of the gear pump (the inlet and outlet pressures are known), the inlet and outlet boundary conditions are set as pressure inlet and pressure outlet, respectively. The specific pressure value is detailed in the orthogonal test scheme described later. The rotation speed of the external gear is set to 2000 r/min, and the direction of rotation is clockwise. According to the definition of the transmission ratio, the rotation speed of the internal gear ring is 1529.4 r/min. The inner and outer cylindrical surfaces of the oil film on the inner wall of the external gear are set as the rotating surface, and the speeds are both 2000 r/min. The inner cylindrical surface of the oil film on the outer wall of the internal ring gear is set as the rotating surface, and the speed is 1529.4 r/min.

(2) Flowing Medium. According to the viscosity range and working conditions of the gear pump oil, 46# mineral oil is selected as the flowing medium. The medium properties of oil at different working conditions are calculated as shown in Table 5.

(3) Turbulence Model. Due to the high speed of the gear pair, and the dual effects of differential pressure flow and shear flow, the oil velocity in the meshing region is very high, and the streamlines are intertwined. In order to simulate this complex flow with the transient flow and curved streamlines, the RNG k − ε turbulence model is used in the rotor region to deal with the streamlines with high strain rate and large bending degree in the internal channel, and the near wall is treated more reasonably.

(4) Cavitation Model. The equilibrium dissolved gas model is selected as the cavitation model. The gas migration is used in the model to determine the mass fraction of noncondensable gas (NCG) dissolved in liquid and assumes that the dissolved gas is in equilibrium. The gas is dissolved according to the balance between the local pressure and the dissolved gas reference pressure.
(5) Number of Time Steps. On the premise of ensuring calculation accuracy and accelerating calculation speed, a variable time step is adopted, and the maximum number of iterations in each time step is set to 100. The number of time steps required for the external gear to rotate one tooth is set to 30. Since the number of external gear teeth is 13, that is, 390 steps are a cycle. According to the external gear speed (2000 r/min), the motion period is 0.03 s.

3.4. Independence Verification of Grid and Convergence Criteria

3.4.1. Grid Independence Verification. It is verified that the increase of grid nodes and grid quality has little effect on the accuracy of the calculation results. The grid model used in this study has good quality, so it is only necessary to verify the influence of the nodes' number on the calculation results. The specific results are shown in Table 6.

Table 6 shows that when the nodes number increases to 1.03539 million, the deviation rate decreases to within 0.1%. Therefore, in order to speed up the calculation speed, the grid model with 1.03539 million nodes is selected for numerical simulation.

3.4.2. Independence Verification of Convergence Criteria. The calculation result basically does not change after the convergence criterion is reduced to a certain value. In this study, the outlet flow rates are calculated when the convergence criteria are 0.1, 0.06, 0.02, 0.01, and 0.001, respectively. The specific results are shown in Table 7.
4. Calculation Results and Analysis

4.1. Internal Flow Field Analysis during the Meshing Process of Gear Pairs

4.1.1. Gas Distribution Region in the Internal Flow Field at the Same Temperature. By comparison, it is found that the gas (including dissolution and blending) in the internal flow channel is all concentrated in the rotor region, so the following gas distribution refers to the rotor region. Since the gas distribution regions in the internal flow field are similar at different temperatures, taking 50°C as an example, the gas distribution diagrams at different pressures are procured after calculation convergence. Taking the working pressure of 75 bar (the gas distribution regions at different pressures are similar) as an example, the gas distribution diagrams of the external gear from entering meshing (time step 1530) to exiting meshing (time step 1560) are procured, as shown in Figure 13.

Figure 13 shows that the gas in the internal flow field is mainly distributed in the trapped oil region, and a small amount of gas is distributed in the gap between the addendum and the crescent diaphragm. With the operation of the gear pair, the gas distribution in the trapped oil region migrates, and the gas content on the external gear tooth surface is significantly higher than that of the internal gear ring tooth surface.

In order to further analyze the gas evolution process inside the gear pump, the pressure distribution contours on the rotor cross section during the gear pair from entering meshing (time step 1530) to exiting meshing (time step 1560) in test 1 (pressure distributions of different tests are similar, taking test 1 as an example) are procured, as shown in Figure 14.

Figure 14 shows that the change of the trapped oil volume in the process of the gear pair from entering meshing to exiting meshing has experienced a process of first decreasing and then increasing, and the corresponding pressure change is first increasing and then decreasing. Compared with Figure 13, the gas phase evolution process in the trapped oil region is from the initial uniform distribution to the dispersed and concentrated distribution, and then to the equilibrium distribution process. At the local low pressure, the gas continuously aggregates and forms unstable growth. With the gradual increase of pressure, the distribution area of the cavitation cloud decreases continuously, and the cavitation intensity decreases persistently. The two figures corroborate each other and further illustrate that the pressure change in the fluid region is the fundamental reason for the gas phase migration in the oil with gas.

4.1.2. Gas Distribution Region in the Internal Flow Field at the Same Pressure. Due to the similarity of the gas distribution regions in the internal flow field at different pressures, taking 100 bar as an example, the gas distribution diagrams at different temperatures are procured after calculation convergence, as shown in Figure 15. In addition, the change of gas distribution during gear meshing is similar to Figure 13(b), which is no longer mentioned here.

Figure 15 shows that the gas distribution in the internal flow field at different temperatures is consistent with different pressures. With the increase in temperature, the gas-containing area in the trapped oil regions gradually expands. The teeth number of gas-containing on the external gear and the internal gear rings is both 2. However, the gas-containing area on the external gear is significantly higher. In addition, with the increase in temperature, the air separation pressure and the saturated vapor pressure of the oil increase significantly, and the figure shows that the cavitation strength is significantly strengthened, and the cavitation distribution area is remarkably expanded.

On the whole, the evolution law of the cavitation flow field shows similar characteristics whether it is at the same temperature or at the same pressure. During the working process of the gear pump, all the gases in the oil are distributed in the rotor area, and the gas distribution area is not fixed. The pressure change in the fluid region is the fundamental reason for the migration of the gas phase in the oil with gas. Therefore, with the passage of time, the gas phase in the oil will evolve, and the specific process is from the initial uniform distribution to the dispersed and concentrated distribution, and then to the process of equilibrium distribution. This evolution law is particularly obvious in the meshing area, which is manifested as the gas phase in the meshing area gradually migrates from the dedendum of the internal gear ring to the addendum of the external gear; at the same time, the gas phase on the adjacent surface gradually migrates from the dedendum of the external gear to the addendum of the internal gear ring.

The cavitation area on the addendum area of the external gear gradually expands to the active tooth surface, and the cavitation intensity gradually increases; until the cavitation area increases to the maximum, the cavitation intensity gradually decreases after reaching the highest; that is, the gas
phase polymerization phenomenon ends. At the same time, the gas phase dissipation area on the dedendum area of the internal ring gear gradually expands to the driven tooth surface and starts to decrease after reaching the maximum; that is, the gas phase dissipation phenomenon ends.

During this period, the cavitation area adjacent to the driven tooth surface starts to expand gradually from the addendum of the internal gear ring, reaches the maximum, and then gradually decays, and finally, the cavitation accumulation phenomenon ends. At the same time, the gas phase dissipation area adjacent to the active tooth surface gradually expands from the dedendum of the external gear. When the gas dissipation area reaches the maximum, the cavitation intensity decreases to the minimum, and the gas dissipation phenomenon ends.

4.2. Comparison of Flow Pulsation at Different Test Conditions. Flow pulsation rate is a key indicator to evaluate the instantaneous flow quality of hydraulic pumps. The instantaneous flow at the outlet of gear pump is always pulsating, which is caused by the periodic change of the meshing point position during the gear pair movement, the uneven internal leakage inside gear pump, the oil

<table>
<thead>
<tr>
<th>Convergence standard</th>
<th>Average outlet flow (L/min)</th>
<th>Deviation rate (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>96.325</td>
<td>0.47</td>
</tr>
<tr>
<td>0.06</td>
<td>96.638</td>
<td>0.15</td>
</tr>
<tr>
<td>0.02</td>
<td>96.706</td>
<td>0.08</td>
</tr>
<tr>
<td>0.01</td>
<td>96.752</td>
<td>0.03</td>
</tr>
<tr>
<td>0.001</td>
<td>96.783</td>
<td>0.01</td>
</tr>
</tbody>
</table>

Figure 13: Gas distribution cloud diagrams in the internal flow field. (a) Gas distribution diagrams at different pressures. (b) Gas distribution diagrams of external gear from entering meshing to exiting meshing at working pressure of 75 bar.
compressibility, and the oil trapped phenomenon. Therefore, the instantaneous flow curves of the SCCGP at different test conditions are procured and compared, as shown in Figure 16.

Figure 16 shows that the instantaneous flow curves of the gear pump in each cycle at different test conditions show continuous and periodic changes, and there are 13 undulating pulsations. The reason is that the oil at the inlet of the pump is transmitted through the meshing transmission of the gear pair, and the gear pair is engaged 13 times in a motion cycle. In addition, the instantaneous flow curves at different test conditions are not the same, which is the result of the comprehensive influence of factors such as the uneven internal leakage, different compression, and oil trapped flow. Then, the pulsation amplitude and the corresponding pulsation rate of instantaneous flow in Figure 16 are procured and plotted as a histogram, as shown in Figure 17.

In Figure 17, the flow pulsation amplitude and the pulsation rate are large as a whole. Due to the little difference among the average flows, the high pulsation amplitude at different test conditions corresponds to the high pulsation rate. The pulsation amplitude and the pulsation rate of test 2 are the smallest, and those of test 7 are the largest. In order to further analyze the influence of factors on the flow pulsation rate and determine the optimal level combination, the orthogonal test analysis table of the flow pulsation rate is constructed, as shown in Table 8.

The relationships between flow pulsation rate and gas content, working pressure, and working temperature in Table 8 are fitted to synthetic curves, as shown in Figure 18. According to the range method, the primary and secondary relationships of the factors affecting the instantaneous flow pulsation rate at outlet in Table 8 are $B > A > C$; that is, the working pressure is the largest factor affecting the pulsation rate, followed by the gas content, and the influence of the working temperature can be ignored.

Figure 18 shows that the flow pulsation rate increases with the increase of gas content and decreases first and then increases slowly with the increase of working pressure. The reason for the former is that with the increase of gas content, combined with Figure 10, the effective bulk modulus of oil decreases significantly, and the compressibility of oil increases. According to Figure 16, the maximum instantaneous flow increases, and the flow pulsation rate increases. The reason for the latter is that with the increase of working pressure, combined with Figure 10, the effective bulk modulus of oil increases slowly, and the internal leakage between the high- and low-pressure cavities inside the gear pump intensifies. When the working pressure is less than 100 bar, the effect of effective bulk modulus on the flow pulsation rate is dominant. When the pressure is greater than 100 bar, the uneven internal leakage is dominant. As a result, the flow pulsation rate decreases first and then increases slowly.
In order to procure a lower flow pulsation rate, the oil with lower gas content is used as much as possible, while maintaining the working pressure at about 100 bar. Furthermore, Table 8 shows that the level combination with the lowest pulsation rate is \(A_1B_3C_1\); namely, the gas content is 0.1%, the working pressure is 100 bar, and the working temperature is 40°C. It has been verified that the flow pulsation rate at this combination is 23.9264%.

4.3. Comparison of Pressure Pulsation at Different Test Conditions. The pulsating flow is output inevitably by gear pump. The pressure pulsation will be stimulated when the flow pulsation encounters the resistance of the system loop. The pulsating pressure not only damages the weak parts in the system but also causes the component vibration and generates fluid noise. Therefore, the instantaneous outlet pressure curves of gear pump at different test conditions are procured and compared, as shown in Figure 19.

Figure 19 shows that the instantaneous pressure curves of the gear pump in each cycle at different test conditions show continuous and periodic changes, and there are 13 fluctuation pulsations. Combined with Figure 16, the trend of the pulsating pressure curves and the corresponding flow curves is completely consistent, and the pressure pulsation is positively correlated with the flow pulsation.

Then, the pulsation amplitude and the corresponding pulsation rate of the instantaneous pressure in Figure 19 are procured and plotted as a histogram, as shown in Figure 20.

In Figure 20, the pressure pulsation amplitude and pulsation rate are small as a whole. Due to the large difference in average pressure, the high pulsation amplitude at different test conditions is not positively correlated with the high pulsation rate. Among them, the pulsation amplitude and the pulsation rate of test 3 are the smallest. The pulsation amplitude of test 9 is the largest, and the pulsation rate of test 4 is the largest. In order to further analyze the influence of factors on the pressure pulsation rate and determine the optimal level combination, the orthogonal test analysis table of pressure pulsation rate is constructed, as shown in Table 9.

The relationships between pressure pulsation rate and gas content, working pressure and working temperature in Table 9 are fitted to synthetic curves, as shown in Figure 21.

Table 9 shows that the primary and secondary relationships of the factors affecting the pressure pulsation rate are \(A > B > C\); that is, the gas content is the largest factor affecting the pulsation rate, followed by the working pressure, and the influence of working temperature can be ignored.

According to Figure 21, in order to procure a lower pressure pulsation rate, the oil with lower gas content is used as much as possible while increasing the working pressure as possible. Furthermore, Table 9 shows that the level combination with the lowest pulsation rate is \(A_1B_3C_2\); namely, the gas content is 0.1%, the working pressure is 125 bar, and the working temperature is 50°C. It has been verified that the pressure pulsation rate at this combination is 0.1412%.

4.4. Comparison of Volumetric Efficiencies and Total Efficiencies at Different Test Conditions. The friction pair in the SCCGP is sealed, lubricated, and transmitted by a fixed small clearance, so the flow loss is unavoidable due to leakage. Meanwhile, the compression loss caused by oil compressibility reduces the outlet flow to varying degrees.

During the movement of gear pair, the friction losses of different degrees are generated between fluid particles, between fluid and parts, and between parts. The product of the friction loss and the flow loss reflects the total efficiency of gear pump. Therefore, the volumetric efficiency and total efficiency of gear pump at different test conditions are procured and plotted as a histogram, as shown in Figure 22.

In Figure 22, the volumetric efficiencies at different test conditions have a large difference, and the total efficiencies
Table 8: Orthogonal test analysis table of flow pulsation rate.

<table>
<thead>
<tr>
<th>Test number</th>
<th>Factor</th>
<th>Test result</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>A</td>
<td>B</td>
</tr>
<tr>
<td>1</td>
<td>0.1</td>
<td>75</td>
</tr>
<tr>
<td>2</td>
<td>0.1</td>
<td>100</td>
</tr>
<tr>
<td>3</td>
<td>0.1</td>
<td>125</td>
</tr>
<tr>
<td>4</td>
<td>0.5</td>
<td>75</td>
</tr>
<tr>
<td>5</td>
<td>0.5</td>
<td>100</td>
</tr>
<tr>
<td>6</td>
<td>0.5</td>
<td>125</td>
</tr>
<tr>
<td>7</td>
<td>1</td>
<td>75</td>
</tr>
<tr>
<td>8</td>
<td>1</td>
<td>100</td>
</tr>
<tr>
<td>9</td>
<td>1</td>
<td>125</td>
</tr>
</tbody>
</table>

k1: 30.3284, 50.9723, 38.0135
k2: 40.1498, 31.1075, 40.6514
k3: 48.4191, 36.8176, 40.2325
R: 18.0907, 19.8648, 2.63796

Figure 18: Relationship between flow pulsation rate and working conditions. (a) Relation with gas content. (b) Relation with working pressure. (c) Relation with working temperature.
have a small difference. High volumetric efficiency is not positively correlated with high total efficiency. The volumetric efficiency of test 1 is the largest, and the total efficiency of test 9 is the highest. In order to further analyze the influence of factors on volumetric efficiency and total efficiency, and determine the optimal level combination, the orthogonal test analysis tables for volumetric efficiency and total efficiency are constructed, respectively, as shown in Tables 10 and 11.

The relationships between volumetric efficiency and gas content, working pressure and working temperature in Table 10 are fitted to synthetic curves, as shown in Figure 23. Table 10 shows that the primary and secondary relationships of the factors affecting the volumetric efficiency are \( C > B > A \); that is, the working temperature is the largest factor affecting the volumetric efficiency, followed by the working pressure, and the influence of gas content is small.

Figure 23 shows that the volumetric efficiency increases slowly with the increase of gas content and then remains basically unchanged. It keeps decreasing with the increase of working pressure and working temperature. The reason for the former is that with the increase of gas content, combined with Figure 10, the effective bulk modulus of oil decreases significantly, and the compressibility of oil increases. According to Figure 16, the maximum value of instantaneous flow increases, and the minimum value basically remains unchanged. In this way, the average flow procured by the instantaneous flow integration increases slowly. The reason for the latter is that with the increase in working
Table 9: Orthogonal test analysis table of pressure pulsation rate.

<table>
<thead>
<tr>
<th>Test number</th>
<th>Factor</th>
<th>Test result</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>A</td>
<td>B</td>
</tr>
<tr>
<td>1</td>
<td>0.1</td>
<td>75</td>
</tr>
<tr>
<td>2</td>
<td>0.1</td>
<td>100</td>
</tr>
<tr>
<td>3</td>
<td>0.1</td>
<td>125</td>
</tr>
<tr>
<td>4</td>
<td>0.5</td>
<td>75</td>
</tr>
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<td>5</td>
<td>0.5</td>
<td>100</td>
</tr>
<tr>
<td>6</td>
<td>0.5</td>
<td>125</td>
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<tr>
<td>7</td>
<td>1</td>
<td>75</td>
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<tr>
<td>8</td>
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<td>100</td>
</tr>
<tr>
<td>9</td>
<td>1</td>
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</tr>
<tr>
<td>k1</td>
<td>0.2426</td>
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</tr>
<tr>
<td>k2</td>
<td>0.5555</td>
<td>0.408</td>
</tr>
<tr>
<td>k3</td>
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<tr>
<td>R</td>
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<td>0.2094</td>
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</table>

Figure 21: Relationship between pressure pulsation rate and working conditions. (a) Relation with gas content. (b) Relation with working pressure. (c) Relation with working temperature.
pressure and working temperature, combined with Figure 9, the absolute viscosity of oil decreases rapidly, and the internal leakage between the high- and low-pressure cavities inside gear pump intensifies.

In order to procure a higher volumetric efficiency, the working temperature is kept as low as possible while reducing the working pressure as much as possible. Furthermore, Table 10 shows that the level combination with the highest volumetric efficiency is \(A_3B_1C_1\); namely, the gas content is 1%, the working pressure is 75 bar, and the working temperature is 40 °C. It has been verified that the volumetric efficiency at this combination is 97.2341%.

The relationships between total efficiency and gas content, working pressure and working temperature in Table 11 are fitted to synthetic curves, as shown in Figure 24.

Table 11 shows that the primary and secondary relationships of the factors affecting the total efficiency are \(A > B > C\); that is, the gas content is the largest factor affecting the total efficiency, followed by the working pressure, and the influence of working temperature is small.

Figure 24 shows the total efficiency increases slowly with the increase of gas content and working pressure, and that increases first and then decreases rapidly with the increase of working temperature. The reason is that it has the same volumetric efficiency with the increase of gas content. According to the definition of total efficiency, although the leakage flow increases with the increase of working pressure, the loss of outlet flow is still lower than the increase of working pressure. As a result, the product of the two still increases slowly. The influence of working temperature on the total efficiency is dual. With the increase in working temperature, combined with Figure 9, the absolute viscosity of oil decreases rapidly. When the viscosity is large, the leakage between the high- and low-pressure cavities is small, but the viscous friction loss between friction pairs is large. When the viscosity is small, the leakage between the high- and

---

**Table 10: Orthogonal test analysis table of volumetric efficiency.**

<table>
<thead>
<tr>
<th>Test number</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>Volumetric efficiency (%)</th>
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<td>75</td>
<td>40</td>
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<td>2</td>
<td>0.1</td>
<td>100</td>
<td>50</td>
<td>93.4002</td>
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<tr>
<td>3</td>
<td>0.1</td>
<td>125</td>
<td>60</td>
<td>87.8714</td>
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<tr>
<td>4</td>
<td>0.5</td>
<td>75</td>
<td>60</td>
<td>94.0668</td>
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<td>5</td>
<td>0.5</td>
<td>100</td>
<td>40</td>
<td>96.6825</td>
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<td>125</td>
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<td>92.7582</td>
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<tr>
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<td>50</td>
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<tr>
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<td>96.6371</td>
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<tr>
<td>k2</td>
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<td>k3</td>
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<td>R</td>
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Table 11: Orthogonal test analysis table of total efficiency.

<table>
<thead>
<tr>
<th>Test number</th>
<th>Factor</th>
<th>Test result Total efficiency (%)</th>
</tr>
</thead>
<tbody>
<tr>
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<td>100 40 75.9422</td>
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<td>75 50 75.6548</td>
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<td>125 40 77.3346</td>
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<tr>
<td>k1</td>
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</tr>
<tr>
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<td></td>
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</tr>
<tr>
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<td></td>
<td>76.3144 75.8205 74.9245</td>
</tr>
<tr>
<td>R</td>
<td>2.3295</td>
<td>1.0161 0.8441</td>
</tr>
</tbody>
</table>

Figure 23: Relationship between volumetric efficiency and working conditions. (a) Relation with gas content. (b) Relation with working pressure. (c) Relation with working temperature.
low-pressure cavities is large, and the viscous friction loss between the friction pairs is small. Therefore, there must be a maximum value of the total efficiency with the change in working temperature, as shown in Figure 24(c), which is 50°C. When the working temperature is lower than 50°C, the viscous friction loss is dominant. When the temperature is higher than 50°C, the internal leakage is dominant.

In order to procure a higher total efficiency, the gas content in the oil should be maintained at about 0.5%, the working pressure should be increased as much as possible, and the oil temperature should be kept at about 50°C. Furthermore, Table 11 shows that the level combination with the highest total efficiency is $A_3B_3C_2$; that is, the gas content is 1%, the working pressure is 125 bar, and the working temperature is 50°C. It has been verified that the total efficiency at this combination is 77.5152%.

4.5. Determination of the Best Working Conditions. In summary, the pressure pulsations under different working conditions are very small. Therefore, the considerations for the four test objectives are focused on flow pulsation rate, volumetric efficiency, and total efficiency. Considering the influence of the level on each target, the best working conditions of the SCCGP can be procured; that is, the values of air content, working pressure, and working temperature are 0.5%, 125 bar, and 50°C, respectively.

5. Experimental Results and Analysis

5.1. Construction of High-Pressure Test Bench. Experiments are the standard for testing the results of numerical calculations. In order to verify the accuracy of the simulation results, a high-pressure (320 bar) positive displacement pump experimental device was designed and constructed. The design principle is shown in Figure 25.

In Figure 25, the electromagnetic spill valve (14) is used as a safety valve, and the proportional relief valve (15) is used...
to change the load in the loop. The liquid level thermometer is used to detect the temperature of the oil in the system, and the torque speed sensor is used to measure the input power of the test pump. The function of the flow meter (17) is to calibrate the measurement result of the flow meter (16). The corresponding sensor information in Figure 25 is listed, as shown in Table 12.

The corresponding test system is constructed according to the design principle. The system is composed of four parts: test bench II, power cabinet I, control cabinet III, and industrial computer IV, as shown in Figure 26. The component numbers in Figure 26 are the same as those in Figure 25. After the test system is started normally, the current or voltage signal is typed into the secondary instrument of the control cabinet, which is converted into the corresponding drive signal through the proportional amplifier to control the action of the corresponding electrical components. The data acquisition software developed by
the researcher team is installed in the industrial computer, which can automatically save all the data detected by the primary instrument.

5.2. Comparison of Experimental and Simulation Results. During the experiment, the hydraulic energy is lost by adjusting the input signal of the proportional relief valve to increase the working temperature. In addition, the operating pressure is set through the electromagnetic spill valve, and the speed of the variable frequency motor is set to 2000 r/min. Because the pulsating frequency ($z_1 n_1 / 60$) of instantaneous flow is very high, the instantaneous flow cannot be measured by the digital flowmeter, and the steady-state flow through the loop can only be measured. The steady-state value detected by the flowmeter within one minute is procured, and the fitted curve is shown in Figure 27.

Figure 27 shows that when the pump shaft speed is set to 2000 r/min, the average flow curve is not a straight line, but an irregularly pulsating curve with an amplitude less than 0.2 L/min. The reasons are many, mainly caused by the control accuracy of the three-phase asynchronous motor.

Next, the output flows under these conditions (the working temperature is 40°C, and the outlet pressures are 75
bar, 100 bar, and 125 bar, respectively; the working temperature is 50 °C, and the outlet pressures are 75 bar, 100 bar, and 125 bar, respectively; the working temperature is 60 °C, and the outlet pressures are 75 bar, 100 bar, and 125 bar, respectively) are measured, respectively, and the corresponding volumetric efficiencies are calculated. In addition, the correlative input powers are measured, respectively, and the homologous total efficiency is calculated. Finally, the experiment and simulation data are compared, and the results are shown in Figures 28–30, respectively.

The above two figures show that at the same working pressure, with the increase in working temperature, the flows of experiment and simulation and the corresponding volumetric efficiencies decrease synchronously. At the same working temperature, with the increase of working pressure, the experiment and simulation data show the same change trend. The experiment and simulation values are very close. The maximum difference of average export flows is about 1.9 L/min, and the maximum difference of volumetric efficiencies is about 1.8%. Overall, the experiment values are slightly lower than the simulation values. The reason is that, first, there are primary and secondary instruments in the experiment, and there are measurement errors; in addition, the speed of the three-phase asynchronous motor is not constant; finally, the oil temperature in the system is not evenly distributed.

Figure 30 shows that the trends of total efficiencies of experiment and simulation are completely consistent. When the working temperatures are 40 °C and 60 °C, respectively, the total efficiencies increase with the increase of working pressure. When the working temperature is 50 °C, the total efficiency increases first and then decreases, and there is a maximum value. The experiment and simulation values increase alternately, and they are very close. The total efficiencies corresponding to the same working pressure and different temperatures do not have a fixed change law. The consistency of the experiment and simulation results verifies the applicability of the simulation model and the correctness of the simulation method.

6. Conclusion

The working conditions of oil are always in a dynamic change process, which will affect the flow characteristics of the SCCGP. In this paper, a tooth profile normal method for solving the tooth profile equations is first proposed, and the volumetric change method is used to determine the geometric displacement equation.

The paper then reviews the mathematical models of the basic properties of oil changing with working conditions. Through analysis, it can be seen that the gas content within 1% has basically no effect on the density and the absolute viscosity of oil. However, with the increase in gas content, the effective bulk modulus decreases significantly. The absolute viscosity decreases rapidly with the increase in working pressure and working temperature, which further confirms that the influence of working temperature on viscosity is much greater than that of working pressure. The effective bulk modulus keeps increasing with the increase of working pressure and working temperature, which verifies that the influence of working pressure on the effective bulk modulus is much greater than that of working temperature. The influence of working pressure and working temperature on density counteracts each other, and the density remains basically unchanged.

In this paper, the orthogonal test scheme is designed and applied it to the study of the flow characteristics of the SCCGP for the first time to optimize its working conditions. The paper elaborates the modeling process of the three-dimensional fluid finite element model and calculates the medium properties of oil under different working conditions. Numerical calculation results show that during the
working process of the gear pump, the gas is mainly distributed in the oil-trapped region and continuously migrates in it, resulting in the gas content of the external gear tooth surface being significantly higher than that of the internal gear ring tooth surface. The trend of the pulsating pressure curve and the flow curve is completely consistent, and the pressure pulsation is positively related to the flow pulsation. Under different combinations of working conditions, the pressure pulsation rate is very low, and the influence of factors on it can be ignored. Flow pulsation rate, volumetric efficiency, and total efficiency are monotonically increasing functions of gas content. The flow pulsation rate decreases first and then increases with the increase of working pressure. There is a minimum value, and the corresponding pressure is about 100 bar. He total efficiency first increases and then decreases with the increase of the working temperature. There is a maximum value, and the corresponding temperature is about 50°C. With he total efficiency increases monotonously with the increase of the working pressure. The influence of working temperature on the flow pulsation rate can be ignored. The volumetric efficiency decreases monotonously with the increase of working pressure and working temperature. The higher the effective bulk modulus is, the lower the uneven internal leakage is, and the lower the flow pulsation rate is.

The optimal working conditions can be procured by integrating various factors; that is, the gas content in the oil should be maintained at about 0.5%. The closer the working pressure is to 125 bar, the better. The oil temperature is maintained at about 50°C as much as possible. At the end of the paper, for the purpose of verifying the modeling, a high-pressure volumetric pump test system was constructed by the author’s research team, and the experimental research was carried out. The simulation data are compared with the experiment value, and the comparison shows the accuracy of the numerical calculation results.

Next, on the basis of this study, the model formulas of the continuous change of the gear pump flow characteristics with working conditions are explored, and they are extended to gear pumps with different profiles to improve the theoretical basis of gear pump research.

### Nomenclature

- $r_1$: Radius of external gear index circle (mm)
- $r_2$: Radius of internal gear ring index circle (mm)
- $r_{a1}$: Radius of external gear addendum circle (mm)
- $r_{a2}$: Radius of internal gear ring addendum circle (mm)
- $r_{j1}$: Radius of external gear dedendum circle (mm)
- $r_{j2}$: Radius of internal gear ring dedendum circle (mm)
- $l$: The vertical distance from the center of external gear to the external gear tooth profile (mm)
- $B$: Tooth width (mm)
- $a$: Center distance (mm)
- $\rho_1$: Distance from the center of external gear to the meshing point (mm)
- $\rho_2$: The distance from the center of internal gear ring to the meshing point (mm)
- $\beta$: Half angle of external gear tooth profile (°)
- $\theta$: Center half angle corresponding to the index arc tooth thickness of external gear (°)
- $\varphi$: Angle between the centerline of tooth thickness of external gear and the y-axis (°)
- $\varphi_1$: Angle between the line from the center of external gear to the point $c_1$ and the y-axis (°)
- $\varphi_2$: Angle between the line from the center of internal gear ring to the meshing point and the y-axis (°)
- $\Delta\varphi_1$: External gear angle (°)
- $\Delta\varphi_2$: Internal gear ring angle (°)
- $\gamma_1$: Angle between the common tangent of tooth profile at point $c_1$ and the $X_1$-axis (°)
- $\Psi_1$: Angle between the common tangent of tooth profile at point $c_1$ and the line from the center of external gear to the point $p_1$ (°)
- $\delta_1$: Initial engagement angle (°)
- $\delta_1'$: End engagement angle (°)
- $Z_1$: Number of external gear teeth
- $Z_2$: Number of internal gear ring teeth
- $\omega$: Angular speed of external gear (rad/s)
- $n_1$: Rotation speed of external gear (r/min)
- $n_2$: Rotation speed of internal gear ring (r/min)
- $i_{12}$: Gear pair transmission ratio
- $k_T$: Temperature-related constant (−8 MPa°C)
- $\rho_1$: Density of air
- $\rho_o$: Density of pure oil
- $\rho_{vo}$: Density of oil vapor
- $\alpha$: Volume fractions of air
- $\alpha_{vo}$: Volume fractions of pure oil
- $p_0$: Standard atmospheric pressure
- $p_{vo}$: Density at pressure $p_0$
- $p_{oi}$: Density of oil vapor at saturated vapor pressure $p_v$
- $p$: Absolute pressure (bar)
- $T$: Thermodynamic temperature (K)
- $p_{o0}$: Operating pressure (bar)
- $\beta$: Volume expansion coefficient (1/°C)
- $\mu$: Viscosity of oil vapor
- $\mu_{oi}$: Viscosity of air
- $\mu_{vo}$: Viscosity of pure oil
- $\mu_o$: Dynamic viscosity corresponding to the working pressure $p$ (Pa.s)
- $V$: Total volume of oil
- $dp/dV$: Change rate of pressure with volume
- $V_o$: Volumes of pure oil
- $V_v$: Volumes of oil vapor
- $V_{oi}$: Volumes of pure oil vapor
- $\lambda$: Variable index
- $E_o$: Effective bulk modulus of pure oil
- $E_v$: Effective bulk modulus of oil vapor
- $E_{oi}$: Effective bulk modulus of pure oil (1700 MPa)
- $E_{oi,0}$: Effective bulk modulus at 0°C and 0 Pa
- $m$: Pressure-related parameter (11.4).

### Data Availability

The numerical and experimental data used to support the findings of this study are included within the article.
Conflicts of Interest
The authors declare that they have no conflicts of interest.

Authors’ Contributions
Guoguo WU was responsible for original draft preparation, investigation, and data curation; Guolai YANG was responsible for supervision and validation; Chuanchuan CAO was responsible for writing, reviewing, and editing the article; and Hongqiang CHAI was responsible for formal analysis and conceptualization.

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