Numerical Analysis of Flow Performance and Design of Gear Pump with Unload Groove

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Abstract. To improve the problem of oil trapping during the operation of external gear pumps, studies on the design of external gear pump teeth and unload groove and the analysis of the effect of their flow characteristics have been carried out. An external gear pump with a new type of unload groove was designed with reference to the general design specification. Based on the computational fluid dynamics (CFD) method, a numerical simulation model of the internal flow field of the gear pump was established, and the simulation analysis of the internal flow characteristics and pressure distribution of the gear pump was carried out to obtain the influence law of unload groove on the performance of the gear pump. The simulation results showed that the unload groove gear pump had a better flow rate, higher capacity efficiency, and less radial pressure than the Default gear pump under the same working conditions and design parameters. The study showed that the external gear pump with unload groove designed in this paper has significantly improved the problems caused by the oil trapping problem compared with the default gear pump.

Keywords: External gear pump; CFD; Unload Groove; Optimization Design

1 Introduction

As design and manufacturing technologies continue to advance, gear pumps have gradually evolved into a significant power equipment in today's context. They are highly regarded for their simplicity in structure, compact size, lightweight, and low sensitivity to contamination, holding a significant share in hydraulic transmission and fluid control technology[1]. They find extensive applications in fields such as marine engineering and shipbuilding[2].

Gear pumps can be classified into two categories: internal meshing gear pumps and external meshing gear pumps. Compared to internal meshing gear pumps, external meshing gear pumps have a simpler structure, smaller size, ease of manufacturing, strong self-priming capability, and lower costs for both maintenance and production. However, external gear pumps still experience issues related to oil trapping[3], resulting in problems such as radial hydraulic imbalance of gears, severe bearing wear, significant flow pulsations, and high noise levels[4]. Therefore, the design and performance analysis of high-performance external meshing gear pumps have remained a focus of research[5].

To address the drawbacks of external meshing gear pumps, Zhang et al.[6] con-ducted simulations of the internal flow field of gear pumps using Fluent software. They identified the characteristics of oil trapping in gear pumps and analyzed the fac-tors influencing it, as well as the issues related to internal leakage and gear stress. Through an analysis of the flow field characteristics of gear pumps, they elucidated the principles, characteristics, affected regions, and causes of oil trapping, providing a theoretical basis for the research and optimization design of external meshing gear pumps. In the effort to improve oil trapping in gear pumps, the design of unloading slots in gear pumps has received extensive research attention[7,8]. These slots, situated on both sides of the gear meshing area, are designed to release the high-pressure oil formed during gear meshing, thereby mitigating the oil trapping phenomenon. Guo et al.[9] obtained the expression for the unloading area using the geometric pattern expansion method, providing a theoretical reference for unloading slot design. Liu et al.[10] conducted comprehensive design of unloading slots and numerical simulation of flow fields, demonstrating that the structurally designed unloading slots can effectively alleviate the oil trapping pressure in the gear meshing area, thus improving the oil trapping issue in gear pumps. It is evident that a well-designed unloading slot has a positive significance in addressing the oil trapping problem in gear pumps[11]. However, traditional unloading slot design methods are largely empirical and can only mitigate the oil trapping phenomenon to some extent. Moreover, the transient oil pressure remains relatively high, and improper design parameters can lead to issues such as reduced flow rates in gear pumps.

This paper aims to propose a design approach for an external gear pump and its unloading grooves based on task requirements. It establishes a numerical simulation model of the internal flow field of the gear pump using CFD methods. The paper conducts simulation analyses on the internal flow characteristics and pressure distribution of different types of gear pumps, and verifies that gear pumps with unloading grooves exhibit superior performance.

2 Gear Pump Model and Parameter Design

2.1 Pump Parameter Design

The research in this paper focuses on external gear pumps. To begin with, it is essential to design the functional parameters of the gear pump according to the task requirements. During the gear pump design process, precise determination of key parameters is imperative, including "required power", "pump shaft input power", "motor output power", "motor speed", "pump drive shaft speed", and "displacement". The accurate determination of these parameters significantly impacts the final performance of the gear pump. The design requirements for the gear pump studied in this paper are as follows: pressure differential $\Delta p = 0.1Mpa$, flow rate $Q = 1000ml/min$, and motor current $I < 1A$.

According to design specifications, the design parameters of the gear pump un-der study are as follows, as shown in Table 1.

Table 1. Gear pump design parameters calculation results.

Parameter name and symbol	Method of calculation	Calculation re-sults
Pump shaft input power P_{shaft}	$\frac{P_{liquid}}{\eta_r \eta_m}$	2.78W
Motor output power P_{motor}	$\frac{P_{shaft}}{\eta_t}$	2.74W
Motor speed n_{motor}	Motor performance curve	7100rpm
Pump drive shaft speed n_{shaft}	$\frac{n_{motor}}{5}$	1420rpm
Displacement q	$\frac{Q_t}{n_{shaft}} = \frac{Q}{\eta_r n_{shaft}}$	0.88ml/r
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Displacement q	$\frac{Q_t}{n_{shaft}} = \frac{Q}{\eta_r n_{shaft}}$	0.88ml/r

In the calculation of the required power P_{liquid}, Δp represents the design pressure differential, and Q represents the design flow rate. In the calculation of the pump shaft input powerP_{shaft}, η_r is the volumetric efficiency, which describes the energy losses due to liquid leakage and compression, and is assumed to be 0.8. η_m is the mechanical efficiency, which describes the friction losses between components or between components and the fluid, and is assumed to be 0.75. In the calculation of motor output power P_{motor}, η_t represents the transmission efficiency of the driving mechanism and is assumed to be 0.9. The calculation of motor speed is based on the motor performance curve, as illustrated in Figure 1. From the graph, the motor operates at the following parameters: output power of 4W, output torque of approximately 5.5 $mN \cdot m$, motor speed of approximately 7100 rpm, and input current of approximately 0.48A, meeting the requirement of a motor operating current below 1A. In the calculation of pump drive shaft speed n_{shaft}, n_{motor} represents the motor speed, and the transmission ratio between the motor and pump shaft is 5:1. The calculation of displacement q is based on the theoretical flow rate of the pump Q_t and the actual flow rate Q.

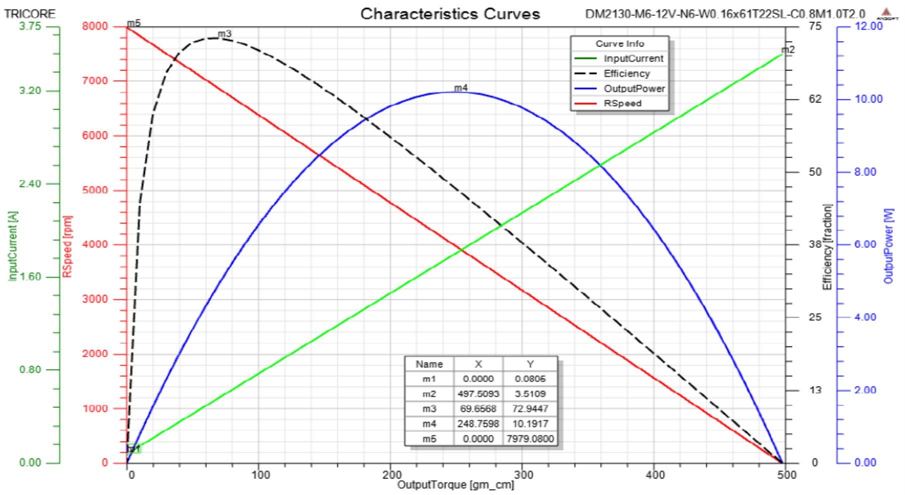


Fig. 1. Motor performance graphs.

After the design is completed, it is necessary to verify the motor's output torque. The input torque of the pump shaft can be calculated as follows:

$$M_{shaft} = \frac{50}{\pi} \Delta pq \frac{1}{\eta_m} \times 10^{-3} = 18.68(mN \cdot m) \tag{1}$$

Where Δp represents the design pressure differential in bars.

This results in the motor's output torque as follows:

$$M_{motor} = \frac{M_{shaft}}{\eta_t} \cdot \frac{1}{5} = \frac{18.68}{0.9 \times 5} = 4.15 < 5.5(mN \cdot m) \tag{2}$$

This complies with the design verification requirements.

2.2 Unload Groove Design

In the design of external gear pumps, to ensure smooth operation, gear overlap generally exceeds 1. This leads to the occurrence of oil trapping during the pump's operation. Oil trapping, during a specific meshing period of the gears, results in the formation of a closed chamber between the two engaging teeth, which shrinks and expands as the gears rotate. When the chamber shrinks, the liquid inside is pressurized, causing a sharp increase in pressure, exerting a significant load on the gear shaft and bearings, leading to gear pump vibration, noise, and reduced operational stability and lifespan. When the chamber expands, the internal pressure drops, creating a vacuum, which can lead to cavitation and vibration noise issues.

In this design, double rectangular unload grooves are symmetrically arranged relative to the gear centerline. The positions of symmetrically arranged double unload grooves should satisfy the following conditions:

(a) When the trapped oil volume begins to decrease, this volume should be connected to the pressure oil chamber.

(b) When the trapped oil volume is at its minimum, it should not be connected to either the pressure oil chamber or the suction oil chamber.

(c) When the trapped oil volume begins to increase, it should be connected to the suction oil chamber.

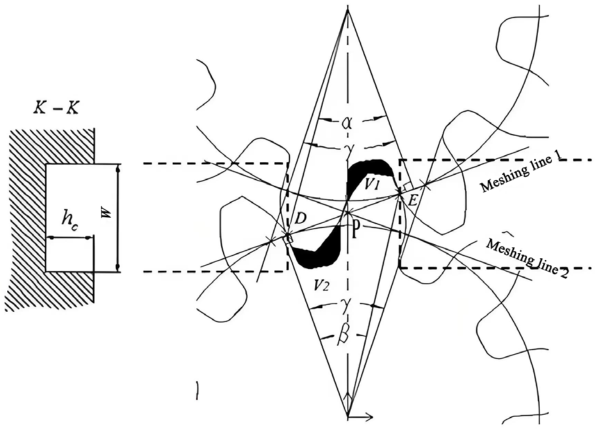


Fig. 2. Symmetrical double rectangular discharge chute.

Figure 2 illustrates the structure of symmetric double rectangular unload grooves with gear tooth clearance. In the figure, the trapped oil volume V_B is at its minimum, and the edges of the two unload grooves align precisely with engagement points D and E. The distance between the two unload grooves ω should ensure that the trapped oil volume V_B remains connected to the pressure oil chamber until it reaches its minimum position. When V_B reaches the minimum position, the trapped oil volume V_B should not be connected to either the pressure oil chamber or the suction oil chamber. After passing the minimum position, V_B should remain connected to the suction oil chamber.

Therefore, the dimension of a has strict requirements. If a is too large, the oil trapping phenomenon cannot be completely eliminated, and if it is too small, it may cause leakage by connecting the suction oil chamber and the pressure oil chamber, reducing the efficiency of the gear pump. The final design parameters for the unload grooves are as follows, as shown in Table 2.

Table 2. Unload grooves design parameters

Parameter name and symbol	Method of calculation	Calculation results
Distance between two unload grooves a	$\pi m \cos^2 \alpha_n$	2.2mm
Distance between high-pressure side unload groove edge and gear centerline a_g	$\frac{1}{2}(\pi m \cos \alpha_n - b \tan \beta) \cdot \cos \alpha_n$	0.116mm

Distance between low-pressure side unload groove edge and gear centerline a_d	$\frac{1}{2}(\pi m \cos\alpha_n + b \tan\beta) \cdot \cos\alpha_n$	2.104mm
Optimal unload groove length c	$2(R - \sqrt{R_f^2 - (\frac{a}{2})^2})$	2.21mm
Unload groove depth h	$h \geq 0.8mm$	0.8mm

In the calculation of the distance between two unload grooves, m represents the module, and α_n is the gear meshing angle (standard gear pressure angle). In the calculation of the optimal length of the unload groove, R represents the radius of the pitch circle (diametral pitch circle), and R_f represents the radius of the root circle. Using the same method, the gear pump unload groove design parameters for mounting high-tooth and low-tooth gears are calculated and shown in Table 3.

Table 3. Design parameters of unload grooves for different gear pumps

Parameter name and symbol	Unit	High-tooth gear	Full-tooth gear	Low-tooth gear
Unload groove spacing a	mm	2.2		
Unload groove length C	mm	2.85	2.21	1.79
Unload groove depth h	mm	0.8		

3 Theoretical Model and Simulation Parameters

3.1 Theoretical Model

The internal flow within a gear pump involves high-speed turbulent flow. The equations used for solving the problem are three-dimensional, unsteady, and incompressible mass and momentum conservation equations[12]:

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho \overline{u_j})}{\partial x_j} = 0 \tag{3}$$

$$\rho \frac{\partial \overline{u_i}}{\partial t} + \rho \overline{u_j} \frac{\partial \overline{u_i}}{\partial x_j} = \rho f_i - \frac{\partial \overline{p}}{\partial x_i} + \mu \frac{\partial^2 \overline{u_i}}{\partial x_j \partial x_j} - \rho \frac{\partial \overline{u_i' u_j'}}{\partial x_j} \tag{4}$$

For simulating the turbulent flow, the RNG $k - \varepsilon$ model is chosen due to its accuracy. The RNG model equations for non-steady flow simulation are as follows:

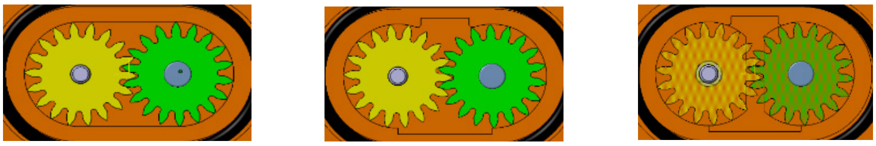
$$\rho \frac{Dk}{Dt} = \frac{\partial}{\partial x_j}\left(\alpha_k u_{eff} \frac{\partial_k}{\partial x_j}\right) + 2u_t \overline{S_{ij}} \frac{\partial \overline{u_i}}{\partial x_j} - \rho \varepsilon \tag{5}$$

$$\rho \frac{D\varepsilon}{Dt} = \frac{\partial}{\partial x_j}\left(\alpha_\varepsilon u_{eff} \frac{\partial_\varepsilon}{\partial x_j}\right) + 2C_{1\varepsilon} \frac{\varepsilon}{k} v_t \overline{S_{ij}} \frac{\partial \overline{u_i}}{\partial x_j} - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} - R \tag{6}$$

Where:$C_{1\varepsilon}$ and $C_{2\varepsilon}$ represent empirical constants; k represents turbulent kinetic energy; ε represents turbulent dissipation rate; α_k and α_ε represent turbulent flow dissipation rates.

3.2 Simulation Parameters and Settings

To validate the performance of the designed unload grooves, two commonly used gear pump types are selected for comparative simulation analysis: the standard gear pump and the inlet-outlet gear pump. The models for each type of gear pump are shown in Figure 3.

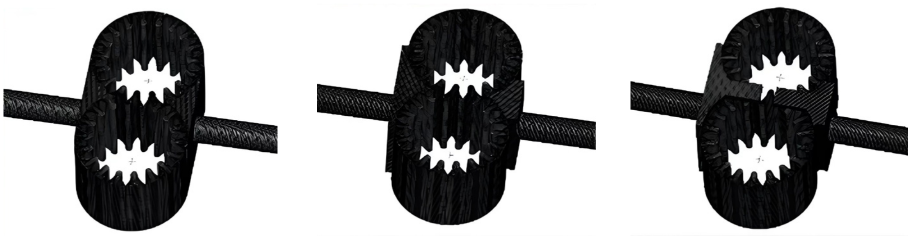


a) conventional gear pump b) inlet and outlet gear pump c) unload groove gear pump

Fig. 3. Various gear pump models.

After importing the simulation models into PumpLinx, simulation parameters are set based on common working conditions for ship gear pumps. The working fluid is water, and both inlet and outlet boundary conditions are set as pressure boundaries. The ambient temperature is 300 K, and the time step for a single driven wheel is set to 30 s. The gear rotation speed is 1420 RPM.

Since the flow inside the gear pump is three-dimensional and unsteady, using a standard grid for calculations would result in short time steps due to the narrow gap between gears, increasing simulation costs. To address this issue, PumpLinx utilizes the Geometry Conformal Adaptive Binary Tree (CAB) algorithm. The software generates Cartesian grids for inlet, outlet, and unload grooves, and generates structured grids around the gears, effectively improving computational efficiency. The meshing for this simulation is done using PumpLinx's built-in automated grid generator, creating high-quality Cartesian hexahedral grids as shown in Figure 4.



a) conventional gear pump b) inlet and outlet gear pump c) unload groove gear pump

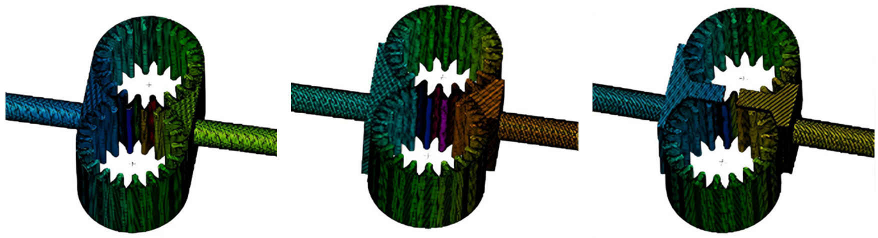
Fig. 4. Simulation model meshing for different types of pump models.

4 Simulation Results and Analysis

During the operation of a gear pump, its performance differences are reflected in several parameters, such as the inlet and outlet flow rates and pressure, which can reflect the pump's conveying performance, and the volumetric efficiency, which can reflect the cavitation condition and flow quality within the gear pump. Therefore, this section will focus on the simulation results of these parameters to explore the performance differences among different types of gear pumps.

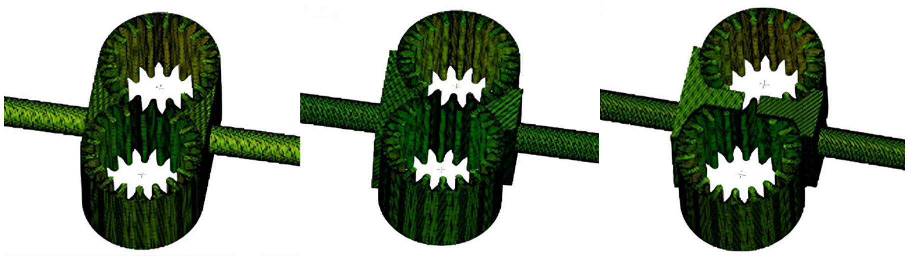
To explore the performance of the designed unload groove gear pump, this study conducted comparative simulations for three types of gear pumps: standard gear pump, inlet-outlet gear pump, and unload groove gear pump, all with the same gear shape and under the same operating conditions. The simulation results are analyzed to evaluate the performance differences among different gear pump types.

After mesh generation and simulation parameter settings, internal flow characteristics of the standard gear pump, inlet-outlet gear pump, and unload groove gear pump were simulated using the PumpLinx software. The final simulation results are presented below. Figure 5 shows the pressure distribution simulation results for the three types of gear pumps, Figure 6 shows the flow rate distribution simulation results, and Figure 7 includes charts showing changes in various parameters during the simulation.



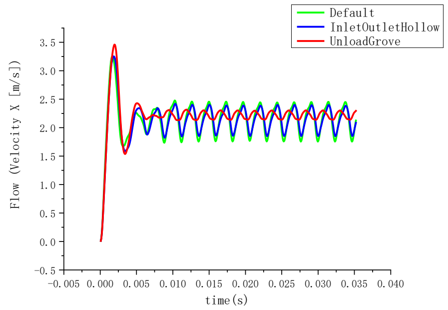
a) conventional gear pump b) inlet and outlet gear pump c) unload groove gear pump

Fig. 5. Gear pump pressure distribution simulation results.

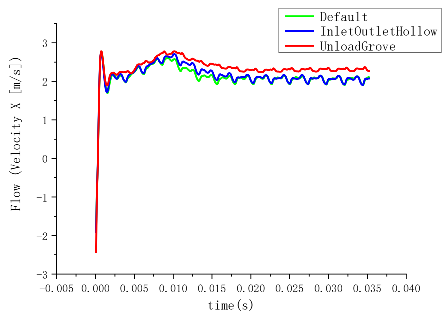


a) conventional gear pump b) inlet and outlet gear pump c) unload groove gear pump

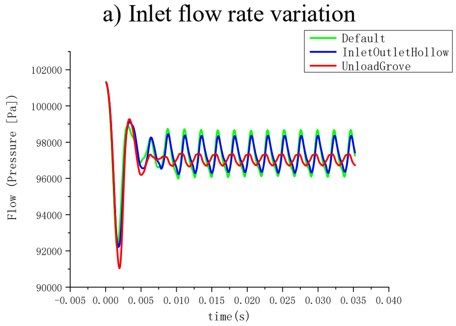
Fig. 6. Gear pump flow rate distribution simulation result.



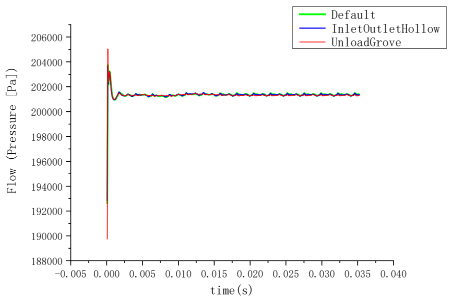
a) Inlet flow rate variation



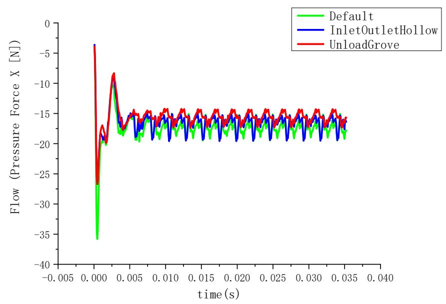
b) Outlet flow rate variation



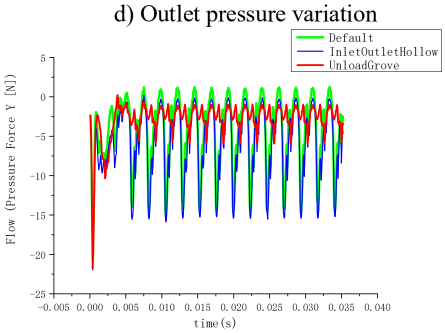
c) Inlet pressure variation



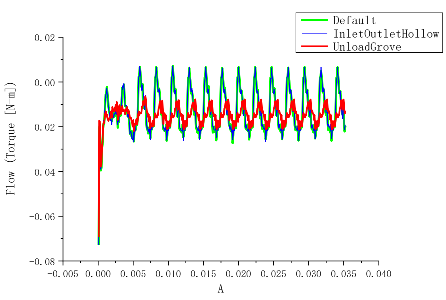
d) Outlet pressure variation



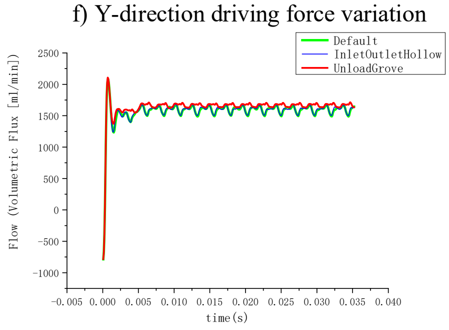
e) X-direction driving force variation



f) Y-direction driving force variation



g) Gear driving torque



h) Flow rate variation

Fig. 7. Comparison of simulation results of different gear pumps.

From Figure 5, it is evident that in a gear pump, the contact area between the two gears is the high-pressure region in the pump. The simulation results show that the liquid pressure in this region in the unload groove gear pump is significantly lower than in the other two types of gear pumps, indicating an improvement in mitigating oil trapping in this area. Figures 7a), 7b), and 7c) show that for all three gear pump types, the unload groove gear pump has significantly smaller variations in inlet and outlet flow rates and pressure.

Additionally, based on gear pump design parameters, the ideal fluid discharge for the gear pump can be calculated using Equation (7), and ideal flow rates can be calculated using Equation (8). These calculated values serve as the basis for calculating the volumetric efficiency of the gear pump for each type and assessing the flow field quality.

$$q_{ideal} = 4\pi R(R_a - R)B \times 10^{-3} = 2h_{ac}\pi Z m^2 B \times 10^{-3} \tag{7}$$

$$Q_{ideal} = q_{ideal} \times n_{shaft} \tag{8}$$

Finally, the simulation results for the three types of gear pumps are summarized in Table 4.

Table 4. Simulation results for different type of gear pump

Simulation parameters	Conventional gear pump	Inlet and outlet gear pump	Unload groove gear pump
Inlet flow rate (m/s)	2.1	2.1	2.2
Outlet flow rate (m/s)	2.2	2.2	2.4
Inlet pressure (Pa)	97500	97500	97000
Outlet pressure (Pa)	201500	201500	201500
X-direction gear driving force (N)	17	17	15
Y-direction gear driving force (N)	7	7	3
Gear driving torque (N-m)	0.015	0.015	0.015
Flow rate (ml/min)	1500	1500	1600
Volumetric efficiency	86.57%	86.57%	92.35%

It is clear that the unload groove gear pump outperforms the other two types of gear pumps in terms of inlet and outlet flow rates, gear shaft loads, and volumetric efficiency. The unload groove type also exhibits smoother operation with smaller peak values in its performance curves. Therefore, compared to the other two types, the unload groove gear pump demonstrates superior performance, lower pressure, and reduced noise.

5 Conclusion

This study explored a method for assisting the design of gear pumps by analyzing the internal flow field through simulations. It resulted in the design of a high-performance unload groove gear pump. The study first performed the basic parameter design for a gear pump based on given performance requirements. Furthermore, it proposed a novel gear pump unload groove design to address the problem of oil trapping in external gear pumps. Standard turbulent flow modeling was used to characterize the flow behavior within the gear pump, and simulations were conducted using PumpLinx software to analyze different types of gear pumps. The key findings are as follows:

(1) A novel unload groove gear pump was designed based on the given performance requirements.

(2) Through simulation and analysis of the internal flow fields of different types of gear pumps operating under the same conditions, it was found that the unload groove gear pump outperformed the other two types of gear pumps in terms of inlet and outlet flow rates, gear shaft load, flow rate, and volumetric efficiency.

Acknowledgments

This work was supported by the National Natural Science Foundation of China (12202265) and the Natural Science Foundation of Fujian Province of China (2022J01131710).

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