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RESEARCH ARTICLE

Flow Field Modeling and Simulation of High-Speed Gear Pump Considering Optimal Radial and End Clearance

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ABSTRACT The end clearance and radial clearance of the gear pump are the most important factors affecting its internal leakage and viscous friction loss, and also the most important factors affecting the total efficiency of a high-speed gear pump. The reasonable value of radial clearance and end clearance is the decisive condition for the high-efficiency performance of high-speed gear pumps. Based on a certain type of high-speed gear pump with a speed of 6000 r/min, the influence of end clearance and radial clearance on internal leakage and viscous friction loss of the gear pump is analyzed under the rated working condition of outlet pressure of 8 MPa. The flow field simulation model of the high-speed gear pump with oil flowing in radial clearance and end clearance is established in PumpLinx. The oil leakage flow rate and pump shaft power under different radial and end clearances are analyzed. The results show that under the condition of 6000 r/min, the phenomenon of insufficient oil filling in the tooth groove is produced, and the radial clearance is 0.16 mm when the total efficiency of the high-speed gear pump is the largest. Under the optimal radial clearance, the working characteristics of gear pumps with different end clearances are analyzed. The end clearance is 0.03 mm when the total efficiency of high-speed gear pumps is the largest. In this paper, the selection of radial and end clearance of high-speed gear pump and the internal flow of different clearances are analyzed, which has important theoretical significance and engineering practical value.

INDEX TERMS External gear pump, flow field modeling, high-speed, radial and end clearance, optimization, total efficiency.

I. INTRODUCTION

External gear pump has the advantages of simple structure, low cost and high reliability. It has been widely used in engineering machinery, aviation and deep sea excavations [1]. Although the external gear pump is widely used, there are still problems with improper values of end and radial clearance, which makes the gear pump less efficient under rated conditions.

Grandall [2] studied the efficiency of hydraulic pumps from two aspects of volumetric efficiency and mechanical

efficiency, constructed the mathematical model of hydraulic pump efficiency, and deduced the theoretical calculation formula. The research results laid a theoretical foundation for the study of hydraulic pump efficiency. Michael et al. [3] established a mathematical expression that correlates the Stribeck value with volumetric efficiency and mechanical efficiency to analyze the efficiency of external gear pumps. The oil churning loss is caused by the resistance of the oil to the movement of the rotating parts of the hydraulic pump. The loss of churning will generate a large amount of heat. Tang [4] established the theoretical calculation model of volumetric efficiency and mechanical efficiency of deep-sea hydraulic gear pump, it is concluded that the churning loss of the gear

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pump is mainly affected by trapped oil pressure and the trend of churning loss and trapped oil pressure in a trapped oil cycle increases firstly and then decreases.

Aiming at the research on the volumetric efficiency of gear pumps, Lv [5] proposed that the leakage of end face clearance of the internal gear pump is the main factor affecting the volumetric efficiency. The axial deformation of the gear pair and floating side plate surface underrated working pressure is obtained by fluid-solid coupling simulation calculation, and the end face clearance considering force deformation is obtained. External gear pump leakage causes include weather sealing performance is good or bad, weather the machining accuracy of parts and installation operation is reasonable, and so on [6], [7]. Hong et al. [8] analyzed the formation mechanism and influencing factors of internal leakage of CBZB2 series gear pumps, and summarized the internal leakage model of gear pumps, which is of great significance for the design of medium and high-pressure gear pumps. There are two forms of end clearance compensation for external gear pumps: elastic side plate compensation end clearance and floating sleeve compensation end clearance [9]. Li and Sun [10] established a trapped oil model of the gear pump and obtained the effect of meshing clearance on the volumetric efficiency of the gear pump.

Aiming at the research on the mechanical efficiency of gear pumps, Li et al. [11] gave the dynamic calculation expression of the mechanical efficiency of gear pumps in a meshing cycle by using computational fluid dynamics simulation. Zardin et al. [12] established a mathematical model to evaluate the mechanical efficiency of the external gear pump. The model considers the main friction losses in the gear pump, including the viscous friction loss at the tip clearance, the clearance between the sleeve and the gear side, the viscous friction loss at the shaft and the bearing, and the gear meshing loss. Borghi et al. [13] studied the influencing factors of the total efficiency of the gear pump and proposed that the position tilt of the bearing seat and the height of the transverse clearance will greatly affect the volumetric efficiency. Wang et al. [14] tested the floating side plate of an external gear pump under multiple working conditions and obtained that the positive pressure distribution of the floating side plate is asymmetric and the floating torque will change with the change of working conditions. The influence of oil temperature on gear pump efficiency mainly depends on the expansion compressibility and dynamic viscosity of oil [15]. In [16], the CFD dynamic simulation model of the linear conjugate internal gear pump was built, and the orthogonal experiment was designed for experimental verification. It was concluded that the gas content, oil temperature, and working pressure of the oil had a significant effect on the total efficiency.

The above research mainly analyzes the influence of factors such as leakage in the gear pump gap and oil temperature on the volumetric efficiency. There are still some deficiencies

TABLE 1. Detailed parameters of high-speed gear pump.

Symbol	Parameter / (Unit)	Value
m	Modulus	3
z	Number of teeth	12
α	Tooth angle / ($^{\circ}$)	20
x	Coefficient of displacement	0.5
B	The breadth of tooth / (mm)	9

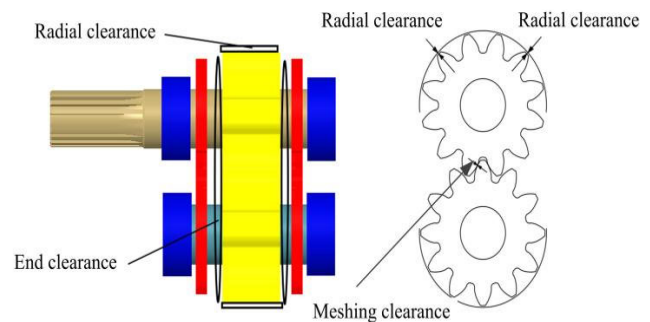


FIGURE 1. Three types of mating clearances for high-speed gear pumps.

in the research on the mechanical efficiency of the gear pump, so it is necessary to study the influence of the radial and end clearance of the gear pump on the mechanical efficiency of the gear pump. In this paper, a high-speed gear pump with a speed of 6000 r/min is taken as the research object, and the value of the end face and radial clearance of the gear pump under rated conditions is studied. When the value of the end face and radial clearance is small, more viscous friction loss will be generated, which makes the efficiency of the gear pump lower. Therefore, it is of great theoretical significance and engineering practical value to determine the optimal value of the end face and radial clearance of a high-speed gear pump underrated working conditions, considering the maximum total efficiency.

II. THEORETICAL ANALYSIS OF EFFICIENCY OF HIGH-SPEED GEAR PUMP

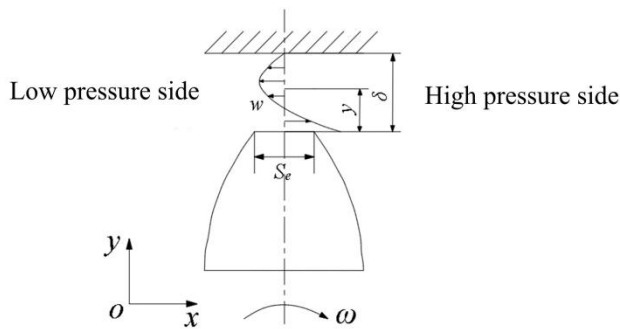
A. GEOMETRIC MODEL

For actual high-speed external meshing gear pumps, the detailed gear parameters are shown in Table 1.

Due to the relative movement between the gears and other parts of a high-speed gear pump, there are inevitably three main types of mating clearances: one is the end clearance between the gear pair side faces and the left and right floating side plate end faces, the other is the radial clearance between the gear tooth tip circular surface and the pump body hole wall surface, and the third is the meshing clearance at the gear pair engagement. The position distribution of the three mating clearances is shown in Figure 1. A summary of relevant structural parameters of high-speed gear pumps is shown in Table 2.

TABLE 2. Structural parameters of high-speed gear pump.

Symbol	Parameter / (Unit)	Value
δ	Radial clearance / (mm)	0.04~0.2
s	End face clearance / (mm)	0~0.2
b	Meshing clearance / (mm)	0.1
S_e	Tooth tip circular tooth thickness / (mm)	1.72
R_e	Addendum radius / (mm)	21.975
R	Pitch radius / (mm)	19.325
R_f	Root circle radius / (mm)	15.75
R_z	Gear shaft radius / (mm)	9

**FIGURE 2.** Oil flow velocity distribution of radial clearance.

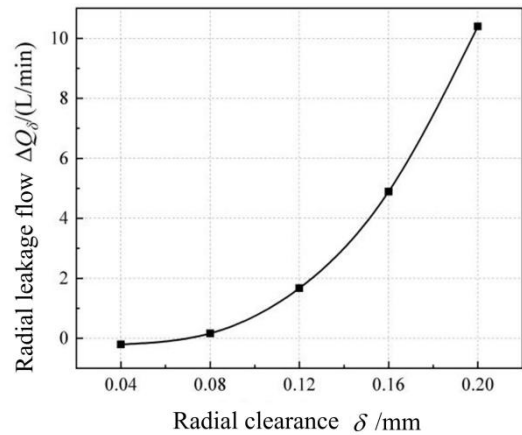
B. CALCULATION OF LEAKAGE FLOW RATE UNDER DIFFERENT RADIAL AND END FACE CLEARANCE

The radial clearance oil leakage of a high-speed gear pump refers to the flow of oil medium from the high-pressure cavity tooth groove along the radial clearance to the low-pressure cavity tooth groove, and its flow direction is opposite to the gear rotation direction. There are two reasons for the oil flow at the radial gap: firstly, the pressure difference oil flow caused by the pressure difference force at both ends of the radial gap; The second is the shear oil flow generated by the relative motion between the two walls that make up the radial gap. After the linear superposition of the two flows, it is obtained that the oil flow at the radial clearance is differential pressure shear flow [17], and its velocity distribution is shown in Figure 2.

The total radial leakage flow rate of the two gears is:

$$\Delta Q_{\delta} = 2B \int_0^{\delta} w dy = B \left(\frac{\Delta p}{6\mu Z_0 S_e} \delta^3 - \frac{\pi n R_e}{30} \delta \right) \times 60 \times 10^3 \quad (1)$$

Among them, δ Represents the radial clearance, m; Δp represents the pressure difference between adjacent cavities of the high-speed gear pump, Pa; Z_0 represents the number of teeth in the pressure transition zone of the gear pump, $Z_0 = 7$; y represents any height from the tooth tip in the y direction, m; S_e represents the thickness of the top circular tooth, m; μ Represents the dynamic viscosity of the oil, Pa $\cdot$ s, where n is the rotational speed of the gear pump, r/min.

**FIGURE 3.** Variation of total radial leakage flow with radial clearance.

There are two ways for oil leakage in the end face clearance: (1) the oil flows from the tooth groove to the gear shaft diameter, and then through the bearing to the low-pressure area; (2) The oil flows through the end face clearance on the side of the gear from the high-pressure zone to the low-pressure zone, and due to the consistent direction of oil leakage and gear rotation at the meshing point, leakage at the meshing point is the main leakage path [18].

The total leakage flow rate of the end faces of the two gears is:

$$\Delta Q_s = \frac{2\pi s^3 \Delta p}{3\mu \ln(R_f/R_z)} \times 60 \times 10^3 \quad (2)$$

In equation (2), R_z represents the radius of the gear shaft, m; R_f represents the radius of the tooth root circle, m; S represents the end face clearance, m. The gear pump consists of a simplified model of four parallel double disc gaps converging flow. It can be seen that the total leakage flow rate of the end face of a high-speed gear pump is in a cubic relationship with the value of the end face clearance, which is proportional to the outlet pressure and inversely proportional to the dynamic viscosity of the oil. Among them, the end face clearance is the most important influencing factor of the total leakage flow rate of the end face.

In summary, under rated operating conditions of 6000 r/min, outlet pressure of 8 MPa, and oil dynamic viscosity of 0.04025 Pa $\cdot$ s, the relationship between radial clearance, end clearance, and oil leakage was analyzed. The structural parameters of the high-speed gear pump in Table 2 were substituted into equations (1) and (2), respectively. The total radial leakage flow rate with radial clearance was obtained as shown in Figure 3, and the total end leakage flow rate with end clearance was obtained as shown in Figure 4.

C. CALCULATION OF VISCOUS FRICTION LOSS UNDER DIFFERENT RADIAL AND END FACE CLEARANCE

In the calculation of the mechanical efficiency of a high-speed gear pump, the change of actual pump shaft input power will affect the mechanical efficiency of the gear pump, and the

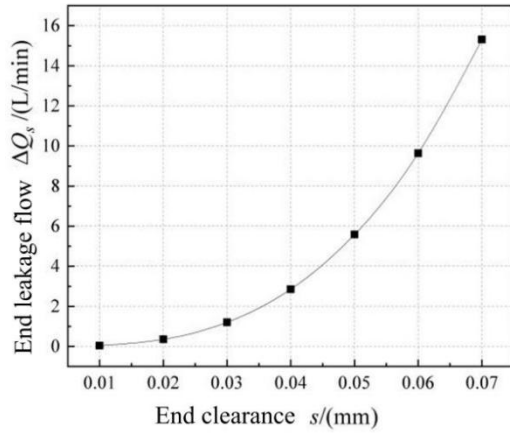


FIGURE 4. Change of total end leakage flow with end clearance.

pump shaft input power of the gear pump is directly affected by the internal mechanical loss of the gear pump. Generally speaking, the mechanical loss of gear pumps mainly consists of five parts: the viscous friction loss between the gear side and the oil, the viscous friction loss between the tooth top and the oil, the friction loss at the gear meshing point, the friction loss between the transmission shaft and bearings, and the friction loss between the transmission shaft and shaft seal. Among them, the latter three types of friction losses account for a small proportion of the total mechanical loss and are ignored when studying mechanical losses [19]. Therefore, this paper mainly discusses the influence of oil friction loss at radial clearance and end clearance on mechanical efficiency.

The expression for the viscous frictional shear stress of oil flow in the radial gap is:

$$\tau_s = \mu \left(\frac{du}{dy} \right)_{y=0} = \frac{\Delta p \delta}{2S_e Z_0} + \frac{\mu v}{\delta} = \frac{\Delta p \delta}{2S_e Z_0} + \frac{\mu \pi n R_e}{30\delta} \quad (3)$$

In equation (3), δ Represents the radial clearance, m; ΔP represents the pressure difference between the high-pressure and low-pressure grooves, Pa. The effective friction area between the tooth tip of a single gear and the wall surface inside the pump body can be obtained as:

$$A = Z_0 B S_e \quad (4)$$

The total radial friction loss power of the gear pair is:

$$\Delta P_\delta = 2\tau_s A v = \frac{1}{15} \pi n R_e Z_0 B S_e \left(\frac{\Delta p \delta}{2S_e Z_0} + \frac{\mu \pi n R_e}{30\delta} \right) \quad (5)$$

Equation (5) shows that when the speed, outlet pressure, and oil dynamic viscosity are fixed, the total radial friction loss power of the gear pump changes with the change of the radial clearance once the gear processing parameters are no longer changed, so it is necessary to study the relationship between the radial clearance and mechanical efficiency in the high-speed gear pump.

The oil flow in the end clearance of a high-speed gear pump can be simplified into four parallel double disk simplified

models, and the total frictional power loss on the end face can be expressed as:

$$\Delta P_s = 4(\Delta P_m + \Delta P_n) \quad (6)$$

In equation (6), ΔP_m represents the frictional loss suffered by the oil in the annular area of a single gear end face from the gear shaft to the tooth root circle; ΔP_n represents the frictional loss suffered by the oil in the annular area of a single gear end face from the tooth root circle to the tooth tip circle. We will discuss them separately as follows.

Friction loss of oil in the annular area of a single gear end face from the gear shaft to the tooth root circle ΔP_m , on a single gear end face, take a small annular area with a radius of r and a difference of dr , and its occupied area $dA = 2\pi r dr$. The linear velocity at radius r on dA is $v = 2\pi nr/60$. Therefore, the expression for the viscous frictional shear stress of the oil flowing in the end face gap is:

$$\tau_s = \mu \left(\frac{du}{dy} \right)_{y=0} = \mu \frac{v}{s} = \mu \frac{\pi nr}{30s} \quad (7)$$

By integrating the radius from the gear shaft to the tooth root circle, it can be concluded that the oil viscous friction loss in the annular area from the gear shaft to the tooth root circle is:

$$\Delta P_m = \int_{R_z}^{R_f} \tau_s v dA = \frac{\mu \pi^3 n^2 (R_f^4 - R_z^4)}{1800s} \quad (8)$$

Friction loss of oil in the annular area of a single gear end face from the tooth root circle to the tooth tip circle ΔP_n , considering that the area where friction occurs in this area is only the part of the gear teeth, it is approximately assumed that the cross-sectional area enclosed by the gear teeth and the gear grooves is equal during calculation. An equivalent small variable is taken to represent the area of friction in this area, and the area occupied by this variable is:

$$dA = \frac{2\pi r dr}{2} = \pi r dr \quad (9)$$

For the convenience of calculation, if the average rotational speed is equal to the tangent speed of the point on the pitch circle, i.e. $v = 2\pi nR/60$, the frictional loss of the oil in the annular area where the oil flows from the tooth root circle to the tooth top circle can be obtained as:

$$\Delta P_n = \int_{R_z}^{R_e} \tau_s v dA = \frac{\mu \pi^3 n^2 R (R_e^3 - R_z^3)}{2700s} \quad (10)$$

Substituting equations (8) and (10) into equation (6), the total friction loss power of the end face is:

$$\Delta P_s = \frac{1}{25} \pi^3 n^2 \frac{\mu}{s} \left[\frac{1}{18} (R_f^4 - R_z^4) + \frac{1}{27} R (R_e^3 - R_z^3) \right] \quad (11)$$

In summary, under the rated operating conditions of 6000 r/min, outlet pressure of 8 MPa, and oil dynamic viscosity of 0.04025 Pa · s, the relationship between radial

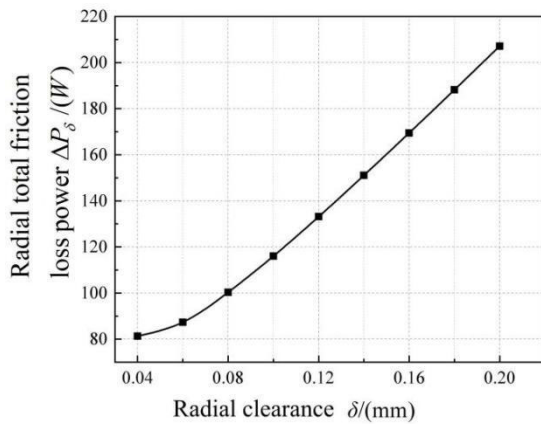


FIGURE 5. Variation of total radial friction loss power with radial clearance.

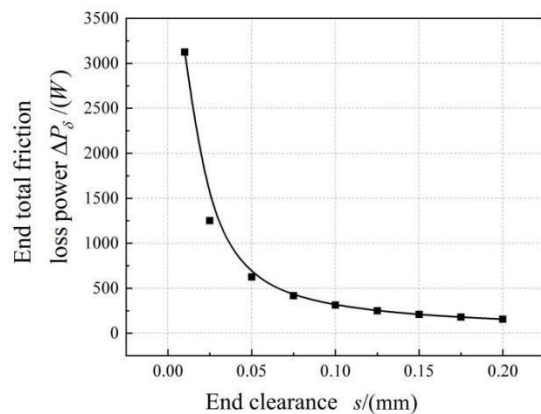


FIGURE 6. Change in total friction loss power of end face with end face clearance.

clearance, end clearance, and oil friction loss was analyzed. The structural parameters of the high-speed gear pump in Table 2 were substituted into equations (5) and (11), respectively. The variation of total radial friction loss power with radial clearance was shown in Figure 5, and the variation of total end friction loss power with end clearance was shown in Figure 6.

Figure 5 shows that the total radial friction loss power also shows an upward trend with the increase of radial clearance, where the radial clearance increases linearly from 0.08 mm to 0.2 mm. This is because the viscous friction shear stress generated by pressure difference flow is much greater than that generated by shear flow, and the increase of radial clearance generates more viscous friction power loss. Figure 6 shows that the total friction loss of the end face decreases rapidly and then gradually with the increase of the end face clearance. The maximum decrease is observed when the end face clearance increases from 0.01 mm to 0.05 mm. This is because the end face clearance and viscous friction shear stress are inversely proportional functions. A smaller end face clearance leads to a larger viscous friction shear stress, resulting in a larger decrease in the total friction loss of the end face.

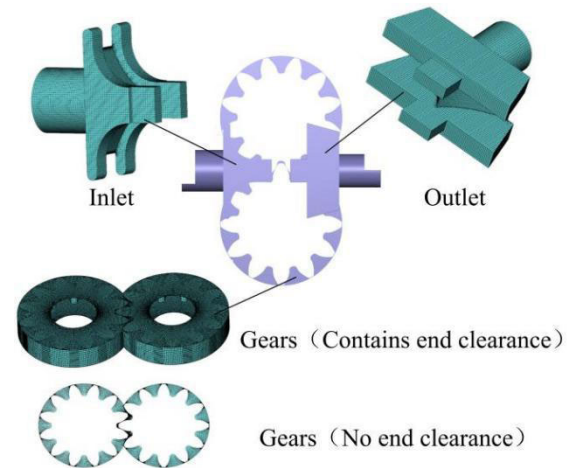


FIGURE 7. Fluid domain modeling of gear pump with and without end clearance.

In conclusion, the value of radial clearance and end clearance under rated conditions is the most critical factor affecting the volumetric efficiency and mechanical efficiency of high-speed gear pumps. Therefore, it is necessary to study the reasonable value of radial clearance and end clearance under the rated condition of the high-speed gear pump.

III. CFD MODELING OF AN EXTERNAL GEAR PUMP

A. FLUID DOMAIN MODEL AND BOUNDARY CONDITION

In actual modeling, the extracted fluid domain is shown in Figure 7. During the operation of a high-speed gear pump, the oil medium is driven by a rotating gear, and the movement state of the oil medium is irregular and extremely complex. RNG k- ϵ turbulence model includes low Reynolds number oil flow, and the calculation results are relatively stable, so this paper selects RNG k- ϵ Numerical simulation using a turbulence model. The default Constant Gas Mass Fraction full cavitation model is adopted. The center distance of the gear is 38.47549798 mm when the gear is installed without backlash. However, to ensure the continuous operation between the two meshing gears, it is necessary to leave an appropriate size of the meshing gap. The center distance of the actual installation is 38.65 mm, and the meshing gap is 0.1 mm.

The layout of monitoring points is shown in Figure 8. Point 1 and point 3 represent the rotation center of driving gear and driven gear respectively. Point 2 and point 4 represent the dynamic monitoring points fixed on the tooth groove of driving gear and driven gear respectively. All monitoring points are located in the center plane of the gear tooth width in the axial direction [20]. The rotor is divided into low pressure area, pressure transition area, high pressure area and meshing area according to the different oil pressure in the area.

The number of rotation cycles of the gear is 5, and the time step of each tooth is 30 steps. The total step size of the simulation calculation is 1800 steps.

The boundary conditions under rated operating conditions are an inlet pressure of 0.101325 MPa, outlet pressure

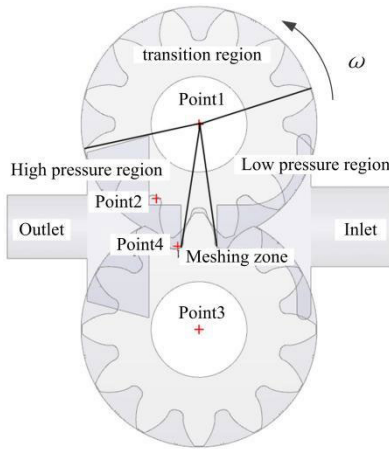


FIGURE 8. Monitoring point arrangement.

of 8 MPa, and rotational speed of 6000 r/min. In the calculation, the temperature rise caused by the oil flow at the gap is ignored. It is assumed that the oil temperature is constant at 40 °C, the oil is 46 # anti-wear hydraulic oil, with a density of 875 kg/m³, a dynamic viscosity of 0.04025 Pa · s, a saturated vapor pressure of 3000 Pa, and a gas mass fraction of 9e-5. The basic parameters of a positively modified gear, in which the modulus is 3, the number of teeth is 12, the tooth shape angle is 20 °, the modification coefficient is 0.5, and the tooth width is 9 mm.

B. GRID INDEPENDENCE VERIFICATION

By changing the meshing accuracy, seven simulation models with different grid numbers are obtained. Taking the outlet flow of the high-speed gear pump as the evaluation index, the average outlet flow of the gear pump under different grid numbers is obtained as shown in Figure 9. As the number of grids increases, the outlet flow rate continues to increase until the number of grids reaches 239615, and the outlet flow rate remains relatively constant. Therefore, the number of grids in the calculation model is controlled at over 240000 to ensure the accuracy of the simulation results.

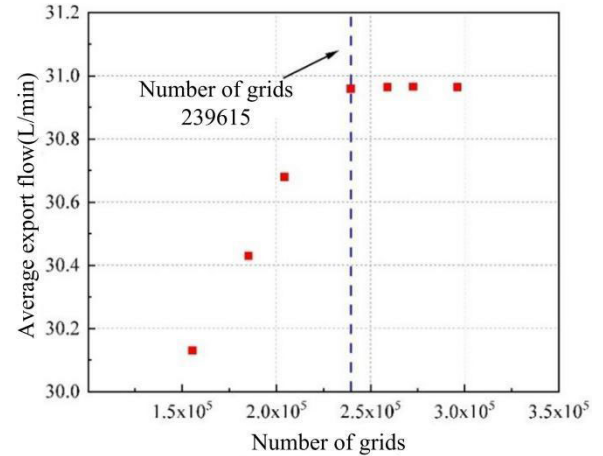


FIGURE 9. Variation of the average outlet flow with the number of grids.

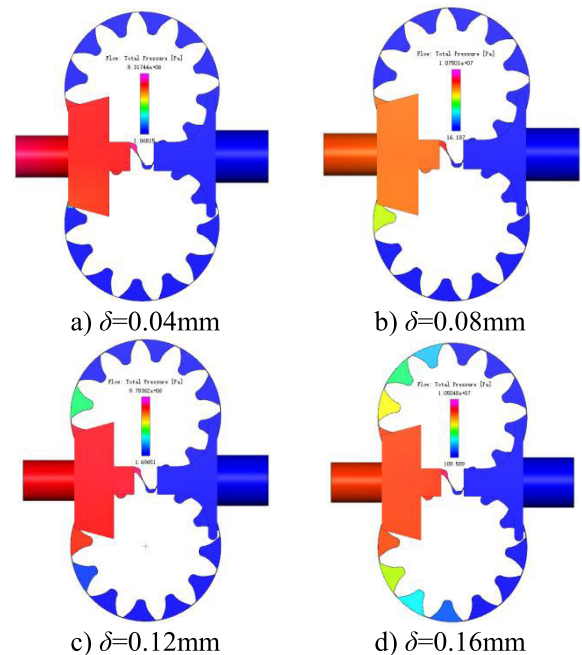


FIGURE 10. Pressure distribution in different radial clearance flow fields.

IV. EFFECT OF RADIAL CLEARANCE ON HIGH-SPEED GEAR PUMP EFFICIENCY

In the design of external gear pumps, the radial clearance is generally selected between 0.05 mm and 0.16 mm [21]. In PumpLink 4.6, the effect of end clearance on total efficiency is ignored in advance, the end clearance value is set to 0 mm and the meshing clearance is set to 0.1 mm. The radial clearance is an ideal model with no eccentricity between the tooth tip surface and the pump's inner wall. The radial clearance is set to 0.04 mm, 0.08 mm, 0.12 mm, and 0.16 mm respectively.

A. PRESSURE FIELD ANALYSIS

Figure 10 shows the change in the pressure distribution in the internal flow field of the high-speed gear pump caused by the

change of the radial clearance at the same simulation time. The left red area indicates the high-pressure oil outlet area, the right blue area indicates the low-pressure oil absorption area and the middle area from red to blue indicates the pressure transition area. The area with the highest pressure is located in the gear meshing area.

When the radial clearance increases from 0.04 mm to 0.16 mm, the number of teeth with radial leakage gradually increases. When the radial clearance increases from 0.04 mm to 0.12 mm, the number of teeth with radial leakage increases from 1 to 2. When the radial clearance increases from 0.12 mm to 0.16 mm, the number of teeth increases from 2 to 4. The specific performance is that the area of the pressure transition zone becomes larger and larger because the radial clearance becomes wider so that the oil from the

high-pressure leakage to the low-pressure reaches the tooth groove farther away from the high-pressure outlet area. It can be seen from Figure 10(d) that the number of teeth on the right side without radial leakage is 3, indicating that the sealing between the top surface of the teeth and the inner wall of the pump is good, and the radial clearance can be designed to be 0.16 mm and above.

The variation of dynamic monitoring point pressure with simulation time during a rotation cycle under different radial clearances is shown in Figure 11, where the drive_point and follower_point represent the dynamic reference points fixed in the primary and secondary gear slots. Due to the driving effect of the driving gear, the dynamic monitoring point of the driving gear represented by the red solid line acts first. The pressure rise curve corresponds to the pressure change of the dynamic monitoring point in the pressure transition zone, and the pressure drop curve represents the transition of the dynamic monitoring point from the meshing zone to the low-pressure oil absorption zone. When the radial clearance increases from 0.04 mm to 0.16 mm, the simulation time required for the dynamic monitoring point to turn over the low-pressure zone decreases, the simulation time required for turning over the pressure transition zone increases, and the simulation time required for turning over the high-pressure zone remains the same. When the dynamic measurement point of the tooth groove turns over to the high-pressure zone, there is a peak, which is because the increase in the radial clearance increases the number of teeth that have radial leakage, resulting in a larger area of the pressure transition zone, and oil trapping in the meshing zone. The trapped oil pressure generated is much higher than the outlet pressure.

Secondly, when the radial clearance is 0.04 mm, the pressure fluctuation amplitude of the grooves in the high-pressure region is large, which is due to the fact that only one gear tooth is involved in radial leakage during the process of gear disengagement, and the sealing of the outlet port is good. Under the action of the squeeze pressure, the oil flow fluctuates greatly, resulting in periodic pressure fluctuations. When the radial clearance increases from 0.04 mm to 0.12 mm, the pressure pulsation amplitude of the high pressure zone tooth slot decreases significantly, which is due to the increased radial clearance leading to more radial leakage of the outlet oil, thereby reducing the pressure pulsation of the outlet zone tooth slot. During the process of increasing the radial clearance from 0.12 mm to 0.16 mm, the simulation time is taken for the dynamic monitoring point of the tooth groove to shift from the low-pressure zone to the high-pressure zone significantly increased, and the pressure showed a stepwise upward trend and the number of steps significantly increased. This is because as the radial clearance increased, the oil leaked to the tooth groove closer to the oil inlet due to the pressure difference, Making the pressure change process of the high-speed gear pump from the low-pressure zone to the high-pressure zone more continuous.

B. VELOCITY FIELD ANALYSIS

As the radial clearance increases from 0.04 mm to 0.16 mm, the velocity distribution of the cross section of the internal flow field of the high-speed gear pump under different radial clearances is shown in Figure 12.

In Figure 12, the left side represents the high-pressure oil outlet area, the right side represents the low-pressure oil suction area, and the middle purple area represents the oil flow rate filled in the groove, which is the highest flow rate in the internal flow field of the pump. Due to the small outlet diameter, the outlet flow rate is significantly greater than the inlet flow rate. When the radial clearance increases from 0.04 mm to 0.16 mm, the number of blue slots with reduced flow velocity in the upper left corner and the lower left corner slots in the velocity distribution map gradually increases. This is because the increase of the radial clearance leads to more oil flowing through the radial clearance to the slots far away from the oil outlet under the action of high pressure at the outlet. The direction of oil leakage is opposite to the direction of gear rotation, resulting in the emergence of a blue low-velocity area with low oil flow velocity in the slots. When the radial clearance is 0.16 mm, the low-velocity blue areas in the upper left corner and the lower left corner slots are the most. Secondly, the flow velocity in the slot near the tooth valley is low, which is due to the high-speed increases in the rotary centrifugal force of the oil, so the oil cannot be fully filled, resulting in a low flow velocity of the oil.

C. ANALYSIS OF THE INFLUENCE OF RADIAL CLEARANCE ON EFFICIENCY

The radial clearance is 0.16 mm. The oil leakage of the radial clearance is visualized. The simulation results are post-processed to obtain the flow velocity distribution of the oil at the radial clearance of the driven gear at a certain simulation time, as shown in Figure 13.

Since the increasing radial clearance will lead to more flow leakage from the high-pressure slot to the low-pressure slot, thereby reducing the volumetric efficiency of the gear pump, it will also affect the viscous friction loss of the oil at the radial clearance.

The radial clearances of the simulation model are set to 0.04 mm, 0.08 mm, 0.12 mm, 0.16 mm and 0.2 mm, respectively. The end clearance is set to 0 mm, and the meshing clearance is 0.1 mm. The variation of the instantaneous flow rate of the outlet of the high-speed gear pump with the simulation time is obtained as shown in Figure 14.

The variation in the total efficiency of the high-speed gear pump with the radial clearance is shown in Figure 15.

With the increase of the radial clearance, the total efficiency of the high-speed gear pump increases first and then decreases. Under the rated working condition of 6000 r/min and 8 MPa outlet pressure, the maximum total efficiency of the high-speed gear pump is 75.8 % when the radial clearance is 0.16 mm.

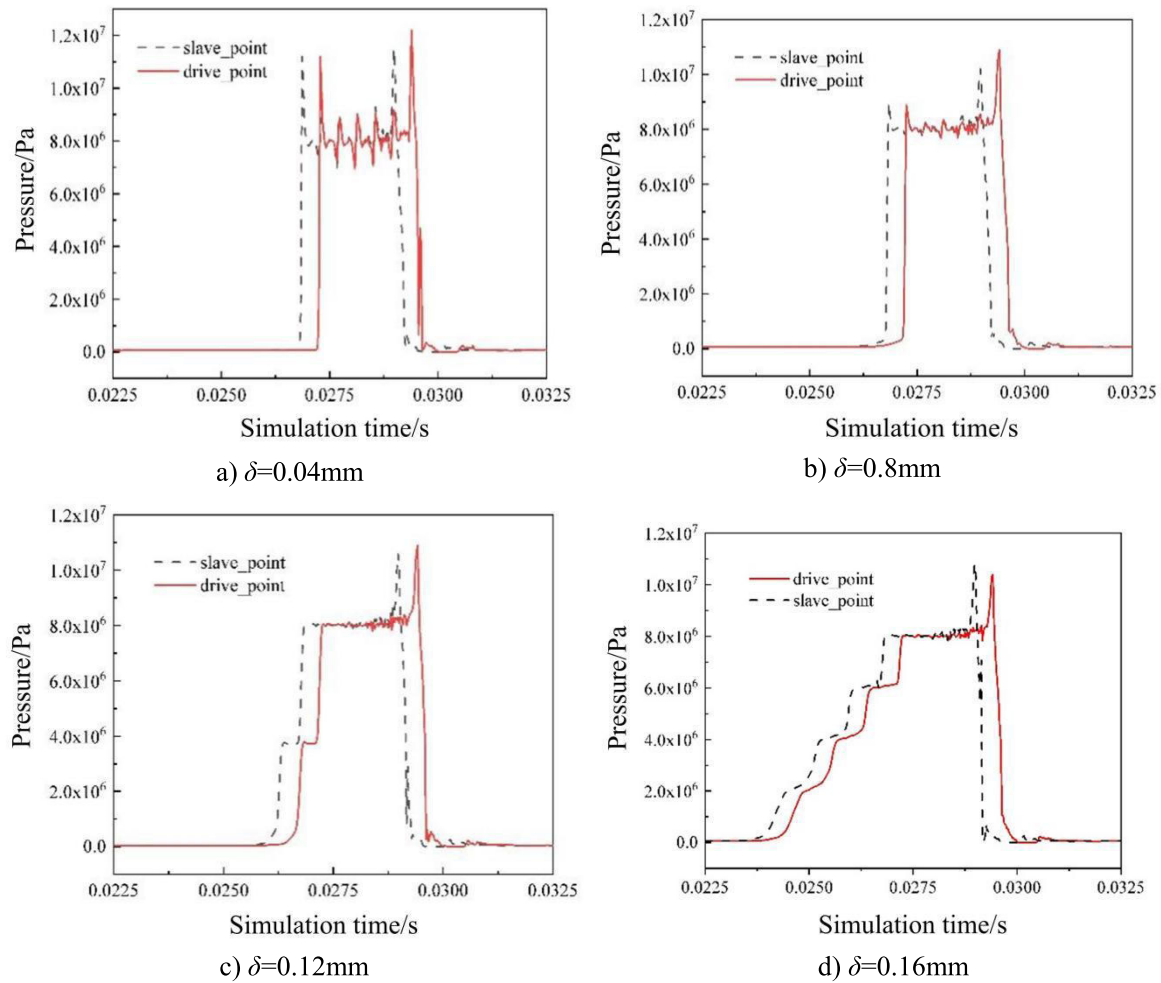


FIGURE 11. Pressure cycle changes of different radial clearances.

V. EFFECT OF END CLEARANCE ON HIGH-SPEED GEAR PUMP

Yang [22] suggests from a practical perspective that the end clearance of a low-pressure large displacement gear pump should be 0.06-0.15 mm when the gear module is 3. The end clearance of the high-speed gear pump simulation model is set to be symmetrically distributed and equal in size. The end clearance is an ideal model with parallel distribution between the gear side and the floating side plate end faces. Set the radial clearance to 0.16 mm, the meshing clearance to 0.1 mm, and the end clearance to 0.05 mm, 0.1 mm, 0.15 mm, and 0.2 mm, respectively, to obtain four sets of simulation models with only different end clearances. Perform simulation calculations according to the rated conditions in Tables 1 and 2, to analyze the impact of end clearances on the internal flow field of the high-speed gear pump.

A. PRESSURE FIELD ANALYSIS

As the end clearance increases from 0.05 mm to 0.2mm, the pressure distribution of the internal flow field of the

high-speed gear pump under different end clearances is shown in Figure 16. When the end gap increases from 0.05 mm to 0.1 mm, the area of the pressure transition zone becomes larger and the pressure gradient changes more smoothly. This is because the end gap widened so that the leakage from high-pressure to low-pressure oil flows through a longer distance to the pump inlet closer to the slot position; when the end clearance increases from 0.1 mm to 0.2 mm, the area of the pressure transition zone does not change much. This is because the number of radial leakage teeth in the pressure transition zone has increased to about 6 so that the area of the pressure transition zone reaches a critical value when the end clearance is 0.1mm. Therefore, the end clearance of the high-speed gear pump should be controlled within 0.1 mm.

To further explain the pressure change of the internal flow field caused by the end gap, the pressure change of the dynamic monitoring point is monitored in real-time, and the change rule of the pressure of the dynamic monitoring point with the simulation time in a rotation cycle under different end gap is obtained as shown in Figure 17. Among them,

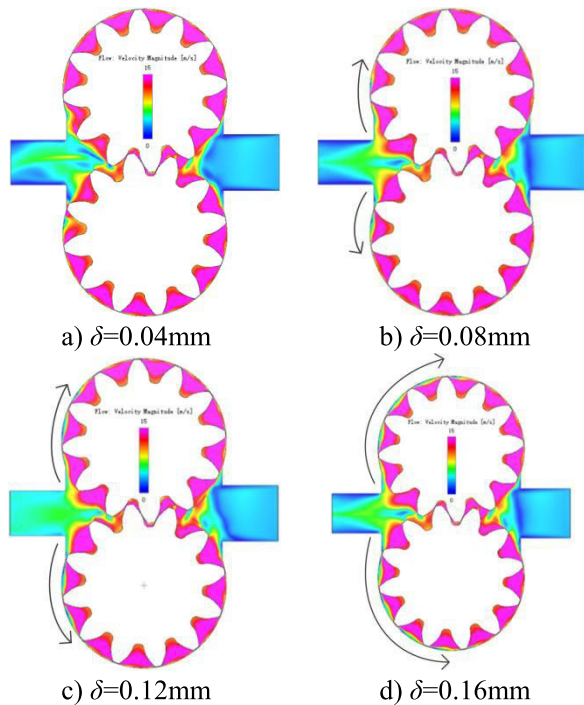


FIGURE 12. Velocity distribution of different radial clearance flow field.

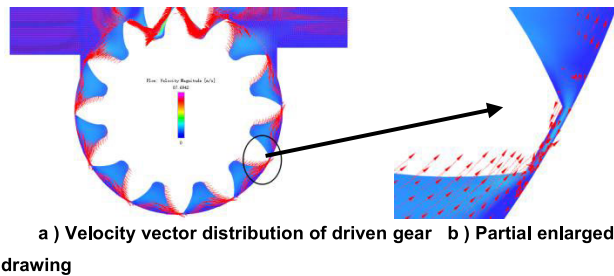


FIGURE 13. Vector cloud diagram of velocity distribution at radial clearance of driven gear.

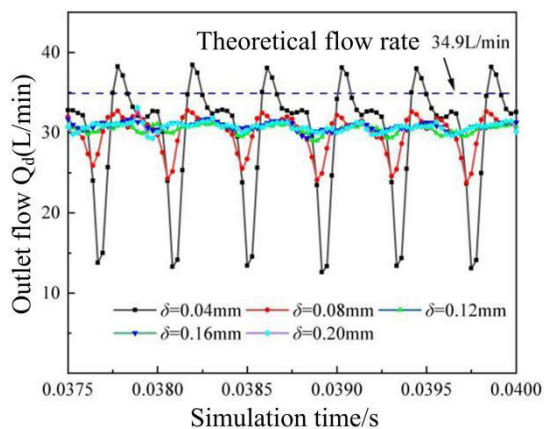


FIGURE 14. Instantaneous flow of pump outlet at different radial clearances.

the drive _ point and follower _ point represent the dynamic parameter measuring points fixed in the main and driven

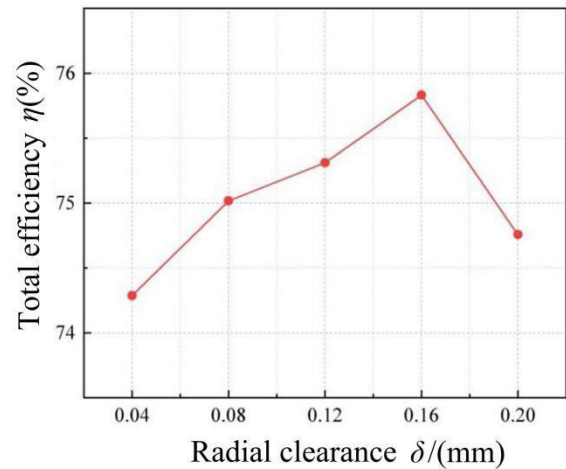


FIGURE 15. Radial clearance and gross efficiency.

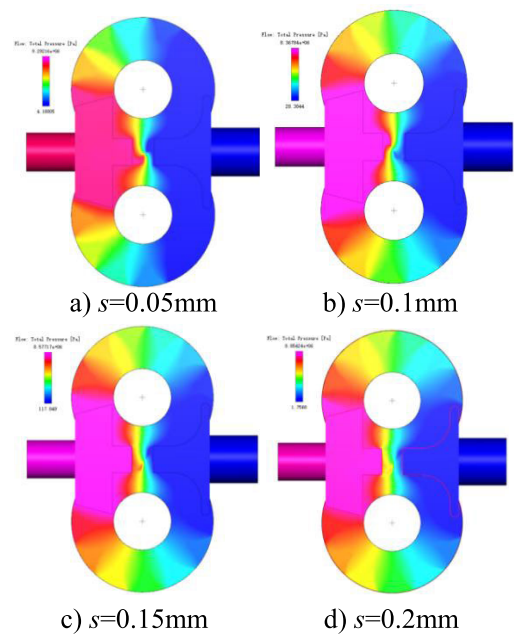


FIGURE 16. Pressure distribution of different end clearance flow fields.

gear slots respectively. When the end gap is 0.05 mm, the pressure of the tooth groove increases by 5 steps, and when the end gap is 0.2 mm, it increases by 7 steps, which indicates that the pressure change in the pressure transition zone is more gentle, and the simulation time required for the tooth groove monitoring point to turn over the pressure transition zone becomes longer. Secondly, compared with the curves of other end clearances, the alveolar pressure change curve of the end clearance of 0.05 mm has an obvious peak when the simulation time is 0.03 s. This is because the dynamic monitoring point enters the gear meshing area at this time. When the end clearance is 0.05 mm, the end face seal is better, and there is only a small amount of end leakage flow, so the trapped oil pressure in the meshing area is much larger than the outlet pressure, and the pressure change curve has

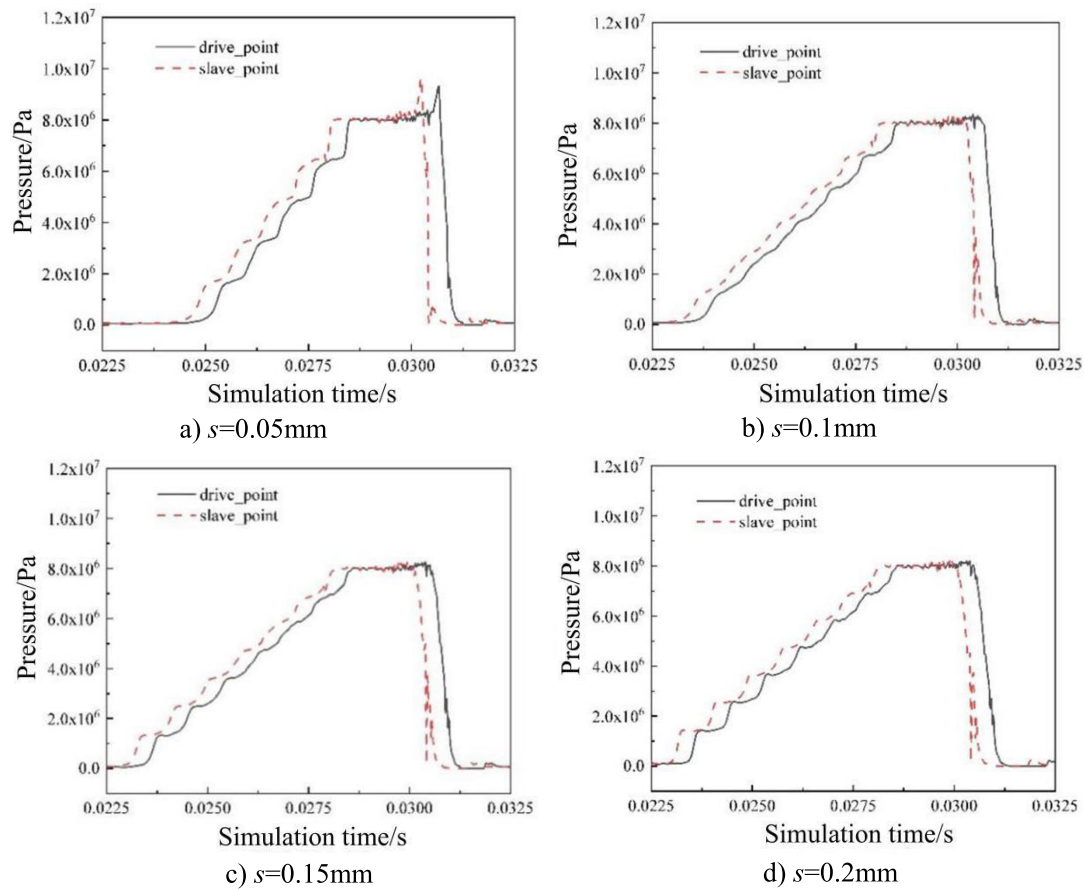


FIGURE 17. Periodic variation of pressure in different end clearances.

no peak because the end clearance is too large. The high-pressure area is filled into the meshing area so that the trapped oil pressure fluctuates approximately near the outlet pressure. Therefore, to prevent the oil exchange between the meshing area and the high-pressure area due to the excessive end clearance. The end clearance should be designed within 0.1 mm.

B. VELOCITY FIELD ANALYSIS

Figure 18 displays the variations in the velocity distribution of the internal flow field of the high-speed gear pump caused by the different end clearance at the same simulation time. As the gear rotates, the oil is continuously filled from the oil inlet area to the gear slot and output from the oil outlet area. The flow rate in the tooth groove near the tooth valley is relatively low, which is due to the increased rotational centrifugal force of the oil at high rotational speeds, which prevents the oil from being filled, resulting in a lower flow rate at this location. In the velocity distribution of the four different radial clearances, the distribution of the flow velocity of the oil in the groove where radial leakage occurs is the same. This is because the radial clearances are all 0.16 mm, and the oil in the radial leakage flows under the same outlet pressure. Secondly, when the end clearance is 0.05 mm, it can be seen

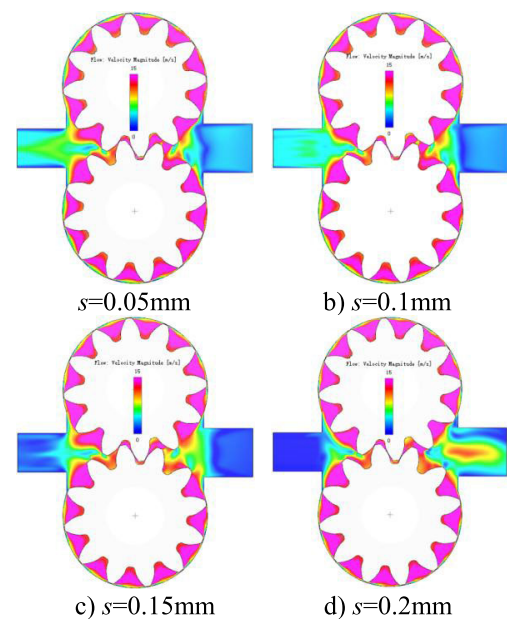


FIGURE 18. Velocity distribution of flow field under different end clearances.

that the flow rate in the inlet area is lower than that in the outlet area, which is due to the smaller diameter of the outlet

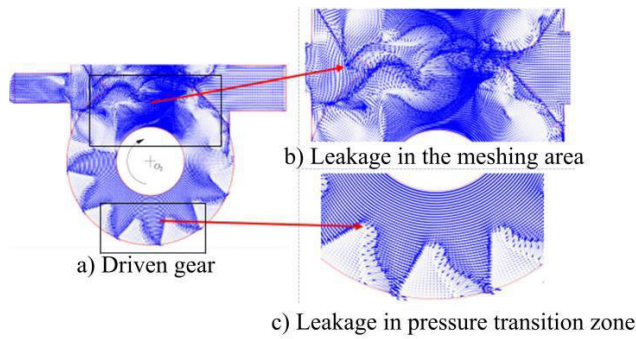


FIGURE 19. Velocity distribution of oil leaking from the end clearance of driven gear.

pipe compared to the inlet and the faster oil flow speed. When the end clearance increases from 0.05 mm to 0.2 mm, the oil flow rate in the outlet area tends to decrease. This is due to the increase in the end clearance, which intensifies the internal leakage of oil from high pressure to low pressure in the gear pump, resulting in a decrease in the outlet flow rate and a decrease in the oil flow rate. When the end face clearance is 0.2 mm, the flow rate in the outlet area is smaller than the flow rate in the inlet area. This is because after a period of operation of the high-speed gear pump, high-pressure oil forms in the outlet area, and at this time, the end face clearance is too large, resulting in a reverse flow of oil from the outlet area to the inlet area through the meshing area. Therefore, the end face clearance of the high-speed gear pump should be taken as less than 0.2 mm.

C. ANALYSIS OF THE INFLUENCE OF END CLEARANCE ON EFFICIENCY

Since the end clearance is symmetrical from top to bottom resembling the numeral “8”, the flow characteristics of the end clearance on one side of the driven gear can be used to represent the oil flow situation of the entire end clearance. Select a high-speed gear pump simulation model with an end clearance of 0.1 mm to visually analyze the oil leakage at the end clearance. After processing the simulation results, the oil flow velocity distribution at the end clearance of the driven gear at a certain simulation time is shown in Figure 19. Figure 19(a) shows the oil flow direction of the driven gear end clearance of the high-speed gear pump. After zooming in on the meshing area and pressure transition area of the left-driven gear, Figure 19(b) and Figure 19(c) are obtained. From the figure, it can be seen that the leakage path of oil in the end face gap is mainly divided into two aspects: one is the leakage from high pressure to low pressure along the meshing zone, because the leakage direction is the same as the gear rotation direction and the path of the leakage channel is relatively short; The second is the leakage from high pressure to low pressure along the pressure transition zone on the side of the gear, where the direction of oil flow is opposite to the direction of rotation of the gear and the path of the leakage channel is long. Therefore, the first is the most important path for end face leakage, but there is no phenomenon of parallel

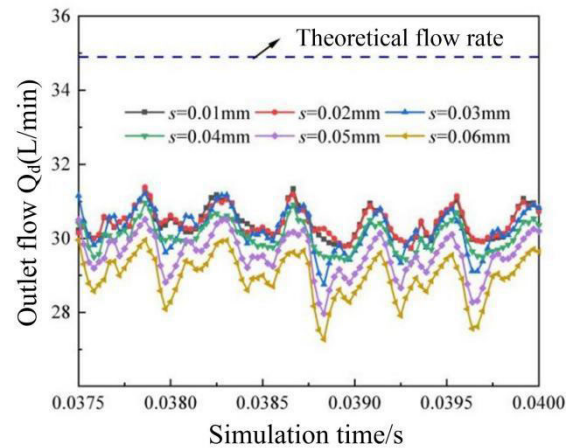


FIGURE 20. Outlet instantaneous flow.

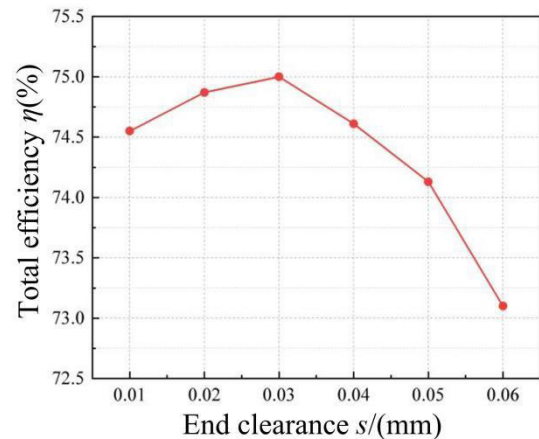


FIGURE 21. End clearance and gross efficiency.

double disk gap confluence from the tooth valley of the gear to the journal leakage. This is due to the fact that excessive rotational speed will cause the oil in the end clearance to undergo shear forces, which will weaken the flow convergence phenomenon in the end clearance. Among them, the continuous change of end clearance can lead to an increase in end leakage, thereby reducing the volumetric efficiency of high-speed gear pumps. The continuous decrease of end clearance can lead to an increase in the oil viscous friction loss at the end clearance, thereby reducing the mechanical efficiency of high-speed gear pumps.

The comparison between the theoretical output flow rate and the actual instantaneous outlet flow rate of a high-speed gear pump under different end face clearances is shown in Figure 20.

Figure 21 shows that the total efficiency of a high-speed gear pump first increases and then decreases with the increase of end clearance. That is, under the rated working condition of 6000 r/min and 8 MPa outlet pressure, the maximum total efficiency of the high-speed gear pump corresponding to an end clearance of 0.03 mm is 75%. At this time, it can be determined that the end clearance equal to 0.03 mm is the optimal end clearance under this working condition. Therefore, when

designing a high-speed gear pump under this working condition, it should be considered to maintain the end clearance around 0.03 mm.

VI. CONCLUSION

In this paper, the influence of oil flow characteristics and high-speed gear pump efficiency in different radial and end clearances is studied under the rated working condition of 6000 r/min and 8 MPa outlet pressure, and the optimal radial clearance and end clearance corresponding to the maximum total efficiency under this working condition are obtained.

(1) As the radial clearance becomes larger, the number of teeth with radial leakage increases, the area of the pressure transition zone continues to expand and the pressure gradient changes more smoothly. Because the high-speed oil cannot be fully filled, the flow velocity near the tooth valley is low. When the radial clearance increases from 0.04 mm to 0.2 mm, the volumetric efficiency increases first and then decreases, the mechanical efficiency shows a decreasing trend, and the total efficiency increases first and then decreases. When the optimal radial clearance is 0.16 mm, the maximum total efficiency of the high-speed gear pump is 75.8 %. Therefore, the radial clearance should be kept near 0.16 mm when designing the high-speed gear pump under this working condition.

(2) The leakage flow of the end clearance in the meshing area is the main way of axial leakage, and the area of the pressure transition zone increases obviously with the increase of the end clearance. When the end clearance is 0.2mm, the flow velocity in the outlet area is smaller than that in the inlet area, and the oil backflow phenomenon occurs. The results of the influence of the end clearance on the efficiency of the high-speed gear pump show that when the end clearance increases from 0.01mm to 0.06mm, the volumetric efficiency of the high-speed gear pump continues to decline, and the larger the end clearance, the more obvious the downward trend. The mechanical efficiency of the high-speed gear pump increases linearly, and the total efficiency of the high-speed gear pump increases first and then decreases. When the optimal end clearance is 0.03mm, the corresponding maximum total efficiency is 75 %, and the high-speed gear pump should be designed under this condition. The end clearance should be kept near 0.03mm.

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