1. Introduction

A piston pump has good tightness and can output considerable pressure, and a traditional reciprocating pump utilizes a crankshaft connected to a rod mechanism to convert the piston’s reciprocating motion to rotational crankshaft motion. The imbalance of the moment of inertia and inertial force induced by the reciprocating mechanism causes increased inertial load on the support members of the piston pump; hence, unstable vibration and noise (Pan et al., 2018; Zloto and Stryjewski, 2017) are easily caused throughout the system. Further, the mechanism of the reciprocating piston pump driven by the crankshaft rotation is complex and unsuitable for high-speed operation, limiting the piston-pump flow and efficiency.

In recent years, much research on improved piston-pump performance has been performed. Thus, Zhang et al. (2015) developed a large-flow piston pump and designed a performance test bench. Further, Xu et al. (2015) proposed a new design method for the valve-plate transition region based on flow area matching and transient reverse flow reduction; this method can be used to design a low-noise open circuit axial piston pump. In addition, Xu et al. (2012) achieved slipper optimization via experimental tests and simulation to provide a method to improve the axial-piston-pump efficiency and reliability.

Although improving the piston pump structure can improve the performance, the effect is subtle. Thus, some scholars have developed rotating piston pumps for improved performance. For example, Willimczik (2000) combined rotating parts with high sealability and a superior control mechanism in a rotary piston pump (RRP) having a piston actuating mechanism with a small angle of inclination. Hence all lateral disparity forces were effectively removed. Further, Li and Chen (1998) and Li (1998) determined the conditions for complete dynamic balance of rotor inertial forces in a single-cylinder rotary piston vacuum pump based on motion analysis, and improved the rotor structure according to the balance conditions to clearly depress the pump vibration.

Herein, the structure principle of a new type of RRP, triangular rotor pump (TRP), is designed and analyzed. The
pump performance for varying shaft speeds ($n$) and outlet dimensions is tested. From the obtained data, the change laws of the test-pump performance parameters (i.e., the flow rate ($q$), pressure ($p$), and efficiency ($\eta$)) are obtained. By modeling the AB-1.25D test pump and performing computational fluid dynamics (CFD) analysis, the characteristics of the pump’s internal fluid structure are analyzed. The experimental data are compared with the numerical simulation results.

2. Structure principle

The main components of TRP proposed here are the rotor, cylinder block, crankshaft, pinion stand, and bearing block. Three-dimensional (3D) and two-dimensional structure diagrams of the two-cylinder TRP are shown in Fig. 1. The pinion stand is fixed at the center hole of the front cover plate. The main journal (MJ) of the crankshaft is concentric with the pinion stand, and the two rod journals (RJ) are concentric with the rotors and distributed symmetrically at 180°. The annular gear is fixed on the rotor and mesh with the fixed gear on the pinion stand, and the gear ratio is 3:2. There are two cylinders, each with two inlets and two outlets. When the motor drives rotation of the MJ, the RJs drive the annular gears on the rotors to mesh with the fixed gears and the rotors rotate around the RJs.

![Fig. 1 Structure of double-cylinder TRP](image)

1: rotor, 2: cover plate, 3: crankshaft, 4: pinion stand, 5: annular gear, 6: cylinder, 7: intermediate plate, 8: inlet and outlet, 9: bearing block

2.1 Principles of new RPP

The rotor divides the cylinder into three working-chambers (W-C). Eccentric movement of the pump W-C is achieved by eccentrically rotating the rotor in the cylinder to change the W-C cavity volume and complete the suction and discharge (Fig. 2). The rotor motion includes counterclockwise rotation of the rotor centroid around the cylinder centroid (rotation of the RJ around the main journal), and clockwise rotation of the rotor around the RJ. The speed ratio is 3:2. In processes (a)−(g) of Fig. 2, the W-C completes suction and discharge by inlet 1, and then completes the second suction and discharge in processes (g)−(l), before returning to a. This is one cycle of the rotor motion, for which the main journal rotates three times and the three W-Cs complete six suction and discharge processes.
2.2 Mathematical Model of Cylinder and Rotor

The sectional line of the cylinder inner cavity is a double arc external cycloid, which is also the envelope of the motion trajectory of the three rotor vertices (Fig. 3).

\[ X = e \cos \beta + L \cos \varphi \]
\[ Y = e \sin \beta + L \sin \varphi \]

(1)
Clearly, the key parameters affecting TRP are $e$ and $L$. Considering the radius of the sealing-strip end ($r_a$) and the angle of the sealing-strip contact point with the radial direction ($\gamma$), the actual cylinder line is

$$
\begin{align*}
X &= e \cos \beta + L \cos \varphi + r_a \cos (\varphi + \gamma) \\
Y &= e \sin \beta + L \sin \varphi + r_a \cos (\varphi + \gamma)
\end{align*}
$$

The TRP of type AB-1.25D with key parameters of $e$: 10 mm, $L$: 75 mm, cylinder thickness ($s$): 43.6 mm, and $r_a$: 2 mm was selected as the test pump. The cylinder line was

$$
\begin{align*}
X &= 10 \cos \beta + 75 \cos \varphi + 2 \cos (\varphi + \gamma) \\
Y &= 10 \sin \beta + 75 \sin \varphi + 2 \cos (\varphi + \gamma)
\end{align*}
$$

### 2.3 Sealing system

The radial seal mainly includes the cylinder seal ring and rotor seal. The rotor is the core component of both the TRP and the liquid sealing part. It is mainly composed of the rotor, ring gear, sealing column, sealing strip, curved seal, wave spring, seal ring and spring (Fig. 4).

The working conditions of the three W-Cs concurrently differ. Thus, the sealing strips installed at the rotor triangular parts are always attached to the cylinder inner wall through combined action of the spring and centrifugal force, and the three W-Cs are separated. The sealing degree determines the pump-body working pressure. When the rotor rotates to the Fig. 5 position, Eq. (1) is differentiated with respect to time ($t$), and $\beta = 3\varphi$ is set; then, the seal velocity components at point M in the X- and Y-directions ($v_x$ and $v_y$, respectively) can be obtained. Differentiating with respect to $t$ again gives the X- and Y-axis components of the sealing-strip acceleration ($a_x$ and $a_y$, respectively) at point M. The angular velocity of the crankshaft is $\omega = d\beta/dt$. According to the trigonometric function of the acceleration components in Fig. 5, the sealing-strip radial ($a_1$) and tangential ($a_2$) accelerations can be obtained:

$$
\begin{align*}
v_x &= -e \sin \beta \frac{d\beta}{dt} - \frac{L}{3} \sin \frac{\beta}{3} \frac{d\beta}{dt} \\
v_y &= e \cos \beta \frac{d\beta}{dt} + \frac{L}{3} \cos \frac{\beta}{3} \frac{d\beta}{dt}
\end{align*}
$$

![Fig. 4 Sealing system](image)
In Fig. 5, \( p_1 \) and \( p_2 \) are the liquid pressures of the adjacent cylinders, \( h \) is the sealing-strip thickness, \( l \) is the sealing-strip length, \( m \) is the sealing-strip mass, and \( \mu \) is the friction coefficient between the sealing strips and side wall. The liquid pressure applied to the seal strips \( (p) \) can be obtained via force analysis:

\[
p = l\left( \frac{h}{2} - r_2 \sin \gamma \right) p_1 + l\left( \frac{h}{2} + r_2 \sin \gamma \right) p_2
\]

The radial inertial force \( F_1 \) is

\[
F_1 = ma_1 = \frac{-L\omega^2 m}{9} - \omega^2 m \cos \frac{2\beta}{3}
\]

and the side friction \( F_f \)

\[
F_f = \mu F_2 = \mu ma_2 = -\mu \omega^2 m \sin \frac{2\beta}{3}
\]

For AB-1.25D, \( l \) is 43 mm and \( h \) is 4 mm. For a certain \( n \), the sealing performance must be ensured, i.e., the core force of the sealing strips \( (F) \) must exceed zero, where \( F = F_1 + F_s - p - F_f \). That is, the spring force \( (F_s) \) must be satisfied:

\[
F_s > (86 - 86\sin \gamma) p_1 + (86 + 86\sin \gamma) p_2 - 10\omega^2 m \left( \mu \sin \frac{2\beta}{3} - \cos \frac{2\beta}{3} - \frac{5}{6} \right)
\]

3. Performance experiment for new double-cylinder RPP
3.1. Experiment system

The test system consisted of a TRP, monitoring sensors, data collection center, high-pressure pipe and storage tank (Fig. 6). The test pump was AB-1.25D type. The monitoring sensors included an FDDCIIIEP2M3A electromagnetic flowmeter for flow parameter collection, an FD80/86 flat model pressure transmitter for inlet/outlet \( p \) monitoring, and a HCNJ-101 dynamic torque sensor to record the input-shaft torque \( (T) \) and \( n \). The sensor data were displayed and recorded in real time by an NHR-8100/8700 color paperless recorder, and stored in a computer.
System control involved adjusting the input-shaft $n$ through a continuously variable transmission (CVT), and control of the pneumatic V-type adjustment ball valve through a valve positioner to change the outlet size and achieve a different output $p$. The outlet size was set to $a$ (7.5 mm) or $b$ (6.5 mm) via the pneumatic V-type adjustment ball valve. $n$ was adjusted from 100 to 200 r/min in 10-r/min 60-s increments via the CVT, for both outlet sizes. The $q$, $p$, and $T$ were monitored and recorded, and the AB-1.25D performance was analyzed (Zhao et al., 2016; Li et al., 2010; Chen et al., 2017; Pei et al., 2017).

3.2 Analysis of $q$, $p$, and $T$

The dynamic change curves of $q$, the differential $p$ between the outlet and inlet ($\Delta p$), and $T$ with time are shown in Fig. 7. All three increased over time: $q$ changed steadily while $\Delta p$ and $T$ fluctuated greatly. With increased $n$, the $\Delta p$ and $T$ fluctuation frequency ($f$) increased and the outlet flow pulse increased accordingly (Table 1). For outlet size $a$ and 200-r/min $n$, $q$, $\Delta p$, and $T$ began to decline with varying degrees of fluctuation. For outlet size $b$ and 170-r/min $n$, the same $q$, $\Delta p$, and $T$ behavior was observed for $n$ of 100–160 r/min. This was due to the excessive load on the motor shaft and the $T$ instability when $n$ reached a certain value, yielding $q$ and $\Delta p$ instability. For outlet size $a$ ($b$) and 190 (160)-r/min $n$, $q$, $\Delta p$, and $T$ reached maxima at the same time, of 8.83 (7.34) m³/h, 1.97 (2.31) MPa, and 260.71 (303.92) N·m, respectively. For the same $n$ and smaller outlet size, $q$ was smaller; however, $\Delta p$ and $T$ were larger, with greater fluctuation.
Under different $n$, the $p$ and $T$ fluctuation degrees can be expressed by non-uniform coefficients $\delta p$ and $\delta T$ (Ismail et al., 2015; Kim et al., 2015; Long et al., 2016):

$$\delta p = \frac{p_{\text{max}} - p_{\text{min}}}{p_m} \times 100\%$$

$$\delta T = \frac{T_{\text{max}} - T_{\text{min}}}{T_m} \times 100\%$$

where $p_{\text{max}} (T_{\text{max}})$, $p_m (T_m)$, and $p_{\text{min}} (T_{\text{min}})$ are the maximum, mean, and minimum pressures (torques), respectively.

The $\delta p$ and $\delta T$ for outlet sizes $a$ and $b$ are shown in Fig. 8. For increasing $n$, both $\delta p$ and $\delta T$ exhibit upward trends for the two different outlet sizes. A more-obvious trend corresponds to a smaller outlet size, and $\delta p$ is larger than $\delta T$ for the same outlet size and $n$. For outlet size $a$, $\delta p$ and $\delta T$ changed little, and when the motor shaft load was very large, the increases in $\delta p$ and $\delta T$ were obvious. At normal $n$, the maximum $\delta p$ and $\delta T$ were 29.7% (41%) and 17.4% (24.3%) for outlet size $a$ ($b$), respectively.

For the $T_m$ recorded at different $n$ in the experiment and the relation $P = nT_m/9550$, the shaft power ($P$) variation with $n$ is shown in Fig. 9. With increased $n$, $P$ increased. The maximum $P$ were similar for the different outlet sizes, at 5.2 (5.1) kW for outlet size $a$ ($b$) and a $n$ of 190 (160) r/min.

Table 1. Pressure ($p$) fluctuation frequency

| $n$ (r/min) | 100 | 110 | 120 | 130 | 140 | 150 | 160 | 170 | 180 | 190 | 200 |
|-------------|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|
| $f$ (Hz)    | 3.3 | 3.7 | 4.0 | 4.3 | 4.7 | 5.0 | 5.3 | 5.7 | 6.0 | 6.3 | 6.7 |

Fig. 7 Experiment data

(c) Torque
3.3 Efficiency of new RPP

Considering the energy flow from the shaft to the fluid, the pump total efficiencies $\eta$ can be expressed as

$$\eta = \frac{\rho g H_m q_m}{\omega T}$$  \hspace{1cm} (12)

Here, $\rho$ is the liquid density, $g$ is the gravitational acceleration, $\Delta p_m$ is the average static pressure difference, $q_m$ is the calculated average flow, $\omega$ is the shaft angular velocity, $T$ is the measured torque at the pump shaft. Based on $\Delta p_m$ and the flow velocity at the inlet and outlet, $v_1$ and $v_2$, the head $H_m$ can be obtained:

$$H_m = \frac{\Delta p_m}{\rho g} + \frac{v_2^2 - v_1^2}{2g}$$ \hspace{1cm} (13)

As the dynamic pressure difference $\rho(v_2^2-v_1^2)/2$ is very small relative to $\Delta p_m$ and can be ignored, Eq. (12) can be converted to:
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\[ \eta = \frac{\Delta p_m q_m}{\omega T} = \frac{(T - T_M) \omega}{\Delta p_m (q_0 - q_{leak})} \frac{\Delta p_m (q_0 - q_{leak})}{T - T_M \omega} = \frac{T - T_M}{\omega} \frac{\Delta p_m}{q_0 - q_{leak}} = \eta_m \eta_h \eta_v \]

(14)

Where \( T_M \) is the torque loss, \( \Delta p_m \) is the theoretical pressure difference, \( q_0 \) is the theoretical flow rate and \( q_{leak} \) is the leakage flow at the inlet which occurs before the energy is transferred from the rotors to the fluids. \( q_{leak} \) is small enough to be ignored due to the existence of the one-way valves. The pump total efficiencies \( \eta \) can be divided into the mechanical (\( \eta_m \)), hydraulic (\( \eta_h \)) (Zhang et al., 2013; Wang et al., 2016) and volumetric (\( \eta_v \)), efficiencies:

\begin{align*}
\eta_m &= \frac{T - T_M}{\omega T} \\
\eta_h &= \frac{\Delta p_m}{\Delta p_m} = \frac{H_m}{H_h} \\
\eta_v &= \frac{q_m}{q_0}
\end{align*}

(15)

The torque loss \( T_M \) can be calculated as

\[ T_M = \frac{9550 L}{n} \]

(16)

The mechanical efficiency \( \eta_m \) reflects the friction loss of mechanical parts such as seals, shaft and bearings. The total mechanical loss \( (L_m) \) is divided into the loss of the sealing friction \( (L_s) \), rotor (seals in rotor surface) \( (L_r) \), bearing \( (L_b) \) and gear meshing \( (L_g) \), which is expressed as (Li et al., 2018)

\[ L = L_s + L_r + L_b + L_g \]

\begin{align*}
L_s &= \mu_s F_s \left( v_s \cos \frac{\beta}{3} + v_s \sin \frac{\beta}{3} \right) - \mu_c F_c \left( v_s \sin \frac{\beta}{3} + v_s \cos \frac{\beta}{3} \right) \\
L_r &= \frac{2 \pi \mu_o \omega_o \left(R_s^2 - R_i^2\right)}{d} \\
L_b &= \frac{2 \pi \mu_i \omega_i ^2 R_s l_e}{c_e} + \frac{2 \pi \mu_i \left(\omega - \omega_i \right) ^2 R_s l_e}{c_e} \\
L_g &= \frac{\mu \gamma \omega e \omega m e o^3}{2 \pi \sin \theta}
\end{align*}

(17)

In Fig. 3, when the rotor annular gear is fixed, the fixed gear drives the cylinder profile to rotate along the annular gear, forming the cylinder profile envelope, which is the three sides of the rotor (Lee et al., 2004). It is difficult to process the rotor profile envelope accurately; thus, three equivalent arcs can be used instead. One equivalent arc is defined by the two rotor vertices in Fig. 3, C and M. The arc’s dot is F and its radius FC can be expressed as

\[ FC = \sqrt{FG^2 + CG^2} = \frac{(L - e)^2 + 3e^2}{L - 4e} = 129.29 \text{mm} \]

The areas \( S_{CEI} \) and \( S_{CAI} \) can be obtained from Eq. (1) and the geometric relationship in Fig. 3:

\[ S_{CEI} = \frac{2}{3} \int \left( e \sin 3\varphi + L \sin \varphi \right) \left( e \cos 3\varphi + L \cos \varphi \right) d\varphi \]
When the rotor is moved to the left position in Fig. 3, the W-C surrounded by CEHA becomes the largest volume, and $S_{\text{CEHA}}$ (the area enclosed by CEHA) is

$$S_{\text{CEHA}} = 2S_{\text{CEA}} = 2(S_{\text{CEH}} - S_{\text{CAL}}) = \pi e^2 + \frac{3\sqrt{3}eL}{2} - \left(\frac{\pi}{6} - \frac{\sqrt{3}}{2}\right)L^2$$

(18)

According to the TRP working principle, the crankshaft performs one revolution, the single cylinder pump completes the suction and discharge of two chamber volumes and the double cylinder pump completes four. The theoretical flow rate ($q_0$) of the double-cylinder eccentric rotary pump is

$$q_0 = 4\pi n S_{\text{CEHA}} = \left(4\pi e^2 + 6\sqrt{3}eLs - \frac{2\pi}{3}L^2s + 2\sqrt{3}L^2s\right)n$$

(19)

In summary, $\eta$ can be calculated by Eq. (14), and $\eta_M$ can be calculated by Eq. (15), (16) and (17), and $\eta_V$ can be calculated by Eq. (15) and (19), and $\eta_H$ can be calculated by $\eta$, $\eta_M$ and $\eta_V$, and $H_{\text{th}}$ can be calculated by (13) and (15). According to the experiment data for $q$, $\Delta p$, and $T$, as well as the above formulas, the $\eta_M$, $\eta_V$, $\eta_H$, and $\eta$ curves with varying $n$ could be obtained as shown in Fig. 10. The following conclusions were drawn from Figure 10:

1. For different outlet sizes, the test-pump $\eta$ tended to decrease slowly with increasing $n$. Before the shaft was overloaded, $\eta_a$ and $\eta_b$ decreased to 64.2% and 61.4%, respectively.

2. At different outlet sizes, $\eta_M$ decreased with increasing $n$ until the load on the shaft was too large. At the same $n$, the mechanical efficiency at outlet size $a$ ($\eta_{Ma}$) was greater than that at outlet size $b$ ($\eta_{Mb}$). The maxima were similar, at 86.2% and 85.9% for $\eta_{Ma}$ and $\eta_{Mb}$, respectively, and then they go down to 73% and 71.9%.

3. At different outlet sizes, $\eta_V$ increased slowly with increasing $n$ until the load on the shaft was too large. This resulted from the good sealing performance of the TRP, with little change in flow loss with increasing $n$. For outlet sizes $a$ and $b$, the volumetric efficiencies ($\eta_{Va}$ and $\eta_{Vb}$, respectively) gradually approached 96.8% and 95.5% maximum, respectively. At the same $n$, $\eta_{Vb} > \eta_{Va}$.

4. At different outlet sizes, $\eta_H$ did not fluctuate much with increasing $n$. The hydraulic efficiencies at outlet sizes $a$ and $b$ ($\eta_{Ha}$ and $\eta_{Hb}$, respectively) were similar, with average values 90.8% and 88.4%, respectively.
4. Numerical simulation of AB-1.25D

4.1 Numerical method

To analyze the flow field structure of the AB-1.25D, Ansys Fluent 18.1 was used to simulate the test pump. Based on the actual AB-1.25D size, its internal flow channel was 3D solid modeled using SolidWorks. ICEM was used to divide the internal flow into non-structural grids (Fig. 11). In this paper, four grid numbers of different quality are selected for grid independence analysis under the design condition, and the information of three grids is shown in Table 2. According to the information in the table, both scheme 2 and scheme 3 can meet the requirements of the calculation accuracy of pressure. However, in order to capture the flow field details, scheme 3 is selected in this paper. The time-averaged Navier-Stokes equation was used as the basic control equation, and the SST turbulence model was used to better solve the strong rotation problem and improve the turbulence calculation accuracy (Chen et al., 2017). The model was subjected to full-flow 3D unsteady-valued value simulation, and the finite-volume method was used to discretize the equations.
Table 2. Pressure (p) fluctuation frequency

| Scheme | Grid number | Mesh quality | Design pressure/MPa | Numerical pressure /MPa |
|--------|-------------|--------------|----------------------|-------------------------|
| 1      | 1741872     | 0.46         | 1.97                 | 1.613                   |
| 2      | 2424316     | 0.58         | 1.97                 | 1.789                   |
| 3      | 3896543     | 0.63         | 1.97                 | 1.832                   |
| 4      | 5842683     | 0.72         | 1.97                 | 1.871                   |

The discrete parameters were set as follows. The pressure base solver was selected and the Pressure Implicit with Splitting of Operators (PISO) pressure-velocity coupling algorithm was used. The convection-term discretization format was quadratic upwind interpolation for convective kinematics (QUICK). Because Fluent adopts collocated grids, the Pressure Staggering Option (PRESTO!) was used for the pressure interpolation. The boundary conditions were set according to the actual test conditions, using pressurized inlets and a pressure-free outlet. A Fluent user defined function was used to define the law of rotor motion, and the wall surface was smooth with no sliding wall. The reference pressure was 101325 Pa and gravity was considered in the calculation process. The calculation convergence accuracy was set to $1 \times 10^{-5}$. The time step was $1 \times 10^{-5}$ s. To verify the experimental data, CFD simulations were performed on the model for different n and outlet sizes, where the rotor rotated for at least 10 cycles per working condition.

4.2 Hydrodynamic field analysis

To analyze the flow structure and pressure distribution in the model pump, the fluid distribution in the pump at outlet size $b$ and $n = 160$ r/min was taken as an example to analyze the fluid structure and pressure distribution on plane A in Fig. 11.

4.2.1 Flow field structure analysis

Fig. 12 shows the velocity vectors on the streamlines and the velocity distribution on plane A of the TRP. Since speed of MJ is 160 r/min, the time required for MJ to make a revolution is $t = 0.375$ s. Suppose that the position (a) in Fig. 12 is $t_0$, and the numerical simulation results are got every $1/4t$. Here, (a)–(g) is a process where the W-C intakes at inlet 1 and discharges at outlet 1 (the crankshaft rotates counterclockwise). The flow velocity is stable in the W-C and there are no long-term vortices. At position a, a clockwise vortex (V₁) appears at inlet 1; this vortex disappears as the W-C volume increases. In process (b)–(c), a counterclockwise small vortex (V₂) forms between the rotor vertex and cylinder wall. When the rotor rotates to position (d), the rotor vertex begins to separate from the cylinder wall and V₂ gradually disappears. At position (d), the largest vortex (V₃) is formed, being clockwise. At the same time, outlet 1 is just connected to the W-C and a clockwise vortex (V₄) remaining in the previous W-C is released. As the rotor vertex makes contact with the cylinder wall, the W-C enters a stable full-pressure state and V₃ and V₄ disappear. Finally, at position (f), the W-C generates a V₄ at outlet 1 until it is released by the next working process.
4.2.2 Distribution and variation of pressure field

Fig. 13 shows the pressure distribution on plane A of the TRP. From the inlet 1 intake to the outlet 1 discharge, the W-C pressure rose continuously and then decreased. From the figure, the highest pressure was at position (h), between (d) and (e) and the beginning of the stable full-pressure state. At different n, the maximum pressure position in the pump appeared at position (h), as shown in Fig. 14. This is because the rotor had just turned over inlet 1 and the W-C turbulence was reduced (process (d)–(e), Fig. 12). The W-C volume changed most rapidly here. As shown in Fig. 14, the maximum W-C pressure occurred at the point of contact of the rotor vertex with the cylinder wall, which again illustrates the importance of the sealing-strip analysis in the Sealing strip section.
Fig. 13 Pressure distribution on plane A (Outlet size b, n=160 r/min)
4.3 Numerical Simulation Results and Analysis

To verify the accuracy of the numerical simulation results, the head $\psi$ and flow $\varphi$ coefficient dimensionless parameters of the TRP were defined according law of similarity (Brennen, 2012):

$$\psi = \frac{p}{\rho u^2}$$

Fig. 14 Maximum pressure positions at different shaft speeds (Outlet size $b$)
Here, $u (= 2\pi en/60)$ is the peripheral speed of the RJ center and $A (= \pi e^2)$ is the area enclosed by the center motion path of the RJ.

The $\psi$, $\phi$, and $\eta$ from the experimental and numerical simulation results for different speeds are shown in Fig. 15 (a) and (b), for the two outlet sizes, respectively. The experimental and simulation results are in good agreement. The $\psi$ and $\eta$ of the numerical simulation fluctuated slightly, and were slightly higher than the experiment data after $n = 110$ r/min. With increased $\phi$, $\psi$ decreased slowly and then dropped suddenly, while the $\eta$ values for the two outlet sizes exhibited different hump changes. For outlet size $a$ ($b$), the $\eta$ maximum appeared at 130–140 (110) r/min.

![Fig.15 Comparison between numerical and experimental dimensionless coefficients](image)

5. Performance comparison

To highlight the TRP performance advantages, the AB-1.25D performance parameters for outlet size $a$ and $n = 190$ r/min were compared with those for 50GDL12-15×7 (a multistage centrifugal pump) and BW60-5 (a piston pump) under the same $P$ (Table 3).

| Type          | $n$/r/min | $P$/kW | $q$/(m$^3$/h) | $p$/MPa |
|---------------|-----------|--------|---------------|---------|
| AB-1.25D      | 190       | 7.54   | 8.83          | 1.97    |
| 50GDL12-15×7  | 2900      | 7.5    | 8.4           | 1.26    |
| BW60-5        |           | 7.5    | 3.6           | 5       |

From the table, compared with the centrifugal pump at the same $P$, the $q$ of AB-1.25D slightly exceeded the 50GDL12-15×6 rated flow and the AB-1.25D $p$ was ~1.56 times the 50GDL12-15×7 rated pressure. Compared with the piston pump at the same $P$, the AB-1.25D $q$ was ~2.33 times the BW60-5 rated flow. In summary, the TRP $q$ can reach that of the centrifugal pump and far exceeds that of the piston pump, and its $p$ exceeds that of the centrifugal pump. Therefore, the TRP is suitable for conditions requiring large $q$ unachievable by the piston pump, or for conditions requiring $p$ unachievable by the centrifugal pump; thus, it can act as a replacement for those pumps in such cases.

6. Conclusion

In this paper, a double-cylinder TRP was designed. Its mechanical structure, working principles, and a mathematical model of the key structure were presented, and the design requirements of the key sealing components were discussed. The AB-1.25D TRP was tested at different outlet sizes and $n$. From a Fluent numerical calculation, the pump internal flow line, velocity vector, and pressure distribution were assessed. The following conclusions were drawn.

(1) For outlet size $a$ ($b$) and $n$ of 190 (160) r/min, $q$, $\Delta p$, $T$, and $P$ reached maxima simultaneously, at 8.83 (7.34) m$^3$/h, 1.97 (2.31) MPa, 265.71 (303.92) N·m, and 5.2 (5.1) kW, respectively.

(2) With increased $n$, the test-pump $\eta$ decreased slowly, with $\eta_a > 64.2\%$ and 61.4\%, respectively. The maximum $\eta_M$ at different outlet sizes were similar, at 86.2\% and 85.9\% for $\eta_M$ and $\eta_{Mb}$, respectively, and then they go down to 73\% and 71.9\%. At the same $n$, $\eta_{Vb} > \eta_{Va}$ at 96.8\% and 95.5\%, respectively, and $\eta_V$ varied little with $n$. With
increased $n$, $\eta_{\text{fl}}$ and $\eta_{\text{mb}}$ fluctuated smoothly, with average values 90.8% and 88.4%, respectively.

(3) Four vortices appeared in the W-C, for different working processes and areas. The W-C $p$ first increased and then decreased, and the maximum $p$ occurred at the beginning of the stable full-$p$ state.

(4) The experimental data for $\psi$, $\varphi$ and $\eta$ for the two outlet sizes agreed well with the numerical calculation, but there were still some discrepancies which came from assembly of equipment, sensors accuracy, numerical iterative and other artificial power losses.

(5) The TRP exhibits superior performance to existing pumps and can be applied in deep mine drainage, long-distance liquid transportation, grouting, etc.

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