Numerical investigation on the impact of the converging angle of the suction chamber on annular jet pumps

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Abstract. The internal flow of an annular jet pump is an annular wall jet developed in a limited space, which is affected by its structure, shape, area ratio and flow rate ratio. In this paper, numerical simulation of an annular jet pump with area ratio $A=1.75$ was conducted based on realizable $k$-$\varepsilon$ turbulence model. And the impact of the converging angle of the suction chamber to the pump performance and the expansion of annular jet is investigated. According to the simulated results, the following conclusion is obtained. With the converging angle of the suction chamber ($\alpha$) being $20^\circ$, the pump performance turns out to be the best. And with the increasing $\alpha$, the maximal axial velocity and the squeezing effect to the annular jet are larger, while the mixing level of the two flows drops. In addition, the half-width of the annular jet, in the suction chamber, is proportional to $\tan(\alpha/2)$ which is the tangent of the half angle of the suction chamber.

1. Introduction

According to the relative location between the nozzle and the suction duct, jet pump can be classified into two categories: center type jet pump and annular type jet pump (CJP and AJP for short). Comparing with the circular nozzle of CJP which is at the center of the suction duct, the nozzle of AJP is annular and surrounds the suction duct. Since the suction duct in AJP is in the center of the pump with large flow diameter, it is of great advantages to convey the flow containing large grains, such as living fish, mineral, nubby food (potato, onion) etc.

Recently, there are considerable investigations on CJP[1], while little for AJP. Shimuzu[2], Elger[3], Namiki[4], Gazzar[5], Long[6] did numerous studies on the performance and internal flow in AJP. H X Liu[8] conducted some pilot studies on the cavitating performance of CJP. Ignoring the converging and diffusing section, Long[7] investigated the expanding of the annular jet in the regular area of AJP and presented the distribution of the characteristic velocity and the velocity half-width.

The function of suction chamber is to enable the ambient flow obtaining a certain radical momentum before the primary and secondary flow mixing completely in the throat. The suction chamber of AJP, comparing with that of CJP, has greater impact on the performance. It is because that the turbulent wall jet in AJP can lead to considerably great friction loss during the process that the axial momentum is transformed into the radical momentum. So, in case of reducing the flow loss and improve the pump performance, focusing on optimizing the structure of the suction chamber is particularly needed. Consequently, as the subsequent investigation of the previous studies[6][7], this paper, on the ground of considering the real structures thoroughly, investigated the impact of the
converging angle of the suction chamber on the pump performance and the expanding of the jet with CFD method.

2. Models and Validation of the Flow Simulation

2.1. CFD Models

The structure of AJP in [2] was adopted as the stimulating prototype which is shown in Figure 1. Nozzle diameter \( D_t = 38 \text{mm} \), outlet pipe diameter \( D_0 = 55 \text{mm} \) and nozzle tip thickness \( t = 2 \text{mm} \) kept constant during the calculating process. Hence, the area ratio \( m \) kept constant at 1.75.

![Figure 1. Configuration of an annular jet pump.](image)

With the cross-section area of the throat and nozzle constant and the direction of the primary and secondary flow in same, the inner flow in AJP can be simplified to be axisymmetric flow. So only half of the flow section is needed for simulating and this will considerably curtail the computing cycles and workloads. During practical modeling, in order to ensure the flow field at inlet and outlet steady, the inlet was lengthened upstream as long as 100mm and an 100mm long straight duct was appended to the outlet of the diffuser.

The calculation domain of AJP is shown in Figure 2. In the section between the nozzle exit and the inlet of the throat, since the primary and secondary flow commence mixing accompanying by momentum shearing and energy exchanging, there exist intensive turbulence shearing force and pressure pulsation. Hence the mesh in this section was refined as shown in Figure 3. The grid number of the mesh was set as 50,000 at first and then was increased to 200,000 to confirm that the simulating result was grid independent.

![Figure 2. Calculation domain of AJP.](image)

![Figure 3. Mesh detail at the nozzle.](image)

![Figure 4. Comparison of pump performance between experimental data and CFD results.](image)
Controlled by RANS equations, the flow in AJP can be treated as steady, uncompressible, turbulence flow. Realizable $k-\varepsilon$ turbulence model which is suitable for modeling the flow with eddy and abruption, can forecast the annular jet expansion ratio accurately. Therefore realizable $k-\varepsilon$ turbulence model was adopted in this paper.

The equations are discretized by control volume technique, and the momentum equations second-order upwind scheme. The SIMPLE algorithm is applied to pressure-velocity coupling. As the average velocity and pressure monitored at the pump exit is steady and the residue is under $10^{-4}$, the computation is completed.

In addition, according to the preceding study [10,11], boundary conditions were set as follows. The inlet boundary chose to be uniform mass flow inlet and the outlet condition is designed as pressure outlet. The wall is no-slip and the boundary type of $x$-axis is set as symmetric.

2.2. Validation of CFD simulation

The experiment data in Shimizu et al.’s work[2] was used to validate the CFD results. Comparison between the numerical results and the experimental data were shown in Figure 4. As shown in the figure, the simulated performance including the pressure ratio $h$ and the efficiency $\eta$ agreed well with the experimental data, which validates the reliability of the CFD methods.

3. Results and Discussions

3.1. Effect of converging angle of suction chamber to performance

In order to investigate the impact of different $\alpha$ on AJP, it was set as 10°, 20°, 30°, 40° respectively with the structural dimension of the nozzle and diffuser constant.

Figure 5 indicates performance curves of AJP ($h$-$q$ and $\eta$-$q$ curve) under different $\alpha$. As shown in the figure, the pump performance with $\alpha=20^\circ$ tends to be the best, especially when $q<0.4$. As for $\alpha=30^\circ$ and $40^\circ$, the corresponding $h$ and $\eta$ both are lower than that with $\alpha=20^\circ$ with a constant $q$, and the larger the converging angle $\alpha$ is, the worse the pump performance is. However, $h$ and $\eta$ with $\alpha=10^\circ$ are relatively lower when $q<0.1$ and generally surpass that with other $\alpha$ as $q$ increasing. Especially when $q>0.4$, those performance curves with $\alpha=10^\circ$ and $20^\circ$ converge into one curve.

![Figure 5. Performance Curves of annular jet pump under different $\alpha$.](image)

Figure 6 shows the distribution of the pressure coefficient ($C_p$) along the axis of AJP under different $\alpha$, when $q=0.5$ which is corresponding to the optimum working condition. For different $\alpha$, $C_p$ differs great at the inlet of the suction chamber ($x/D_t=0$). When $\alpha$ is small, $C_p$ in the suction chamber varies slightly, and vice versa, which is because the length of the suction chamber is different. It is obvious that when $\alpha$ is as large as $40^\circ$ the corresponding $C_p$ drop to the minimum value in a considerable short distance along the axis, which will exert great flow losses. Therefore, with a smaller $\alpha$, the loss that caused by the structural sudden change tends to be smaller, while the frictional head loss caused by the incremental length of suction chamber becomes larger.
3.2. Effect of converging angle of suction chamber to jet expansion

For AJP, the jet expansion in suction chamber is susceptible to $\alpha$. As a result of the contracting cross sectional area, the maximal axial velocity $u_m$ subjects to a tendency of increasing and reaches to its maximal value at the junction of suction chamber and nozzle. The axial velocity profile at $x/L_c=0$, 0.5 and 1 under different $\alpha$ is presented in Figure 7.

As shown in the figure, at the cross section of $x/L_c=0$, the whole annular jet is squeezed to a certain extent which increases with the increasing $\alpha$, and the wall effect, near the wall, has been destroyed. The velocity distribution of the annular jet at $x/L_c=0.5$ tends to be uniform and $u_m$ hovers at the center of the jet, and Table 1 indicated $u_m/u_j$ at each cross sectional area when $q=0.5$. In addition, $u_m$ reaches to its peak value at the cross section of the throat entrance ($x/L_c=1$). As shown in Table 1, with $\alpha$ increasing, the mixing of the primary and secondary flow, which means that the expending and entraining of the annular jet, is weakened, even though $u_m$ is enlarged simultaneously. Specifically, the maximum value of $u_m/u_j$ is as high as 1.55 with $\alpha=40^\circ$, while it is only 1.14, when $\alpha$ drops to 10°. However it is when $\alpha=10^\circ$ that the mixing of the primary and secondary flow is the most intensive.

### Table 1. $u_m/u_j$ at each cross sectional area, $q=0.5$.  

| $\alpha$ ($^\circ$) | $x/L_c$ | $u_m/u_j$ | $y/D_t$ |
|---------------------|--------|-----------|---------|
| 10                  | 0      | 1.089     | 0.648   |
|                     | 0.5    | 1.098     | 0.561   |
|                     | 1      | 1.144     | 0.458   |
| 20                  | 0      | 1.115     | 0.642   |
|                     | 0.5    | 1.156     | 0.555   |
|                     | 1      | 1.320     | 0.459   |
| 30                  | 0      | 1.138     | 0.637   |
|                     | 0.5    | 1.186     | 0.560   |
|                     | 1      | 1.428     | 0.474   |
| 40                  | 0      | 1.186     | 0.637   |
|                     | 0.5    | 1.233     | 0.564   |
|                     | 1      | 1.554     | 0.479   |

Figure 8 present the distribution of axial velocity profile under different $\alpha$ at the cross sectional area of $x=80$mm. For the AJP with $\alpha=10^\circ$, this profile is in the suction chamber, while in the throat, for the AJP with other different $\alpha$. As shown in Figure 8, the width of the annular jet is nearly the same for each $\alpha$ and the expanding and entraining of the annular jet cease at $y/D_t=0.2$. The velocity distribution of the annular jet under $\alpha=20^\circ$, 30° and 40° are nearly the same. Hence the rate of the jet expanding is similar at the same cross sectional area for AJPs with different $\alpha$. Moreover, as $\alpha$ is increasing, $u_m$ at this profile tends to be greater. Therefore, the suction chamber serves as a throat to some extent.
The length scale of span wise flow is represented by the half width of the jet \( y_{1/2} \). For confined turbulent wall jet, Bandyopadhyay\[9\] propose that all the length scales vary linearly along the axial direction. The expanding ratio in the section of \( x/R>18 \) is shown as the following equation:

\[
\frac{dy_{1/2}}{dx} = 0.046
\]

\[\alpha=10^\circ\]
\[\alpha=20^\circ\]
\[\alpha=30^\circ\]
\[\alpha=40^\circ\]

**Figure 8.** The axial velocity profile at the cross sectional area of \( x=80\text{mm} \).

**Figure 9.** The length scale of \( u_{1/2} \) under different \( \alpha \).

However this is different from confined turbulent annular wall jet. In AJP, since the velocity at centerline keeps on varying along the axis, the variation of \( y_{1/2}\) can be divided into two sections, which is shown in Figure 9. The first section is the expanding section in suction chamber, in which the internal wall is defined as the origin of \( u_{1/2} \). The length scale of \( u_{1/2} \) under different \( \alpha \) and the axial location see a good linear relationship, and the corresponding expanding rate \( (dy_{1/2}/dx) \) is directly related to \( \alpha \) (shown as in Table 2) which can be approximately indicated by the following equation:

\[
\frac{dy_{1/2}}{dx} = \tan\left(\frac{\alpha}{2}\right)
\]

**Table 2.** Relationship between the length scale \( y_{1/2} \) and \( \alpha \).

| \( \alpha \) (°) | \( dy_{1/2}/dx \) | \( \arctan(dy_{1/2}/dx) \) (°) |
|----------------|------------------|-------------------------------|
| 10             | 0.1139           | 6.50                          |
| 20             | 0.1958           | 11.08                         |
| 30             | 0.2812           | 15.71                         |
| 40             | 0.3634           | 19.97                         |

The second section is in the throat. As shown in Figure 9, the length scale \( y_{1/2} \) of \( u_{1/2} \) under all different \( \alpha \) is on a straight line, and experiences a linear relationship with \( x/D_t \). Therefore, in the throat, \( \alpha \) impacts little on the distribution of \( y_{1/2} \) and \( dy_{1/2}/dx \) is a constant.

According to the preceding analysis, the expanding and mixing of the annular jet in AJP is affected by the suction chamber and the throat. The half-width of jet is proportional to the tangent of the half converging angle of the suction chamber.

4. Conclusions

This paper numerically simulated the inner flow of the annular jet pump by CFD techniques. The impact of converging angle of the suction chamber to its performance and the expansion of annular jet is mainly studied. The main results can be summarized as follows: (1) The converging angle of the suction chamber has great impact on the pump’s performance and the pump performance tends to be optimum when \( \alpha=20^\circ \); (2) In the suction chamber, the expanding rate of \( y_{1/2} \) under different \( \alpha \) is
proportional to \( \tan(\alpha/2) \); (3) In the throat, \( \alpha \) impacts little on the distribution of \( y_{1/2} \) and \( dy_{1/2}/dx \) is a constant.

These preceding conclusions built a solid foundation of the structural optimization for annular jet pump and comprehension of inner flow in annular jet pump.

**Notation:**
- \( x \)-the length along central axial of jet pump, set the origin at outlet of the nozzle
- \( y \)-the distance away from the wall of mixing throat or the suction chamber in the radial direction
- \( u \)-the velocity
- \( u_1, u_2 \)-velocity of primary flow and secondary flow at outlet of nozzle
- \( p \)-static pressure
- \( u_m, u_c \)-maximal velocity and axial velocity at arbitrary cross section
- \( u_{1/2} \)-characteristic half-velocity, \( u_{1/2} = (u_m + u_c)/2 \)
- \( y_{1/2} \)-half-width of jet, corresponding to the characteristic length of \( u_{1/2} \)
- \( C_p \)-pressure coefficient, \( C_p = (p-p_0)/(0.5\rho u_0^2) \)
- \( \alpha \)-converging angle of suction chamber
- \( m \)-area ratio, \( m = A_{th}/A_j \)
- \( A_{th} \)-cross sectional area of throat
- \( A_j \)-cross sectional area of nozzle outlet
- \( q \)-flow rate ratio, \( q = Q_s/Q_o \)
- \( Q_s \)-secondary flow rate
- \( Q_o \)-primary flow rate
- \( h \)-pressure ratio,
\[
h = \left[ \left(p_s + \frac{y^2}{2g}\rho_s g + z_s\rho_s g\right) - \left(p_i + \frac{y^2}{2g}\rho_i g + z_i\rho_i g\right) \right] \times \left[ \left(p_s + \frac{y^2}{2g}\rho_s g + z_s\rho_s g\right) - \left(p_i + \frac{y^2}{2g}\rho_i g + z_i\rho_i g\right) \right]^{-1}
\]
- \( \eta \)-efficiency, \( \eta = hq(1-h)^{-1} \)

Subscripts: c-mixed flow; s-secondary flow; o-primary flow

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