NUMERICAL STUDY ON SWIRLING FLOW AND SEPARATION PERFORMANCE OF SWIRL VANE SEPARATOR

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Full knowledge of gas-liquid flow in a swirl vane separator is essential for separator design and performance improvement. In this paper, three-dimensional numerical investigations on swirling flow and separation performance are carried out. The detailed flow field, pressure loss, and grade efficiency are obtained numerically. Results show that a long and narrow wake is induced downstream of the swirl vanes and the static pressure in this region is lower than that near the external wall. The pressure loss in the swirl vane part occupies a large proportion of the total pressure loss. The gas velocities, including axial and tangential components, decrease a relatively small amount downstream of the swirl vanes and the strong swirling flow can persist for a long distance until the top outlet. The radial velocity is at least one order less than the other two velocities and has little influence on the droplet separation. The distribution of the droplet size has a great influence on separation performance and there exists a critical value for both grade efficiency and pressure loss. The critical droplet size depends strongly on the separator design and operation conditions.

KEY WORDS: swirl vane separator, swirling flow, flow field, separation performance

1. INTRODUCTION

Swirl vane separators in nuclear reactors are reasonable for separating more than 80% of liquid water from the steam-water mixture to assure quality steam to dryers and turbines (Green and Hetsroni, 1995). The advantage of this type of steam separator is its geometrical simplicity and high efficiency. However, the flow field inside the separator is extremely complicated due to the strong turbulence, three dimensionality, and rotational motions. Since the separator performance is directly related to the distribution of gas-liquid flow, for the purpose of separator design and performance improvement, a deep understanding of gas-liquid swirling flow and its relationship with separator performance is essential.

In the PSI ARTIST project (Güntay et al., 2004; Ogino et al., 2008), the flow field in the full-sized swirl vane separator within a downscaled steam generator was measured by using the laser Doppler velocimeter (LDA) technique. Also, the phenomenon of droplet retention in the separator was studied. Kataoka et al. (2008, 2009a,b) conducted a downscale model test of a swirl vane separator using air and water as working fluids. The two kinds of flow patterns inside the separator, including annular and churn flow, were investigated and their results showed that the separation efficiency is sensitive to the flow patterns. On this basis, the flow pattern, liquid film thickness, separation efficiency, and distributions of droplet diameter in an air-water mixture flow were measured to understand the characteristics of the two-phase swirling flow and to establish an experimental database applicable to the model test. Xiong et al. (2014) studied the effects of flow pattern on the separation efficiency of a swirl vane separator and concluded that the separation efficiency in annular flow is higher than that in mist and churn flow. Liu and Bai (2016) analyzed the features of the droplet trajectory in a swirl vane separator and established theoretical models for critical droplet
diameter, grade efficiency, and overall separation efficiency, respectively. Also, the scaling laws which enable a comparison between the downscaled separator and a full-sized separator under similar conditions were developed. By using the computational fluid dynamics (CFD) numerical method, Chaki and Murase (2006) investigated the swirling flow characteristics in the swirl vane section and introduced a new drag coefficient model to improve the calculation of centrifugal force. Xiong et al. (2014) studied the droplet distribution effect on the separator performance and found that the separation efficiency is not sensitive to the size of the big water droplets, but it is affected significantly by the microscale water droplets. In addition, a considerable number of studies focused on the optimization of swirl vanes (swirler) were conducted to obtain the ideal geometries and features. For example, Ikeda et al. (2003) proposed a swirler having an almost uniform cross-sectional area in the flow direction, by which they decreased the pressure loss by about 20%. Nakao et al. (2001) measured pressure loss and separation performance for a swirler with lowered vane angle and lowered hub diameter. They confirmed that the reduction of vane angle and hub diameter is effective in decreasing pressure loss. Matsubayashi et al. (2012) designed an improved swirler, which consists of a small hub and six modified vanes. This swirler can effectively reduce pressure loss while keeping high separation performance.

Although many investigations have been carried out to study the separation efficiency and pressure loss of the swirl vane separator, gas-liquid swirling flow and reasonable methods for predicting the flow field are still not fully understood, which are indispensable for separator design and performance improvement. To address this, in the present work, three-dimensional numerical calculations are carried out with different inlet droplet groups. The results provide the details of the pressure and velocity distributions and the decay law of the swirling flow is also obtained. Moreover, an estimating separation efficiency and pressure loss taking note of the influence of droplet diameter are developed. The numerical results not only show the gas-liquid swirling flow characteristics of the swirl vane separator but also have practical significance for the separator design and performance improvement.

### NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tr>
<td>$c$</td>
<td>constant</td>
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<tr>
<td>$C_D$</td>
<td>drag coefficient</td>
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<tr>
<td>$C_l$</td>
<td>lift force coefficient</td>
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<tr>
<td>$d_1$</td>
<td>diameter of inlet pipe (m)</td>
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<td>$d_2$</td>
<td>diameter of central hub (m)</td>
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<td>$d_3$</td>
<td>diameter of top outlet (m)</td>
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<tr>
<td>$d_p$</td>
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<tr>
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<td>Euler number</td>
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<td>$F_{vm}$</td>
<td>visual mass force (N)</td>
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<td>Reynolds number</td>
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<td>$S_w$</td>
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### Greek Symbols

<table>
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<tr>
<th>Symbol</th>
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<td>volume fraction</td>
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<tr>
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<tr>
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<td>density (kg ⋅ m$^{-3}$)</td>
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<tr>
<td>$\tau$</td>
<td>shear stress (Pa)</td>
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2. NUMERICAL SIMULATION

2.1 Geometric Construction

Figure 1 shows the configuration of the modeled swirl vane separator. It has one inlet at the bottom and three different outlets at the top. The steam-water mixture enters the separator through the inlet pipe and the water is driven toward the cylinder wall by the centrifugal force generated by the swirl vanes. When reaching the inner wall, the droplet deposits on it and forms a liquid film which is finally removed from the steam through the outlets. Generally, a majority of the liquid film exits from outlet 3 and another small part exits from outlet 2. The steam, including some unseparated droplets, leaves the separator through outlet 1. The diameter of the inlet pipe is 501 mm ($d_1$) and 1350 mm in height ($h$). The diameter of the cylinder is 544 mm ($D$) and 1500 mm in height ($H$). The diameter of the central hub is 174 mm ($d_2$) and there are four swirl vanes fixed on the hub. The height of the swirl vanes is 400 mm ($l$) and the vane angle is fixed to $30^\circ$ with respect to the radial direction. The diameter of the top outlet is 420 mm ($d_3$) and 287 mm in height ($L$). In the calculation, the origin point of the coordinate is located at the center of the inlet pipe; i.e., $z = 0$ at the inlet of the pipe.

Considering the geometric complexity of the swirl vanes, the separator model is divided by using the unstructured tetrahedral grids in the swirl vane part and using structured hexahedral grids in the pipe part. A fine grid is created all along the surface of the swirl vanes and cylinder wall to capture the free velocity gradient. Three different grid systems, i.e., 15 million mesh elements, 20 million mesh elements, and 25 million mesh elements have been used for a grid independence test to analyze the total pressure loss of the separator. Result shows that 15 million mesh elements already satisfy grid independence.

In practical working conditions, the inlet volume fraction of the liquid phase is generally less than 10% (Griffith, 1997), which makes the dilute dispersion assumption possible. Then the steam-water flow at the inlet can be treated as a homogeneous mist flow, i.e., continuous steam and evenly dispersed liquid droplets in the steam core. The Euler-Euler multiphase model is adopted in the present work to simulate the steam-water flow. In this model, the droplets are assumed to have a sphere shape and the collision, coalescence, and breakup of the droplets are not considered due to the low liquid volume fraction. Depending on the power density of the reactor core, the droplet size ranges from 0

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**FIG. 1:** Schematic of the modeled swirl vane separator
to 200 \mu m \text{ (Kolev, 2006)}. In view of this, numerical calculations are carried out on this geometry with seven different droplet diameters at inlet \((d_p = 1, 10, 50, 70, 100, 120, 150 \mu m)\).

### 2.2 Description of Numerical Model

#### 2.2.1 Basic Equations

For three-dimensional steady incompressible flows, mass and momentum conservation equations for gas and liquid flow are given as follows:

\[
\nabla \cdot \left( \alpha_q \rho_q \vec{u}_q \right) = 0, \tag{1}
\]

\[
\nabla \cdot \left( \alpha_q \rho_q \vec{u}_q \vec{u}_q \right) = -\alpha_q \nabla p + \alpha_q \rho_q \vec{g} + \nabla \cdot \left[ \alpha_q \left( \tau_q + \tau'_q \right) \right] + M_q, \tag{2}
\]

where \(\alpha_q\) is the volume fraction; \(\rho_q\) the density; \(\vec{u}_q\) the flow velocity; \(p\) the pressure shared by all phases; \(\tau_q\) and \(\tau'_q\) the shear stress and turbulent shear stress, respectively. The subscript \(q\) means \(q\) phase. The \(\alpha_q\) satisfies

\[
\alpha_c + \alpha_d = 1, \tag{3}
\]

where the subscripts \(c\) and \(d\) stand for the continuous gas phase and dispersed liquid phase, respectively. The term \(M_q\) represents the interface force and satisfies

\[
M_q = F_{drag} + F_{lift} + F_{vm}, \tag{4}
\]

where \(F_{drag}\), \(F_{lift}\), and \(F_{vm}\) are drag force, lift force, and virtual mass force, respectively.

The drag force \(F_{drag}\) is calculated according to the following equations (Gosman et al., 1992):

\[
F_{drag} = A_d (\vec{u}_c - \vec{u}_d) + A_d \frac{\nabla \cdot \vec{u}_c}{\alpha_d \alpha_c \alpha_\alpha} \nabla \alpha_d, \tag{5}
\]

\[
A_d = \frac{3 \alpha_d \rho_c C_D}{4 d_p} |\vec{u}_c - \vec{u}_d|, \tag{6}
\]

\[
C_D = \left\{ \begin{array}{ll}
24 \frac{1 + 0.015 \text{Re}_d^{0.687}}{\text{Re}_d} & \text{Re}_d \leq 1000 \\
0.44 & \text{Re}_d > 1000
\end{array} \right., \tag{7}
\]

\[
\text{Re}_d = \frac{\rho_c d_p |\vec{u}_c - \vec{u}_d|}{|\vec{u}_c|}, \tag{8}
\]

where \(C_D\) is the drag coefficient and \(d_p\) is the mean diameter of the liquid droplet.

The lift force \(F_{lift}\) is calculated as (Lance and Bateille, 1991):

\[
F_{lift} = -C_l \alpha_d \rho_c (\vec{u}_c - \vec{u}_d) \times (\nabla \times \vec{u}_c), \tag{9}
\]

where \(C_l\) the lift coefficient, equals 0.5.

The virtual mass force \(F_{vm}\) is calculated as (Auton et al., 1988)

\[
F_{vm} = 0.5 \alpha_d \rho_c (\vec{u}_c \cdot \nabla \vec{u}_c - \vec{u}_d \cdot \nabla \vec{u}_d). \tag{10}
\]
2.2.2 Turbulence Model

Considering the three-dimensional swirling flow induced by the swirl vanes, the RNG $k$–$\varepsilon$ turbulence model is adopted to simulate the steam-water flow in the swirl vane separator. The effect of swirl on turbulence is included in the RNG $k$–$\varepsilon$ model, which enhances the accuracy for swirling flows.

In the RNG $k$–$\varepsilon$ model, transport equations for the turbulence kinetic energy $k$ and turbulence dissipation $\varepsilon$ are solved:

$$\frac{\partial}{\partial x_i} \left( \rho k u_i \right) = \frac{\partial}{\partial x_j} \left( \alpha_k \mu_{e,ff} \frac{\partial k}{\partial x_j} \right) + G_k - \rho \varepsilon + S_k,$$

(11)

$$\frac{\partial}{\partial x_i} \left( \rho \varepsilon u_i \right) = \frac{\partial}{\partial x_j} \left( \alpha_\varepsilon \mu_{e,ff} \frac{\partial \varepsilon}{\partial x_j} \right) + C_{\varepsilon_1} \frac{\varepsilon}{k} G_k - C_{\varepsilon_2} \rho \varepsilon^2 \frac{\varepsilon}{k} - R_\varepsilon + S_\varepsilon,$$

(12)

where quantities $\alpha_k$ and $\alpha_\varepsilon$ are the inverse effective Prandtl numbers for $k$ and $\varepsilon$, respectively. $G_k$ represents the generation of turbulence kinetic energy due to the mean velocity gradients. $C_{\varepsilon_1}$ and $C_{\varepsilon_2}$ are model constants ($C_{\varepsilon_1} = 1.42, C_{\varepsilon_2} = 1.68$). $\mu_{e,ff}$ is the effective viscosity. $S_k$ and $S_\varepsilon$ are user-defined source terms. Other parameters in Eqs. (11) and (12) are given by (Ansys Fluent, 2011)

$$S_{i,j} = \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right), \quad \beta = 0.012, \quad \eta_0 = 4.38$$

2.3 Boundary Conditions and Solver Parameters

The steam and water are used as working fluids and the calculations are conducted under the same condition as that in actual nuclear reactor operation, i.e., a pressure of 7.13 MPa and a temperature of 287°C. At the inlet, a velocity inlet boundary condition is applied where the steam-water mixture superficial velocity $U_b$ is 5.27 m/s. The turbulence intensity is calculated by $I = 0.16 \text{Re}^{-1/8}$, where Re is the Reynolds number based on the inlet pipe diameter. Then a turbulent intensity of 2.3% is defined at the inlet. The initial liquid phase volume fraction $\alpha_l$ is 0.096. The boundary conditions at the separator outlets are prescribed as a fully developed pipe flow and all are pressure-out conditions. The outlet pressures $p_{out}$ are set to 7.13 MPa. No-slip conditions are assumed at the wall. For the grid nodes near the wall, they are approximated and treated using standard wall function.

The steam-water flow in a swirl vane separator is simulated by commercial Fluent code. The pressure-velocity coupling uses phase-coupled SIMPLE and a first-order upwind interpolation scheme is adopted in all numerical simulations. In the process of computation, the default relaxation factors are reduced appropriately to solve the convergence difficulties and the mass conservation of each phase at the inlet and outlets is monitored to determine that the calculation has reached a steady state.

3. RESULTS AND DISCUSSION

3.1 Swirling Flow Field

3.1.1 Static Pressure

Figures 2(a) and 2(b) show the typical contour plots and radial distributions of the steam static pressure. It can be seen that as the swirl vanes induce a rotational motion, the steam static pressure decreases from the cylinder wall to the center in the radial direction, and a low-pressure zone appears in the center. The pressure gradient is large along
the radial direction and there exists a high-intensity vortex at the exit of the swirl vanes. At the region downstream of the swirl vanes, the axial variation of the static pressure is fairly low, indicating that the static pressure loss is small in the separator cylinder.

The static pressure loss within a separator is contributed to both local losses and frictional losses. The local losses include the contraction loss at the inlet, the expansion loss at the outlet, and local loss mainly in the swirl vanes part, while the frictional losses include the swirling loss due to the wall friction and the dissipation loss of the dynamic energy of the steam. Figure 3 shows the area-average static pressure distribution of the steam flow. Results indicate that the steam expands at the exit of the swirl vanes due to the increasing flow area ($z/D = 3$), leading to a small static pressure recovery ($z/D = 3$ to 4). Then the static pressure decreases a little along the axial direction of the separator cylinder ($z/D = 4$ to 5) while it decreases sharply in the swirl vane part ($z/D = 2$ to 3), which is the main pressure loss in the swirl vane separator.

**FIG. 2:** Static pressure distribution of steam flow: (a) typical contour plots; (b) radial distributions

**FIG. 3:** Area-average static pressure distribution of steam flow along axial direction
3.1.2 Velocity Magnitude

Figures 4(a) and 4(b) show the streamlines of the steam flow and the radial distributions of the steam velocity magnitude. As the swirl vanes induce a rotational motion that creates a radial pressure gradient, a strong swirling flow and a narrow wake region are formed downstream of the swirl vanes, which can be kept a long distance until the top outlet. Due to the high-intensity vortex and wake flow, the streamline of the steam flow curves and the velocity magnitude in the core region is relatively low, resulting in the appearance of the flow reversal and several circulating zones near the swirl vanes. This phenomenon is unfavorable for the droplet separation because, for the small droplets, their centrifugal force is not large enough to overcome the radial resistance and then the small droplets will be entrained by the downward steam stream in this region and flow upstream, thus increasing the eddy loss and decreasing the separation efficiency.

3.1.3 Velocity Component

Figures 5(a) and 5(b) show the tangential and axial velocity component distributions of the steam flow. The profile of tangential velocity exhibits a hump-type distribution which is an expected quasi-forced and quasi-free combination of the Rankine-type vortex. The tangential flow has a three-region structure consisting of core, annular, and wall regions. The core region is featured by a forced vortex motion and the flow in this region is stabilized due to the radial pressure gradient and the turbulence is suppressed. In the annular region, which is similar to the free vortex motion, the flow is unstable and strong turbulence is generated. In the wall region, the velocity gradient is quite steep and the value of the tangential velocity equals zero on the wall surface. Since the steam-water flow downstream of the swirler is generally an annular flow with low flow resistance, the variation of the tangential velocity along the axial direction is small in the separator cylinder shown in Fig. 5(a). For the axial velocity of the steam flow, it has the shape of an inverted “W” type distribution, which is the same as the experimental result of Kapulla et al. (2008). The reverse flow appears in the axial velocity near the central axis due to the radial and axial pressure gradient in the central zone, which is consistent with the experimental study of Ogino et al. (2008). Compared with the tangential velocity distribution, the axial velocity in the central core is more flat and the velocity magnitude also reaches a peak value near the cylinder wall.

As the tangential velocity profile can be described as a Rankine vortex shown in Fig. 6, then the tangential velocity distribution in the radial direction can be expressed as

\[ u_{\theta} = \frac{C}{r^n}, \]  

\[ u_{\theta} = \frac{C}{r^n}, \]  

(14)

**FIG. 4:** Velocity magnitude distribution of steam flow: (a) streamlines of steam flow; (b) radial distributions
where $C$ is a constant, $r$ is the radius, and exponent $n$ depends on $r$. In the core region near the axis, $n$ is close to $-1$ (forced vortex), whereas $n$ approaches 1 (free vortex) near the wall.

Figures 7(a) and 7(b) show the radial velocity distribution and the velocity comparison between radial, axial, and tangential components. It can be seen that the steam radial velocity is about an order lower than the other two velocity components, which has also been confirmed by Chen et al. (1995) and Hoffmann (2002), who studied the down-exhaust cyclone separator for gas-solid separation. Consistent with our previous theoretical study on the gas-liquid separation in a swirl vane separator, the steam radial velocity is found to have little influence on the radial motion of the droplets (Liu and Bai, 2016).

### 3.1.4 Swirl Intensity

For engineering purposes, it is important to understand the decay law of the swirling flow in a separator, because this application requires sufficiently large swirl intensity to maintain separation performance. The intensity of the
swirling flow can be quantified by the computation of the corresponding swirl number; as defined in Eq. (15), a nondimensional angular momentum flux, is the ratio of angular to axial momentum flux (Gupta et al., 1984).

\[ S_w = \frac{\int \rho_m u_z u_\theta r dA}{R_o \int \rho_m u_z^2 dA}, \]  

where \( u_z \) and \( u_\theta \) are axial and tangential velocities, respectively. \( \rho_m \) is the mixture density, which can be calculated as

\[ \rho_m = \alpha_g \rho_g + \alpha_l \rho_l. \]  

Figure 8 shows the swirl intensity distribution of steam-water flow along the axial direction of the separator. The trend in the figure indicates that near the exit of the swirl vanes (\( z/D = 3.0 \) to 4.0), the steam expands and the axial velocity decreases, leading to large swirl intensity in this region. Then due to the friction of wall and viscous dissipation, the swirl intensity weakens as it moves downstream. On the whole, the decay of the swirl intensity is small in the separator cylinder, indicating that the strong swirling flow induced by the swirl vanes can persist for a long distance and result in high droplet separation efficiency.

### 3.1.5 Liquid Phase Volume Fraction

Figure 9 shows the volume fraction distribution of the liquid phase. As the swirl vanes create a redistribution of the downstream velocity field from axial to tangential, accompanied by a transfer of axial to angular momentum, the higher-density droplets first deposit on the upper swirl vane wall surface and then are thrown to the cylinder wall as a rising, thin liquid film. At a distance downstream of the swirl vanes, the steam and water distribute uniformly in the cylinder and an obvious stratification phenomenon appears, indicating that the steam and liquid droplet separate from each other successfully. Most of the liquid film exits from outlet 3 and another small part exits from outlet 2. The steam, including some unseparated droplets, leaves the separator through outlet 1. This steam-water flow pattern is qualitatively the same as that in the real swirl vane separator.
3.2 Separation Performance

3.2.1 Grade Efficiency

Figures 10(a) and 10(b) show the effect of droplet diameter on the separator performance, i.e., grade efficiency and pressure loss. In general, the grade efficiency increases quickly when the inlet droplet diameter is less than approximately 45 μm, and when diameter reaches the critical value (about 65 μm), the grade efficiency keeps a maximum value of 100% as the droplet diameter is further increased. Our previous theoretical model for the grade efficiency showed that the critical value of droplet diameter is about 52 μm under the same conditions (Liu and Bai, 2016). The main reason for the discrepancy is that in the theoretical model development, the narrow wake induced by the swirl vanes was assumed to have the same size as the central hub, which, however, is smaller than the hub diameter as shown in the present numerical result [see Fig. 4(a)]. Since the central hub directly determines the droplet...
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3.2.2 Pressure Loss

The pressure loss is defined as the difference between the static pressure at the inlet and outlet. As shown in Fig. 10(b), when increasing the inlet droplet diameter, the pressure loss increases sharply at first and decreases slightly afterward. As the small droplets usually have a good following with the steam flow, the degree of turbulence is relatively small downstream of the swirl vanes and the pressure loss is mainly caused by the wall friction. However, when droplet diameter increases, the centrifugal force acting on the droplets is large enough that most of the droplets can be separated. At first, the droplets deposit on the upper swirl vane wall surface and then are thrown to the cylinder wall as a rising, thin liquid film (see Fig. 9). The impact of the swirl vanes on the large droplets increases the turbulence in the swirl vane passage, leading to an increase of the eddy loss and kinetic energy loss. Therefore, there exists a critical droplet size for both grade efficiency and pressure loss. The critical droplet value depends strongly on the separator design and operation conditions. This is similar to the studies of Baskakov et al. (1990) using reverse flow cyclones for gas-solid separation and Chen et al. (1999) using a down-exhaust cyclone separator for gas-solid separation.

4. CONCLUSIONS

In this paper, three-dimensional numerical investigations on steam-water swirling flow and separation performance have been carried out for the swirl vane separator. From the results, the following conclusions can be obtained:

1. A narrow wake flow is induced downstream of the swirl vanes which can persist for a long distance until the top outlet. The static pressure in this region is lower than that near the external wall, resulting in the appearance of flow reversal and several circulating zones near the swirl vanes, which is unfavorable for the small droplet separation.
2. The radial velocity of steam flow is at least one order less than the other two velocity components and has little influence on the droplet separation. The axial and tangential velocity components decrease is relatively small in the separator cylinder, thus leading to a high separation efficiency.

3. The pressure loss in the swirl vanes part occupies a large proportion of the total pressure loss while the pressure loss downstream of the swirl vanes is relatively small.

4. The distribution of the droplet size has a great influence on the separation performance and there exists a critical value for both grade efficiency and pressure loss.

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