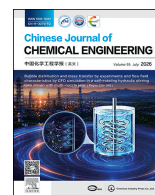




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Full Length Article

Flow field characteristics analysis of venturi ejector improved by swirler using CFD simulation

Yunlong Wang¹, Jijun Ge², Zheng Kuang¹, Guoxuan Li¹, Pan Xu^{1,*}, Rongshan Bi^{1,*}¹ College of Chemical Engineering, Qingdao University of Science and Technology, Qingdao 266042, China² Qingdao Shenwo Chemical Technology Co., Ltd., Qingdao 266073, China

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ABSTRACT

The ejector, a key component of the jet loop reactor for efficient gas–liquid mass transfer, has its structural design and optimization as the core of process intensification. In this work, based on the self-designed down-jet venturi ejector, water–air is used as the working medium, and a swirler structure is added to the nozzle. Combined with the computational fluid dynamics (CFD) method, the changes of the internal flow field of the ejector in the presence of the swirler are analyzed in depth. The effects of liquid circulation rate (Q_L), nozzle size, and swirler angle on the performance of the ejector are investigated. The results show that under the operating conditions of $0.55 \text{ m}^3 \cdot \text{h}^{-1}$, the combination of a 3 mm nozzle and 30° swirler can significantly improve the ejector's suction volume (Q_G) and gas–liquid interface area density (IAD). Q_G reaches $3.22 \text{ m}^3 \cdot \text{h}^{-1}$, and IAD is $5246.8 \text{ m}^2 \cdot \text{m}^{-3}$. This study reveals the internal hydrodynamic mechanism of the swirler to enhance the performance of the ejector. It provides theoretical guidance for the design and operation optimization of high-performance swirl ejectors.

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1. Introduction

In the chemical reaction process, efficient gas–liquid mass transfer is the core challenge to achieve process intensification. The jet loop reactor effectively alleviates the problems of low mixing efficiency and high energy consumption of traditional reactors due to its unique jet circulation mixing mechanism [1–3]. By enhancing the interphase contact at the micro scale, the reactor simultaneously achieves ultra-high mass transfer rate and uniform distribution of the reaction field. The key is to use the high-speed jet generated by the ejector to form a strong shear circulation flow field in the confined channel, so as to achieve efficient gas–liquid two-phase entrainment and mixing [4,5]. As the core driving component of the reactor performance improvement, the design and performance optimization of the ejector are crucial [6].

The ejector mainly comprises the nozzle, the suction chamber, the contraction section, the throat, and the diffusion section. Its high-efficiency mass transfer function is derived from the synergistic effect of various components in the working process [7,8]. The primary fluid is ejected at high speed through the nozzle to form a local low-pressure zone, thereby ejecting the surrounding

fluid. The suction chamber ensures the smooth entry of the entrained fluid, stabilizes the flow field, and provides the basis for subsequent mixing. The contraction section further enhances the jet effect by reducing the cross-sectional area of the flow, accelerating fluid mixing, and promoting turbulence development. The turbulence intensity is significantly enhanced in the throat, so the gas–liquid two-phase can be fully contacted and broken, thus significantly improving the interphase mass transfer efficiency [9,10]. Finally, the diffusion section gradually expands the cross-sectional area of the flow channel, effectively converts the kinetic energy of the fluid into pressure energy, restores the system pressure, and stabilizes the gas–liquid mixture to ensure the completion of the efficient mass transfer process. Based on its efficient gas–liquid mass transfer characteristics, ejectors are widely used in chemical engineering [11–13], environmental governance [14–16] and energy systems [17–19] to enhance the reaction and transfer process, optimize the efficiency of waste gas wastewater treatment, and promote the reaction rate and product conversion performance in the energy conversion process. In order to further improve its mixing effect, working performance, and reduce energy consumption, it is of great research value and application significance to deeply understand the internal flow and mixing characteristics of the ejector, and to improve the mass transfer efficiency and overall performance by optimizing the structural parameters and operating conditions.

* Corresponding authors.

E-mail addresses: xupan5226@163.com (P. Xu), birongshan@163.com (R. Bi).

Aiming at the performance research and optimization of the down-jet ejector, early research mainly relied on experimental methods to carry out much of the work. For instance, Dutta and Raghavan [20] compared the mass transfer and hydrodynamic characteristics of ejectors equipped with a straight throat and a venturi throat in the loop reactor. The results show that the venturi ejector can achieve better primary dispersion in the diffusion section, especially in the case of low ejector's suction volume (Q_G), and it can achieve excellent dispersion of bubbles. In addition, the study also found that the growth rate of the gas–liquid interface area of the venturi ejector was significantly higher than that of the straight throat ejector. Ben Brahim *et al.* [21] studied the momentum transfer characteristics of this type of ejector with water, ethylene glycol, and oil as the working fluids and air as the entrained fluid. They established the total loss factor and the corresponding entrainment rate correlation through theoretical derivation. Havelka *et al.* [22] focused on the optimization of the structure of the swirler on the performance of the ejector. The experiment confirmed that introducing the swirler in the nozzle can significantly improve the Q_G . However, although experimental methods can obtain reliable macroscopic performance data, they often consume a lot of workforce and material costs, and it is difficult to obtain fine flow field information inside the ejector. This limits the in-depth understanding of the mass transfer mechanism.

In order to overcome the above limitations and further explore the complex flow mechanism inside the ejector, computational fluid dynamics (CFD) technology has become a valuable research tool in this field due to its increasingly mature computing power and economic efficiency [23,24]. The CFD method has significantly promoted understanding of the key physical processes inside the ejector, such as gas–liquid two-phase mixing characteristics, pressure and velocity distribution, and flow state evolution [25]. For instance, Zhang *et al.* [26] used the CFD-PBM coupling method to study the ejector's flow and mass transfer characteristics, successfully predicted the bubble size distribution, and established a correlation between the gas–liquid mass ratio and the local volume mass transfer coefficient in the throat center. Xu *et al.* [27] systematically investigated the influence of nozzle structure parameters on the performance of the ejector by using CFD technology. The calculation results show that the optimal scheme is that the nozzle outlet velocity is $21.8 \text{ m} \cdot \text{s}^{-1}$, the nozzle inlet velocity is equal to the outlet velocity of the centrifugal pump, and the nozzle contraction angle is 20° . Zhou *et al.* [28] innovatively designed a new type of venturi ejector with double inlet holes. CFD simulation and experimental verification show that opening the second inlet can effectively optimize the flow field distribution in the ejector, promote gas–liquid mixing and bubble dispersion, and increase the gas–liquid contact area. These studies fully demonstrate the powerful ability of CFD in revealing the complex multiphase flow and mass transfer mechanism inside the ejector, and lay the foundation for its application to more complex structural optimization.

In terms of swirler optimization, although early experiments have confirmed that it has a significant effect on improving the Q_G value of the ejector, there are still obvious deficiencies in the current research. On the one hand, the experimental method is difficult to reveal the fine flow field characteristics and phase interaction mechanism inside the ejector under the action of swirler. On the other hand, although CFD technology has been applied to the study of ejectors, there is still a lack of systematic research on the internal flow structure, turbulence characteristics and mass transfer law under the coupling of swirl field and jet

field. In particular, the influence of key geometric parameters and operating parameters of swirlers has not been fully elucidated. In order to fill the above research gaps, this study designed the structure of the down-jet venturi ejector and the supporting swirler. With water and air as the working medium, the advanced CFD numerical simulation method was used to systematically study the complex gas–liquid two-phase flow characteristics and mass transfer law inside the ejector with the swirler structure. The effects of key operating parameters and geometric parameters on the performance of the ejector were investigated, which provided a solid theoretical basis and data support for the accurate design and optimization of high-performance swirl ejectors.

2. Governing Equation and Calculation Specification

2.1. Multiphase flow model

In this study, the working fluid of the ejector is water and the entrained fluid is air. In order to simulate the gas–liquid two-phase flow in the ejector, the Eulerian–Eulerian (E-E) multiphase flow model is selected [29]. The E-E model regards the gas–liquid two-phase as a continuous medium, and describes its flow characteristics by solving their respective mass and momentum conservation equations. The model can effectively deal with the interaction between gas and liquid phases, especially for the case of high flow rate and strong interaction between two phases in the ejector.

In this model, the pressure field is shared by each phase, and the mass and momentum control equations of each phase are solved independently. Eq. (1) and Eq. (2) show the specific forms of mass and momentum control equations.

$$\frac{\partial(\alpha_i \rho_i)}{\partial t} + \nabla \cdot (\alpha_i \rho_i \mathbf{u}_i) = 0 \quad (1)$$

$$\frac{\partial(\alpha_i \rho_i \mathbf{u}_i)}{\partial t} + \nabla \cdot (\alpha_i \rho_i \mathbf{u}_i \mathbf{u}_i) = -\alpha_i \nabla P - \nabla(\boldsymbol{\tau}_D + \boldsymbol{\tau}_t) + \mathbf{F}_i + \alpha_i \rho_i \mathbf{g} \quad (2)$$

where i represents different phase fluids, α_i , ρ_i , $\mathbf{u}_i$, and P_i represent the volume fraction, density, velocity, and pressure of the i -th phase fluid, respectively. $\boldsymbol{\tau}_D$ and $\boldsymbol{\tau}_t$ represent the viscous stress tensor and the Reynolds stress tensor, respectively. $\mathbf{F}_i$ represents the interaction force applied by the additional interaction described in the two-fluid model, including drag force, lift force, virtual mass force, surface tension, *etc.* [26]. $\mathbf{g}$ represents the gravitational acceleration.

2.2. Turbulence model

In the simulation of the ejector, the commonly used turbulence models include the standard $k-\epsilon$ model and the realizable $k-\epsilon$ model. The standard $k-\epsilon$ model may have inaccurate results when simulating complex flows such as turbulent boundary layer, recirculation zone, and strong shear flow. Especially in some flow structures, the standard model's prediction often deviates from the experimental value. In order to solve these limitations, the realizable $k-\epsilon$ model is improved based on the standard $k-\epsilon$ model. The realizable $k-\epsilon$ model modifies the turbulent dissipation rate (ϵ) equation and introduces a new turbulence generation term. Through this improvement, the realizable $k-\epsilon$ model can simulate high Reynolds number turbulent flow more accurately, avoiding the non-physical understanding of the standard $k-\epsilon$ model in some flow scenarios. Considering the complexity and

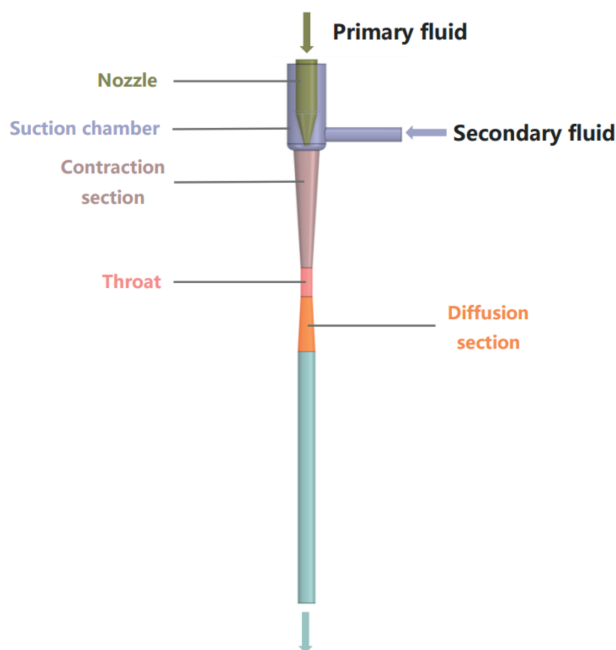


Fig. 1. The geometric construction of the ejector.

turbulence characteristics of the gas–liquid two-phase flow after adding the swirler to the venturi ejector, especially in the case of a strong vortex, complex boundary layer, and high velocity gradient, the realizable $k-\varepsilon$ model is more suitable for the turbulence simulation of this study than the standard $k-\varepsilon$ model [7,26,30]. The calculation formulas are as follows,

$$\frac{\partial(\rho k)}{\partial t} + \nabla \cdot (\rho \mathbf{u} k) = \nabla \cdot \left(\mu + \frac{\mu_t}{\sigma_k} \nabla k \right) + P_k - \rho \varepsilon + S_k \quad (3)$$

$$\frac{\partial(\rho \varepsilon)}{\partial t} + \nabla \cdot (\rho \mathbf{u} \varepsilon) = \nabla \cdot \left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \nabla \varepsilon \right) + \frac{\varepsilon}{k} (C_{1\varepsilon} P_\varepsilon - C_{2\varepsilon} \rho \varepsilon) + S_\varepsilon \quad (4)$$

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \quad (5)$$

where P_k and P_ε are the generating terms, S_k and S_ε are the source terms, and μ and μ_t are the dynamic viscosity and turbulent viscosity, $\sigma_k = 1.0$, $\sigma_\varepsilon = 1.2$, $C_{1\varepsilon} = 1.44$, $C_{2\varepsilon} = 1.9$, and $C_\mu = 1.44$. Based on this model selection, the standard wall functions are employed as the near-wall treatment to model the flow in the viscous sub-layer and buffer region, ensuring a balance between computational accuracy and efficiency (see Fig. 1).

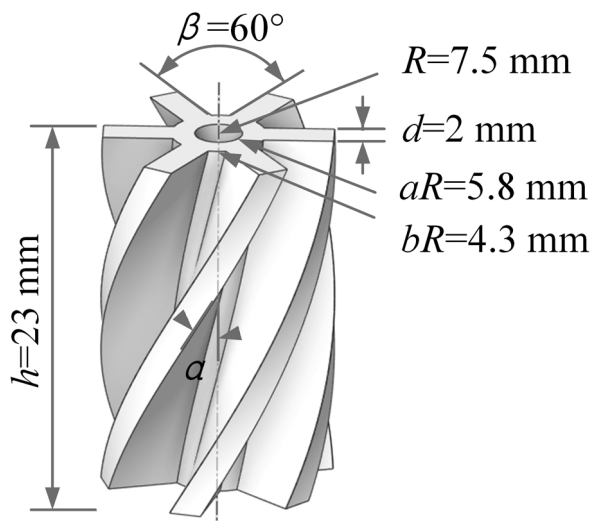


Fig. 3. Swirler structure and dimension diagram.

2.3. Mesh partition and simulation setting

The detailed structure size of the ejector and the relative position of the shaft center line are shown in Fig. 2. The swirler is placed inside the nozzle (Fig. 3). The poly-hexcore grid type is used to construct the unstructured grid system. In order to accurately capture the flow characteristics of the key area, the local mesh refinement processing was performed on the nozzle (including the built-in swirler) and the ejector contraction section. The final ejector mesh diagram is shown in Fig. S1 (in Supplementary Material).

In the solution setting, water is defined as the primary phase, and air is the secondary phase. The simulated ambient pressure is atmospheric, and the temperature is 20 °C. The physical parameters of water and air under this condition are shown in Table S1. The boundary conditions are set as follows: the water inlet adopts the velocity inlet, the air inlet adopts the pressure inlet, and the downstream outlet of the ejector adopts the pressure outlet. The internal flow is considered a steady and incompressible flow (it has been verified in the Supplementary Material). In order to solve the pressure–velocity coupling equation, the Phase-Coupled SIMPLE algorithm is selected. The detailed setting parameters of the solver are listed in Table S2. Residuals for the governing equations reach within 1.0×10^{-4} relative error as the convergence criterion. Q_G attains a stable level, and the mass flow rate balances between entering and leaving the model as an additional convergence condition.

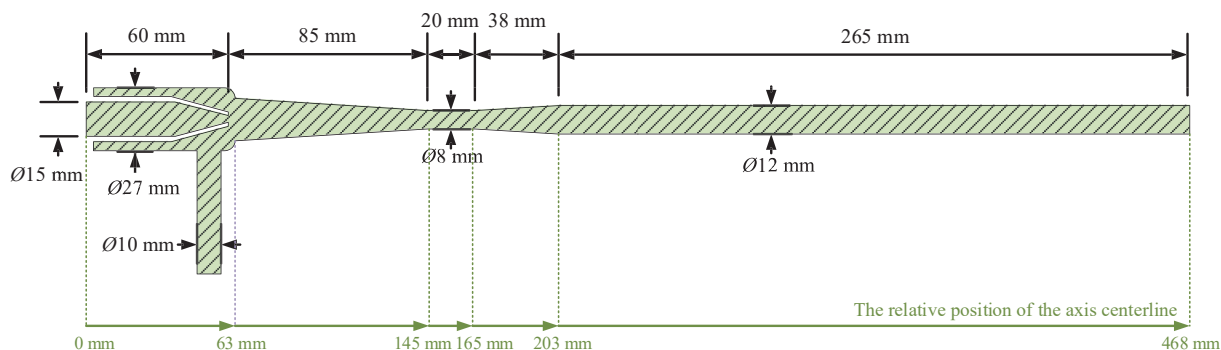


Fig. 2. Internal dimensions of the ejector.

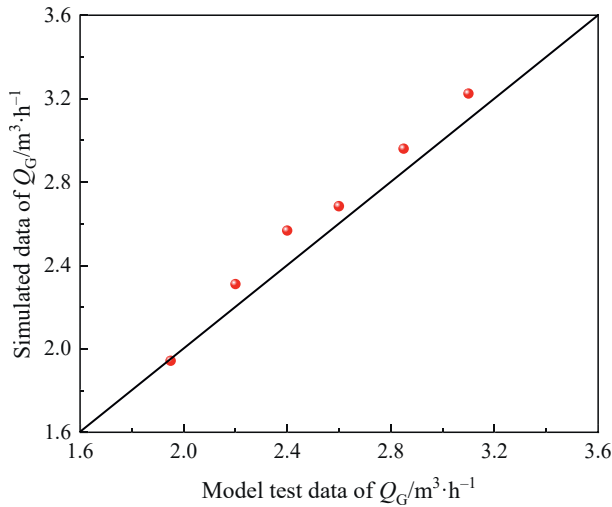


Fig. 4. Comparison of simulation and model test data.

2.4. Mesh independence study and experiment validation

In order to verify the influence of mesh generation on the numerical simulation results, for the ejector structure with a nozzle diameter of 3 mm and a swirl angle of $\alpha = 30^\circ$, the grid independence test was carried out under the condition of liquid

circulation rate (Q_L) of $0.55 \text{ m}^3 \cdot \text{h}^{-1}$ (the results are summarized in Table S3 and Fig. S2). When the total number of grids increases from 838670 to 1012616, the Q_G value and axial pressure distribution of the ejector change little and tend to be stable. Considering the calculation accuracy and efficiency, the number of grids, 838670, was finally determined for subsequent simulation calculations. In order to verify the accuracy of the numerical simulation, the numerical simulation of the ejector was carried out under the condition of Q_L in the range of $0.45\text{--}0.55 \text{ m}^3 \cdot \text{h}^{-1}$. Fig. 4 is the comparison error diagram between the simulated and experimental values of the ejector Q_G . It can be seen from Fig. 4 that the average relative deviation is only 4.2%. The results show that CFD simulation can effectively reflect the actual situation and verify the rationality and reliability of the numerical simulation method.

3. Results and Discussion

3.1. Influence of Q_L

The liquid circulation rate Q_L is a process parameter of the ejector, determining the supply rate of the working fluid to the ejector. In order to explore the influence of Q_L on the flow characteristics of the ejector, under the structural parameters of a 3 mm nozzle and a 30° swirler, three kinds of Q_L of 0.47, 0.51, and $0.55 \text{ m}^3 \cdot \text{h}^{-1}$ were selected to carry out numerical simulation research on the ejector. Fig. 5(a)–(d) show the Q_G of the ejector (a), the ejection coefficient (μ) and the interface area density (IAD) (b),

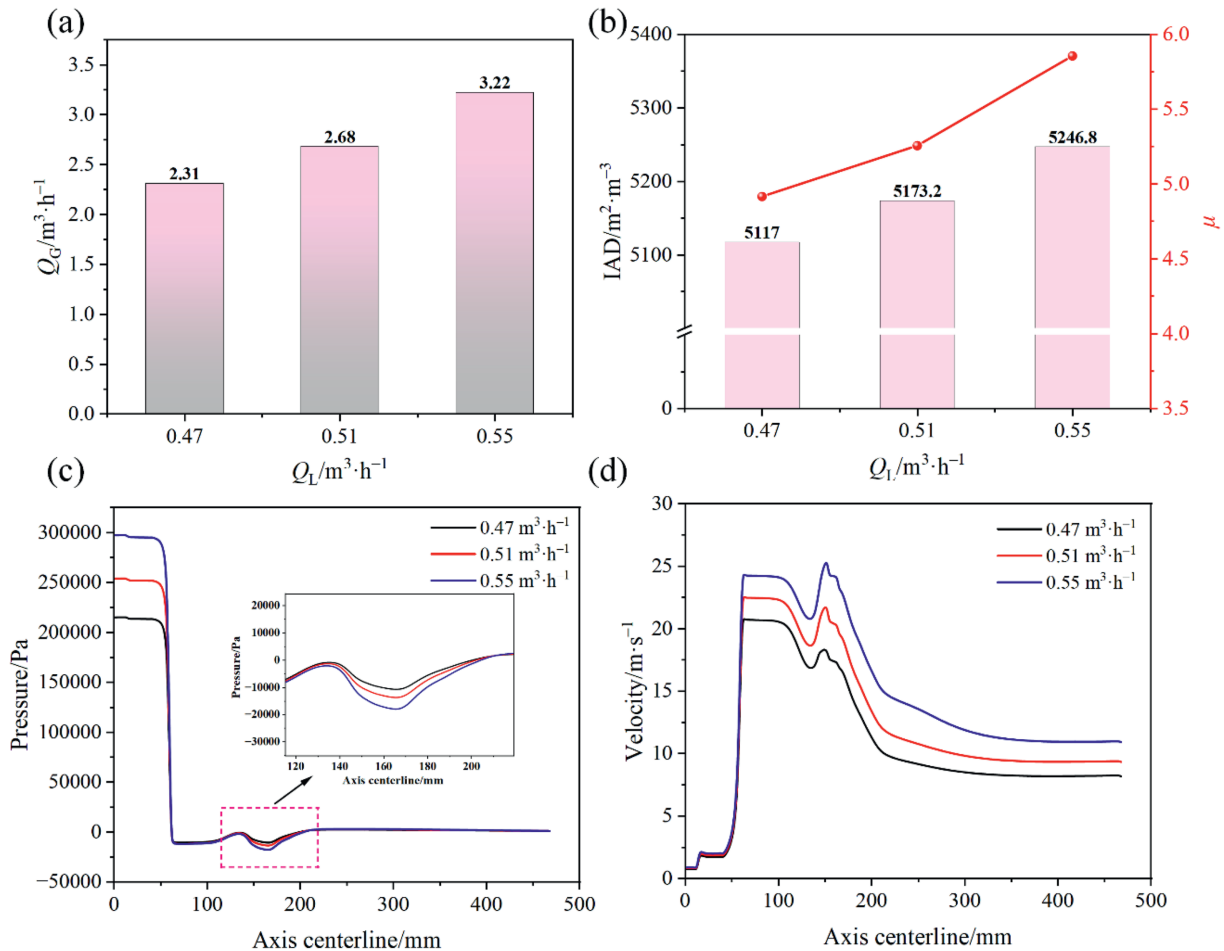


Fig. 5. Ejector performances under different Q_L : (a) Q_G , (b) IAD and μ , (c) axial pressure distribution, (d) axial velocity distribution.

from the nozzle outlet to the lower outlet of the ejector (b), the pressure distribution (c) and velocity distribution (d) of the central axis (see Fig. S3) under different Q_L . Here, μ is defined as the ratio of the volume flow rate of the entrained fluid to the working fluid, that is, $\mu = Q_G/Q_L$. IAD is the contact interface area between the two phases in the unit mixed volume, namely $IAD = A/V_{total}$. From Fig. 5(a), it can be seen that with the increase of Q_L , Q_G shows a gradually increasing trend. This change is mainly due to the increase of Q_L , which increases the input energy of the system, increases the kinetic energy carried by the working fluid, and then significantly enhances the turbulent kinetic energy in the nozzle outlet area (see Fig. S4). It can be seen from Fig. S4 that when Q_L increases from 0.47 to 0.55 $\text{m}^3 \cdot \text{h}^{-1}$, the turbulent kinetic energy at the nozzle outlet increases from 0.253 to 0.328 $\text{m}^2 \cdot \text{s}^{-2}$. The enhancement of turbulence promotes the gas phase medium to be entrained more effectively, thus strengthening the ejection effect and gradually increasing Q_G . The results in Fig. 5(b) show that μ and IAD also increase with the increase of Q_L , but the increasing trends of them are different: when Q_L increased from 0.51 to 0.55 $\text{m}^3 \cdot \text{h}^{-1}$, the increase of μ was 65% higher than that of Q_L from 0.47 to 0.51 $\text{m}^3 \cdot \text{h}^{-1}$, while the increase of IAD was only 29.5%. This indicates that although the increase of Q_L helps to improve the gas phase suction capacity of the ejector, the improvement effect on IAD is relatively limited, which may be due to the restriction of the internal flow characteristics of the ejector on the interface renewal and maintenance.

As shown in Fig. 5(c), the pressure of the fluid drops sharply after ejecting from the nozzle. After entering the contraction section, the pressure gradually rises and peaks at the end of the contraction section, that is, 10 mm before the inlet of the throat. The peak pressure increases with the decrease of Q_L . When Q_L is 0.55 $\text{m}^3 \cdot \text{h}^{-1}$, the minimum pressure is approximately -18 kPa. After that, the pressure begins to decrease and drops to the lowest value at the outlet of the throat, and the greater the Q_L , the smaller the minimum value. Then, in the diffusion and straight pipe sections, the pressure slowly rises, finally reaching the outlet pressure of 980 Pa. The characteristic difference of the velocity distribution of the central axis of the ejector mainly appears in the downstream area of the nozzle (Fig. 5(d)). After the fluid is ejected from the nozzle, the central axis velocity gradually decreases in the contraction section. It reaches the local minimum value in the region 10 mm before the throat inlet. After that, the velocity increases and the peak velocity appears in the throat. It is worth noting that the peak position gradually moves backward with the increase of Q_L . When Q_L is 0.47, 0.51 and 0.55 $\text{m}^3 \cdot \text{h}^{-1}$ respectively,

the peak positions are located at 3, 5 and 7 mm behind the throat entrance. At Q_L of 0.55 $\text{m}^3 \cdot \text{h}^{-1}$, the peak velocity reaches the maximum velocity of the central axis of the ejector, that is, 25.3 $\text{m} \cdot \text{s}^{-1}$. After the peak, the velocity continues to decline along the central axis until the ejector outlet.

3.2. Influence of nozzle size

The nozzle size belongs to the core structural parameters of the ejector, which determines the flow section and velocity of the working fluid. In order to explore the influence of nozzle size on the flow characteristics of the ejector, three nozzle sizes of 2, 3, and 4 mm were selected for numerical simulation using a 30° swirler at Q_L of 0.55 $\text{m}^3 \cdot \text{h}^{-1}$. The performance parameters of the ejector under different nozzle sizes are shown in Fig. 6(a). Under the same Q_L conditions, compared with the 3 mm and 4 mm nozzles, the Q_G generated by the ejector using the 2 mm nozzle is significantly higher, which is 2.8 times that of the 3 mm nozzle and 8.4 times that of the 4 mm nozzle, respectively. However, the ΔP required for the 2 mm nozzle also increases significantly, reaching 5.4 times that of the 3 mm nozzle and 17.5 times that of the 4 mm nozzle. This performance difference is due to the fundamental influence of the nozzle size on the flow field characteristics. For smaller nozzles, at a constant Q_L , the narrow flow area forces the working fluid to pass at a higher flow rate. According to the principle of fluid mechanics, this high-speed jet enhances its turbulence kinetic energy (see Fig. S5). The turbulent kinetic energy of the 4 mm nozzle is 0.104 $\text{m}^2 \cdot \text{s}^{-2}$, and when the nozzle diameter is reduced to 2 mm, the turbulent kinetic energy is greatly increased to 1.695 $\text{m}^2 \cdot \text{s}^{-2}$. This enhancement effect induces a more significant entrainment effect in the ejector, thereby significantly increasing the Q_G value. However, the high flow rate also brings significant fluid friction loss and turbulent dissipation. In order to overcome these resistances and maintain the same Q_L through the small aperture, a larger inlet driving pressure must be applied, resulting in a significant increase in the ejector ΔP .

Fig. 6(b) is the gas–liquid distribution cloud diagram on the front axial plane before the lower outlet of the ejector throat (see Fig. S6). The diagram shows that the 2 mm nozzle has a large outlet kinetic energy and a small outlet cross-sectional area. The width of the fluid diffusion after passing through the swirler is small, and the main position of gas–liquid mixing is concentrated in the throat area. With the increase of nozzle size, the position of gas–liquid two-phase mixing gradually moves forward to the throat. Especially in the case of a 4 mm nozzle, due to the large

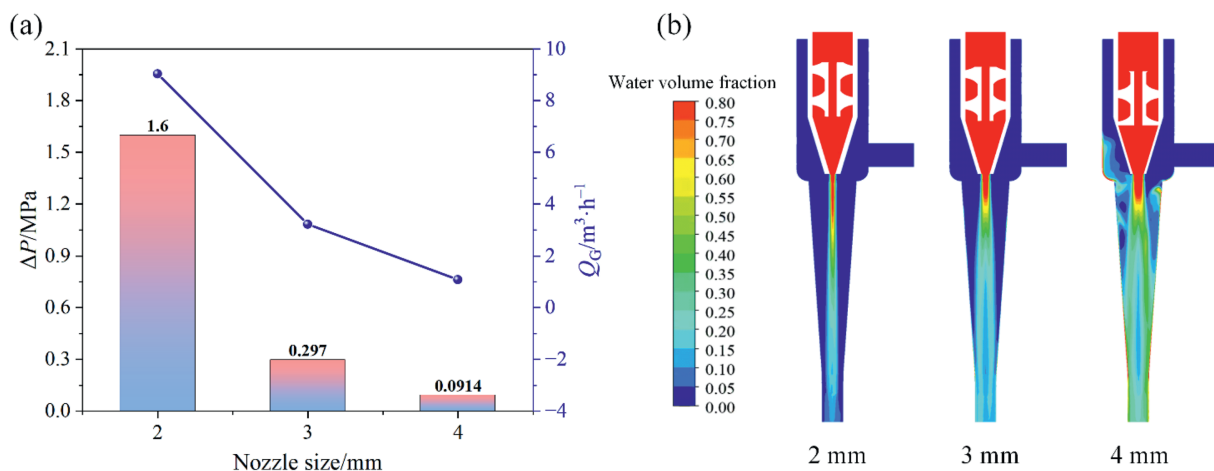


Fig. 6. (a) ΔP and Q_G of the ejector under different nozzle sizes. (b) The gas–liquid distribution contour map before the ejector throat exit.

flow area and the tangential velocity of the fluid after passing through the swirler, the diffusion angle of the fluid increases, resulting in the fluid hitting the wall in the contraction section, so that the gas–liquid mixing is initially completed in the contraction section. In addition, part of the fluid will be mixed upward, increasing kinetic energy dissipation. Larger kinetic energy loss will lead to a decrease in the Q_G of the ejector under this condition, which is also one of the reasons for the lower Q_G of the 4 mm nozzle after the ejector is installed with the swirler. For the 3 mm nozzle, the fluid began to hit the wall at 10 mm before the throat inlet, and the fluid squeezed and mixed, which was consistent with the phenomenon that the pressure reached the peak at this position in Fig. 5(c).

The IAD and μ of the ejector under different nozzle sizes are shown in Fig. 7(a). Based on the 3 mm nozzle, when the nozzle size is reduced from 3 to 2 mm, μ increases by 2.8 times, but IAD only increases by 9.6%. When the 3 mm nozzle is expanded to a 4 mm nozzle, μ is reduced by 3.0 times, and IAD is reduced by 21.8%. It can be seen from these results that this nonlinear change reveals the differential regulation mechanism of nozzle size on performance. The smaller nozzle greatly improves the ejection efficiency by enhancing the kinetic energy of the working fluid. However, the concentration tendency of the high-speed jet limits the interphase

dispersion effect, resulting in a weak IAD increase. The larger nozzle weakens the intensity of turbulent kinetic energy due to the decrease in flow velocity, and the ejection capacity and mixing efficiency are significantly attenuated simultaneously. In contrast, the IAD of the 3 mm nozzle is relatively high, which means that this size can provide a more uniform gas–liquid distribution in the ejector and maintain a better mixing effect. This conclusion is further supported by the gas–liquid distribution cloud images at different heights after the nozzle exit (Fig. 7(b)). The 2 mm nozzle has evident stratification due to the high kinetic energy of the working fluid driven by the high pressure drop and the insufficient residence time of the gas–liquid two-phase. The liquid phase is concentrated in the central area of the flow channel. Due to the large flow cross-sectional area of the 4 mm nozzle, the swirl intensity of the fluid after passing through the swirler is excessively developed, enriching the liquid phase in the throat section to the wall area. The turbulent vortex generated by the 3 mm nozzle with the swirler is more balanced in spatial distribution and intensity, which promotes the gas–liquid two-phase to be thoroughly mixed near the throat without an obvious phase separation interface. This uniform dispersion state eventually shows a higher IAD.

Fig. 7(c) shows the velocity distribution of the central axis of the ejector under different nozzle diameters. The overall flow

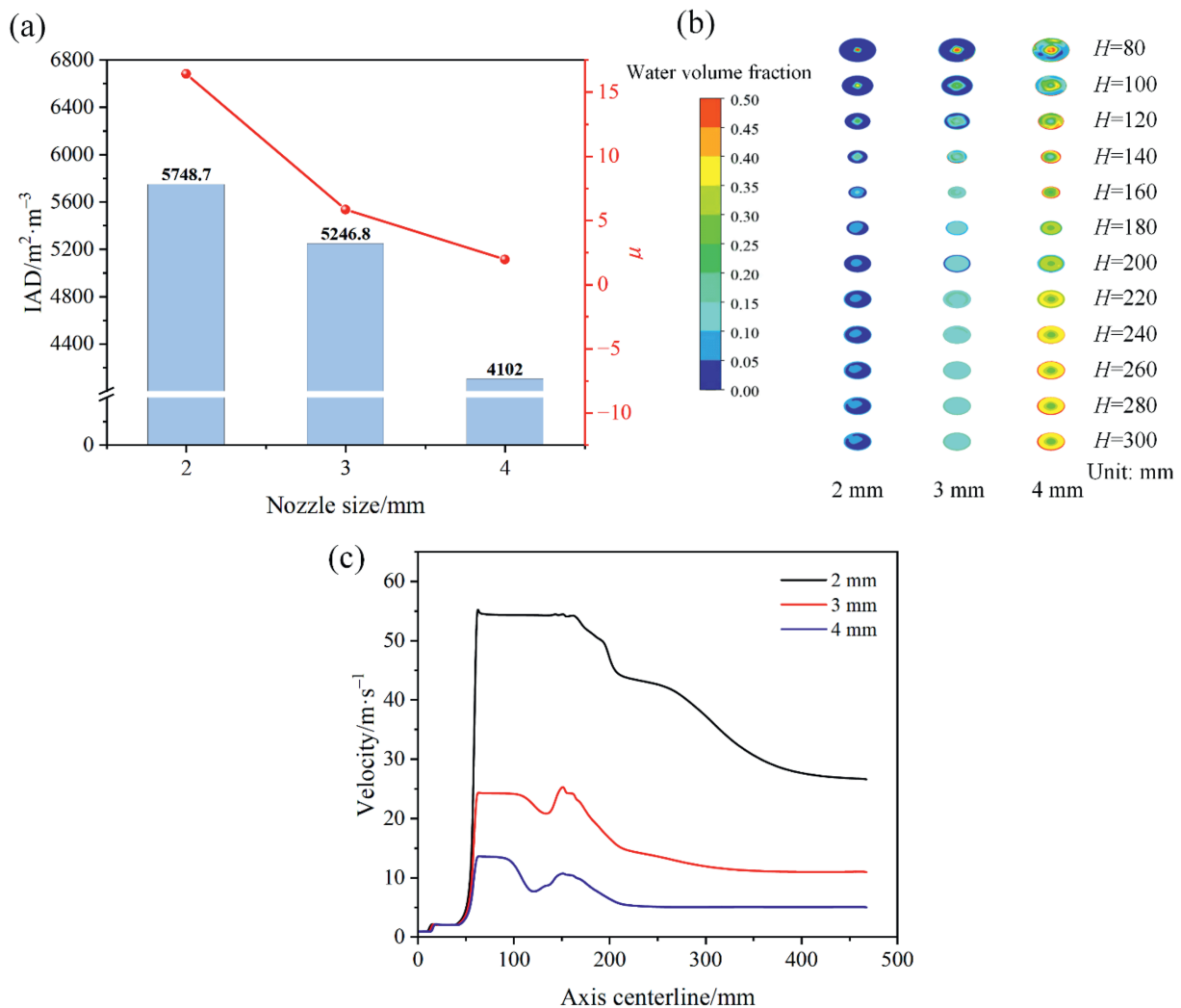


Fig. 7. (a) IAD and μ of the ejector under different nozzle sizes. (b) Gas–liquid distribution cloud map at different heights after the nozzle outlet. (c) Axial velocity distribution of the ejector under different nozzle sizes.

velocity of the 2 mm nozzle is significantly higher than that of the 3 mm and 4 mm nozzles, and the maximum flow velocity is located at the nozzle outlet, which is about $55 \text{ m} \cdot \text{s}^{-1}$. The velocity begins to decay gradually behind the downstream outlet of the throat. In comparison, the starting position of the velocity attenuation of the 4 mm nozzle moves forward to the middle of the contraction section. The attenuation position of the 3 mm nozzle is between the two, which moves backward relative to the 4 mm nozzle. In addition, unlike the 2 mm nozzle, the 3 and 4 mm nozzles have a speedy recovery phenomenon in the throat section. The higher flow rate generated by the 2 mm nozzle leads to a significant shortening of the residence time of the gas–liquid two-phase, which also helps to explain why, although the μ of the 2 mm nozzle is higher, its IAD does not increase significantly compared to the 3 mm nozzle.

3.3. Influence of swirler angle

The swirler is installed inside the nozzle, and its core function is to change the flow state of the working fluid and transform it from axial flow to rotational flow. As a key design variable, the angle of the swirler directly determines the magnitude of the rotational strength and tangential velocity components obtained by the working fluid. In order to explore the influence of the angle of the swirler on the flow characteristics of the ejector, under the condition of Q_L of $0.55 \text{ m}^3 \cdot \text{h}^{-1}$ and nozzle diameter of 3 mm, the numerical simulation of the ejector without swirler and the ejector with different angle swirlers (0° , 15° , 30° , 45° , 60°) is carried out respectively. The swirler's structure with different angles is shown in Fig. S7. Fig. 8(a) shows the change of Q_G when the ejector is matched with different angle swirlers. Compared with the case without a swirler, except for the 0° swirler, the addition of swirlers at all angles significantly improves the Q_G of the ejector, and with the increase of the swirler angle, the Q_G of the ejector increases first and then decreases. Among them, the 30° swirler has the most significant effect on the Q_G of the ejector, and its corresponding Q_G value is about 2.75 times higher than that without a swirler. Fig. 8(b) shows the λ_2 criterion for the water flow field in the converging section and throat of the ejector under different swirler angles. λ_2 criterion is defined as the second eigenvalue of the strain rate tensor within the velocity gradient tensor. Its physical significance lies in the fact that when $\lambda_2 < 0$, it

indicates that rotational motion dominates the fluid flow at that point. The larger the value of $|\lambda_2|$, the higher the vortex intensity at the corresponding location. As shown in the figure, the trend of $|\lambda_2|$ with respect to the swirler angle aligns with the trend of Q_G , with the maximum $|\lambda_2|$ value corresponding to the 30° swirler. This indicates that under this configuration, the vortex generated inside the ejector exhibits the highest intensity and most vigorous rotation. High-intensity vortices more effectively promote momentum exchange between the working fluid and the entrained fluid, driving greater gas participation in the flow. Simultaneously, they induce a stronger, broader low-pressure core zone within the ejector, thereby providing a larger suction-driven pressure differential. Furthermore, the enhanced vortex shear and mixing action also helps promote bubble breakup and dispersion. Consequently, the 30° swirler achieves optimal ejector performance by forming the strongest vortex structure.

Fig. 9 show the gas–liquid and pressure distribution cloud maps before the outlet of the lower throat of the ejector at different angles. It can be seen from Fig. 9(a) that the flow pattern inside the ejector is similar under the no swirler and 0° swirler conditions. The liquid is sprayed vertically downward along the nozzle, and the gas–liquid two-phase mixture is mainly mixed in the throat area. The gas–liquid mixing position under a swirler of 0° is lower than that without a swirler. In contrast, the other four angles of the swirler significantly changed the internal flow state of the ejector. As the swirl angle increases, the width of the liquid jet gradually expands. At a slight swirl angle (such as 15°), the jet width has not exceeded the throat diameter, and the gas–liquid mixing is still mainly concentrated in the throat area. Although the liquid jet has begun to have a specific diffusion effect, the conversion degree of axial momentum to angular momentum is limited, and the more substantial turbulence effect has not been thoroughly excited. There is still room for improvement in turbulence intensity and mixing efficiency. Therefore, in this case, the Q_G of the ejector is lower than that of the 30° swirler. In the case of a larger swirl angle, the width of the liquid jet increases significantly, which causes the jet to collide with the wall in the contraction section and mix in advance. The mixing position gradually moves up with the increase of the swirl angle. For example, under a 60° swirler condition, the mixing area has been extended to the middle of the contraction section. In addition, as the swirler angle increases, the jet rotation intensity increases.

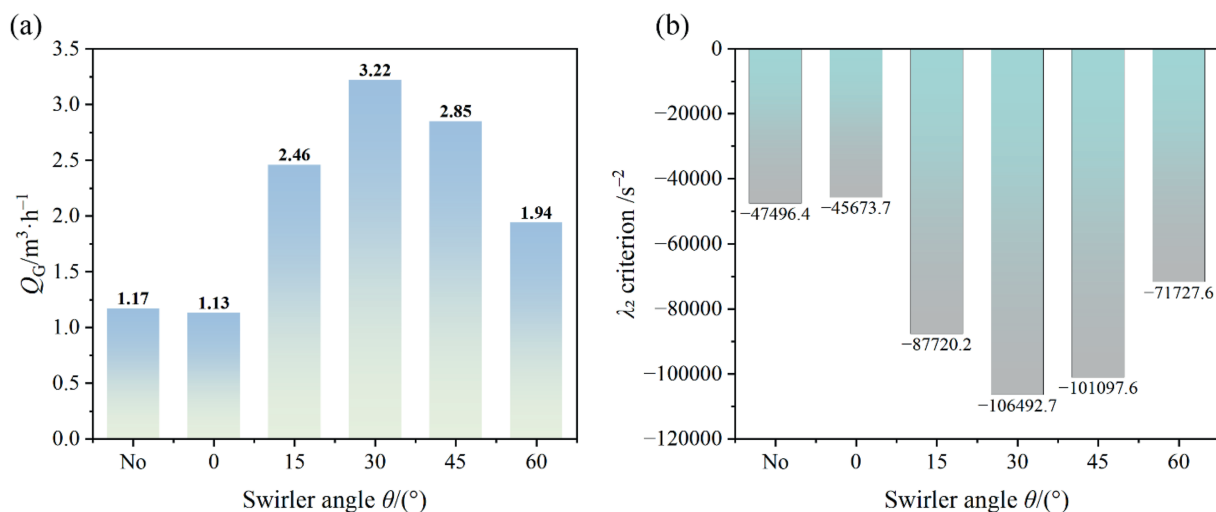


Fig. 8. (a) Q_G of the ejector under different swirler angles. (b) The λ_2 criterion for the water flow field in the converging section and throat of the ejector under different swirler angles.

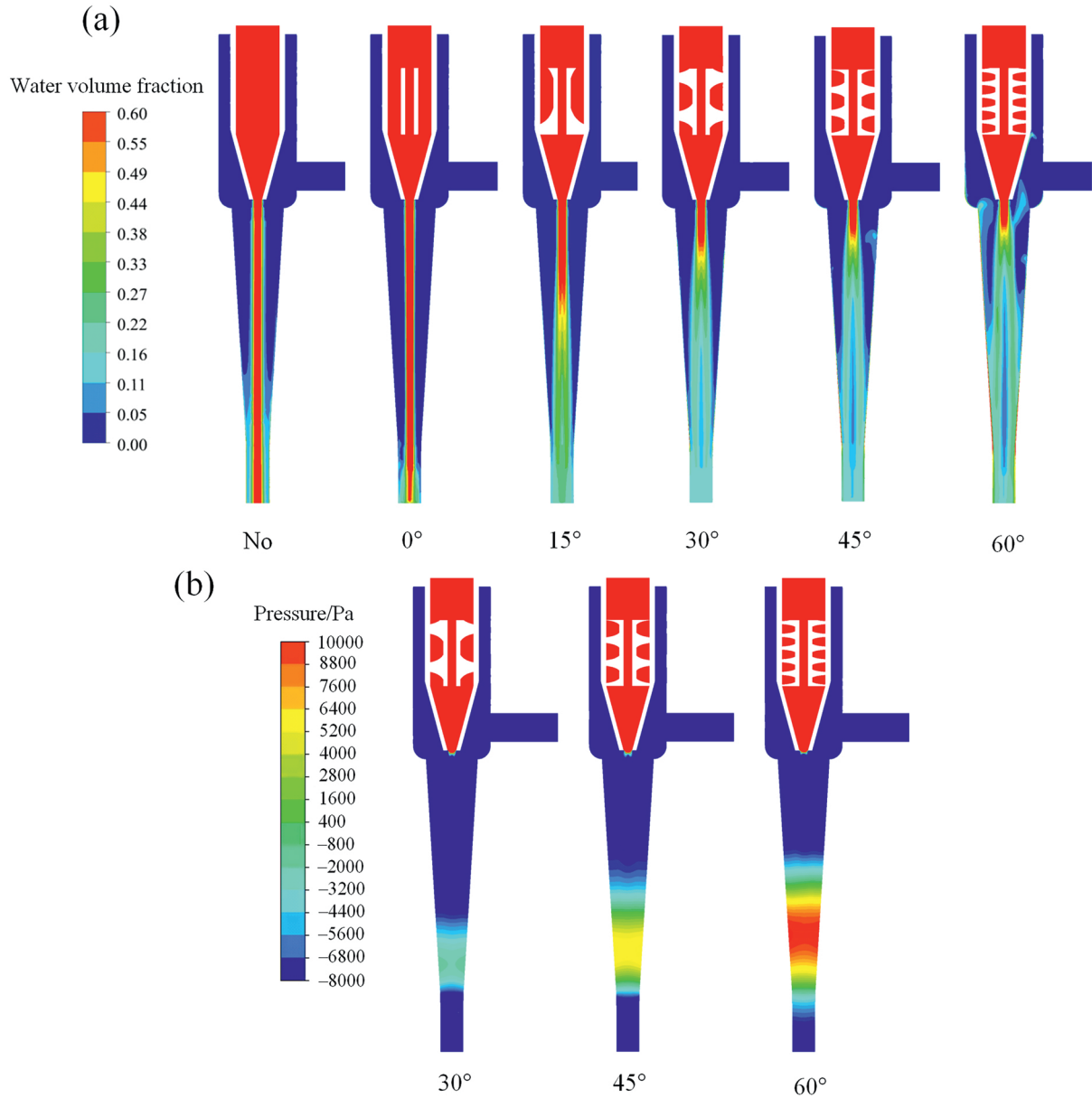


Fig. 9. (a) Gas–liquid distribution contour map upstream of the ejector throat exit under different swirler angles. (b) Pressure in the ejector contraction section under different swirler angles.

Combined with the expansion of the jet width, the liquid collides with the wall and forms a rotating flow along the wall. The flow velocity in the central area decreases and the vacancy phenomenon gradually appears, which is more significant with the increase of the swirl angle. The pressure distribution in Fig. 9(b) further shows that, as the angle of the swirler increases, the impact of the fluid on the wall is strengthened, and the pressure in the impact zone also increases. Under the condition of a 30° swirler, the impact position is still negative pressure. In contrast, under 45° and 60° swirlers, the impact position has turned into positive pressure, and the impact point obviously moves upstream to the contraction section. The above results show that when the swirler with a larger angle converts the axial momentum into angular momentum, the conversion rate is higher, resulting in increased energy dissipation and weakened turbulence effect. Therefore, the

Q_G corresponding to 45° and 60° swirlers is lower than that under the condition of a 30° swirler, and as the swirler angle continues to increase, Q_G further decreases.

Fig. 10(a) and (b) show the axial pressure and velocity of the ejector with different angle swirlers. It can be seen from Fig. 10(a) that the axial pressure increases with the increase of the swirler angle before the nozzle outlet. Under the condition of no swirler, the axial pressure is close to that of 0° swirler, but the value is slightly higher. After the liquid is ejected, the axial pressure in the contraction section and the throat area drops sharply, from positive pressure to negative pressure. Under no swirl or slight swirl angle (such as 15°), the axial pressure increases slowly from the contraction to diffusion sections. The pressure corresponding to the 0° swirler is the lowest, followed by the no swirler, and the 15° swirler is relatively high. When the angle of the swirler further

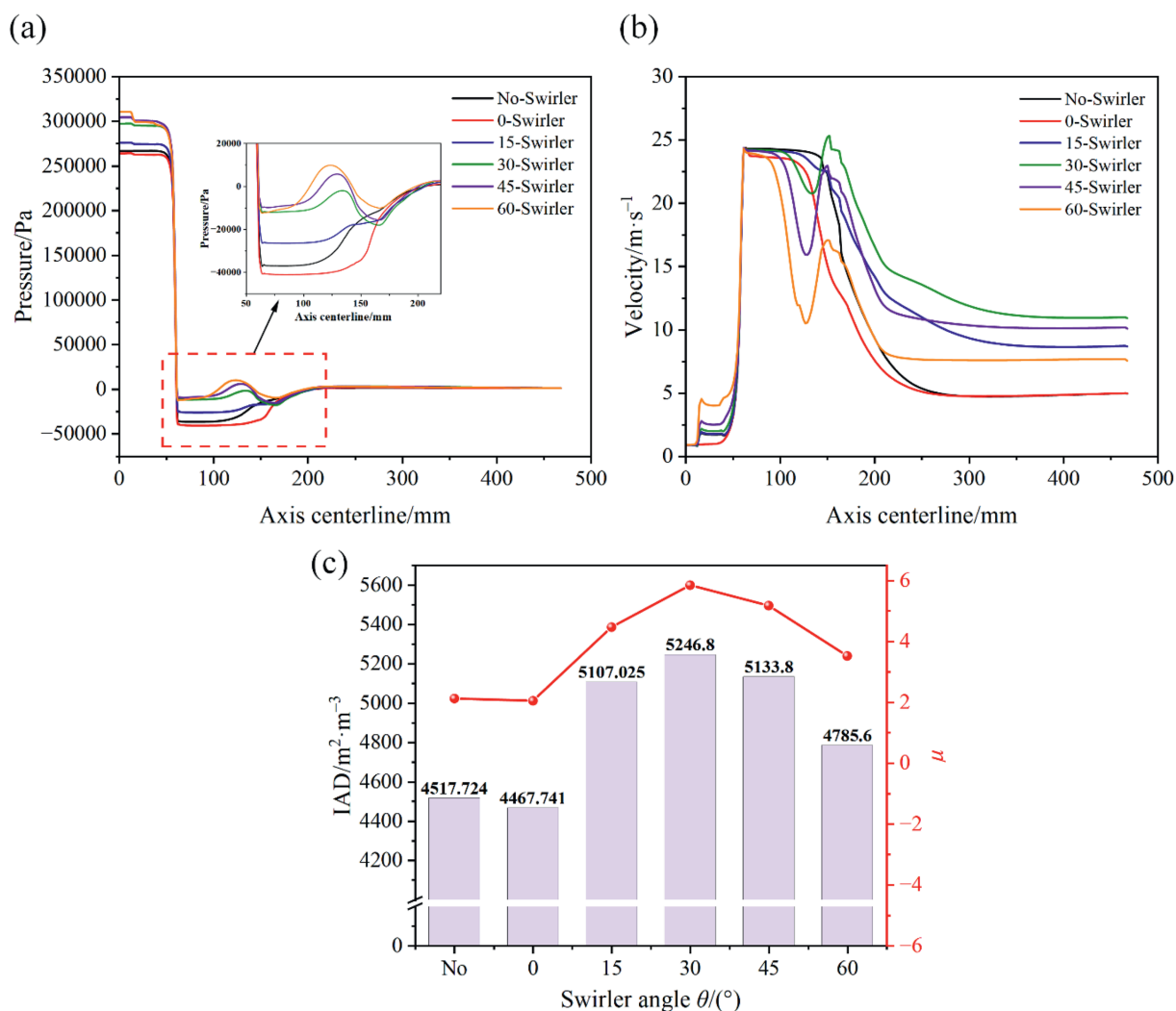


Fig. 10. Ejector performances under different swirler angles: (a) axial pressure distribution, (b) axial velocity distribution, (c) IAD and μ .

increases ($\geq 30^\circ$), the axial pressure in this section increases with the increase of the swirl angle. In addition, a prominent pressure peak can be observed in the contraction section, and the peak increases with the increase of the swirl angle. The position of its appearance is also gradually moved forward. The pressure peaks of 30° , 45° , 60° swirlers are located at 11.5, 15.5 and 22 mm in front of the throat inlet, respectively, and the corresponding pressure values are -2045 , 5657 and 9713 Pa. The pressure distribution trend is consistent with the results shown in Fig. 9(b). It can be seen from Fig. 10(b) that the axial velocity in the nozzle can be increased by installing a swirler in the ejector before the nozzle exit, and the velocity increases further with the increase of the swirl angle. After the fluid enters the ejector, the axial velocity decreases gradually under the condition of no swirler, 0° swirler and 15° swirler. When the swirler angle further increases, a local minimum value of the axial velocity can be observed in the contraction section. The minimum value is more significant with the increase of the swirl angle, and the position gradually moves forward. For example, the velocity of the 60° swirler at this position is reduced from $25 \text{ m}\cdot\text{s}^{-1}$ at the nozzle outlet to $10.5 \text{ m}\cdot\text{s}^{-1}$. This phenomenon is consistent with the flow behavior shown in Fig. 9(a). The liquid rotates along the wall at the corresponding position, resulting in a local vacancy in the central area, and the flow rate decreases accordingly. The larger the angle of the swirler,

the stronger the rotation effect of the fluid, which in turn enhances this vacancy phenomenon, resulting in a further decrease in the axial velocity, and the location of this phenomenon also advances to the nozzle end. After entering the middle section of the throat, due to the decrease in the throat's flow area, the liquid's flow is compressed, the original vacancy phenomenon disappears, and the flow rate of the fluid increases, resulting in a peak value of the axial velocity. Finally, in the straight pipe section of the ejector, the axial velocity gradually tends to be stable, and the greater the Q_C value of the ejector, the greater the axial velocity. This is due to the same liquid Q_L and the same flow area. The increase in Q_C means an increase in flow rate, which increases the axial velocity.

Fig. 10(c) shows the effects of different angle swirlers on the IAD and μ of the ejector. It can be seen from the figure that, except for the 0° swirler, the other four angle swirlers significantly increased the μ and IAD of the ejector, and as a whole, they increased first and then decreased with the increase of the swirler angle. Among them, the 30° swirler has the most significant lifting effect. Compared with the non-swirler, the μ value is increased by 2.75 times, and the IAD is increased by 16.1%. It is worth noting that the performance of 15° and 45° swirlers differs. Although the μ value of the 15° swirler is 13.7% lower than that of the 45° swirler, its IAD is only 0.5% lower than that of the latter. The internal flow mechanism can explain this difference. Compared with

the 15° swirler, the 45° swirler has a larger angle, which makes the axial momentum of the fluid more converted into angular momentum, the swirl intensity is enhanced, the turbulence is higher, and the μ value is increased. However, the fluid tends to rotate downward near the wall due to the strong swirling flow. The flow velocity and turbulence are mainly concentrated in the near-wall region, and the bubble breaking capacity in the central region is insufficient. Therefore, although the μ value is high, the IAD is not significantly improved. In summary, the optimal swirler angle needs to strike a balance between enhanced mixing and enhanced bubble breaking. In the study of different angle swirlers, the 30° swirler shows the best comprehensive performance, which can not only significantly improve the μ value but also achieve a significant improvement in IAD, reflecting its advantages in improving the ejector's mixing and bubble breaking ability.

4. Conclusions

Based on the self-designed down-jet venturi ejector, this work introduces the swirler structure into the nozzle. Combined with the CFD method, the change of the internal flow field of the ejector in the presence of the swirler is analyzed in depth. The effects of circulation, nozzle size, and swirler angle on the performance of the ejector are studied. In the Q_L range of the study, with the increase of Q_L , Q_G and IAD showed a gradual increase trend, among which Q_G increased significantly, while IAD increased relatively smoothly. Reducing the nozzle diameter can effectively improve Q_G and IAD. However, a nozzle diameter that is too small will lead to increased concentration of high-speed jets and a fluid residence time that is too short, limiting the dispersion effect between gas and liquid, resulting in a limited increase of IAD. At the same time, its ΔP also increased significantly. Under the same conditions, the ΔP of the 2 mm nozzle was 5.4 times that of the 3 mm nozzle, while the IAD increased by only 9.6%. Compared with the case without a swirler, except for the 0° swirler, the introduction of 15°, 30°, 45°, and 60° swirlers significantly improved the Q_G and IAD of the ejector, and both of them increased first and then decreased with the increase of the angle of the swirler. Among them, the 30° swirler significantly improves the ejection effect and mixing performance. When the Q_L is $0.55 \text{ m}^3 \cdot \text{h}^{-1}$ and the nozzle diameter is 3 mm, the Q_G value is 2.75 times higher than that without a swirler, and the IAD is increased by 16.1%, showing excellent comprehensive performance.

This study provides an important quantitative basis and theoretical support for the structural design and operation optimization of high-performance swirl ejectors. The structure of the swirler has great potential in enhancing the gas–liquid mixing and reaction process, and has good engineering application prospects. However, this study is based on the isothermal simulation of water–air system, without considering the influence of fluid physical properties, non-isothermal effect and chemical reaction on the flow-mass transfer coupling behavior in the actual chemical process. Future research can be extended to the performance analysis of high viscosity and multi-component systems, and the flow-mass transfer-reaction synergistic mechanism can be further revealed in combination with reaction kinetics. Thereby promoting the efficient application of this technology in industrial reactors.

CRedit Authorship Contribution Statement

Yunlong Wang: Writing – original draft, Software, Methodology, Investigation, Data curation, Conceptualization. Jijun Ge: Validation, Software, Investigation, Data curation. Zheng Kuang: Validation, Formal analysis, Data curation, Conceptualization.

Guoxuan Li: Writing – review & editing, Software, Methodology, Funding acquisition, Data curation. Pan Xu: Writing – review & editing, Investigation, Funding acquisition, Formal analysis, Conceptualization. Rongshan Bi: Writing – review & editing, Supervision, Methodology, Conceptualization.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Supplementary Material

Supplementary data to this article can be found online at <https://doi.org/10.1016/j.cjche.2026.01.024>.

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