Effects of Nozzle Geometry on Cavitation in Nozzles of Pressure Atomizers*

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Abstract

The effects of nozzle geometry on cavitation in the nozzle of pressure atomizers and the liquid jet are examined using various two-dimensional (2D) nozzles with different geometries. Then, whether or not the conventional cavitation numbers can be used to predict the formation of supercavitation, in which liquid jet atomization is enhanced, is examined. As a result, we confirm that (1) the thickness of the cavitation zone increases with the ratio C_u of the cross-sectional area upstream of the nozzle to that of the nozzle, (2) the spray angle increases with C_u , (3) the formation of supercavitation can be predicted using the cavitation number σ_c' in which the effects of the flow contraction and the frictional pressure drop are taken into account, and (4) the conventional cavitation numbers σ , σ_2 and σ_3 cannot predict the formation of supercavitation in nozzles with different geometries.

Key words: Cavitation, Nozzle, Cavitation Number, Injector, Atomization, Flow Contraction

1. Introduction

It has been pointed out that cavitation occurs in a nozzle of pressure atomizers, and influences atomization of a liquid jet discharged from the nozzle⁽¹⁾. Hence, efforts have been made to visualize cavitation in nozzles⁽²⁾⁻⁽¹¹⁾. The observations confirmed that liquid jet atomization is enhanced when cavitation is developed in a nozzle⁽²⁾⁻⁽⁶⁾, i.e., in the supercavitation regime^{(4),(6)}. An indicator which can be utilized to predict the formation of supercavitation is, therefore, of great use in designing pressure atomizers.

In the previous study⁽¹²⁾ we confirmed that cavitation in a nozzle is strongly affected by the flow contraction in the nozzle. Since the contraction coefficient C_c of a non-cavitating flow in a nozzle is affected by the ratio C_u between the cross-sectional area upstream of the nozzle A_u and that of the nozzle A_N ⁽¹³⁾, C_u may affect cavitation in a nozzle. The effects of the difference in C_u on cavitation, however, have not been clarified yet.

Pressure atomizers with various nozzle geometries, e.g., hole nozzles and slit nozzles, have been developed. Efforts have been devoted to optimize nozzle geometries, since it is difficult to predict cavitation in the nozzles with different geometries. Various cavitation numbers have been proposed and used as an indicator of cavitation in nozzles^{(1)-(3),(6)-(11)}. Although we have confirmed that one of the conventional cavitation numbers σ ⁽¹⁴⁾ can be used to predict development of cavitation in nozzles of fixed configurations⁽⁶⁾, it has been suggested that most of the cavitation numbers do not quantitatively predict development of cavitation in nozzles with different configurations. This can be because most of the

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cavitation numbers do not take into account the effects of flow contraction. Hiroyasu et al.⁽²⁾ have proposed a cavitation number σ_c which takes into account the pressure P_c and the velocity V_c in the vena contracta. Whether or not the cavitation number σ_c can quantitatively predict the formation of supercavitation, however, has not been verified yet.

In this study cavitation in various two-dimensional (2D) nozzles with different upstream areas A_u is visualized to investigate the effects of the difference in C_u on cavitation and the liquid jet. Then, the experimental data on development of cavitation in the 2D nozzles and various cylindrical nozzles⁽⁵⁾ are used to examine whether or not the cavitation numbers can be utilized to predict the formation of supercavitation in nozzles with various geometries.

2. Experimental setup and conditions

Schematic of the experimental setup is shown in Fig. 1. Filtered tap water at room temperature was supplied from the plunger pump and discharged from the 2D nozzle into ambient air at atmospheric pressure (back pressure $P_b = 0.101$ MPa). The liquid flow rate was measured using flowmeters. The uncertainty in measured flow rate was less than 3.7 %. The injection pressure P_u was measured at 100 mm upstream the nozzle using Bourdon pressure gauges. Images of cavitation and liquid jet were taken using a digital camera and a flash lamp (Nissin Electronic, MS-1000 & LH-15M, duration = 4 μ s). The concentration of oxygen dissolved in the water was measured using a dissolved oxygen probe (Hach company, HQ30d) and was about 9 mg/L.

Fig. 1 Experimental setup

Fig. 2 Schematics of 2D and cylindrical nozzles

Schematics of the 2D and cylindrical nozzles are shown in Figs. 2 (a) and (b), respectively. The dimensions of the 2D nozzles and those of the cylindrical nozzles⁽⁵⁾ are summarized in Tables 1 and 2, respectively. The 2D nozzles with various widths W_N , lengths L , thicknesses t and upstream widths W_u were used. Photos of cavitation in the cylindrical nozzles with various inner diameters D_N , lengths L and upstream diameters D_u were also used to examine the applicability of cavitation numbers to various nozzle geometries. The C_u and C_c in the tables are defined by

$$C_u = \frac{A_u}{A_N} = \frac{V_N}{V_u} \quad (1)$$

$$C_c = \frac{A_c}{A_N} = \frac{V_N}{V_c} \quad (2)$$

where A is the cross-sectional area, and V the mean velocity in the cross section. The subscripts u , N and c denote the upstream region, the nozzle, and the vena contracta, respectively. How to evaluate C_c will be described later.

Table 1 Dimensions of the 2D nozzles

$W_N \times t$ [mm]	$C_u = A_u/A_N = W_u/W_N$	L / W_N	$C_c = A_c / A_N$
4.21 x 0.98	7.60	3.8	0.68
2.23 x 0.47	7.17	3.6	0.68
3.81 x 0.95	2.91	4.2	0.69
4.06 x 0.95	1.45	3.9	0.77

Table 2 Dimensions of the cylindrical nozzles⁽⁵⁾

D_N [mm]	$C_u = A_u/A_N = (D_u/D_N)^2$	L / D_N	$C_c = A_c / A_N$
2.0	64	4	0.61
1.0	100	4	0.61
0.5	100	4	0.61
1.0	4	4	0.625
2.0	64	20	0.61
0.5	100	20	0.61

3. Effects of nozzle geometry on cavitation and liquid jet

Figures 3 (a) and (b) show cavitation and a liquid jet in the 2D nozzle of 4.21 mm in width W_N and those in the half-scale nozzle, respectively. Figures 3 (c) and (d) are those in the 2D nozzles with smaller C_u . At $C_u = 1.5$, hydraulic flip is not formed. Except for this case, an increase in V_N (from left to right) results in the following transition⁽⁶⁾: from no cavitation (dimensionless cavitation length $L^* = 0$, where L^* is the ratio of the streamwise length L_{cav} of the cavitation zone to the nozzle length L , i.e., $L^* = L_{cav} / L$), developing cavitation (cavitation bubbles appear in the upstream half of the nozzle, $0 < L^* < 0.7$), supercavitation (cavitation films are formed in the upstream half and cavitation clouds are shed and collapse near the exit, $0.7 < L^* < 1$), to hydraulic flip (the reattachment of a separated boundary layer does not occur in the nozzle, $L^* = 1$). Liquid jet atomization is enhanced in the supercavitation regime for all the nozzles. Images of cavitation in a cylindrical nozzle of 4 mm in diameter and a liquid jet are shown in Fig. 4 for comparison. Cavitation and liquid jet regimes show the same trends for 2D and cylindrical nozzles. Hence, once we can find an indicator for the formation of supercavitation, it is useful for a

(a) $W_N = 4.21$ mm, $C_u \approx 7.6$, $L/W_N \approx 4$, $W_N/t \approx 4$

(b) $W_N = 2.23$ mm, $C_u \approx 7.2$, $L/W_N \approx 4$, $W_N/t \approx 4$

(c) $W_N = 3.81$ mm, $C_u \approx 2.9$, $L/W_N \approx 4$, $W_N/t \approx 4$

(d) $W_N = 4.06$ mm, $C_u \approx 1.5$, $L/W_N \approx 4$, $W_N/t \approx 4$

Fig. 3 Effects of nozzle size and shape on cavitation (2D nozzle)

Fig. 4 Cavitation and liquid jet in a cylindrical nozzle ($D_N = 4$ mm, $C_u = 64$, $L/D_N = 4$)

rational design of pressure atomizers. It should be noted that the spray angle $\theta^{(6)}$ in the supercavitation regime is smaller in the case of smaller C_u , i.e., the spray angle θ is 16° for $C_u = 7.6$ and 2.9 , while $\theta = 9^\circ$ for $C_u = 1.5$.

The values of the conventional cavitation number σ defined by Eq. (3) are shown in Figs. 3 and 4^{(6),(14)}.

$$\sigma = \frac{P_b - P_v}{\frac{1}{2} \rho V_N^2} \quad (3)$$

where P_v is the vapor saturation pressure and ρ the liquid density. Cavitation is developed as σ decreases for all the nozzles. As shown in Figs. 3 (a) and (b), supercavitation is formed at $\sigma \simeq 0.7$ in the nozzles of similar configurations. The agreement confirms that the formation of supercavitation in nozzles of similar configurations can be predicted using σ . On the other hand, as shown in Figs. 3 and 4, supercavitation is formed for smaller σ when C_u is smaller, e.g., supercavitation is formed at $\sigma = 1.15$ for $C_u = 64$, at $\sigma = 0.69$ for $C_u \simeq 7.6$, and at $\sigma = 0.26$ for $C_u \simeq 1.5$. The differences in σ clearly demonstrate that we cannot use σ to predict the formation of supercavitation in nozzles with different configurations.

Images of cavitation near the inlet of the 2D nozzles in supercavitation are shown in Fig. 5. The thickness W_{cav} of the cavitation zone increases with C_u . When W_{cav} is large, the velocity V_c at vena contracta is large and the pressure P_c at vena contracta is low. This is the reason why supercavitation is formed at smaller σ in the case of smaller C_u . Hence, the flow contraction plays an important role in development of cavitation.

Fig. 5 Effect of C_u on cavitation thickness ($W_N \simeq 4$ mm, $L = 16$ mm)

4. Various cavitation numbers

In addition to the cavitation number σ defined by Eq. (3), various cavitation numbers have been proposed and used as an indicator for development of cavitation in nozzles. Bergwerk ⁽¹⁾, Soteriou et al. ⁽³⁾ and many other researchers ⁽⁷⁾⁻⁽⁹⁾ have used the cavitation number σ_2 :

$$\sigma_2 = \frac{P_u - P_b}{P_b - P_v} \quad (4)$$

Nurick ⁽¹⁰⁾ and Payri et al. ⁽¹¹⁾ used the cavitation number σ_3 defined by

$$\sigma_3 = \frac{P_u - P_v}{P_u - P_b} \quad (5)$$

Note that the cavitation numbers σ , σ_2 and σ_3 do not take into account the effects of flow contraction, and they have been used not only for nozzles but for long orifices ^{(15),(16)}.

Figure 6 shows the time-averaged liquid velocity distribution obtained by using a Laser Doppler Velocimetry (LDV) system in a cavitating nozzle of 4 mm in width ⁽¹²⁾. Flow contraction is formed at the inlet, and the velocity distribution near the exit is uniform. Based on the velocity distribution, we can obtain the pressure distribution along the axis of a nozzle shown in Fig. 7. Cavitation inception takes place in the separated boundary layer where the static pressure takes the lowest value.

Hiroyasu et al. ⁽²⁾ proposed the cavitation number σ_c , which is defined as

$$\sigma_c = \frac{P_c - P_v}{\frac{1}{2}\rho V_c^2} \quad (6)$$

The definition of σ_c is similar to that of σ defined by Eq. (3). The pressure P_c and the velocity V_c at the vena contracta are used in σ_c . To evaluate P_c in Eq. (6), they used the Bernoulli's equation:

$$P_c + \frac{1}{2}\rho V_c^2 = P_b + \frac{1}{2}\rho V_N^2 + \frac{\lambda(L - L_{cav})}{D_H} \frac{1}{2}\rho V_N^2 \quad (7)$$

where λ is the friction factor, and D_H the hydraulic equivalent diameter of a nozzle. Substituting Eqs. (2) and (7) into Eq. (6) yields

$$\sigma_c = C_c^2 \left[\frac{P_b - P_v}{\frac{1}{2}\rho V_N^2} + \frac{\lambda(L - L_{cav})}{D_H} + 1 \right] - 1 \quad (8)$$

They assumed that (i) $C_c = 0.6$ and (ii) $\lambda(L - L_{cav})/D_H = 0$, and carried out a validation test of σ_c defined by Eq. (9).

$$\sigma_c = 0.36 \left[\frac{P_b - P_v}{\frac{1}{2}\rho V_N^2} + 1 \right] - 1 \quad (9)$$

Although they found that cavitation is likely to be developed for low σ_c , they failed in a quantitative prediction of cavitation regime using σ_c given by Eq. (9). They suggested that quantitative verifications of the validity of σ_c are needed. No quantitative verification, however, has been carried out yet.

Fig. 6 Time-averaged liquid velocity distribution in the 2D nozzle ($W_N = 4$ mm) ⁽¹²⁾

Fig. 7 Pressure distribution in a cavitating nozzle

5. Applicability of the cavitation numbers

Whether or not the cavitation numbers σ , σ_2 , σ_3 and σ_c can predict the formation of supercavitation in nozzles of various geometries is tested. The dimensionless cavitation length L^* is used to represent the formation of supercavitation, i.e., $0.7 < L^* < 1$.

Considering the applicability of σ_c to long nozzles in which the frictional pressure drop is not negligible, we evaluate σ_c by

$$\sigma_c = C_c^2 \left[\frac{P_b - P_v}{\frac{1}{2} \rho V_N^2} + \frac{\lambda L}{D_H} + 1 \right] - 1 \quad (10)$$

The friction factor λ in Eq. (10) is calculated using

$$\lambda = \beta \frac{0.3164}{Re^{0.25}} \quad (11)$$

$$Re = \frac{D_H V_N}{\nu_L} \quad (12)$$

where β is given by ⁽¹³⁾

$$\beta = \begin{cases} 1.00 & \text{for cylindrical nozzles} \\ 1.15 & \text{for 2D nozzles of } W_N/t \cong 4 \end{cases} \quad (13)$$

The C_c for a cylindrical nozzle can be given as a function of C_u ⁽¹³⁾. Since there are no appropriate correlations of C_c for 2D nozzles, C_c for the 2D nozzles is given using the measured W_{cav} :

$$C_c = \begin{cases} 0.61 & \text{for cylindrical nozzles of } C_u \geq 10 \\ 0.625 & \text{for cylindrical nozzles of } C_u = 4 \\ 1 - \frac{2W_{cav}}{W_N} & \text{for 2D nozzles} \end{cases} \quad (14)$$

where W_{cav} is the cavitation thickness measured at $0.1L$ downstream of the nozzle inlet, at which the cavitation thickness and the time-averaged streamwise velocity reach constant values in the streamwise direction. Note that we have verified the validity of Eq. (14) for 2D nozzle based on liquid velocity distribution measured by using LDV ⁽¹²⁾.

Figure 8 shows the relations between L^* and the cavitation numbers, σ , σ_2 , σ_3 and σ_c . As shown in Fig. 8 (a), supercavitation ($0.7 < L^* < 1$) takes place for larger σ in the cylindrical short nozzles (the length-to-diameter ratio $L/D_N = 4$) than in the 2D nozzles. Since C_c of the cylindrical short nozzle is smaller than that of the 2D nozzle, the discrepancy in σ indicates that the degree of flow contraction (C_c) has to be taken into account to predict the formation of supercavitation in various nozzles. Supercavitation is formed for larger σ in the cylindrical short nozzles ($L/D_N = 4$) than in the long nozzles ($L/D_N = 20$). The difference indicates the importance of the frictional pressure drop. As shown in Figs. 8 (b) and (c), the similar discrepancies among the nozzles with different geometries can be seen for σ_2 and σ_3 . (Note that σ_2 shows the opposite trends due to its definition.) These discrepancies clearly indicate that σ , σ_2 and σ_3 , which do not take into account the effects of the flow contraction and frictional pressure drop, cannot predict the formation of supercavitation for nozzles of different geometries.

To the contrary, the relation between L^* and σ_c does not strongly depend on the nozzle geometry, that is, supercavitation is formed at $\sigma_c = -0.13 \pm 0.05$ for all the nozzles. Thus, the formation of supercavitation in nozzles with different geometries can be predicted using σ_c defined by Eq. (10).

The failure in the prediction of the formation of supercavitation using σ_c defined by Eq. (9) by Hiroyasu et al. ⁽²⁾ may be due to (i) the disregard of the frictional pressure drop, (ii) error in C_c due to dull edges of the nozzle inlet and (iii) measurement errors in V_N .

It should be noted that actual pressure at the vena contracta P_c is always higher than P_v , since hydraulic flip is formed before P_c becomes lower than P_v . In hydraulic flip, the actual pressure distribution differs from the pressure distribution shown in Fig. 7 and P_c becomes P_b . However, σ_c calculated by Eq. (10) is found to be useful for predicting supercavitation.

Fig. 8 Relation between $L^* (= L_{cav} / L)$ and the cavitation numbers

Note that both σ_c given by Eq. (8) and that by Eq. (10) are found to give good predictions. Since Eq. (10) does not include L_{cav} in its RHS, we can calculate its value more easily.

It should be also noted that C_c of the 2D nozzle was evaluated using Eq. (14), and the values of C_c of non-cavitating flows in cylindrical nozzles were used for those of the cavitating cylindrical nozzle. Hence, the agreement between the 2D and cylindrical nozzles implies that the thickness of the separated boundary layer can be regarded as that of cavitation zone.

Based on the concept of the traditional cavitation number⁽¹⁴⁾, the cavitation number σ_c' can be defined:

$$\sigma_c' = \frac{\frac{1}{2}\rho V_c^2 + P_c - P_v}{\frac{1}{2}\rho V_c^2} \quad (15)$$

Since $\sigma_c' = \sigma_c + 1$, σ_c' can be expressed as

$$\sigma_c' = C_c^2 \left[\frac{P_b - P_v}{\frac{1}{2}\rho V_N^2} + \frac{\lambda L}{D_H} + 1 \right] \quad (16)$$

Supercavitation is to be formed for $\sigma_c' \simeq 0.87 \pm 0.05$, and cavitation inception takes place for $\sigma_c' \simeq 1.05 \pm 0.15$. Note that cavitation inception occurs when σ_c' is close to unity.

6. Conclusions

The effects of nozzle geometry on cavitation in the nozzle of pressure atomizers are examined using two-dimensional (2D) nozzles of various geometries. Then, whether or not the conventional cavitation numbers can be utilized to predict the formation of supercavitation, at which liquid jet atomization is promoted, in various nozzles with different geometries is examined using the experimental data on development of cavitation in the 2D and cylindrical nozzles. As a result, the following conclusions are obtained:

- (1) The thickness W_{cav} of the cavitation zone in the supercavitation regime increases with the ratio C_u of the cross-sectional area upstream of the nozzle to that of the nozzle.
- (2) In the supercavitation regime, the spray angle θ increases with C_u .
- (3) We can predict the formation of supercavitation in nozzles with various geometries using the cavitation number σ_c' , in which the effects of the flow contraction and frictional pressure drop are taken into account.
- (4) The other conventional cavitation numbers σ , σ_2 and σ_3 are not appropriate for predicting the formation of supercavitation in nozzles with various geometries.

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