

Redefinition of standard equation for discharge coefficient of throat-tapped flow nozzle

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Abstract

Throat-tapped flow nozzle, which is widely used in power plants, are examined for several geometric parameters, throat-tap and upstream-tap diameter, surface roughness of nozzle and so on. It is found that the largest influence factor is the throat-tap diameter. According to the experimental result, new equations for the throat-tapped flow nozzle are proposed. All experimental discharge coefficient is within $\pm 0.5\%$ of the proposed equations.

1. Introduction

Throat-tapped flow nozzle which is one of differential pressure flow meter is widely used in electric power plant. For example, ASME PTC 6 [1] requires using the throat-tapped flow nozzle to evaluate the steam turbine in fire power plant. Furthermore, the throat-tapped flow nozzle is used in nuclear power plant to manage a thermal output. Thus, the role of the throat-tapped flow nozzle is very important for electric power plants.

The standard equation for discharge coefficient of the throat-tapped flow nozzle is defined in ASME PTC 6 and PTC 19.5 as following,

$$C_{PTC6} = k_t - \frac{0.185}{Re_d^{0.2}} \left(1 - \frac{361239}{Re_d} \right)^{0.8} \quad (1)$$

where, Re_d is Reynolds number based on a diameter of the throat area of the nozzle. The nominal value of the first term of the right-hand side is 1.0054. ASME PTC 6 requires that k_t is decided by the fitting of actual calibration data and it satisfies the range; $1.0029 < k_t < 1.0079$. This means that the discharge coefficient obtained by the actual calibration should be within $\pm 0.25\%$ from Eq.(1) with $k_t=1.0054$. However, we obtained over ranged experimental discharge coefficients in past examinations [2],[3]. In Fig. 1, the experimental discharge coefficients, C_x given by authors [2] and

Reader-Harris et al.[3] are shown. This figure indicates the following two issues. The first is the influence of the diameter of the throat-tap; d_t . The discharge coefficient increases with the diameter of the throat-tap. ASME PTC 6 defines that Eq. (1) is available for $d_t = 3.125 - 6.25$ mm. The difference between each throat-tap is 1.5% at maximum and it is not small to ignore the influence of the diameter of the throat-tap.

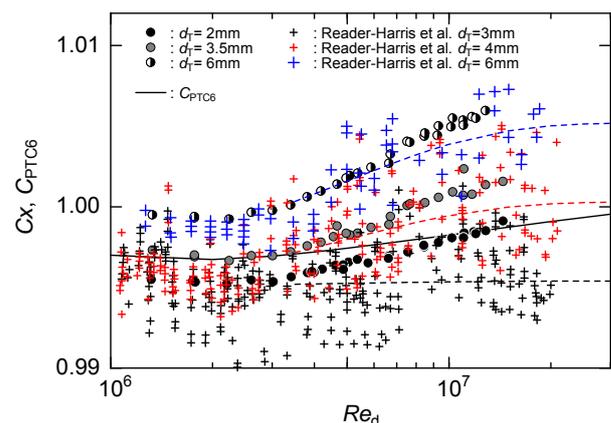


Figure 1: Discharge coefficients for throat-tapped flow nozzle by Furuichi et al. [2] and Reader-Harris et al [3].

The second issue is the experimental result of the discharge coefficient for high Reynolds number region. The increasing ratio of the discharge coefficient with Reynolds number is larger than Eq.(1). Consequently, the experimental discharge coefficient is largely deviated from Eq.(1) for larger

tap diameter, although it is within 0.25% for low Reynolds number region. The throat-tapped flow nozzle is generally used in power plants and the Reynolds number is up to 10^7 . This large difference at high Reynolds number region should be noticed with carefully.

According to the experimental results, we have proposed new equation of the discharge coefficient of throat-tapped flow nozzle in the previous reports [4]. In this paper, additional experimental results for the throat-tapped flow nozzle are presented and proposed the final equation of the discharge coefficient. As mentioned in previous, authors have been done many researches for the throat-tapped flow nozzle [2],[4]-[6]. This paper is a summary of these research works.

2. Experiments

The cut model of the throat-tapped flow nozzle is shown in Fig. 2. The parameters influenced to the discharge coefficient are throat diameter; d , pipe diameter; D , throat-tap diameter; d_t , upstream-tap diameter; d_u and surface roughness. In the series of the experiments, we examined for many nozzles with different geometric parameters shown in Table 1.

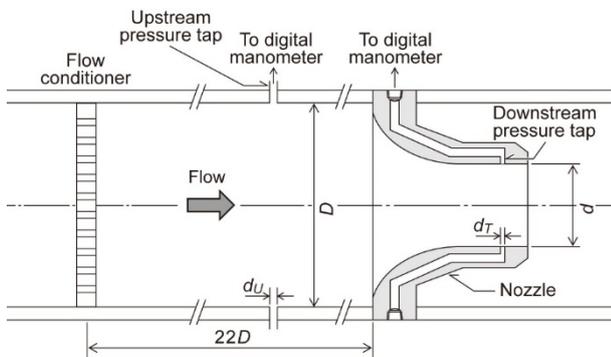


Figure 2: Schematic of throat-tapped flow nozzle.

Table 1: Parameters of examined throat-tapped flow nozzle.

Pipe diameter D (mm)	100, 200, 350	
Throat diameter d (mm)	50, 99, 165	
Diameter ratio β	app. 0.5	
Throat-tap diameter d_t (mm)	2, 3.5, 4, 5, 6, 7	
d_t/d	0.012 – 0.1	
Upstream-tap diameter d_u (mm)	2, 4, 5	
Surface roughness	Ra (μm)	0.10, 0.80
	Rt (μm)	0.60, 2.5

All experiments were carried out using the national standard of flow rate of water in NMIJ. The uncertainty of flow rate is different each experiment and it is from 0.040% to 0.10%. The uncertainty of the discharge coefficient is from 0.080% to 0.12% for all experiments.

3. Results

3.1 Influence of throat-tap diameter and d_t/d

The discharge coefficients for different d_t/d are shown in Fig. 3. C_{PTC6} is calculated from Eq.(1) with $k_t=1.0054$. We have examined three throat-tapped flow nozzles with different throat diameter, d as shown in Table 1. As mentioned in the introduction, the discharge coefficient increases with the size of the throat-tap diameter. This phenomenon, namely tap effect, is caused by the secondary flow in the pressure tap. The physical background of the tap effect has been clarified well in the previous papers [4][5]. From the results shown in Fig. 3, the discharge coefficient of the throat-tapped flow nozzle is influenced by the parameter d_t/d , namely dimensionless tap diameter. The discharge coefficients for $d_t/d = 0.035, 0.036, 0.040$ are almost same although the throat diameter and tap diameter are different each other. From this result, the equation for the discharge coefficient of the throat-tapped flow nozzle should be included the parameter, d_t/d .

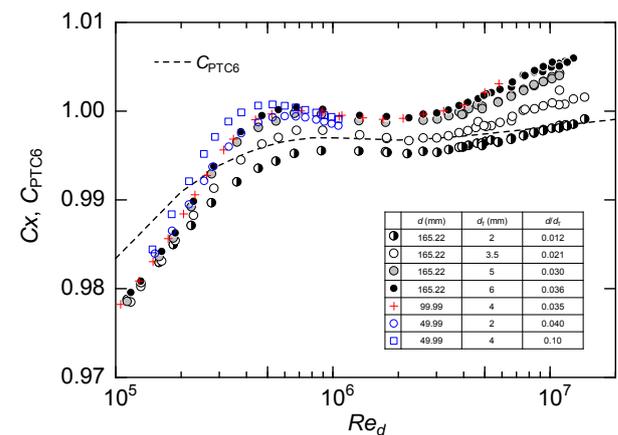


Figure 3: Discharge coefficients for different size of throat diameter d and throat-tap diameter d_t .

3.2 Influence of upstream-tap diameter

In generally, the measured pressure using wall pressure tap is larger than true pressure because of the tap effect [7]. Consequently, the differential pressure increases with increasing the throat-tap diameter and decreases with upstream-tap diameter. The discharge coefficients for different

upstream-tap diameter are shown in Fig. 4. The result shown in Fig.4 supports this phenomenon. The discharge coefficients for $d_U=2$ mm are larger than $d_U=5$ mm. However, the difference between two upstream-tap diameters is 0.02% at most. The uncertainty of the discharge coefficient is estimated to be 0.08% in this experiment. Therefore, it can be concluded that the influence of the upstream-tap diameter to the discharge coefficient is negligibly small.

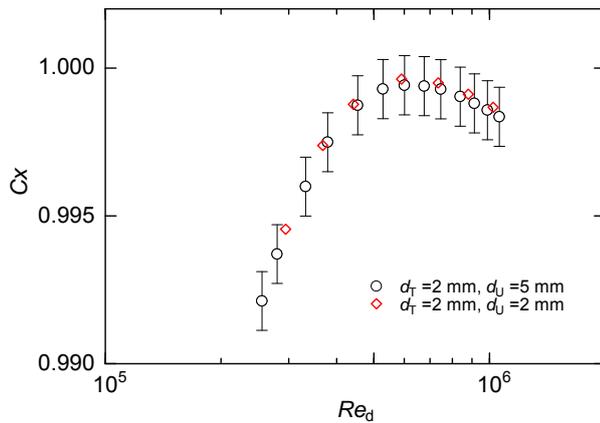


Figure 4: Influence of upstream-tap diameter.

3.3 Influence of roughness of surface

The discharge coefficient might be a function of the development of the boundary layer in the nozzle. Equation (1) is obtained by the theory of the boundary layer development by Murdock et al [8]. In this analysis, the surface finish of the nozzle is assumed to be hydraulic smooth. To satisfy the hydraulic smooth surface, the larger Reynolds number is, the smaller roughness required is. The relation between the required surface roughness and Reynolds number is following [9].

$$\frac{k_s}{d} < \frac{10\sqrt{2}}{Re_d\sqrt{\lambda}} \quad (2)$$

Where, k_s is the height of surface roughness, ν is the kinematic viscosity and λ is the friction factor. If the roughness does not satisfy this requirement, the thickness of the boundary layer on the throat-tap is larger than the hydraulic smooth surface. Figure 5 gives the results for different roughness of the nozzle. The mean roughness of R1 is $Ra=0.10$ μm and R2 is 0.80 μm . Both roughness value satisfy Eq.(2) for examined Reynolds number range, however the discharge coefficient is clearly influenced by the roughness at higher Reynolds number region, $Re_d > 7 \times 10^7$. In here, when Rt , which is the maximum roughness, is used instead FLOMEKO 2019, Lisbon, Portugal

of Ra in Eq.(2), the critical Reynolds number for R2 is $Re_d \approx 7 \times 10^7$. To evaluate the roughness of the nozzle, the relation between Rt and Reynolds number is better. It should be noted that the influence of the roughness is not related to the throat-tap diameter.

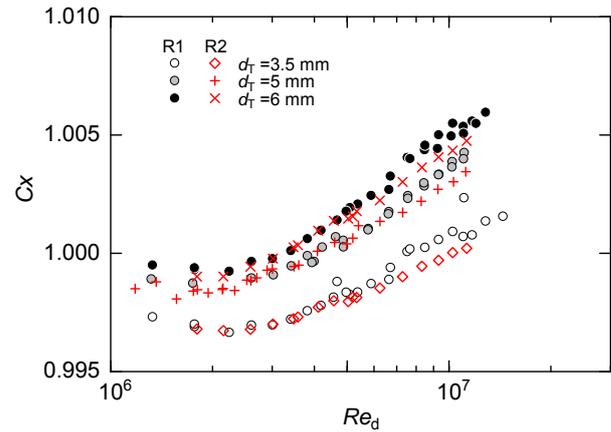


Figure 5: Influence of surface roughness of the nozzle

3.4 Influence of flow conditioner

ASME PTC6 requires to install the flow conditioner at upstream of the flow nozzle as shown in Fig. 2. The Mitsubishi type flow conditioner, which is plate type one, is installed at $22D$ upstream in the experiments presented above sections. The flow conditioner is useful to eliminate the influence the upstream flow condition, however it induces generally large pressure loss. In this section, the influence of the flow conditioner is examined in terms of the necessity of it.

The examinations with the flow conditioner, without the flow conditioner, and with a half-moon plate were carried out. The half-moon plate is to simulate the downstream flow field of a single elbow. The pipe layouts for the experiments are shown in Fig. 6. The flow conditioner is installed downstream of the DN600 pipe with $20D$ and the reducer to DN400 pipe. In the same position, the half-moon plate is installed instead of the flow conditioner. The discharge coefficients for each examination are shown in Fig. 7. From the comparison between with and without the flow conditioner, the discharge coefficient is not influenced by the flow conditioner in this pipe layout. Since the straight pipe section and reducer is installed at upstream of the flow conditioner, the velocity profile in there might be well developed and axisymmetric. Therefore, to disturb the flow field, the half-moon plate was installed in the same position of the flow conditioner and measured the discharge coefficient. The result is deviated app.

0.06% from the result without the half-moon plate. This deviation might not be large compared with the measurement uncertainty. When the upstream length is over than $22D$, the influence of the upstream condition appears to be not necessary to consider. This is reasonably equivalent with the requirement of straight pipe in ISO-5167.

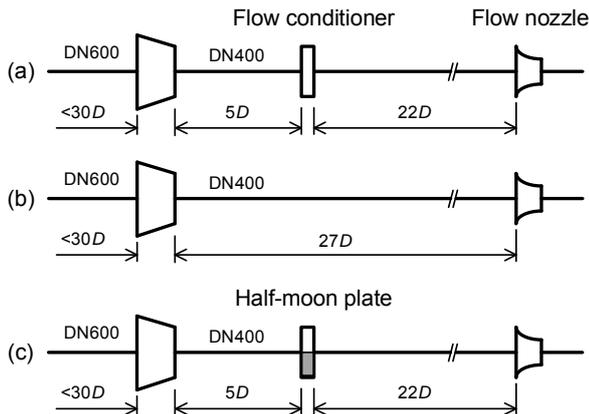


Figure 6: Pipe layouts.

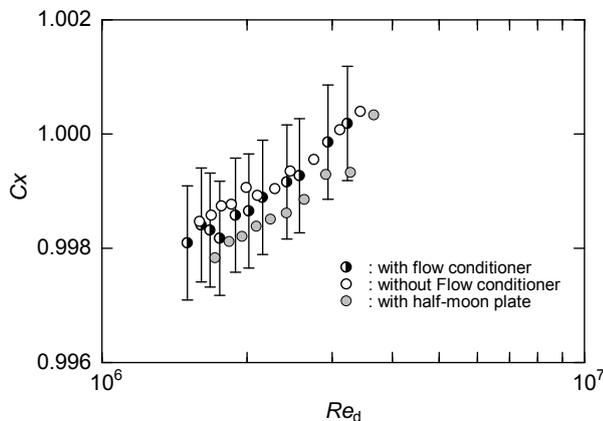


Figure 7: Influence of flow conditioner.

4. Discussion

We have already established the equations of discharge coefficient as the function of Reynolds number and d_t/d by combining the theoretical and experimental analysis [4]. As shown in Fig. 3, the discharge coefficient is clearly related to dimensionless throat-tap diameter. If the flow nozzle is manufactured correctly, the discharge coefficient should exhibit a theoretical or semi-empirical curve when plotted against the Reynolds number. This is important to confirm the extrapolation of the discharge coefficient. However, sometimes we face to the different results depending of manufacturing procedure. Figure 8 FLOMEKO 2019, Lisbon, Portugal

shows the example which does not lie on the result in Fig.3 and the proposed equations. Furthermore, the result in Fig. 8 is clearly strange for the tap effect. The discharge coefficient should increase with the throat tap diameter, however it increases as $d_t = 5, 6, 4,$ and 7 mm at $Re_d = 1.0 \times 10^6$. Although the reason of this result is not clear, but it might depend on the manufacturing finish of the throat tap. Therefore, the equation of the discharge coefficient for general use should be defined to cover such inconsistent data.

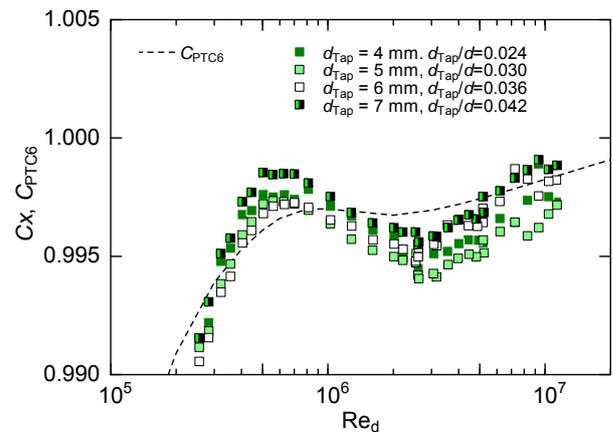


Figure 8: Experimental results of another throat-tapped flow nozzle.

The discharge coefficients obtained in the series of the experiments are plotted in Fig. 9. The all results for the parameters in shown in Table 1 are shown in there. We have done wide range of Reynolds number, from $Re_d = 5 \times 10^5$ to 1.4×10^7 . In the figure, Eq.(1) of ASME PTC6 with $\pm 0.25\%$ criteria is also shown. The behaviour of the experimental discharge coefficients is not so different from ASME equation, however, many of data is outside of the $\pm 0.25\%$ criteria, especially at high Reynolds number region. Since ordinal Reynolds number is up to $Re_d=10^7$ in most of plants, the equation for the discharge coefficients should be re-considered according those experimental data.

We propose the following new equations to give the discharge coefficient of the throat-tapped flow nozzle,

$$C_{P-1} = 1.0042 - \frac{8.41}{Re_d^{0.5}} \quad \text{for } Re_d < 1.3 \times 10^5 \quad (3)$$

$$C_{P-2} = 0.9558 - \frac{8.41}{Re_d^{0.5}} + 0.00492 \ln(Re_d) \quad \text{for } 1.3 \times 10^5 < Re_d < 4.0 \times 10^5 \quad (4)$$

$$C_{P-3} = 1.0090 - \frac{8.41}{Re_d^{0.5}} \quad \text{for } 4.0 \times 10^5 < Re_d < 8.0 \times 10^5 \quad (5)$$

$$C_{P-4} = 1.0090 - \frac{0.255}{Re_d^{0.2}} \left(1 - \frac{400000}{Re_d}\right)^{0.8} \quad \text{for } 8.0 \times 10^5 < Re_d < 3.0 \times 10^6 \quad (6)$$

$$C_{P-5} = 0.9823 - \frac{0.255}{Re_d^{0.2}} \left(1 - \frac{400000}{Re_d}\right)^{0.8} + 0.0018 \ln(Re_d) \quad \text{for } 3.0 \times 10^6 < Re_d \quad (7)$$

These equations are given by substitute $d_T/d=0.24$ for the proposed equation in the previous paper [4]. The proposed equations are shown in Fig. 9. All experimental discharge coefficients are within $\pm 0.5\%$ of the proposed equations.

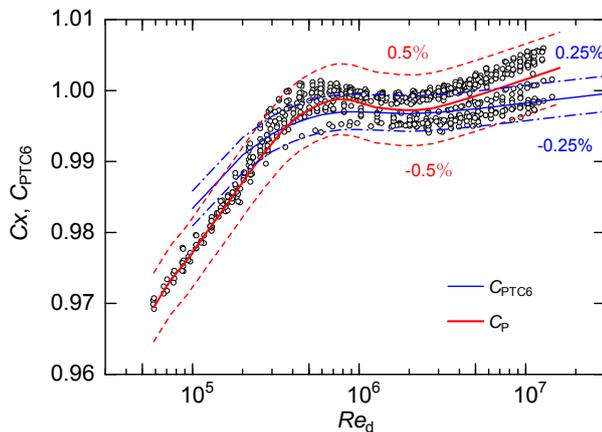


Figure 9: Examined data and proposed equation.

5. Conclusion

This paper presents experimental discharge coefficient for the several geometric parameters; throat-tap diameter, upstream-tap diameter, roughness of surface of nozzle and flow conditioner. The most influence parameter for the discharge coefficient is throat-tap diameter and the influence of the other parameters is generally negligible small. According to this result, new equations of the discharge coefficient for the throat-tapped flow nozzle is proposed. Although they are separated for five Reynolds number range, all experimental data in NMIJ is within $\pm 0.5\%$ of them.

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