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Original Paper (Invited)

Experimental and Numerical Investigations on Performances of Darriues-type Hydro Turbine with Inlet Nozzle

Daisuke Matsushita[1], Kei Tanaka[2], Satoshi Watanabe[1], Kusuo OKuma[1] and Akinori Furukawa[3]

[1]Department of Mechanical Engineering, Kyushu University
744 Motooka, Fukuoka, 819-0395, Japan
matu@mech.kyushu-u.ac.jp, fmnabe@mech.kyushu-u.ac.jp
[2]Graduate School of Engineering, Kyushu University
[3]Oita National College of Technology, 1666 Maki, Oita, 870-0152, Japan
a-furukawa@oita-ct.ac.jp

Abstract

Low head hydropower is one of realistic renewable energies. The Darrieus-type hydro turbine with an inlet nozzle is available for such low head conditions because of its simple structure with easy maintenance. Experimental and numerical studies are carried out in order to examine the effects of gap distances between the runner pitch circle and two edges of inlet nozzle on turbine performances. By selecting narrower gaps of left and right edges, the performance could be improved. From the results of two dimensional numerical simulations, the relation between the performance and flow behaviors around the Darrieus blade are discussed to obtain the guideline of appropriate inlet nozzle design.

Keywords: low head hydropower, Darrieus turbine, inlet nozzle-edge gap, performance test

1. Introduction

Needless to say, the global warming and energy problems are getting into serious ones. The most preferable solution is to utilize renewable energies. Hydropower is one of them, but extra-low head hydropower less than 2m is almost undeveloped yet. Figure 1 shows a selection chart for hydro turbines, where H and Q are effective head and flow rate, respectively. As there has been no turbine suitable for utilization of low head power (H<2m), we are developing a Darrieus-type hydro-turbine system for such extra-low head hydropower, and have demonstrated its effectiveness through laboratory experiments [1-3].

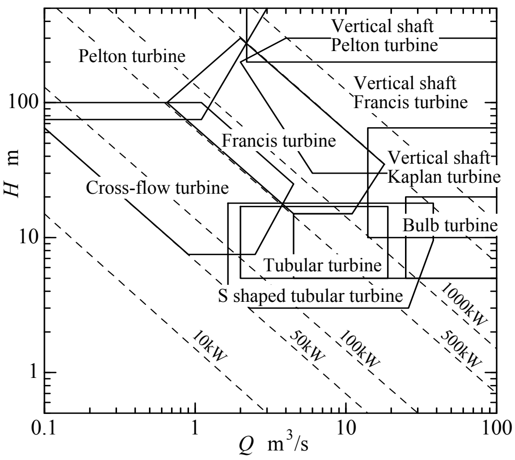

Fig. 1 Selection chart of hydro turbines

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Corresponding author: Daisuke Matsushita, Professor, matu@mech.kyushu-u.ac.jp

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The Darrieus-type hydro-turbine is one of cross-flow type turbines, which mainly consists of several numbers of two-dimensional blades rotating faster than the oncoming flow. Since the torque, i.e. the shaft power, is generated, in principle, by the tangential component of lift force working on each blade, hydraulic efficiency of this type of hydro-turbine is relatively higher than those of other cross-flow types. Moreover, in the applications for small rivers, irrigation channels and ditches, in which whole the flow stream can be guided into the hydro-turbine, it was found in our previous studies that the power could be increased by just putting the narrow inlet nozzle in front of the runner. In this case, we could remove even the runner casing and draft-tube without deteriorating the hydraulic efficiency, which would contribute the cost-reduction of whole hydro turbine system; the cost-effectiveness is one of the most important issues for the utilization of small hydropower.

In the present study, experimental and numerical studies are carried out in order to draw out more favorable effect of inlet nozzle. Since it has been experimentally found that the symmetric two-dimensional nozzle with the ratio of nozzle width S_{in} to the diameter of runner pitch circle D of S_{in}/D=0.8 reveals the best performance for Darriues-type runner among both the symmetric and asymmetric nozzles with S_{in}/D=0.7 - 1.0. Then, the effects of the gap distance between the runner pitch circle and left and right edges of the nozzle on turbine performances are herein investigated by using the symmetric nozzle with S_{in}/D=0.8. It is found that, smaller the gap between the nozzle edges and the runner pitch circle is, higher the generated power as well as the efficiency becomes. In order to understand the flow mechanism for the performance improvement, the unsteady two-dimensional numerical simulation with a commercial CFD code, ANSYS-CFX, is carried out. The flow behaviours around the Darrieus blade near the nozzle edge are discussed.

2. Working Principle of Darrieus-type Hydro Turbine

The working principle of Darrieus turbine in parallel duct is depicted in Fig. 2. The resultant force F with lift F_l and drag F_d varies in one revolution because the relative velocity W and the attack angle α are dependent on the rotating position θ, defined in Fig. 2, in the constant operation condition of oncoming absolute velocity V and the peripheral speed of blade U as the Darrieus turbine is cross-flow type. The generated power in one revolution of the blade L_t and the blade efficiency η_t are evaluated theoretically from the following expressions.

$$L_t = \int_0^{2\pi} F_u U d\theta \Big/ 2\pi \tag{1}$$

$$\eta_t = \int_0^{2\pi} F_u U d\theta \Big/ \int_0^{2\pi} (F_u U + F_d W) d\theta \tag{2}$$

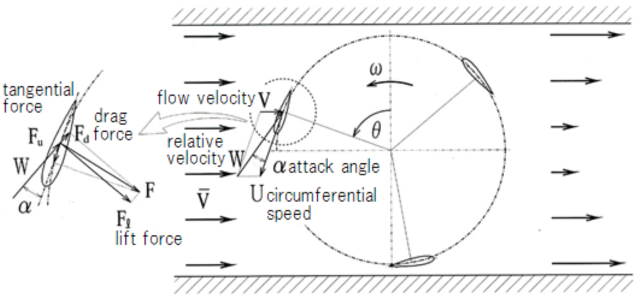


Fig. 2 Working principle of Darrius turbine

The turbine geometry preferable for higher η_t is as follows according to the previous investigation [4]. A symmetric blade section NACA0018 is adopted with consideration for stall, cavitation characteristics and camber effect, which is unique due to a circular rotating motion of Darrieus blade. A rotating symmetric Darrieus blade, having the leading and trailing edges out of the blade pitch circle, is equivalent to a stationary cambered blade in a uniform flow. In contrast, a cambered Darrieus blade with the camber line along the pitch circle is equivalent to a stationary symmetric blade.

Figure 3 shows the time variations of measured torque coefficient C_t=$T_1/(\rho V^{*2}BD)$ [5], where T_1 is instantaneous torque of one Darrieus blade evaluated from adding measured torque and torque loss due to rotating arm before installing blades, ρ the fluid density, V^* the average velocity at inlet section with the width S_{in}. The effect of narrow inlet nozzle with S_{in}/D =0.80 can be easily found in Fig.3. The open symbol □ is in the case of parallel walls with S_{in}/D = S/D =1.08 and the solid symbols ● and ♦ are in the cases of runner casing width with S/D =1.08 and 1.35 under S_{in}/D =0.80 constant. By installing the inlet nozzle with S_{in}/D =0.80, C_t at upstream region of θ=π/6 to 5π/6 becomes higher than that with S_{in}/D =1.08. Additionally, C_t at the upstream region keeps large values even in the case with S/D =1.35, in contrast to the apparent deterioration of C_t at downstream region (θ=π to 2π).

A key point for designing high efficiency turbine is how to extract the power efficiently at the upstream region in one revolution especially in the case with the inlet nozzle (S_{in}/D =0.80). A setting attitude of symmetric blade on the pitch circle is known to sensitively affect the efficiency, and the blade being set tangent to the pitch circle at its 1/2 chord point gives the best efficiency from the authors experiences. Blade number is decided through taking account of self-starting as mentioned later, though one blade is recommended for higher efficiency.

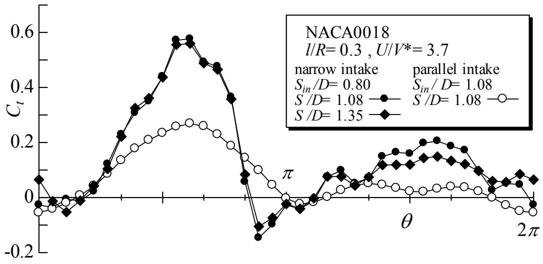


Fig. 3 Time variations of torque in one rotation of one Darrieus blade

Effects of runner casing width S/D on turbine efficiency have been investigated in both cases with parallel channel (S_{in}/D = S/D) and with the inlet nozzle (S_{in}/D =0.80$\neq S/D$). Figure 4 depicts the changes of the best efficiency, where the efficiency η is defined as the ratio of shaft output power to the supplied water power [4]. The symbols of open ○ and solid ● show the results without and with inlet nozzle, respectively. In the case without inlet nozzle (parallel), decreases extremely with increase of S/D. On the other hand, in the case with inlet nozzle with S_{in}/D =0.80 (narrow), is kept almost constant, being independent of S/D. For this reason, the adoption of inlet nozzle conduces to possibility for the removal of the draft tube and side-walls of the runner casing.

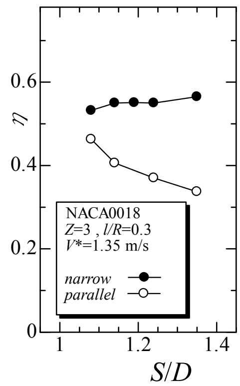


Fig. 4 Turbine efficiency against width of turbine casing walls

3. Experimental Setup

Figure 5 shows the experimental apparatus for performance measurement of Darrieus turbine. Water is supplied from an underground water reservoir 1 to the test open channel 6 through a pump 2. The water in open channel flows through the strainer 7 into the vertically installed test Darrieus runner 8, and then discharges to the underground water reservoir 1 through the downstream weir 12. The width of the open channel is W=1.2m. Head is kept by upstream weir with height is 0.8m of maximum and movable downstream weir 12. Water levels of upstream and downstream of the runner are calculated by measured static pressure on the bottom of the channel. The flow rate is controlled by a control valve 6 and is measured by the orifice flow meter 4. The local velocity at the nozzle is measured by portable propeller type velocimeter (Kenek, VTR-200-20N) at the center of the inlet nozzle. Electric current and voltage as the outputs of generator 9 are measured by a multi-meter 10 before the electric loads 11.

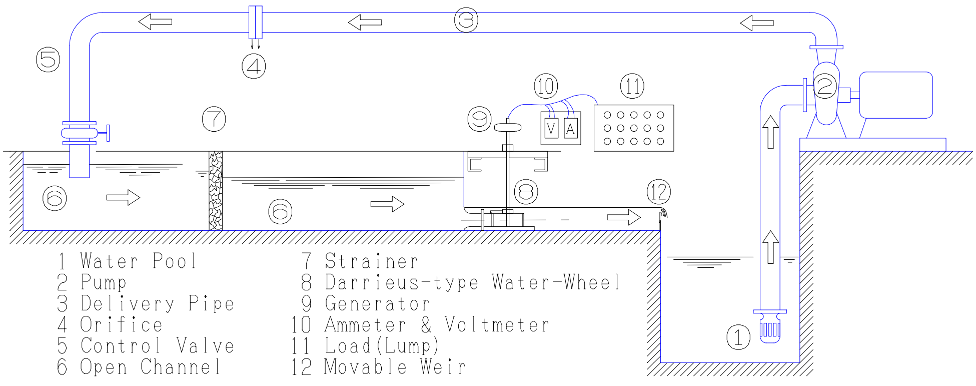


Fig. 5 Schematic view of experimental setup

The geometry and arrangement of the Darrieus runner and the inlet nozzle are shown in Fig. 6. The runner has 5 blades of NACA0018 (Z=5) with the chord length of l=75mm along a pitch circle with the diameter of D=500mm (radius of R=250mm). The blade span is B=300mm. The nozzle width S_{in} is decided by the relation of S_{in}/D=0.8, that geometry gives the best characteristics of the Darrieus turbine with the inlet nozzle in our past research [4]. From the same reason, the ratio of the chord length against the runner diameter l/R=0.3 and the overlapped centerlines of both of the runner and the nozzle are selected. The runner rotates counter clockwise way in the top view. Figure 6 also shows the definition of the radius of right and left nozzle edges, R_{na} and R_{nb} from the rotating axis. The right and left gap distances, ($R_{na}-R$) and ($R_{nb}-R$) between the runner pitch circle and each nozzle edge are changed in this experiment.

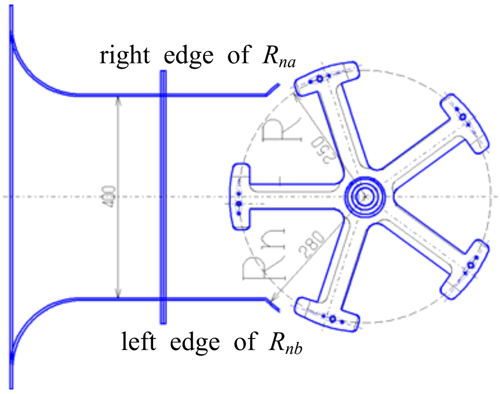


Fig. 6 Arrangement of runner and original inlet nozzle

The normalized turbine performances such as a head coefficient C_h, a power coefficient C_p and a turbine efficiency η are evaluated by,

$$C_h = \frac{H}{V_n^2/2g} \tag{3}$$

$$C_p = \frac{2P}{\rho Q V_n^2} \tag{4}$$

$$\eta = \frac{P}{\rho g Q_n H} = \frac{C_p}{C_h} \tag{5}$$

where ρ denotes the fluid density, g the acceleration of gravity, and P the generated power. Averaged velocity at the nozzle exit V_n is calculated by $V_n=Q/(BS_{in})$.

4. Numerical simulation

For numerical simulation, commercial CFD tool (ANSYS CFX) is used. The numerical model consists of two domains, a stationary domain for the flow channel and a rotational domain for the runner part, which are connected via so-called transient rotor stator interface. The grids in both domains are made of unstructured mesh, but for the limitation of computational resources, the two dimensional simulation is carried out with adapting symmetry condition at upper boundary and bottom boundary. Total grid numbers are 200,000 for the channel domain and 350,000 for the runner domain. Inlet boundary condition is the uniform inflow condition and constant pressure at the exit. One calculating step is set to correspond to 1 degree of the runner rotation. Variables for evaluation of turbine performances are calculated after the converged periodic flows are obtained.

The normalized turbine performances such as head coefficient C_{hc}, power coefficient C_{pc} and turbine efficiency η_c can be evaluated by,

$$C_{hc} = \frac{2(P_{in} - P_{out})}{\rho V_n^2} \tag{6}$$

$$C_{pc} = \frac{2T\omega}{\rho Q V_n^2} \tag{7}$$

$$\eta = \frac{T\omega}{(P_{in} - P_{out})Q} \tag{8}$$

where P_{in} and P_{out} are total pressure at 1.5D upstream and downstream position against the center of the runner.

5. Results and discussions

5.1 Effects of same nozzle gap with R_n=R_{na}=R_{nb}

Figure 7 shows the influence of the same nozzle gap with R_{na}=R_{nb} (=R_n) on the turbine performances. Horizontal axis is the speed ratio U/V_n defined as the blade rotating speed ratio U per the flow velocity at the inlet nozzle V_n. The vertical axes are the head coefficient, the power coefficient and the turbine efficiency. The original gap used in the previous studies corresponds to R_n/R=1.12, and R_n/R=1.06 is the minimum gap by the structural limitation. Solid plots ● denote experimental data and double circle plots ◎ numerical results under the condition of U/V_n=3.0.

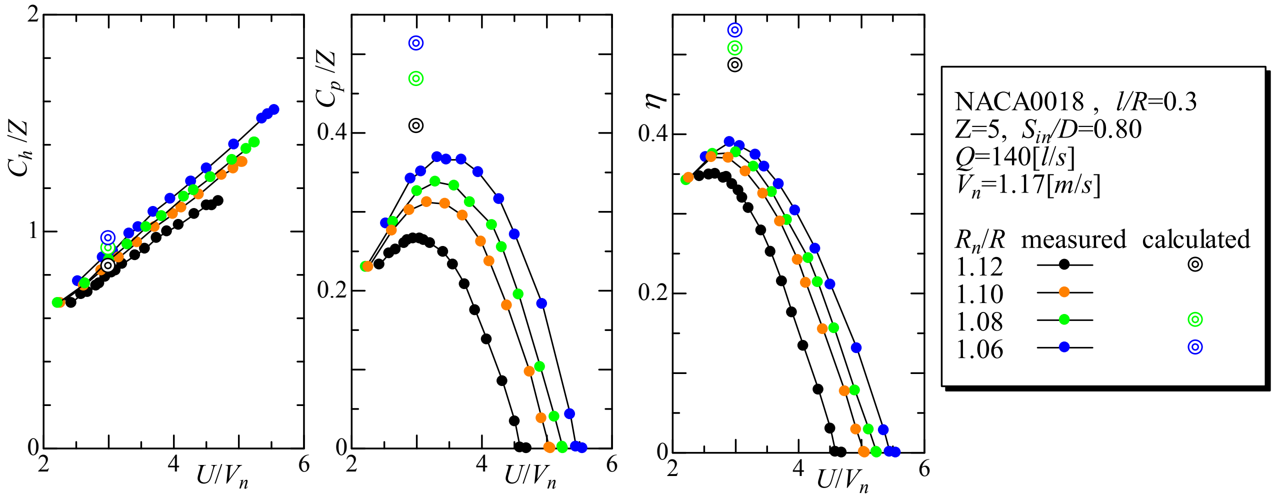


Fig. 7 Influence of nozzle gap on turbine performance (symmetry gap with R_n=R_{na}=R_{nb})

As reducing the gap, the power coefficient and the turbine efficiency increase and the ratio of U/V_n for the maximum efficiency also increases. This means that the turbine operation shifts more high rotational speed condition. Generated torque calculated by CFD is larger than the experimental one, mainly due to taking no account of the efficiency of the generator, but agrees much qualitatively with the measured one.

Figure 8 shows a simulated torque fluctuation for one revolution of the five-bladed runner. The horizontal axis is rotating angle θ defined in Fig. 2, indicating the position of the one of the blades. The vertical axis is the generated torque by the runner T (solid line) and the torque generated by one Darrieus blade T_1 (dashed line). Two vertical lines show the positions of each edge of the nozzle. It is found that the generated torque increases as the blade approaches to the edge of the nozzle, then, decreases quickly to around zero. Smaller gap between the nozzle edge and the runner pitch circle causes the weakened leakage flow through the gap, yielding pressure increase on the pressure side of blade due to the stagnation effect, and then the generated torque (the power) is increased.

Figure 9 shows the pressure distribution and the vector map for five-bladed runner with the inlet nozzle in the case where the blade is near the left edge of the nozzle, where one blade generates large torque as has been seen in Fig. 8. When one blade approaches to the left edge, the next blade locates inside of the right side of the nozzle. The narrow gaps take large static pressure around both of the nozzle edges, indicating the flow stagnation due to Darrieus blade. Then by the flow blockage effect of the blades, incoming flow distorts to the left side inside the nozzle, i.e. large amount of the fluid flows toward the region where the blade can generate larger torque. Actually, the time averaged velocity distributions at the nozzle exit, as shown in Fig. 10, indicate the decrease of the incoming velocity (X-component) near the nozzle right wall but the increase around the nozzle center. This distorted flow can contribute to the further increase of generated torque with the reduction of nozzle gap.

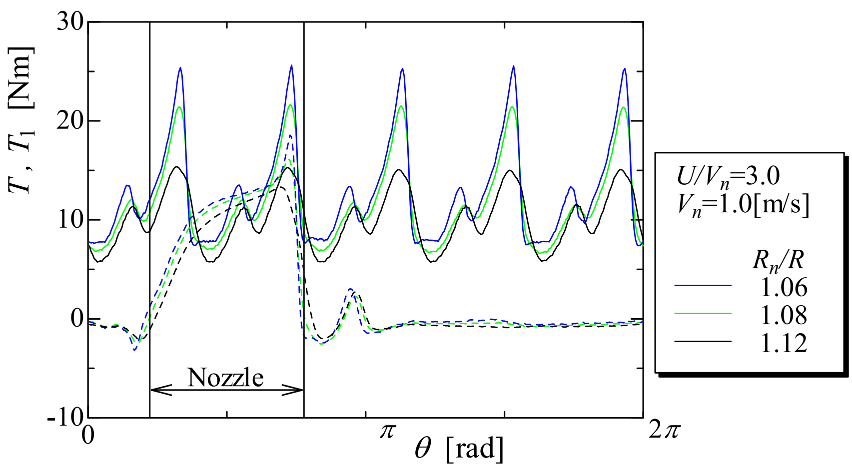
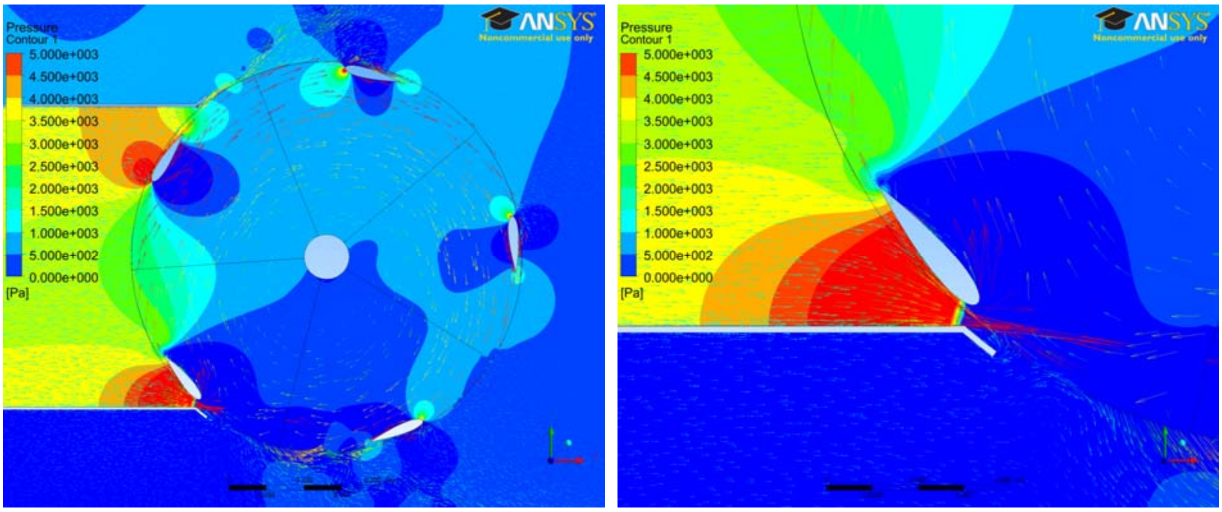
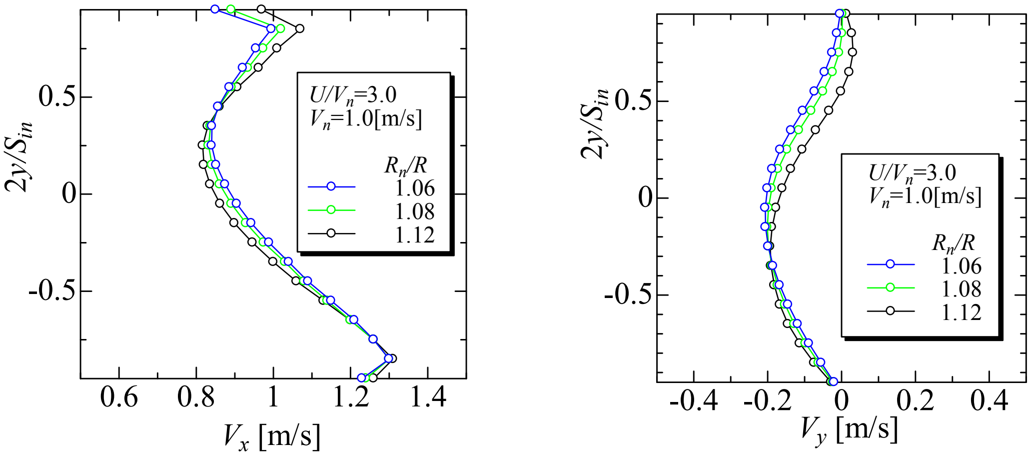


Fig. 8 Torque fluctuation for one revolution of runner



(a) Whole of the runner (b) Detail around right edge of the nozzle

Fig. 9 Pressure distribution



(a) X direction (b) Y direction

Fig. 10 Velocity distribution at exit of the nozzle

5.2 Effects of different nozzle gap with $R_{na} \neq R_{nb}$

In order to clarify the effect of the nozzle gap separately of the right and left nozzle edges on the turbine performances, these two gaps are independently changed. Two cases are examined; the case A is with the smaller gap only at right side (R_{na}/R, R_{nb}/R)=(1.08, 1.12), while the case B is the smaller gap only at left side (R_{na}/R, R_{nb}/R)= (1.12, 1.08). Figure 11 shows the turbine performance for various gaps, where the black open plots denote the original condition, the green open plots the symmetry nozzle case of R_{na} =R_{nb}, the red solid plots the case A of asymmetry nozzle and the blue solid plots the case B. Double circle plot shows the result of CFD in the case of U/V_n =3.0. Looking at the maximum efficiency condition with U/V_n =3.0, the case B takes almost the same value of the original gap although the case A takes a similar value with the narrow gap condition of R_{na}/R=R_{nb}/R=1.08 with the symmetry gap. From the calculated pressure distribution map (which is not shown here), it was confirmed that the high static pressure around the left side of the edge made the generated torque high in the case B. Figure 12 shows the time revolutions of generated torque by whole the runner T (solid curves) and one blade T_1 (dashed curves). The generated torque T_1 in the case B takes remarkably larger maximum value than in the other cases, but it takes smaller value in the other blade locations inside the nozzle, which limits the increase of torque, i.e. the generated power in the case B.

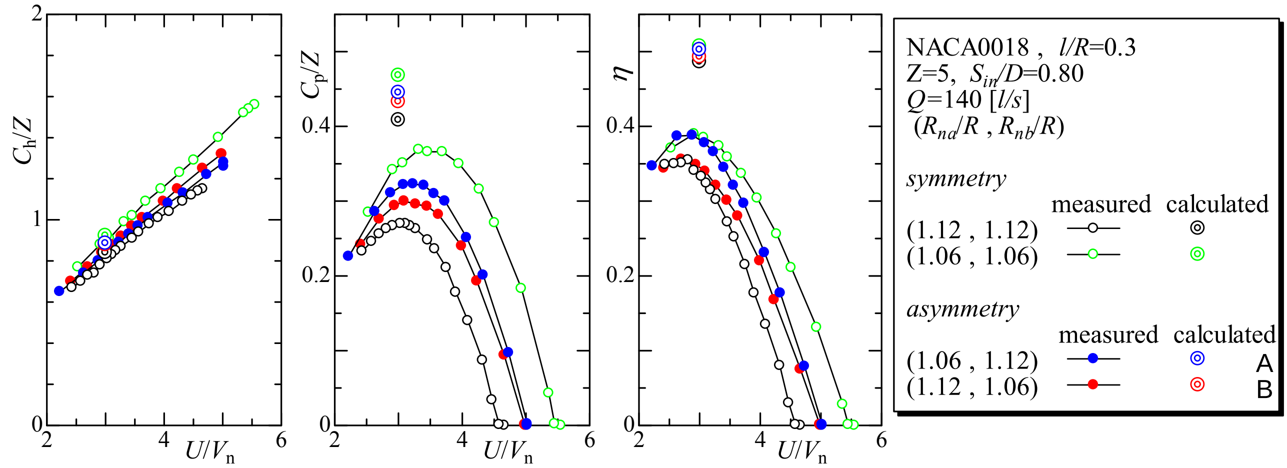


Fig. 11 Influence of nozzle gap on turbine performance (asymmetry gap)

Figure 13 shows the calculated pressure distributions at the same blade location for the cases A and B. In the case B, the pressure rise occurs in the whole nozzle passage although it occurs at only left side in the case A. The narrow gap at right side in the case A induces the blockage effect, and then the flow increases at the maximum torque locations in the center region of nozzle exit by flow redistribution. Figure 14 shows the time averaged velocity distributions at the nozzle exit, from which it can be confirmed that the larger flow distortion occurs in the case A rather than in the case B.

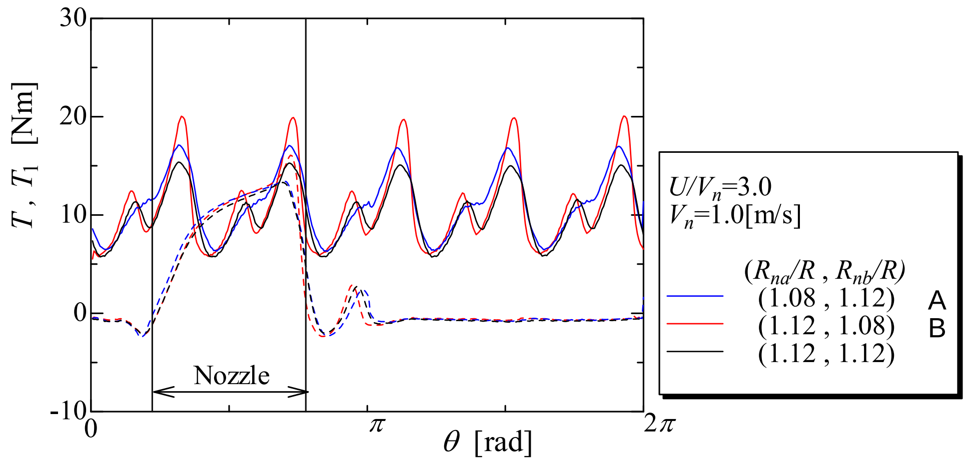
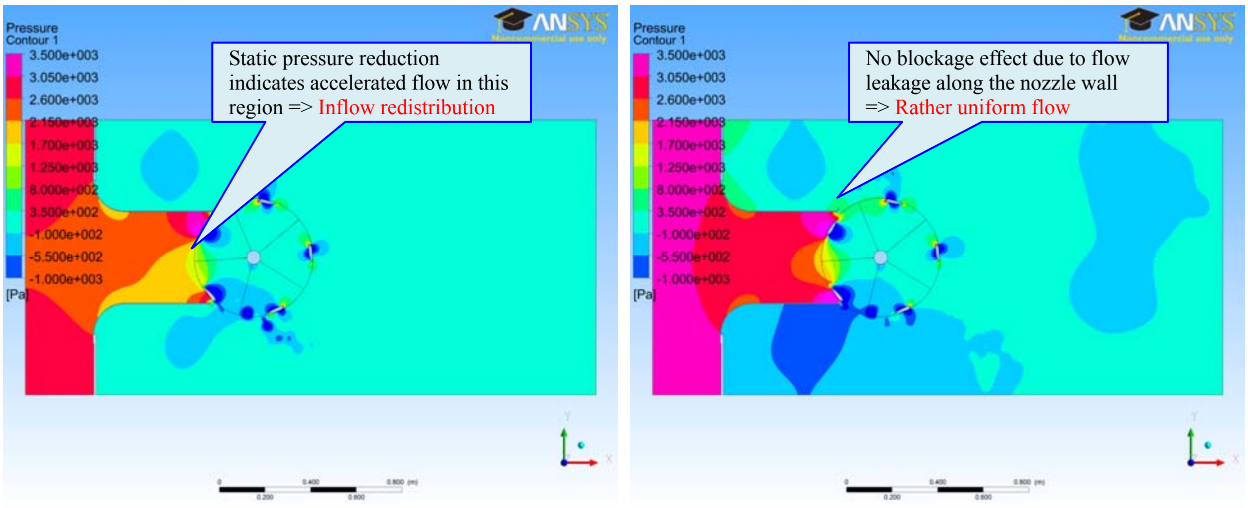


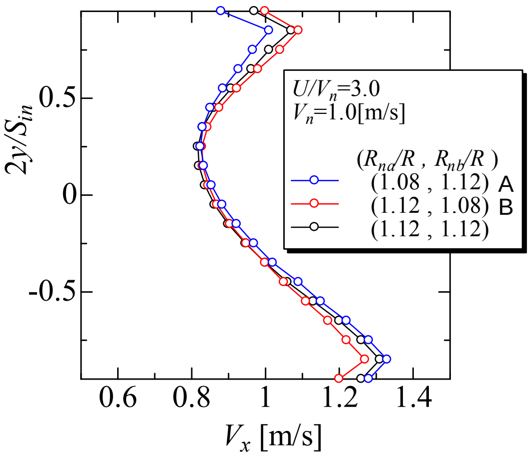
Fig. 12 Torque fluctuation for one revolution of runner



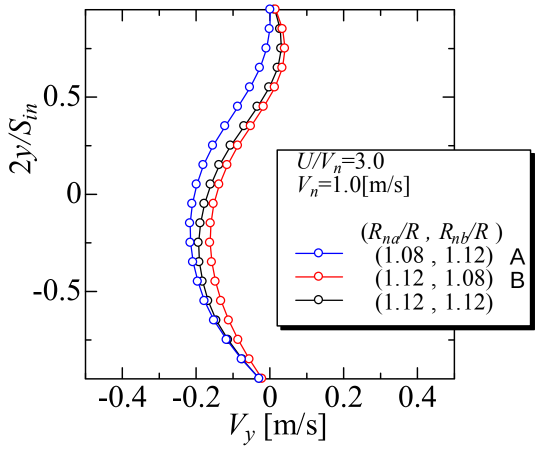
(a) case A

(b) case B

Fig. 13 Pressure distribution



(a) X direction



(b) Y direction

Fig. 14 Velocity distribution

Through the comparisons among examined geometrical cases of the inlet nozzle, it is clear that the nozzle gap which is defined as a distance between the nozzle edge and the runner affects the torque variation in one revolution of the blade as well as the flow distortion inside the nozzle. The turbine performance predicted by the present numerical simulation gives qualitative agreement with the experimental results. There is a large quantitative discrepancy on the estimation of the power coefficient, which is mainly due to the generator efficiency; it is not considered in the numerical simulations.

6. Summary

An influence of the gap between the nozzle edge and the runner is investigated for Darrieus-type hydro turbine with the inlet nozzle in a parallel open channel. The nozzles with optimized width and the several combinations of types of gaps are considered. The results can be summarized as follows.

Reducing the gap between the nozzle edge and the runner pitch circle can increase the turbine efficiency. It is known that the blade can produce the large torque with high efficiency when the blade locates at the upstream side of the runner. The reduction of nozzle gaps increases the generated power in the high efficiency blade locations, which contributes the improvement of turbine efficiency. This can be applied for the both cases, identical nozzle gaps and different nozzle gaps for the two nozzle edges.

In the case with same nozzle gaps between two nozzle edges, the reduction of the nozzle gap for the edge in which side the Darrieus blade enters the inlet nozzle passage is more effective for the performance improvement than that for the edge in the other side.

By reducing the nozzle gap, the blockage effect of the Darriues blade itself locating inside the inlet nozzle induces the inflow redistribution; the incoming velocity increases around the center of the nozzle, where the blade can generate large torque with higher efficiency. This flow redistribution is a principal factor for the improvement of the turbine efficiency with the reduction of nozzle gaps.

Nomenclature

B	Span length of Darrieus blade [m]	T	Generated torque of runner [Nm]
C_h	Head coefficient	T_1	Generated torque of one blade [Nm]
C_p	Power coefficient	U	Peripheral speed on runner pitch circle [m/s]
D	Diameter of runner pitch circle [m]	V_n	Average velocity of inlet nozzle [m/s]
g	Acceleration of gravity [m/s^2]	Wc	Width of channel [m]
H	Total head between up- and downstream [m]	W	Relative velocity [m/s]
L	Chord length [m]	Z	Number of Darrieus blades
P	Generated power [W]	α	Attack angle [deg]
P_{in}, P_{out}	Upstream and downstream total pressures [Pa]	ρ	Fluid density [kg/m^3]
Q	Flow rate [L/s]	η	Turbine efficiency
S_{in}	Inlet nozzle width [m]		

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