Tentative Study on Performance of Darriues-Type Hydroturbine Operated in Small Open Water Channel

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Abstract. The development of small hydropower is one of the realistic and preferable utilizations of renewable energy, but the extra-low head hydropower less than 2 m is almost undeveloped yet for some reasons. The authors have developed several types of Darrieus-type hydro-turbine system, and among them, the Darrieus-turbine with a wear and a nozzle installed upstream of turbine is so far in success to obtain more output power, i.e. more shaft torque, by gathering all water into the turbine. However, there can several cases exist, in which installing the wear covering all the flow channel width is unrealistic. Then, in the present study, the hydraulic performances of Darrieus-type hydro-turbine with the inlet nozzle is investigated, putting alone in a small open channel without upstream wear. In the experiment, the five-bladed Darrieus-type runner with the pitch-circle diameter of 300 mm and the blade span of 300 mm is vertically installed in the open channel with the width of 1,200 mm. The effectiveness of the shape of the inlet nozzle is also examined using two types of two-dimensional symmetric nozzle, the straight line nozzle (SL nozzle) with the converging angle of 45 degrees and the half diameter curved nozzle (HD nozzle) whose radius is a half diameter of runner pitch circle. Inlet and outlet nozzle widths are in common for the both nozzles, which are 540 mm and 240 mm respectively. All the experiments are carried out under the conditions with constant flow rate and downstream water level, and performances are evaluated by measured output torque and the measured head difference between the water levels upstream and downstream of the turbine. As a result, it is found that the output power is remarkably increased by installing the inlet nozzle, and the turbine with SL nozzle produces larger power than that with HD nozzle. However, the peak efficiency is deteriorated in both cases. The speed ratio defined by the rotor speed divided by the downstream water velocity at the peak efficiency is larger in both cases with the inlet nozzle, partly due to the increase of inflow velocity into the turbine. In order to understand the cause of the differences of power, i.e. torque characteristics of the turbine with SL and HD nozzles, two-dimensional CFD simulation is carried out. It is found that the instantaneous torque variation is important for the overall turbine performances, indicating the possibility of further performance improvement through the optimization of nozzle geometry.

1. Introduction

Small hydropower is one of prospective renewable energies in Japan. Especially, there are a plenty of available sites in low head conditions such as small rivers, agricultural water channels and many kinds
of waterway near the urban area. Furthermore, the extra-low head hydropower less than 2m is almost undeveloped yet. Figure 1 shows a selection chart of hydro-turbines, where \( H \) and \( Q \) are the effective head and the flow rate, respectively. As there has been no turbine suitable for utilization of low head power \((H<2m)\) which is focusing on, developing a Darrieus-type hydro-turbine system has been developed for such extra-low head hydropower, and have demonstrated its effectiveness through laboratory experiments [1-3].

The Darrieus-type hydro-turbine is one of the cross-flow type turbines, which mainly consists of several numbers of two-dimensional blades which rotates faster than the incoming 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 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 [2-4] that the power could be increased by just putting the narrow inlet nozzle in front of the runner. In this case, the hydro-turbine system does not require the runner casing and the draft-tube for keeping the high hydraulic efficiency, which would contribute the cost-reduction of the whole hydro-turbine system; the cost-effectiveness is one of the most important issues for the utilization of small hydropower.

The authors have developed several types of Darrieus-type hydro-turbine system, and among them, the Darrieus-turbine with a wear and a nozzle installed upstream of turbine is so far in success to obtain more output power, i.e. more shaft torque, by gathering all water into the turbine. However, there can several cases exist, in which installing the wear covering all the flow channel width is unrealistic. Then, in the present study, the hydraulic performances of Darrieus-type hydro-turbine with the inlet nozzle is investigated, put alone in a small open channel without upstream wear. The experimental and numerical studies are carried out in order to draw out more favorable effect of inlet nozzle in such situation. Since it has been experimentally found that the symmetric two-dimensional nozzle with the ratio of nozzle width \( S_n \) to the diameter of runner pitch circle \( D \) of \( S_n/D=0.8 \) reveals the best performance for Darriues-type runner among nozzles with \( S_n/D=0.7 - 1.0 \). Then, the effects of the shape of the nozzle on turbine performances are herein investigated. 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 runner and two types of nozzles are discussed.

2. Experimental setup

2.1. Working principal of Darrieus-type hydro turbine
The working principle of Darrieus turbine in a parallel duct is depicted in fig. 2. Since the Darrieus turbine is a cross-flow type, the resultant force $F$ with the lift and drag components, $F_l$ and $F_d$, varies during the revolution because the relative velocity $W$ and the attack angle $\alpha$ are dependent on the rotating position $\theta$, as shown in this figure, for the constant operation condition of incoming absolute velocity $V$ and the peripheral speed of blade $U$. The generated power in one revolution of the blade $L_t$ and the blade efficiency $\eta_t$ are theoretically expressed by the following expressions.

$$L_t = \int_0^{2\pi} F_u U d\theta / 2\pi$$  \hspace{1cm} (1)

$$\eta_t = \int_0^{2\pi} F_u U d\theta / \int_0^{2\pi} (F_u U + F_d W) d\theta$$  \hspace{1cm} (2)

**Figure 2.** Working principle of Darrieus turbine

The turbine geometry preferable with higher efficiency $\eta_t$ is determined as follows according to the previous investigation [4]. A symmetric blade section NACA0018 is adopted taking account of stall, cavitation characteristics and the camber effect. The latest is a unique character of the Darrieus blade due to its circular rotating motion; 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 $C_t = T_1 / (2 \rho V^2 B R^2)$ [5], where $T_1$ is the instantaneous torque of one Darrieus blade evaluated from subtracting the torque loss due to the rotating arm from the measured torque, $\rho$ the fluid density, $V^*$ the average velocity at the inlet section with the width $S_{in}$. The effect of the narrow inlet nozzle with $S_{in}/D=0.80$ can be easily found in the figure. By installing the inlet nozzle with $S_{in}/D=0.80$, $C_t$ at the blade locations in the upstream region of $\theta=\pi/6$ to $5\pi/6$ becomes higher than that with $S_{in}/D=1.08$. Additionally, $C_t$ in the upstream region keeps large values even in the case with $S/D=1.35$, in contrast to the apparent deterioration of $C_t$ in the downstream region ($\theta=\pi$ to $2\pi$).

**Figure 3** Time variations of torque in one revolution of one Darrieus blade
A key point for designing the high efficiency turbine is how to extract the power efficiently at the upstream region in one revolution of the runner especially in cases with the inlet nozzle \((S_o/D = 0.80)\). A setting attitude of the 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 is known to give the best efficiency from the authors experiences. Blade number is five which is decided through taking account of a view point of the self-starting characteristics.

2.2. Experimental apparatus

Figure 5 shows the experimental apparatus for the performance measurement of the present Darrieus turbine. Water is supplied from an underground water reservoir \(\circ1\) to a test open channel \(\circ6\) through a pump \(\circ2\). The water in the open channel flows through a strainer \(\circ7\) into the vertically installed test Darrieus runner \(\circ8\), and then discharges to the underground water reservoir \(\circ1\) through a downstream weir \(\circ11\). The width of the open channel is \(W=1.2m\). The downstream water level is set by the movable downstream weir \(\circ11\). Water levels of upstream and downstream of the runner are calculated from the measured static pressure at the bottom of the channel. The flow rate is controlled by a control valve \(\circ5\) and is measured by an orifice flow meter \(\circ4\). The test Darrieus runner is driven by a motor \(\circ10\) and the torque is measured by a torque meter \(\circ9\) (Unipulse, UTM-30Nm). The local velocity could be also measured by a portable propeller type velocimeter (Kenek, VTR-200-20N), which would not be presented in this paper.

![Figure 4 Schematic view of experimental setup](image-url)

The geometry and the arrangement of the Darrieus runner and the inlet nozzle are shown in fig. 5. The runner has five blades \((Z=5)\) of NACA0018 with the chord length of \(l=45\)mm and the blade span of \(B=300\)mm, which are set tangent to a pitch circle with the diameter of \(D=300\)mm (radius of \(R=150\)mm).
at their mid-chords. The nozzle exit width $S_{n}$ is decided from the known preferable relation of $S_{in}/D=0.8$, which 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 clockwise way in the top view.

The objective of the installation of the inlet nozzle is to gather the water flow into the Darrieus runner. In the present study, two types of two-dimensional symmetric nozzles as shown in fig. 5 are examined, which are named the straight line nozzle (SL nozzle) and the half diameter curved nozzle (HD nozzle); SL nozzle consists of straight walls with the converging angle of 45 degrees, whereas HD nozzle consists of curved walls, whose radius are a half diameter of the runner pitch circle. The inlet and outlet widths of the nozzles are in common, which are $W_{n}=540\text{mm}$ and $S_{in}=240\text{mm}$ respectively. Then, the horizontal blockage of the nozzle to the channel is expressed by $W_{n}/W=0.4$.

![Figure 5 Geometry and arrangement of the runner and the inlet nozzle](image_url)

As for the non-dimensional performance parameters, the power coefficient $C_{p}$ ($C_{pg}$, $C_{pgn}$), the head coefficient $C_{h}$ and the turbine efficiency $\eta$ are defined as follows

$$C_{pg} = \frac{T_{\omega}}{\frac{1}{2}\rho A_{r} \bar{V}^{3}} \quad \text{(parallel, without nozzle)}$$

$$C_{pgn} = \frac{T_{\omega}}{\frac{1}{2}\rho A_{n} \bar{V}^{3}} \quad \text{(with HD/SL nozzle)}$$

$$C_{h} = \frac{H}{\bar{V}^{2}/2g}$$

$$\eta_{g} = \frac{T_{\omega}}{\rho g Q H}$$

where $H$ denotes the total head difference between upstream and downstream of the runner assuming the uniform velocity distributions, $g$ the acceleration of gravity, $Q$ the flow rate, $T$ the measured torque, and $\omega$ the angular rotational speed of the runner. For the reference velocity, the area averaged velocity $\bar{V}$ downstream of the runner is used, which are calculated by $\bar{V}=Q/(h_{d}W)$ where $h_{d}$ is the measured downstream water level. For the calculation of the power coefficients, two reference area $A_{r}$ and $A_{n}$ are used for $C_{pg}$ and $C_{pgn}$ respectively; $A_{r}$ is the projected area of the runner and $A_{n}$ is the area of inlet section of the nozzle. It should be noted that, in the efficiency $\eta$, flow losses other than those in the hydro-turbine such as the mixing loss downstream of the runner are included.
The experiments for the performance evaluation are done at the constant flow rate $Q$ and the constant downstream water level $h_d$.

3. Numerical simulation

In order to discuss the effectiveness of the nozzle design from the flow fields, unsteady numerical simulation using a commercial CFD code ANSYS-CFX is carried out. The numerical model consists of two domains, a stationary domain for the flow channel and a rotational domain for the runner part as shown in fig. 6, 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 approximately 1,200,000 for the channel domain and 270,000 for the runner domain. The inlet boundary condition is the uniform inflow condition while the constant pressure condition for the outlet boundary. A time step is set to correspond to one degree of the runner rotation. The solution after the converged periodic flows are obtained is used for the discussion of the flow field as well as for the performance evaluations.

Since the present simulation is two-dimensional, the definitions of the turbine performances such as the power coefficient $C_{pc}$, the head coefficient $C_{hc}$, and the turbine efficiency $\eta_c$ are different from those in the experiment, which are defined as

$$C_{pc} = \frac{T\omega}{\frac{1}{2}\rho Q \bar{V}^2}$$

$$C_{hc} = \frac{2(P_{in} - P_{out})}{\rho \bar{V}^2}$$

$$\eta_c = \frac{T\omega}{(P_{in} - P_{out})Q}$$

where $P_{in}$ and $P_{out}$ are mass averaged total pressure at the inlet and outlet boundaries. For this performance evaluation, the time-averaged values during one revolution of the runner are used.

4. Results and discussion

4.1. Effects of installation of inlet nozzle on turbine performance

Figure 7 shows the effects of HD and SL nozzles on the turbine performances. The horizontal axis is the speed ratio $U/\bar{V}$ defined as the blade rotating speed ratio $U$ per the averaged flow velocity at the downstream region $\bar{V}$. The flow rate $Q$ is 450L/s and the downstream water level is $h_d/B=1.43$, which is deep enough to have all the blades under the water.

By comparing the performances between with and without inlet nozzles, it notice that the power coefficient $C_{pgn}$ with the inlet nozzles is much larger than $C_{pg}$ without nozzles despite that the reference area in the denominator of the power coefficient is larger in $C_{pgn}$. The head coefficient $C_h$ is also larger.
in the case with the inlet nozzle, indicating that the Darrius runner can extract more energy from the water stream with the inlet nozzle than that without nozzles. However, the efficiency is remarkably deteriorated by installing the inlet nozzle, which means that the energy losses significantly increase. It is presumed that the mixing loss after the runner occupies substantial part of energy losses, in addition to the energy loss in the runner. It can be also noticed that the speed ratio with the maximum power coefficient and the maximum efficiency is larger in the cases with both inlet nozzles. From this fact, the actual flow velocity into the runner is increased by the inlet nozzle, meaning that the water flow into the runner is increased as expected by installing the inlet nozzle.

Next, by comparing the performances between with SL and HD nozzles, it is found that the power coefficient $C_{pp}$ is apparently larger with SL nozzle while the difference of head coefficient $C_h$ is small, resulting in the better efficiency in the case with SL nozzle. This indicates the possibility of the performance improvement by further optimization of the nozzle geometry.

![Figure 7 Turbine performances obtained by experiment](image1)

![Figure 8 Turbine performances with SL nozzle for the various downstream water levels](image2)

Figure 8 shows the turbine performances with SL nozzle for the various downstream water levels $H_d/B$. It can be found that, with sufficient downstream water level than some critical value, the
normalized turbine performances can be kept, which are useful for the design and planning of the turbine applied for the actual sites. If the water level decrease below its critical value, the power coefficient is remarkably deteriorated, resulting in the efficiency deterioration. The same tendency was found for the other two cases, those are with HD nozzle and without nozzles.

In the next section, the reason for the better performance obtained with SL nozzle using the results of the numerical simulations will be discussed.

4.2. Numerical performances and flow fields

Figure 9 shows the comparisons of the turbine performances with inlet nozzles between the numerical simulations and the experiments. It should be noted that the quantitative comparison cannot be discussed since the numerical simulation is two-dimensional without considering the water surface. In the numerical simulation, the evaluated performances fluctuate even though the time averaged values are used, so that the amplitudes of fluctuations by \( \pm \) and \( x \) are also plotted. From those figures, the CFD reproduces qualitatively the experimental results; the power coefficient is apparently larger in the case with the SL nozzle, while the difference of the head coefficient is not very large. Then, the present numerical simulation is plausible for the qualitative discussions about the effect of the inlet nozzle.

![Figure 9](image.png)

**Figure 9** Comparisons of turbine performances between numerical simulations and experiments

Figure 10 shows the phase variations of the instantaneous torque generated by the runner during its one revolution in the cases with SL and HD nozzle. The results at the speed ratio \( U/V \) with the maximum power coefficient are chosen for comparisons. In fig. 10 (b), the instantaneous generated torque per one blade (Z1) is plotted in addition to that by the whole runner (Z5). It is found that the high torque is obtained when one of the blades locates in the water stream from the nozzle, namely \( \theta - n \times 72^\circ = 42^\circ \sim 138^\circ \).

From the torque generated by one blade, it is found three major differences between SL and HD nozzles. In the region (i), the instantaneous torque with HD nozzle is larger than that with SL nozzle. However, in the region (ii) where the maximum torque is observed for the both nozzles, the generated torque is much larger for SL nozzle, which might be the main contribution of more generated power for SL nozzle. This is probably related to the flow direction into the nozzle in that region; the incoming flow in this region is considered to be along with the nozzle wall. In the region (iii), the negative torque is observed for HD nozzle, whereas for SL nozzle the negative torque is diminished, which is also considered to contribute to more generated power for SL nozzle.
Figure. 10 Torque coefficient variations obtained by experiment and numerical simulation at the speed ratio with the maximum power coefficient

Figure. 11 Instantaneous pressure distributions around runner with SL nozzle (top) and HD nozzle (bottom)

Figure 11 shows the instantaneous pressure distributions when one of the blades locates at (a) $\theta=90^\circ$, (b) $130^\circ$ and (c) $155^\circ$. At $\theta=90^\circ$, it is difficult to recognize the difference of the pressure distribution around the blade between SL and HD nozzles. However, at $\theta=150^\circ$, where the generated torque takes the maximum, the large pressure on the pressure side is acting on the blade especially in the case with
SL nozzle, which might result in the larger peak of torque variation for SL nozzle. Assuming that the incoming flow is almost along the nozzle wall, the angle of attack is larger for SL nozzle, resulting in the larger generated torque at this position. At $\theta=155^\circ$, large low pressure region can be found near the trailing edge on the pressure side (inner side) for HD nozzle, which seems to be the reason for the negative generated torque as described before.

The above discussion again indicates the possibility of the performance improvement by further optimization of the nozzle geometry and that the converging angle near the runner seems to be very important, which will be investigated in our future study.

5. Conclusions

In the present study, the authors have experimentally and numerically investigated the hydraulic performances of Darrieus-type hydro-turbine with the inlet nozzle, put alone in a small open channel without upstream wear. Main results are as follows.

1. The inlet nozzle is effective to collect the water stream and then to increase the generated power for the vertical-axis type Darrius runner. The consumed head is also increased, resulting in the deterioration in the total efficiency. It should be noted that the energy losses outside of the runner are included in the input power.

2. The normalized turbine performances are available as far as the runner is installed sufficiently below the water surface, which are useful for the design and the planning of the hydro-turbine system for the actual low head sites.

3. Among two examined types of the nozzle, the straight line nozzle (SL nozzle) and the half diameter curved nozzle (HD nozzle), SL nozzle contributes to more generated power with the higher efficiency. This indicates the possibility of the performance improvement by further optimization of the nozzle geometry.

4. Two-dimensional numerical model constructed in the present study well reproduces the effects of the inlet nozzle on the turbine performances qualitatively.

5. The instantaneous torque generated by one blade and the surrounding flow field indicate that the converging angle of the nozzle near the runner is important for the turbine performances.

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