Influence of meridional plane shape on performance and internal flow of high head contra-rotating small hydroturbine

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Abstract. It is requested that the hydroturbine be a small size and have high performance. Therefore, we adopted contra-rotating rotors, which can be expected to achieve radial compactification and high performance. However, a conventional contra-rotating rotor was composed of two axial flow rotors, so it had a specification suitable for high flow rate and low head. In order to achieve small hydroturbine suitable for low flow rate and high head, we propose a new type of contra-rotating rotors, which are composed of a hybrid rotor and a centrifugal rotor. In the present paper, we focus on the meridional plane shape at the connection between the front rotor and the rear rotor of the hydroturbine composed of these rotors as a first step of this research, and investigate performance by the numerical analysis. As a result, we clarified that the relationship between the performance and internal flow when the blade width at the connection between the front rotor and the rear rotor changed.

1. Introduction

There is a strong demand to change energy sources from fossil fuels to renewable energy such as hydropower, wind power, solar energy and so on. Small hydropower generation is one type of alternative energy, and there is a significant potential for small hydroturbines. Small hydropower facilities that generate about 100[kW]-1000[kW] have spread widely. However, it causes environmental destruction by foundation construction and the set-up of draft tubes. On the other hand, there are a lot of places that can generate about 100[W]-1[kW] (pico-hydropower) in agricultural water and a small stream. Small hydropower installations are expected to have lower environmental impact, and the spread of small hydropower installations which effectively use these small hydropower resources is important. Therefore, darrieus and gyro-type turbines, which were suitable for the design specification of low head in agricultural water and a small river, were investigated and the performance characteristics and the optimum design parameter were discussed [1,2]. Further, a small-cross flow turbine used for a small stream as an environmentally friendly pico-hydroturbine and a savonius turbine with low cost were suggested, and the performance characteristics were discussed [3,4]. Efficiency of small hydroturbines is lower than that of a large one. Then, there are demands for small hydroturbines to be able to keep high performance. Therefore, we adopted contra-rotating rotors,
which could be expected to achieve high performance. However, it is difficult to adopt a complicated mechanism, because the hydroturbine in this study is small. Therefore, it assumed that the energy applied to the contra-rotating rotors will be recovered by two independent generators. The final goal of this study is the development of a small hydroturbine like electrical goods, which has high portability and makes effective use of the unused small hydropower energy resource.

In previous research, we selected some places in Tokushima Prefecture in Japan where small hydropower can be generated, and conducted field tests of head, flow rate, water quality and capacity utilization [5]. Then we investigated performance of the test model by the numerical analysis and a prospect to apply this small hydroturbine for pico-hydropower was confirmed [6]. After that, we designed and manufactured an experimental apparatus to verify the performance of it and investigated the internal flow condition by the numerical analysis results [7].

In this research, based on the previous research results, we improved the conventional contra-rotating small hydroturbine and considered a high performance small hydroturbine with low specific speed suitable for low flow rate and high head. In order to achieve it, the new contra-rotating small hydroturbine was adopted a centrifugal rotor as the rear rotor and a hybrid rotor composed of an axial flow rotor and a mixed flow rotor as the front rotor. As a first step of this research, we investigated the performance of the new contra-rotating small hydroturbine by the numerical analysis. In the previous paper, we reported the influence of deflection angle and number of blades which affects the performance of the hydroturbine [8,9]. In the present paper, we report the influence of meridional plane shape, which affects the performance of the hydroturbine. Among them, we focus on the blade width at the connection between the front rotor and the rear rotor, which affects the performance of contra-rotating rotor greatly. The relationship between the meridional plane shape and performance is clarified by the numerical analysis results and the suitable meridional plane shape at the connection between the front rotor and the rear rotor for the new contra-rotating small hydroturbine is considered.

2. Rotor Geometry and Design Parameter

A meridional plane image of this hydroturbine is shown in Fig.1. In order to achieve higher turbine head than the conventional contra-rotating small hydroturbine composed of axial flow rotors, the centrifugal rotor was adopted as the rear rotor for this contra-rotating small hydroturbine. Also in contra-rotating type, it is desirable that the output powers of the front and rear rotors are equal. Therefore, we proposed a hybrid rotor which was composed of an axial flow rotor and a mixed flow rotor as the front rotor, to make almost the same high design head of the rear rotor using the centrifugal type. The fluid flows axially into the hydroturbine, and that is bent in an inward direction of the radius by mixed flow part of the front rotor. The fluid is bent to axial direction again by the rear rotor, and flows out from the hydroturbine. By doing so, high turbine head can be expected because a centrifugal action can be used effectively. A guide vane was not set at the inlet of the front rotor because this hydroturbine was designed as compact as possible.

The design parameters of this hydroturbine are shown in Table 1. The design flow rate and the design turbine head are \(Q_d=5[l/s]\) and \(H_d=3.8[m]\) respectively. They were determined based on the output power, head and flow rate assumed in the pipe of agricultural water and the small-scale water-supply system. The design heads of front and rear rotor are \(H_{dF}=1.9[m]\) and \(H_{dR}=1.8[m]\) respectively, the loss head from the outlet of the front rotor to the inlet of the rear rotor was assumed 0.1[m]. Each design rotational speed of the front and rear rotors of the hydroturbine is the same as \(N_d=N_{dR}=1000[\text{min}^{-1}]\).

The rotor and primary dimensions of this hydroturbine are shown in Fig.2 and Table 2 respectively. The hydroturbine diameter is small under 100[mm]. Each primary dimension was determined by the design output power, head, flow rate and rotational speed. The rear rotor was designed so that the inlet angle matched downstream flow of the front rotor and swirling flow did not remain downstream of the rear rotor at the design flow rate. The blade angle of each rotor was designed to change in a linear function between the inlet angle and the outlet angle.

The specific speed in this hydroturbine and the general specific speed range in each hydroturbine type are shown Fig.3. A horizontal axis is the specific speed \(N_{SP}\). Each horizontal line shows the
general specific speed range in each hydroturbine type. The specific speeds of each part in this hydroturbine were relatively low within the specific speed range of the general hydroturbine.

In the contra-rotating rotor, the connection between the front rotor and the rear rotor is important, because the circumferential velocity occurred by the front rotor is recovered by the rear rotor. A conventional contra-rotating rotor was composed of two axial flow rotors, and the meridional plane shape was simple with no change in the flow path cross sectional area. However, this new contra-rotating rotor has a complicated meridional plane shape that has a bend and a change in cross sectional area, because using centrifugal action by radius difference. Therefore, in order to investigate the influence of the meridional plane shape, the casing shape is fixed this time and the outlet blade width of the front rotor and the inlet blade width of the rear rotor were changed by changing the hub shape of each rotor. When a design method of large hydroturbine was applied to this hydroturbine, the suitable blade widths at the outlet of the mixed flow part and the inlet of the centrifugal rotor were \( B_{\text{outF}} = 9.0 \) and \( B_{\text{inR}} = 5.0 \) respectively. In this investigation, assuming a case that the connection between the front rotor and the rear rotor is smoothly, we changed the blade widths as \( B_{\text{outF}} = B_{\text{inR}} = 5.0, 6.0, 7.5 \) and \( 9.0 \) (4 types). Then, the meridional velocities depend on the blade width. In the front rotor, the blade angles of the outlet were designed so that Euler's heads were constant regardless of the meridional velocity change by the blade width. Therefore, in the rear rotor, the blade angles of the inlet were designed to match the downstream flow of the front rotor, which changed by the blade angle of the front rotor.

Table 1 Design parameters of hydroturbine

|                           | Flow rate [m\(^3\)/s] | 0.005 |
|---------------------------|------------------------|-------|
|                           | Rotational speed [min\(^{-1}\)] | 1000  |
| **Front rotor**           |                        |       |
| Axial flow part           | Shaft power [W]         | 23    |
|                          | Turbine head [m]        | 0.8   |
|                          | Specific speed [min\(^{-1}\), kW, m] | 200   |
| Mixed flow part           | Shaft power [W]         | 34    |
|                          | Turbine head [m]        | 1.1   |
|                          | Specific speed [min\(^{-1}\), kW, m] | 164   |
| **Rear rotor**            |                        |       |
| Centrifugal part          | Shaft power [W]         | 53    |
|                          | Turbine head [m]        | 1.8   |
|                          | Specific speed [min\(^{-1}\), kW, m] | 110   |

Fig.1 Meridional plane image of hydroturbine
Table 2 Primary dimensions of hydroturbine rotor

|                      | Diameter [mm] | Blade width [mm] | Blade angle [deg] |
|----------------------|---------------|------------------|-------------------|
|                      | Inlet         | Outlet           | Inlet             | Outlet           | Inlet         | Outlet           | Inlet             | Outlet           |
| Front rotor          |               |                  |                   |                   |               |                  |                   |                   |
| (Hybrid type)        |               |                  |                   |                   |               |                  |                   |                   |
| Axial flow part      | Inlet 56      | Outlet 66        | Inlet 37.8        | Outlet 22.5      | Inlet 56      | Outlet 66        | Inlet 22.5        | Outlet 22.0      |
|                      | Outlet 76     |                  | Outlet 33.4       | Outlet 22.0      | Outlet 76     |                  | Outlet 29.8       | Outlet 21.2      |
| Mixed flow part      | Inlet 56      | Outlet 10        |                    |                   |               |                  |                   |                   |
|                      | Outlet 50     |                  |                    |                   |               |                  |                   |                   |
| Rear rotor           |               |                  |                   |                   |               |                  |                   |                   |
| (Centrifugal type)   |               |                  |                   |                   |               |                  |                   |                   |
| Centrifugal part     | Inlet 44      | Outlet 15        | Inlet 22.5        | Outlet 22.0      | Inlet 5.0, 6.0, 7.5, 9.0 (4 types) | Outlet 11.5 |
|                      | Outlet 38     |                  | Outlet 26.5       | Outlet 21.2      | Outlet 11.5  |                  | Outlet 65.9       | Outlet 64.8      |

* : Change to match meridional velocity change by blade width change so that the Euler's head is constant.
** : Match downstream flow of front rotor, which changed by blade angle of front rotor.

Fig. 2: Overviews of hydroturbine rotor

Fig. 3: Specific speed of hydroturbine rotor
3. Numerical Analysis Method and Condition

In the numerical analysis, the commercial software ANSYS-CFX 2019 R2 was used under the condition of 3D steady flow condition. Fluid was assumed to be incompressible and isothermal water and the equation of the mass flow conservation and Reynolds Averaged Navier-Stokes equation were solved by the finite volume method. The Shear-Stress-Transport was used as the turbulence model and the Automatic was utilized as the near wall treatment. The numerical grids of the whole numerical regions used for the numerical analysis are shown in Fig.4. A numerical model was designed assuming the test section of the experimental apparatus in future works. The numerical region has an inlet pipe, outlet pipe and hydroturbine which is composed of the front and rear rotor. The Stage (Mixing-Plane) was used to connect the interfaces in these regions. The lengths of each inlet and outlet pipe region are $5D_F$ and $5D_R$ respectively. A tip clearance is $1[\text{mm}]$. Each front and rear rotor’s shaft was modeled in the numerical analysis and rotated in the front and rear rotor’s rotational direction respectively. The constant velocity perpendicular to the inlet boundary and the constant pressure having only axial velocity were given as the boundary condition at the inlet and outlet respectively. The numerical grid points of each region were about 0.7 million for the inlet and outlet pipe region, about 2.0 million to 3.4 million for the front rotor region and about 2.1 million to 2.5 million for the rear rotor region respectively. The numerical grid points of each rotor were different by the blade width. The calculations were performed under these conditions, and the convergence of the calculations was good.

![Numerical grids](image)

4. Results and Discussions

4.1. Performance curves

Performance curves of this hydroturbine under the different meridional plane shape obtained by the numerical analysis are shown in Fig.5. In this investigation, the rotational speed of each rotor are constant as $N_F=N_R=1000[\text{min}^{-1}]$. A horizontal axis is the flow rate. The vertical axes are the total pressure efficiency, the turbine head $H$ and shaft power $P$ respectively. The total pressure efficiency of the hydroturbine is calculated from the ratio of the total shaft power $P_F+P_R$ to the load of the hydroturbine. The load of the hydroturbine is obtained by the multiplication of the flow rate $Q$ and the mass flow averaged total pressure change of the hydroturbine between the inlet and the outlet. The mass flow averaged total pressure change is obtained by the multiplication of the fluid density $\rho$, gravitational acceleration $g$ and the turbine head $H$. The turbine head $H$ of the hydroturbine was obtained from the mass flow averaged total pressure change between $1.0D_F$ upstream from the front rotor leading edge and $1.0D_R$ downstream from the rear rotor trailing edge. Therefore, the total pressure efficiency $\eta$ of the hydroturbine is expressed in the following equation.

$$\eta = \frac{P_F+P_R}{\rho g Q H}$$
In the hydroturbine shown in Fig.5, when the blade width was small, the total pressure efficiency was lower at less than 1.0\(Q_d\). On the other hand, regardless of the blade width, the total pressure efficiency was almost the same at greater than 1.0\(Q_d\). The maximum total pressure efficiency was obtained at 0.8\(Q_d\), which was lower than the design flow rate.

Next, total pressure efficiency curves of the front rotor and the rear rotor obtained by the numerical analysis are shown in Fig.6 and Fig.7 respectively. Each axis is the same as Fig.5(a). The turbine head \(H_F\) of the front rotor and the turbine head \(H_R\) of the rear rotor were obtained from the mass flow averaged total pressure change between 1[mm] upstream from the leading edge and 1[mm] downstream from the trailing edge of each rotor. As with the previously described calculation method of the hydroturbine, the total pressure efficiency of the front rotor \(\eta_F\) and the total pressure efficiency of the rear rotor \(\eta_R\) are expressed in the following equation respectively.

\[
\eta_F = \frac{P_F}{\rho g Q F H_F}, \quad \eta_R = \frac{P_R}{\rho g Q R H_R}
\]

In the front rotor shown in Fig.6, when the blade width was large, the total pressure efficiency was higher at greater than 0.8\(Q_d\). In the rear rotor shown in Fig.7, when the blade width was small, the total pressure efficiency was greatly lower at less than 1.0\(Q_d\). The relationship between the performance and internal flow in each blade width was investigated by the numerical analysis results.

4.2. Internal flow of rear rotor

The relative velocity vectors around the blade of the rear rotor at 0.6\(Q_d\) where the most different performance was shown are shown in Fig.8. The position is \(B/B_c=0.7\). \(B\) is arbitrary blade width position from the hub of rotor and \(B_c\) is a width between the hub of rotor and the casing. The color of vectors shows the relative velocity, and the length of vectors is constant regardless of the relative
velocity. The rotational direction of the rear rotor is the upper side of the paper. The left side of the paper is the inlet side of the rear rotor, and the right side of the paper is the outlet side of the rear rotor.

In $B_{\text{inR}}=9.0\,\text{[mm]}$, the fluid was flowing out along the blades. On the other hand, in $B_{\text{inR}}=5.0\,\text{[mm]}$, the flow separation occurred at the pressure surface side. Therefore, it is considered that one of the reasons the total pressure efficiency decreased is the flow separation.

The static pressure distributions around the blade of the rear rotor at $0.6Q_d$ are shown in Fig.9. The position is $B/B_c=0.7$. The rotational direction and the flow direction are the same as Fig.8. In $B_{\text{inR}}=5.0\,\text{[mm]}$, the adverse pressure gradient existed at the pressure surface side. By contrast, in $B_{\text{inR}}=9.0\,\text{[mm]}$, the adverse pressure gradient did not exist. Therefore, it is considered that the flow separation occurred by the adverse pressure gradient.

The static pressure distributions on the meridional plane of the rear rotor at $0.6Q_d$ are shown in Fig.10. A line on the meridional plane shows $B/B_c=0.7$. The static pressure decreased greatly around the bend on the casing side and the area extended into the rear rotor. When the blade width $B$ is narrow, the percentage of the low static pressure region to the flow path area was relatively larger. Then it was strongly influenced by the low static pressure on the casing side, and the adverse pressure gradient existed on $B/B_c=0.7$ line in $B_{\text{inR}}=5.0\,\text{[mm]}$. Therefore, it is considered that the total pressure efficiency was low because the flow separation occurred when the blade width $B_{\text{inR}}$ was narrow.
5. Concluding remarks

The influence of the meridional plane shape on the performance of the new contra-rotating small hydroturbine composed of the hybrid rotor and centrifugal rotor were investigated by the numerical analysis results. As a result, the following conclusions were obtained.

1. When the blade width was small, the total pressure efficiency of this hydroturbine was lower at less than 1.0$Q_d$. Because the total pressure efficiency of the rear rotor was greatly lower.
2. It was strongly influenced by the low static pressure on the casing side of the rear rotor when $B_{inR}=5.0$[mm], and the adverse pressure gradient existed on the tip side. Then the flow separation occurred at the pressure surface side at 0.6$Q_d$. Therefore, it is considered that the total pressure efficiency of the rear rotor was greatly lower at less than 1.0$Q_d$.

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Fig.10 Static pressure distributions on meridional plane of rear rotor ($Q=0.6Q_d$)