Proposal of a new hydraulic turbine capable of high efficiency operation over a wide range of flow rate and effective head

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Abstract. Hydroturbines should keep a high efficiency in the widest range of operation conditions. However, conventional turbine has been designed to reduce swirl at a runner outlet at its design point to maximize efficiency, and is unsuitable for partial load operation due to the growth of swirl at the outlet. The swirl of the runner outflow causes loss and unstable flow at the downstream draft tube. To prevent those problems, we developed a new type of hydraulic turbine capable of a wide range of operation conditions, specifically flow rate and effective head, with a high efficiency. The designed new turbine is equipped with a parallel diffuser and a volute diffuser downstream of the runner, instead of a draft tube. The outflow swirl is arranged to have forward turning constantly against runner rotation. It is possible to decrease downstream loss, and to mitigate the vortex core behaviour this way.

1. Introduction

In recent years, hydropower generation is playing key a role among renewable energy generations in electricity supply. Those renewables are solar power generation and wind power generation, whose output often fluctuates according to the weather. Now that the sites for large-scale hydroelectric power plants have almost all been developed in Japan, attention is being paid to small hydropower generation [1].

Due to economic efficiency and the small output of small hydropower plants, they should be usable for a wide range of operation conditions. In this research, the flow rate and the effective head are targeted as variable conditions for the operation. A conventional turbine, nevertheless, is designed to minimize the swirl of the runner outflow at its design point to provide the highest efficiency, and is unsuitable for operation at off-design point. Especially at partial loads, the swirling outflow of the runner induces a vortex core in the draft tube (DT). The vortex core influences losses and unstable flow inside the DT, and this consequently leads to problems such as efficiency reduction, degradation of reliability due to noise and oscillation, and unreliable power output due to water pressure pulsation [2][3]. Hence, when the discharge is insufficient, the turbine operation would be ceased. To continue operation with this small flow rate, it is indispensable to reduce the generation of the vortex core at the DT [4]. The research was performed using Computational Fluid Dynamics (CFD) analyses and experiments on flow instability in the DT [5]. At this time, we introduced a volute diffuser (VD) instead of a DT, attempting to prevent those problems from developing. The volute diffuser enables the runner outflow to flow in...
the radial direction and pressure to recover at the region. A larger amount of angular momentum is given at the runner inlet and the outflow swirl is arranged to have constant forward turning against the runner rotation. The turbine equipped with a VD and a new type of runner aims for high efficiency under a wide range of operation points.

2. Plan for installing the new turbine
We attached a VD at downstream side of the runner, instead of the commonly used DT. Figure 1 and figure 2 show the runner and the diffuser in the conventional and the new turbine, respectively.

In the conventional turbine, the angular momentum is enforced by guide vanes (GV) and is retrieved as an energy at runner vanes (RV). As mentioned above, at a design point, there is no swirl at the runner outflow, since it provides the highest efficiency. The outflow changes from clockwise to counter clockwise responding to operation points. In the new turbine, the outflow swirl is set to be constant. The RV outflow into VD is swirling all the time.

The new turbine we are developing is to be installed in Yoroibata hydroelectric power plant (located in Akita Pref., Japan).

The new runner has been planned to have a good response to variable flow rate and effective head. There are two established hydraulic turbines in the Yoroibata power plant. Those are suspended in 1/3 of the year, whereas the maintenance discharge for downstream is still necessary. The third power plant equipped with the new runner is planned to have operation in the gap term, and to supply electricity with the plant. The target operating range is 15% to 110% for the flow rate and 50% to 150% for head. The turbine is expected to keep approximately 40% to 90% efficiency in those intervals.

Figure 1. The runner and the DT of a conventional turbine
Figure 2. The runner and the VD of a new turbine (Planned at the plant)

3. Design

3.1. Turbine specification
Table 1 shows the turbine specification of new design. SV in table 1 indicates Stay Vanes located upstream of the GV. $N_s$ in the table is defined as

$$N_s = N P^{0.5} H e^{-1.25} \text{ (m-kW)}$$

(1)

The maximum flow rate $Q$ is set to 0.834 (m$^3$/s), at Guide Vane Opening (GVO) of 110%. Additionally, we defined 90% GVO as the design point.
Table 1. Specifications of the new turbine

| Items                        | Values | Units |
|------------------------------|--------|-------|
| Specific speed $N_s$         | 142    | m-kW  |
| Maximum effective head $H_{e_{max}}$ | 54.85  | m     |
| Rated net head $H_e$         | 47.4   | m     |
| Maximum flow rate $Q$        | 0.834  | m$^3$/s |
| Design point rotation speed $N$ | 1000   | rpm   |
| Power $P$                    | 313    | kW    |
| Swirl direction              | Forward swirl | -    |
| Casing type                  | Spiral  | -    |
| Diffuser type                | Parallel, Volute | -    |
| Number of SV $Z_s$           | 18     | -     |
| Number of GV $Z_g$           | 18     | -     |
| Number of RV $Z_r$           | 16     | -     |
| Inlet diameter $D_1$         | 0.4366 | m     |
| Outlet diameter $D_2$        | 0.4089 | m     |
| The height of vane $B_g$     | 0.0715 | m     |

3.2. Swirl direction

The new design concept keeps the runner outflow in constant swirl direction. In this paper, we defined “Forward Swirl (FS)” as a swirl of which direction is equivalent to that of the runner rotation, and “Reverse Swirl (RS)” as the contrary.

When adopting the normal design to a Francis turbine, the inflow and outflow directions are set to be radial and axial, respectively. There is no swirl at the design point, which means the outflow velocity should be minimized. On contrary, broad fluctuation of loss occurs at the DT, since the flow involves a wide range of absolute outflow angle $\alpha_2$ depending on the operating conditions. The absolute angle $\alpha$ is expressed by using meridional velocity $V_m$ (m/s), peripheral velocity $U$ (m/s), and peripheral component of relative velocity $W_U$ (m/s) as below.

$$\alpha = \frac{V_m}{U-W_U} \text{ (degree)}$$  \hspace{1cm} (2)

In the FS design, two flow passages are possible: radial outflow with radial inflow, or with axial inflow. They both have narrow range of $\alpha_2$, and accordingly, the loss range at downstream region of the runner will become smaller. Such losses are, for instance, disk friction loss due to the large diameter of the runner, and friction loss due to its high-speed flow.

In the RS design, the flow passage is radial outflow with axial inflow. Similar to the FS design, it has a narrow range of loss transition at the region. The range of $\alpha_2$ is also narrow when changing the operation point. Consequently, the loss range upstream the RV will decrease. The smaller velocity at upstream is helpful to decrease loss in this region. In contrast, the loss at the downstream region of the RV becomes greater due to its high-speed. A wide range of absolute outflow angle induced by swirl’s reversal should also be taken into consideration.

Figure 3 to figure 5 each represents the velocity triangle and the meridional plane of every design. Factors $U$, $V$, and $W$ (m/s) each represents peripheral speed, absolute velocity, and relative velocity. It should be noted that the values on the figure are only advisory.
To select the runner design among the above three, we carried out an elementary parametric study based on one dimensional model calculations. The model focuses on a single streamline which begins at the GV and passes through the VD. The model inputs are the passage configuration, $N$ and $He$. This method regards a real passage as represented-simple flow path, and calculate theoretical head and loss with assumed flow rate $Q$.

From the study, we concluded that FS is the most suitable design. The loss in FS design was comparatively lower than that of normal design, whereas the loss in RS design was unstable depending on the slip factor $k$. It was ascertained that FS design has the smallest range of losses and $\alpha_2$ at different operation points.
4. Computational models and setup

4.1. Turbine design
Based on those turbine design and specifications from the previous chapter, we prepared the models of both conventional and new turbines. Figure 6 and figure 7 shows the overview of the runners of each turbine. The conventional Francis runner was designed to fulfil the specification on table 1, except for the factors peculiar to that of the new runner.

The new runner has greater shaft diameter at the crown side, and its meridional blade length is significantly longer than that of the conventional runner. This is to convert the flow into radial direction at the crown side.

The new design constantly remains peripheral component of outflow absolute velocity $V_{u2}$ (m/s), whereas it is arranged to be zero at design point in conventional design. In terms of hydraulic turbine, the theoretical head $H_{th}$ is expressed as

$$H_{th} = \frac{U_1V_{u1} - U_2V_{u2}}{g} \text{ (m)}$$  \hspace{1cm} (3)

where $g$ (m/s$^2$) is gravity acceleration. To guarantee a sufficient $H_{th}$ for the new turbine, it is essential to supply a high enough peripheral velocity with runner inflow ($U_1$ (m) and $V_{u1}$ (m)). Therefore, the turbine inlet diameter increases to raise the high peripheral velocity.

![Figure 6. The overview of the runner of the conventional turbine](image)

![Figure 7. The overview of the runner of the new turbine](image)

4.2. Volute diffuser design
The swirling outflow from the RV flows into parallel diffuser (PD) and VD. For analyses, we prepared two types of VD: Folded type (FD) diffuser and speed-ring type (SR) diffuser. They both are constant speed diffusers. Figure 8 and figure 9 show the section view of each VD.

![Figure 8. The speed-ring type VD](image)

![Figure 9. The folded type VD](image)

4.3. Computational setup
The computational analyses had been performed by using ANSYS-CFX17.2 commercial code. We carried out steady state computation by using the Reynolds-Averaged Navier-Stokes equation (RANS) method with a turbulence model, the Shear Stress Transport $k$-$\omega$ (SST $k$-$\omega$) [6][7], which performs well for hydraulic turbine [8]. A regular grid was prepared for cascades and the PD. An unstructured mesh was also prepared for the VD. The total number of nodes and elements are approximately 2.4 million, 6 million, respectively. Analyses run in periodic boundary with boundary conditions of total pressure as
inlet and static pressure as outlet. We carried out analyses in distinct GV opening: 110% (as maximum), 90% (as design point), 60%, 45% and 30%.

Figure 10 and figure 11 each shows computational model and mesh configuration in the analytical region.

![Computational domains](image1)
![Mesh configuration in cascades](image2)

**Figure 10.** Computational domains  
**Figure 11.** Mesh configuration in cascades

5. Loss examination in the new turbine

5.1. Loss distribution

Figure 12 shows the calculation result of flow rate transition against every operation point.

![Flow rate transition](image3)

**Figure 12.** GV opening and flow rate characteristic

The loss distribution of each diffuser design is shown in figure 13 and figure 14. The loss coefficient $\theta_{\text{loss}}$ is defined as the percentage of the loss head in the component divided by the effective head.

Since all components except for the VD are common for those two models, the loss in those components are expected to be similar. The loss at SR diffuser surpasses that of the FD diffuser at every GVO. It is noteworthy that the PD loss increases at partial load and at over load.
5.2. Loss in the runner

The runner was designed to have the smallest loss at design point, and the greater loss would arise in partial load conditions. Figure 15 shows the distribution of the rothalpy loss near the leading edge, the mid chord and near the trailing edge.
The rathalpy loss is defined as below.

\[ \zeta_R = \frac{l_1-l}{g \mu_e} \]  \hspace{1cm} (4)

\( I \) is defined as

\[ I = \frac{P_s}{\rho} + 0.5W^2 - 0.5U^2 \]  \hspace{1cm} (5)

Where \( P_s \) is the static pressure (Pa), and \( \rho \) is density (kg/m³). Subscript 1 indicates the inlet of the component. The rathalpy is commonly used to evaluate the total enthalpy in turbo machine without including the enthalpy rise enforced by rotation. The SS and the PS in the figure 15 are the suction side and the pressure side on the RV.

The rathalpy loss converge around the suction surface on the shroud side. As we move onto partial load operation, the loss region expands and moves toward the pressure surface on shroud side.

5.3. Loss in the parallel diffuser

The loss in the bend part of the PD increases in size under partial load conditions. This stems from the bias of the meridional velocity \( V_m \) (m/s) distribution as shown in figure 16. The distribution around the design point is roughly uniform. When the runner is in partial load conditions, the swirl in RV outflow inclines toward the shroud side. Therefore, the flow separates from the crown side and induces a greater loss at the PD. When the runner is operated in over load conditions, the high \( V_m \) region extends to both the crown and hub side and consequently induces vast friction loss.

5.4. Loss comparison with two volutes

The distributions of total pressure loss, \( C_{ps} \), and \( C_v \) in each diffuser’s outlet are shown in Figure 17 and figure 18. \( C_{ps} \) and \( C_v \) represents pressure coefficient (static pressure) and flow coefficient, respectively. The total pressure loss \( \zeta, C_{ps} \) and \( C_v \) are defined as below,

\[ \zeta = \frac{P_1}{\rho g H_e} \]  \hspace{1cm} (-)  \hspace{1cm} (6)

\[ C_{ps} = \frac{P_{S1} - P_S}{\rho g H_e} \]  \hspace{1cm} (-)  \hspace{1cm} (7)

\[ C_v = \frac{V}{\sqrt{2g H_e}} \]  \hspace{1cm} (-)  \hspace{1cm} (8)

In the FD diffuser, the results suggest that there is no remarkable disposition in radial direction (figure 17). At 110% GVO, there is vector in counter clock wise direction which contradicts results in the other openings. At this GVO, the distribution of \( V_m \) leans toward the crown side, whereas the others lean toward the shroud side (figure 16). Therefore, at this GVO, the leaned inflow and the circulating flow inside the VD collide, which bring about the incline of loss.

In SR diffuser, as is the case of FD diffuser, the distribution of \( V_m \) converges around the outlet of the PD (figure 16). As we move onto partial load operation, the vortex leans toward the inner side, and
the flow collision occurs (figure 18). From those results, it is concluded that the FD diffuser is superior to the SR diffuser from point of view of loss distribution.

![GVO Total pressure loss \( \zeta \) (\%)](image)

![Cps (\%)](image)

![Cv (\%)](image)

**Figure 17.** Internal flow of volute with FD diffuser (at volute outlet section)
Figure 18. Internal flow of volute with SR diffuser (at volute outlet section)
6. Loss comparison with conventional turbine

6.1. Outflow absolute angle $\alpha$

Figure 19 and figure 20 show the $\alpha_2$ transition against the operation points in each turbine. In general, $\alpha_2$ becomes larger than 90 degrees at large GVO. The figure 19 corresponds to this, 110% GVO has obtuse $\alpha_2$ through the runner. In contrast, $\alpha_2$ of the new runner has been kept acute at all GVO, even in over load operation. The result shows that FS design has mitigated the transition of $\alpha_2$, and has enabled the turbine to operate within constant $\alpha_2$ distribution at 90% GVO. It was ascertained that the new turbine equipped with FS design runner is practical for wide range of GVO.

![Figure 19. Distribution of $\alpha_2$ (the conventional runner)](image1)

![Figure 20. Distribution of $\alpha_2$ (the new runner)](image2)

6.2. Loss distribution

Figure 21 and figure 22 show the loss transition with every operation point. In the figure, the coefficient $Q_{11}$ and loss value are normalized by the values at the design point values. Subscript $d$ indicates the value at the design point. The coefficient of flow rate $Q_{11}$ is defined as below.

$$Q_{11} = \frac{q}{D^2H^{0.5}} \quad (-)$$

The new turbine has smaller loss at the diffusers at partial load operation compared with the conventional turbine. The result shows that replacement of the DT by the VD is practical to decrease the loss and is suitable for operation in the targeted range.

The loss at the GV is negatively proportional with $Q_{11}$ in both turbines. It indicates that impact loss, rather than friction loss, is dominant at GV in partial load condition. Impact loss derives from incongruence of the inflow against the leading edge of the GV [8]. This incongruence also induces collision loss with adjacent GV flow passages. Meanwhile, the loss at the casing and SV decreases proportionally with $Q_{11}$. This indicates that the main factor of loss in the casing and the GV is to be the friction loss.

The VD and the PD have restricted the development of larger loss at partial load conditions while the DT loss increases due to strong swirl.
6.3. Performance of the new turbine
Figure 23 shows the computational results of the new turbine. Every hydraulic efficiency is normalized by that of the highest in the conventional design. A marked improvement of the efficiency in partial load operations can be observed. At the design point, where the dimensionless $Q_{11}$ is 1, the hydraulic efficiency of the new design is lower than that of the conventional turbine. This is caused by the remnant of the swirl in forward direction at the design point.

Incidentally, this new turbine is yet to be optimized, thus the result should be evaluated as incomplete. Further improvement of runner efficiency is expected. Nevertheless, the result is enough to ascertain the usability of the new turbine. To develop the turbine, further analyses for instability flow is required.

Figure 23. Loss distribution in the model (with SR volute)
7. Conclusion
(1) The new turbine equipped with volute diffuser keeps high efficiency at partial load conditions compared with the conventional turbine.
(2) The new runner with radial outflow and the forward swirl design has a smaller range of absolute outflow angle $\alpha_2$ at the runner outlet. The runner has larger meridional blade length and diameter.
(3) The folded type volute diffuser is superior to the speed-ring type diffuser from the point of view of loss distribution.
(4) At partial load operation, greater loss has occurred in the parallel diffuser since the flow separates from crown side. At over load operation, the loss at volute diffuser increases due to impact loss between the parallel diffuser outflow and circulating flow inside volute diffuser. Those flows were induced by the distribution of meridional velocity $V_m$ in parallel diffuser and swirling flow.
(5) Further optimization and flow instability computation are required to reveal the flow behavior.

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