Characteristics and internal flow of a low specific speed pump used as a turbine

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Abstract. PAT (Pump as Turbine) has gradually been a popular solution for the small-scale hydropower generation. Because of the rich assortment and the low initial investment. However, PAT is designed as a pump but not a turbine. Therefore, a lot of research was conducted about the performance prediction and the loss distribution of pump and turbine mode. In this paper, we focused on the internal flow and loss mechanisms of the best efficient point of both modes. The characteristic and internal flow were investigated by experimental and computational approaches. The reasons of efficiency reduction in turbine mode and loss mechanisms were clarified by the computational results. The separation flow region occurs on the suction side or pressure side in non-BEP, and the swirling flow counter-rotates the PAT occurring at the outlet in BEP might be the reasons of the efficiency reduction. Moreover, because of the camber line shape of PAT, the separation flow occurs near the inlet could not reattach soon which induces the loss region. Furthermore, the secondary flow near the blade surface and the flow from the pressure side to suction side interfere with each other which induced the loss region.

1. Introduction

Hydropower has been used as a very old renewable energy for human beings for electric power generation. It is now regarded as the main power generation solutions in many countries [1]. Large-scale hydropower plants are installed for obtaining sufficient energy. However, large hydropower plants require huge initial investment and the environmental destruction due to the submergence upstream from the dam is concerned [1]. Therefore, small-scale hydropower generation has attracted attention and is expected as a solution to solve energy problems in the rural and distant areas [2].

Small-scale hydropower generation mainly uses the river and agricultural water. Which means the installation place is narrow and power output is small. Therefore, the development of low-cost and compact hydro turbines is required. However, the implement is difficult because new development consumes much time and money. Hence, it has been proposed to apply existing pumps to work as a turbine which is called PAT (Pump as Turbine) [3]. Less initial investment and the availability of preparing various sizes and specifications productions according to demands are the advantages of PAT. However, the operating characteristics and the internal flow in the turbine mode before installation is necessary to be clarified, since the PAT is designed as a pump. Thus, a large number of research on performance prediction and loss distribution of pump and turbine mode has been
conducted by many researchers. Stepanoff, Derakhshan, Yang, Williams et al \[1,4,5,6,7\] developed the theoretical method to predict the parameters in the BEP (Best Efficient Point) by the head and discharge correction factors (\(h\) and \(q\)). Chapallaz, Rawal and Kshirsagar, Yang et al \[6,8,9\] have investigated the loss distributions of PAT by theoretical and computational methods. The paper which investigated the relationship between the loss mechanisms in pump and turbine modes is not sufficient yet. Therefore, in this paper, the internal flow and loss mechanisms of the pump and turbine modes are investigated and clarified by computations.

2. Experimental Setup
The overview of the test apparatus used in the experiments of pump and turbine mode are shown in figure 1. And the model PAT made of acrylic resin is shown in figure 2. It is designed as a pump which is a scale model for one stage of a multistage boiler feed pump \[10\]. The specific information is shown in table 1.

![Figure 1. Overview of the test apparatus](image1)

![Figure 2. Acrylic model PAT](image2)
Table 1. Detailed parameters of PAT

| Items                      | Values  | Units          |
|----------------------------|---------|----------------|
| Specific speed \( N_s \)   | 218.2   | rpm, m³/min, m |
| Number of diffuser vanes \( Z_d \) | 9       | -             |
| Number of impeller vanes \( Z_i \) | 7       | -             |
| \( D_1 \)                 | 0.09234 | m             |
| \( D_2 \)                 | 0.1500  | m             |
| \( B_2 \)                 | 0.1432  | m             |
| Impeller outlet vane angle \( \beta_2 \) | 22.50   | degree        |
| Diffuser inlet vane angle \( \alpha_3 \) | 11.50   | degree        |

In the table 1, the specific speed was defined as

\[
N_s = \frac{NQ^{1/2}}{H^{3/4}} \quad \text{[rpm, m³/min, m]} \tag{1}
\]

where \( N, Q, H \) are the rotational speed, discharge and head.

In the experiment, air was used as the working fluid and the Mach number is 0.07. The wall static pressure at the upstream and downstream of the PAT was measured by dynamic pressure sensors and the discharge was measured by a hot wire probe. The rotational speed was fixed to 3000 min⁻¹ which was measured with a tachometer. Since model PAT is integrated with the motor, torque could only be calculated by measuring the load current flowing through the motor using a clamp meter. All the data was collected by the data collection device (DAQ) and organized by LabVIEW.

When carrying out the performance test, the operating point was determined by using the discharge coefficient \( \phi \) and head coefficient \( \psi \) which are defined as

\[
\phi = \frac{Q}{\pi D_2 b_2 U_2} \tag{2}
\]

\[
\psi = \frac{P_{t2} - P_{t1}}{\rho U_2^2} \tag{3}
\]

where \( U_2, P_{t2}, P_{t1} \) are the circumferential speed of impeller outlet, total pressure of the downstream of pump and total pressure of the upstream of pump. The design point of pump mode is \( \phi \) equals 0.077. In the pump mode, a valve was used at the downstream of the PAT to adjust the discharge. In the turbine mode, the discharge was adjusted by a blower. And a surge tank was set between the blower and PAT to alleviate the discharge and rectify the flow upstream.

3. Computational Framework

The computation was performed by a commercial code ANSYS CFX 17.2 which solves the Navier-Stokes equation using the finite volume method. The figure 3 shows the analysis domains: inlet pipe, impeller, diffuser vane, motor, outlet pipe. The impeller and diffuser vane were analyzed as a periodic boundary using one flow path to shorten the analysis time. All the meshes used in the computation were hexahedral grids created by Turbo Grid and ICEM CFD. The detailed mesh lines in cascades is shown in figure 4. The mesh information of each domain is compiled in table 2.

The computation was a steady-state analysis by Reynolds-Averaged Navier-Stokes equation (RANS) with the SST \( k-\omega \) (Shear Stress Transport) turbulence model to investigate the steady-state
performance and internal flow of PAT. The SST k-ω turbulence model is possible to evaluate the separation flow near the blade surface [11,12,13], especially the flow near the PAT inlet in turbine mode.

The mixing plane model was used for the interfaces between the stationary parts and the rotating parts. Moreover, the General Grid Interface (GGI) [14] is applied on all interfaces that the topology of meshes at both sides of interfaces does not have to match with each other. The inlet averaged total pressure 0 Pa and the mass flow value was set as the boundary conditions of the pump mode. In the turbine mode, the mass flow value was set in the inlet and the outlet averaged static pressure was 0 Pa. The rotational speed of PAT was set to 3000 min⁻¹.

![Figure 3. Analysis domains](image)

(red: impeller, blue: diffuser vane, green: motor, grey: inlet/outlet pipe)

| Flow domains   | No. of elements |
|----------------|-----------------|
| Inlet-pipe     | 5289822         |
| Impeller       | 588700          |
| Diffuser vane  | 607100          |
| Motor          | 1777524         |
| Outlet-pipe    | 2104109         |
| Total          | 10367255        |

| Table 2. Detailed mesh information |

![Figure 4. Mesh configuration in cascades](image)

4. Characteristic of PAT

Figure 5 shows the head characteristics based on experimental and computational results of PAT, and figure 6 shows efficiency characteristics. Figures compare the results of experiments and computations in pump mode and turbine mode, respectively.

The approximate curves of head characteristics show that the experimental and computational results qualitatively agree with each other. In the pump mode, the results show high consistency on the large flow side compared to the design point that \( \phi \) equals 0.077. However, the computational results are smaller than experimental ones and the deviation is large on the low flow side. In the turbine mode, the experimental values are generally larger than the computational ones. The results do not show high consistency as the pump mode, but large deviations do not occur.
The approximate curves of efficiency characteristics show that the experimental and computational results qualitatively agree with the trend of change. The mechanical efficiency $\eta_m$, leakage efficiency $\eta_l$ and disk friction efficiency, which are included in the experimental results, are predicted one-dimensionally by the formulae from Kurokawa [15]. Hence, figure 6 refers to a comparison of hydraulic efficiency. In addition, all efficiency values are normalized by the maximum efficiency of pump mode $\eta_{imax}$. The experimental and computational results show quantitatively large deviations in both modes. Also, the experimental results are unstable, since the values do not overlap well with the approximate curve. The reason of the instability of experimental results might be the measurement method of torque. In this experiment, torque was calculated from the lead current value measured from the clamp meter by using the characteristics of the motor. Therefore, the test facility is necessary to be improved to obtain the data with better accuracy.

Besides, the efficiency decreases by approximately 30% at each operating point when pump mode changes to the turbine mode. The reason will be explained by the computational results below.

Figure 5. Head curves of PAT
5. Internal Flow and Loss Mechanisms of PAT

5.1. Flow distributions at the upstream and downstream of PAT

The distributions of upstream and downstream flow of PAT are investigated to clarify the reason of the efficiency reduction in turbine mode compared with pump mode. The examined operating points are $\phi$ equals 0.06 (part-load), 0.12 (BEP), 0.18 (over-load). Figure 7 shows the inlet and outlet velocity triangles in the turbine mode for defining the velocity and flow angle. Figure 8, 9 and 10 show flow distributions at the inlet and outlet when the $\phi$ equals 0.06, 0.12, 0.18, respectively. All the values of velocity in figures are normalized by the equation below.

$$V^* = \frac{V}{(2gH)^{0.5}}$$ 

Firstly, the inlet flow angle $\beta_1$ is smaller than the metal angle when $\phi$ equals 0.06. It could be inferred that the flow collides with the suction side of the blade, and a separation flow might occur on the pressure side. The metal angle means the blade angle of PAT. Moreover, the swirling flow corotates the PAT occurring at the outlet.

Besides, the inlet flow angle $\beta_1$ is also smaller than the metal angle when $\phi$ equals 0.12. It could be inferred that the flow impinging on the suction side of the blade which induces a separation flow on the pressure side. Moreover, the swirling flow counter-rotates the PAT occurring at the outlet. This
operating point is the BEP, which corresponds to the design point of a normal turbine. The outlet flow of the design point is assumed to be with no swirl in a normal turbine. However, the swirling flow counter-rotates the PAT is accompanied on the BEP of PAT. Therefore, the cavitation surge should be worried when cavitation occurs during the operation.

Finally, the inlet flow angle $\beta_1$ is larger than the metal angle when $\phi$ equals 0.18. It could be thought that the flow collides with the pressure side of the blade, and a region of the separation flow might be formed on the suction side. However, since the separation flow is induced on the suction side, the separated flow would reattach to the suction surface due to the pressure field. And the separation bubble might be formed. In addition, the swirling flow counter-rotates the PAT occurring at the outlet.

All in all, the reasons of the efficiency reduction in turbine mode could be summarized as the points below. Firstly, when the PAT is in part-load points, the separation flow region occurs on the pressure side of the blade and the swirling flow corotates the PAT occurring at the outlet. Secondly, when the PAT is in BEP, the separation flow region also occurs on the pressure side of the blade and the swirling flow counterrotates the PAT occurring at the outlet. Finally, when the PAT is in over-load points, the separation flow region occurs on the suction side of the blade and the swirling flow counter-rotates the PAT occurring at the outlet. Therefore, the separation flow region occurs on the suction side or pressure side in non-BEP, and the swirling flow counter-rotates the PAT occurring at the outlet in BEP might be the reasons of the efficiency reduction in turbine mode.

![Figure 7. Velocity triangles in turbine mode](image-url)
Figure 8. Flow distribution when $\phi = 0.06$

Figure 9. Flow distribution when $\phi = 0.12$

Figure 10. Flow distribution when $\phi = 0.18$
5.2. Loss evaluation methods

In order to investigate the loss mechanism of PAT, the internal flow of the pump mode when \( \phi \) equals 0.077 will be compared with the flow of the turbine mode when \( \phi \) equals 0.12 (BEP of turbine mode). The rothalpy is used to define the loss coefficient in the flow field of PAT. The rothalpy \( I \) is defined as

\[
I = \frac{P_s}{\rho} + \frac{1}{2}(W^2 - U^2) \quad [m^2/s^2]
\]

which is conserved in isentropic and non-viscous flow [17]. Therefore, the reduction of rothalpy in the viscous flow means the loss induced by viscosity. The loss coefficient \( \zeta_i \) is defined as

\[
\zeta_i = \frac{I_{in} - I}{0.5U_{in}^2}
\]

Moreover, the secondary flow is used to evaluate the flow in the cross-section perpendicular to the main flow. The strength of it is defined by the equation below.

\[
SKE = \left( \frac{V_{sec}}{V_{ave}} \right)^2
\]

The loss might be induced by the secondary flow has already been known from some studies [13,18]. Besides, the inspection cross sections used to evaluate the flow are shown in figure 11.

![Figure 11. Inspection cross sections](image)

5.3. Loss distributions at BEP

Figure 12 shows the loss distributions with streamlines in the flow direction. In pump mode, the loss area concentrates on the suction side of impeller blades. In hub-span, the loss concentrates near the trailing edge of blades where the separation and backflow regions occur. The separation and backflow regions are induced because the flow velocity decreases. In mid-span, the separation and backflow region start from the mid-chord of blades and spreads toward the middle of the flow path which induces the loss area. And in shroud-span, the backflow area occurs from the leading edge (LE) of blades and gradually spreads to downstream. However, the high loss regions only remain near the section side.

In the turbine mode, the loss region also concentrates mainly on the suction side. However, in hub-span and mid-span, the loss region not only occurs near the suction side but also near the inlet of pressure side which induced by the separation flow. Normally, in turbine flow, the separation flow at the inlet of pressure side might reattach quickly due to the pressure field and forms a separation bubble. In that case, the loss region might not spread downstream. However, in the case of PAT, since it has a camber line with a reverse curvature from that of a normal turbine, the separation flow is difficult to reattach and the loss region expands to the downstream. Next, in shroud-span, the loss region induced from the vicinity of the inlet and spreads downstream. A high loss region occurs near the suction surface. In order to examine its identity, the flow in the cross-section perpendicular to the mainstream is investigated.
Figure 12. Loss distributions with streamlines in the flow direction

Figure 13 and 14 show the loss and the secondary flow distribution in the cross-section perpendicular to the mainstream of the pump mode and the turbine mode, respectively. Firstly, in the pump flow, the loss region concentrates on the suction side. The region expands generally from LE to trailing edge (TE). Moreover, according to the secondary flow vectors and their strength, the secondary flow near the suction side is strong. It could be confirmed that only the region where the separation and backflow area interfere with the strong secondary flow area coincides with the loss region. Therefore, the secondary flow in the middle of the flow path is strong near the TE, but the loss region is not induced.

Secondly, in the turbine flow, the loss region is concentrated near the shroud on the suction side. Near the LE and Mid-chord, no loss region could be confirmed near the pressure surface. Hence, the loss region spreading from the inlet of the pressure surface to the downstream might only due to the separation flow in the mainstream direction, not because of the secondary flow which is perpendicular to the mainstream. Next, near the shroud on the suction side of the Mid-chord, the secondary flow near the blade surface and that from the pressure side to the suction side strongly interfere with each other. That region coincides with the loss region. Moreover, near the TE, the secondary flow further strengthened, and the interference region with strong secondary flow also spreads to the hub side. Finally, a large loss region is induced near the whole of the suction side.
6. Conclusions

The characteristic and internal flow of PAT were investigated by experimental and computational methods. The efficiency decreases by approximately 30% at each operating point when pump mode changes to the turbine mode. The reasons of efficiency reduction in turbine mode and loss mechanisms were clarified by the computational results. The separation flow region occurs on the suction side or pressure side in non-BEP, and the swirling flow counter-rotates the PAT occurring at the outlet in BEP might be the reasons of the efficiency reduction in turbine mode. Besides, since PAT has a camber line with a reverse curvature from that of a normal turbine, the separation flow occurs near the inlet is difficult to reattach and the loss region expands to the downstream. This loss region might only due to the separation flow in the mainstream direction, not because of the secondary flow which is perpendicular to the mainstream. In addition, the secondary flow near the blade surface and that from the pressure side to the suction side strongly interfere with each other. That region coincides with the loss region.

Therefore, the effective approach to increase the efficiency might be suppressing the separation area near the inlet of PAT and generate the non-swirling flow on the BEP by processing the inlet geometry of PAT.

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References

[1] Jain S V and Patel R N 2014. Investigations on pump running in turbine mode: A review of the state-of-the-art. Renewable and Sustainable Energy Reviews, 30, 841-868.

[2] Derakhshan S and Nourbakhsh A 2008. Experimental study of characteristic curves of centrifugal pumps working as turbines in different specific speeds. Experimental thermal and fluid science, 32(3), 800-807.

[3] Ramos H and Borge A 1999. Pumps as turbines: an unconventional solution to energy production. Urban Water, 1(3), 261-263.

[4] Stepanoff A J 1948. Centrifugal and axial flow pumps.

[5] Derakhshan S and Nourbakhsh A 2008. Theoretical, numerical and experimental investigation of centrifugal pumps in reverse operation. Experimental Thermal and Fluid Science, 32(8), 1620-1627.

[6] Yang S S, Derakhshan S and Kong F Y 2012. Theoretical, numerical and experimental prediction of pump as turbine performance. Renewable Energy, 48, 507-513.

[7] Williams A A 1994. The turbine performance of centrifugal pumps: a comparison of prediction methods. Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy, 208(1), 59-66.

[8] Chapallaz J M, Eichenberger P and Fischer G 1992. Manual on pumps used as turbines. Vieweg.

[9] Rawal S and Kshirsagar J T 2007. Numerical simulation on a pump operating in a turbine mode. In Proceedings of the 23rd International Pump Users Symposium. Texas A&M University. Turbomachinery Laboratories.

[10] Sano T, Nakamura Y, Yoshida Y and Tsujimoto Y 2002. Alternate blade stall and rotating stall in a vaneed diffuser. JSME International Journal Series B Fluids and Thermal Engineering, 45(4), 810-819.

[11] Iino M, Tanaka K, Miyagawa K and Okubo T 2003. Numerical analysis of 3d internal flow with unstable phenomena in a centrifugal pump. In Proceedings of the 7th Asian International Conference on Fluid Machinery, Fukuoka, Japan, October (pp. 7-10).

[12] Iino M, Tanaka K, Miyagawa K and Okubo T 2003. Numerical simulation of hysteresis on head/discharge characteristics of a centrifugal pump. In Proceedings of the ASME FEDSM 4th ASME JSME Joint Fluids Engineering Conference, Honolulu, HI, USA (pp. 6-10).

[13] Liu Z H, Shinji M and Miyagawa K 2017. Internal flow and loss mechanisms of specific speed 160 m-kW shroudless hydro turbine. In the 1st symposium of Asian Working Group of IAHR Committee on Hydraulic Machinery and Systems (AWG-IAHR), Beijing, China.

[14] Beaudoin M and Jasak H 2008. Development of a generalized grid interface for turbomachinery simulations with OpenFOAM. In Open source CFD International conference (Vol. 2). Berlin.

[15] Kurokawa J. Simple formulae for volumetric efficiency and mechanical efficiency of turbomachinery, JSME Transaction, Part B 56(531) (1990) 3389–3396 (in Japanese).

[16] Miyagawa K 2002. Flow instability in an elbow draft tube for a Francis pump-turbine. In Proc. 21st IAHR Symposium,(2002).

[17] Lyman F A 1992. On the conservation of rothalpy in turbomachines. In ASME 1992 International Gas Turbine and Aeroengine Congress and Exposition (pp. V001T01A078-V001T01A078). American Society of Mechanical Engineers.

[18] Miyagawa K and Iwasaki Y 2000. Study on internal flow and new design technology for a Francis turbine runner. In 20th IAHR Symposium on Hydraulic Machinery and Systems, Charlotte.