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마이크로 수력 에너지원의 수평축 스크류 터빈 : 설계 타당성 연구

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Horizontal-Axis Screw Turbine as a Micro Hydropower Energy Source: A Design Feasibility Study

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Abstract >> Micro hydropower is a readily available renewable energy source that can be harvested utilizing hydrokinetic turbines from shallow water canals, irrigation and industrial channel flows, and run-off river stream flows. These sources generally have low head (<1 m) and low velocity which makes it difficult to harvest energy using conventional turbines. A horizontal-axis screw turbine was designed and numerically tested to extract power from such low-head water sources. The 3-bladed screw-type turbine is placed horizontally perpendicular to the incoming flow, partially submerged in a narrow water channel at no-head condition. The turbine hydraulic performances were studied using Computational Fluid Dynamics models. Turbine design parameters such as the shroud diameter, the hub-to-shroud ratios, and the submerged depths were obtained through a steady-state parametric study. The resulting turbine configuration was then tested by solving the unsteady multiphase free-surface equations mimicking an actual open channel flow scenario. The turbine performance in the shallow channel were studied for various Tip Speed Ratios (TSR). The highest power coefficient was obtained at a TSR of 0.3. The turbine was then scaled-up to test its performance on a real site condition at a head of 0.3 m. The highest power coefficient obtained was 0.18. Several losses were observed in the 3-bladed turbine design and to minimize losses, the number of blades were increased to five. The power coefficient improved by 236% for a 5-bladed screw turbine. The fluid losses were minimized by increasing the blade surface area submerged in water. The turbine performance was increased by 74.4% after dipping the turbine to a bottom wall clearance of 30 cm from 60 cm. The final output of the novel horizontal-axis screw turbine showed a 2.83 kW power output at a power coefficient of 0.63. The turbine is expected to produce 18,744 kWh/year of electricity. The design feasibility test of the turbine showed promising results to harvest energy from small hydropower sources.

Key words : Screw turbine(스크류 터빈), Computational fluid dynamics(전산 유체 역학), Multiphase open channel flow(다상 개수로 유동), Micro hydro power(마이크로 수력)

1. Introduction

Due to the increasing energy demands and the over-reliance on fossil fuel sources, renewable energy is progressively gaining people's attention. Hydropower resources are highly valued across the world since they are clean, renewable, and inexhaustible. Hydropower turbines are generally chosen as per the available head and the specific speed to make full use of the hydropower[1]. Large hydropower plants require huge constructions such as dams, reservoirs, penstock etc. and contribute to several environmental impacts during the construction and operation[2]. In recent years, extracting energy from low-head sources such as streams or are increasingly popular as it does not require heavy constructions and also can be installed in remote areas to meet the green energy requirement in remote villages[3].

One of the earliest hydraulic machines is the Archimedes screw. A helical array of blades is wrapped around a central cylinder supported within a fixed trough. The kinetic energy of the water is converted into mechanical energy as the water flows through the blades. The torque thus generated rotates the turbine shaft generating electricity through a generator[4]. Several researchers have studied the working of Archimedes screw and designed according to the site requirements. Saroinsong et al.[5] studied a 3-blade Archimedes screw for a low head application and found that the highest efficiency of the turbine is obtained at a shaft slope angle of 25° and at 50 rpm. Syam et al.[6] studied the performance of a single-blade screw turbine in laboratory and obtained a maximum power of 116 watts at a flowrate of 0.02 m^3/s at 236 rpm with 57% efficiency. Helmizar et al.[7] studied the effect of hub-to-shroud diameter at various flow rates and observed a maximum efficiency of 40.23% at a ratio of 0.42. Platonov et al.[8] conducted a numerical study using computational fluid dynamics (CFD) and found that a screw turbine with 4 or more pitches start operating inefficiently due to several stagnation zones.

The Archimedes screw being an efficient machine, however, require a minimum head of more than 1 m is required for the turbine to rotate and generate power since they are placed at an angle between the upstream and downstream channel[9]. For an ultra-low head requirement (<1 m), the most sought turbine is the water wheel type turbines with large diameters[10]. For a size constraint hydropower site, that require a smaller turbine at ultra-low heads, the Archimedes screw turbine is modified to install it horizontally without any inclination against the incoming flow. The several stages of the turbine design began with an initial screw turbine design inspired from an inducer-type guide vane devised by the authors for flow correction in a multistage centrifugal pump[11]. The screw-type guide vane design was remodeled to develop an undershot turbine that can be used to extract energy from shallow flowing waters. The initial design is then tested for various parameters to obtain a feasible design. The turbine is then customized based on the real site operating condition to produce power with a high efficiency.

The novel approach towards the Archimedes screw turbine to obtain power from shallow waters at ultra-low heads are presented in this paper. The effects of various parameters are tested using CFD tools and the turbine designs are modified to obtain an efficient machine. Rigorous fluid dynamic approaches are undertaken to simulate the turbine in real case scenarios so as to accurately predict the feasibility of such a turbine in real world applications.

2. Description of the model

A 3-bladed screw turbine with an initial diameter of 2m and a width of 2.75 m is designed to operate at an industrial water channel of 2.75 m width and 2.5 m depth. The initial turbine dimensions were selected based on the test site dimensions such that the turbine has the equal width of the channel. The turbine is placed horizontally perpendicular to the incoming flow through the channel while being half-submerged in the water. A hub-to-tip ratio of 0.5 and a pitch-to-cord ratio of 0.223 is chosen for this design. The turbine design and specifications are given in Table 1 and Fig. 1.

The initial turbine model is tested for a number of parameters in a steady-state setting to obtain the feasible design in a no-head scenario. Later, the turbine is scaled up for real site operation at low-head.

3. Numerical methods

3.1 Numerical model

The screw turbine performances are studied by solving three-dimensional continuity and momentum equations in a commercial CFD solver. The steady-state, single-phase, incompressible flows are modelled using Reynolds-averaged Navier-Stokes (RANS) equations while the unsteady-state free surface multiphase flows are modelled using unsteady RANS and free surface equations. These equations are very well documented in various literatures[12]. The turbulent behavior of the fluid during the turbine operations are captured using the k-ω shear stress transport (SST) turbulence model. The SST model uses an integrated feature to move from the k-ω model near the wall region to the k-ε model away from the boundary layer[13]. SST model has been well proved to accurately predict the turbulent characteristics of open-channel turbines[14].

The 3D model of the turbine is generated using Bladegen software while the fluid domains are designed using Space Claim module of ANSYS software. Ansys meshing tool is used for the grid generation. The CFD model along with the governing equations are solved using CFX-solver and post-processed using CFD-Post modules. The unsteady open channel flow requires to solve multiphase equations for air and water with buoyancy and drag force terms. The two fluids are separated by the difference in density of both the fluids. The interphase transfer between the air and water is solved using the standard free surface model. A default interface drag force coefficient of 0.44 is used for the momentum transfer. The inhomogeneous multiphase simulation however uses a homogenous turbulence model for both the

Table 1. Initial Screw turbine configuration

Parameter	Value	Description
Ro (m)	1	Outer radius
Ri (m)	0.5	Inner radius
N	3	Number of blades
m	1.39	Number of helix turns
Ri/Ro	0.5	Hub-to-shroud ratio
t (m)	0.004	Blade thickness
L (m)	2.75	Blade length
S (m)	1.57	Pitch
C (m)	7.04	Cord length
S/C	0.223	Pitch-cord ratio

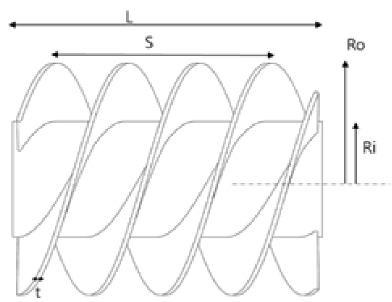


Fig. 1. Initial screw turbine design

fluids for a stable computation. A transient rotor-stator interface is selected at the interface between the submerged part of the turbine and the channel. For the unsteady calculations, the turbine rotated at 3° for every time step. The domains are initialized with hydrostatic pressure for water and ambient pressure for air. The hydrostatic pressure ensures the increase in pressure with the depth of the water. A velocity of 1 m/s is used at the inlet while the outlet is an opening with the hydrostatic pressure distribution. A step function is used to define the volume of fluid throughout the domains. The depth of the upstream water is defined through the step function for the volume of fluid (VoF) of water as follows:

$$vofWaterUp = step((0[m]-y/(1[m])) \qquad (1)$$

Where -y is the direction of gravity and y below 0[m] is defined as water for a 50% submerged turbine. The VoF of air at upstream is defined as '1-vofWaterUp'. A similar equation is also used for the downstream water depth. The water level is adjusted for no-head and low head conditions using the step function.

The steady and the unsteady boundary conditions of the fluid domains are shown in Figs. 2, 3 respectively. The steady state domain uses velocity inlet and 0 Pa

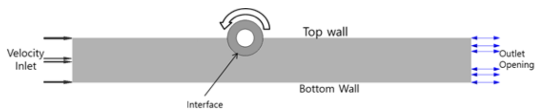


Fig. 2. Steady state single-phase boundary conditions

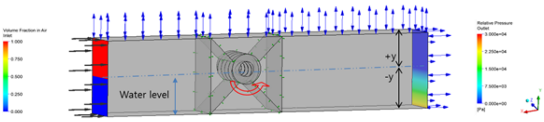


Fig. 3. Unsteady state multiphase boundary conditions

opening outlet condition. The interface between the turbine and domain is chosen as stage averaging. The top and bottom faces of the domains are chosen to be wall since air is not modelled in the steady state simulations. In the unsteady setting, the top surface and the outlet is defined as opening with ambient and hydrostatic pressures, respectively. The inlet volume fraction of air is shown in red while the water is shown in blue. The hydrostatic pressure is shown at the outlet in the increasing order of water depth. The turbine rotates in an anti-clockwise direction when viewed from the side. The timestep and total time for the unsteady simulations is set according to the rpm of the turbine. A total of five to eight revolutions of the turbine were required to obtain a stable output.

The turbine is studied for its performance at various Tip Speed Ratios (TSR) using the unsteady multiphase model. The TSR is a non-dimensional parameter which is defined using the following equation:

$$TSR = \frac{r\omega}{V} \qquad (2)$$

Where r is the shroud radius (m), ω is the angular velocity (rad/s) and V is the average turbine inlet velocity (m/s). The efficiency is significantly influenced by the TSR.

Furthermore, the power coefficient term 'Cp' is used to characterize the ratio of energy extracted by a water wheel to energy available in the stream in order to provide a more accurate assessment of the potential to extract energy from the channel. The power coefficient Cp is defined as[15]:

$$Cp = \frac{\tau\omega}{\frac{1}{2}\rho A V^3} \qquad (3)$$

Where τ is the torque generated by the blades (N.m), ρ is the density of water (kg/m^3) and A is the blade's swept area immersed in water (m^2).

3.2 Numerical Grid

The numerical grid required for the simulations are generated and tested for the its grid independency to ensure that the results obtained do not fluctuate with the grid size. The grid independency tests are carried out using the steady-state single phase model since the optimum grid thus obtained would satisfy the unsteady multiphase simulations too. To take the boundary layers into account, several layers of hexahedral meshes are placed along the blade surface such that the y+ values are kept below 30.

The most accurate approach for a grid convergence research is the Grid Convergence Index (GCI), which is calculated using the Richardson extrapolation method[16]. For a key variable collected from three independent sets of grids with considerable resolutions, an approximate relative error (e_a) and fine grid convergence index (GCI_{fine}) are determined. The torque generated by the turbine is taken as the key variable in this study. The grid convergence index can be calculated from:

$$GCI_{fine} = \frac{1.25e_a}{r-1} \tag{4}$$

where, r is the grid refinement factor.

The GCI_{fine} for the optimum mesh is obtained as 0.323% with an approximate relative error of 1.58%. Since the torque attained for the fine grid has a GCI value of less than 1%, the resulting grids may be deemed to be optimal, and additional grid refining is not required. The number of nodes for the optimal grids is 2.34 million. Fig. 4 shows the final grid of the turbine.

4. Parametric study

The initial turbine model is tested for various design parameters through the parametric study. The steady-state single-phase simulation models are used for this study. The shroud diameter, the hub-to-shroud ratio and the submergence depths are studied to obtain a feasible turbine model. The shroud diameter is varied between 1m to 2m, the hub-to-shroud ratio is varied from 0.3 to 0.7 and the submergence depth is varied from 20% to 50%. In the shroud diameter parametric study, the turbine is kept at 50% submergence depth. Fig. 5(a) shows the decrease in power as the turbine diameter decreases. This is a known factor and hence the shroud diameter of 2 m is selected for the next study. In the hub-to-shroud ratio study, the turbine diameter is kept constant at 2 m while the hub diameter is changed according to the ratio. Here too, the turbine is kept at 50% submergence depth. Fig. 5(b) shows that the highest power is obtained when the hub diameter is half of the shroud diameter (hub-to-shroud ratio=0.5). This is because, in the high ratios, due to low blade height, the contact area of the blade with the water is short thereby producing little torque. This is not the case with low ratio turbines. The contact area between the

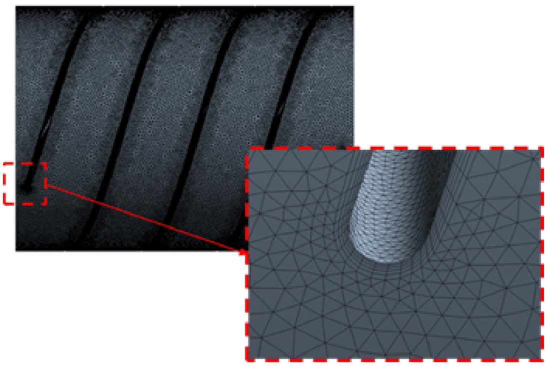
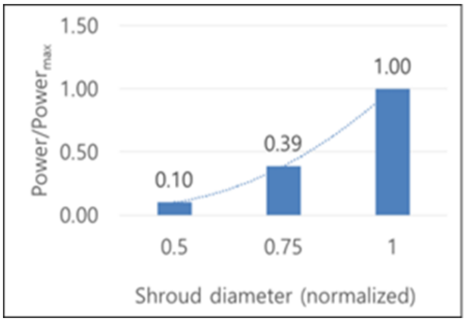
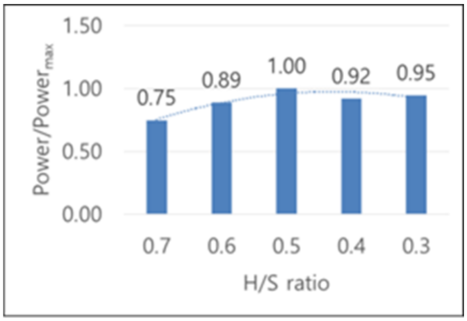


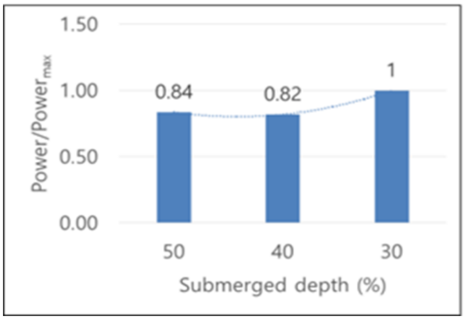
Fig. 4. Optimal grid of the turbine obtained by GCI method



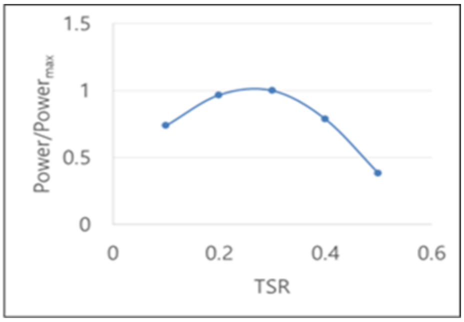
(a) Shroud diameter



(b) Hub-to-shroud ratio



(c) Submergence depth



(d) TSR

Fig. 5. Parametric Study (a) shroud diameter (b) hub-to-shroud ratio (c) submergence depth (d) TSR

water and the blades are high but the drag force acting on the blades counter the torque acting on the blades thereby producing lower output.

In the submergence depth study, the turbine shroud diameter of 2 m and the hub-to-shroud ratio of 0.5 is chosen. Fig. 5(c) shows that the maximum useful power is produced when the turbine is 30% submerged in water. The more the turbine is submerged in the water, the more the drag force act upon the blades which slows down the blades rotational speed thereby generating lower power output. The turbine submerged less than 30% has only the blade tips submerged in water. The rest of the turbine body has no interference with the water. This means that more than 70% of the turbine is exposed to air than water. Thus the turbine rotates only due to the surface velocity of the water rather due to the hydrostatic pressure force. The turbine submerged by more than 30% has large drag forces acting on the blades which affect the power production. Therefore, it is ideal to keep the turbine submerged at 30%. Therefore, a turbine of 2 m shroud diameter and 1 m hub diameter is selected and placed on water at 30% submergence depth for the TSR parametric study.

The steady state parametric studies were conducted for determining the turbine dimensions and the effect of submergence depth on the turbine. The power coefficient of the turbine at various tip speed ratios are studied by solving the unsteady multiphase model. The TSR range from 0.1 to 0.5 showed positive power production with TSR ratio of 0.3 delivering maximum power as shown in Fig. 5(d). The power produced is inadequate (Cp=0.029) at the no-head condition in terms of economic recovery. The 2 m turbine submerged 50% in a 2.5 m depth water channel loses 1.5 m of the kinetic energy that can be extracted from the channel. The large distancebetween the bottom wall of the channel and the turbine gives away a large volume of the incoming water to flow under the turbine which contributes to the low power coefficient. In order to produce higher power and to obtain larger Cp values, the turbine is required to operate at a minimum head with a short bottom wall clearance. The turbine configurations determined from the parametric studies are used as a reference to the scaled-up prototype turbine design for the real site.

5. Prototype turbine design for the real site operating condition

The screw turbine design is scaled-up to be used at an industrial water channel at a head of 0.3 m. The upstream water depth at full operating condition of the plant is maintained at 2.2 m while the downstream water level is obtained at 1.9 m thereby creating a head of 0.3 m. The clearance between the turbine and the bottom wall of the channel is required to be kept under 0.6 m for extracting maximum kinetic energy from the channel. The turbine being 30% submerged in water with a bottom clearance of 0.6 m, the turbine diameter must be at least of 4.21 m. Therefore, the screw turbine is scaled-up by 210% by keeping the same cord length. The prototype design is tested for various TSR ratios using the multiphase model. The turbine produced positive power for TSR between 0.1 to 1. TSR greater than one showed a negative power which indicates that the turbine is forcefully rotated than being rotated by the incoming flow. Fig. 6 shows a maximum Cp of 0.18 is obtained at TSR of 0.3 similar to the model turbine. There is a significant increase in power coefficient between the model and the prototype. The Cp value increased from 0.029 to 0.182. The power produced by the prototype is still inadequate for the economic recovery. The losses due to the large distance between the bottom wall clearance and the turbine has been reduced significantly with the prototype model but there are additional losses associated with the turbine design. Since the cord length of both the model and the prototype designs are kept constant, the number of helix turns of the blades reduced from 1.39 to 0.65 thereby increasing the blade pitch from 1.57 m to 3.3 m for model and prototype, respectively. The large pitch between the helix turns increased the gap between the blades, permitting large volume of water to flow in between the blades without transferring kinetic energy of the water to the turbine.

The losses associated with large pitch size of the turbine can be reduced by decreasing the turbine pitch and cord length. The turbine pitch-to-cord ratio is reduced from 0.52 to 0.469 by decreasing the cord length by 54% to obtain a new turbine configuration. The newly modified turbine however had large gaps between the blades which would contribute to additional losses. The number of blades is increased from

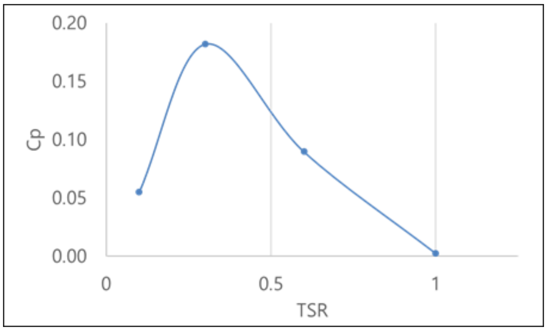


Fig. 6. TSR vs. Cp of prototype model

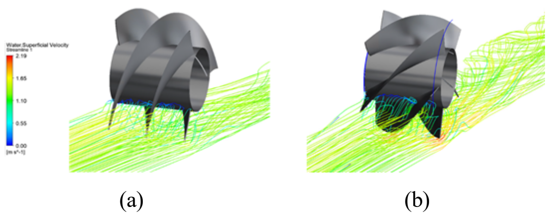


Fig. 7. Velocity streamlines of water through (a) the 3-bladed turbine and (b) the 5-bladed turbine

three to five blades such that the incoming flow does not pass through the turbine without converting its kinetic energy into turbine torque. Fig. 7 shows the water superficial velocity streamlines passing through the two turbine models. The 3-bladed turbine have large spaces between the blades which allows huge volume of water to pass through them without energy transfer to the turbine. On the other hand, the modified 5-bladed turbine absorbs substantial energy from the water flowing through the blade thereby increasing the power coefficient.

The TSR test of the modified 5-blade turbine showed a maximum Cp of 0.43 at a TSR of 0.6. A large increase in Cp of 236% is obtained by modifying the blade. The turbine performance chart also shifted from the maximum Cp value at TSR 0.3 to TSR 0.6 owing to the loss correction modifications. The power coefficient is plotted against TSR for both the turbine models in Fig. 8.

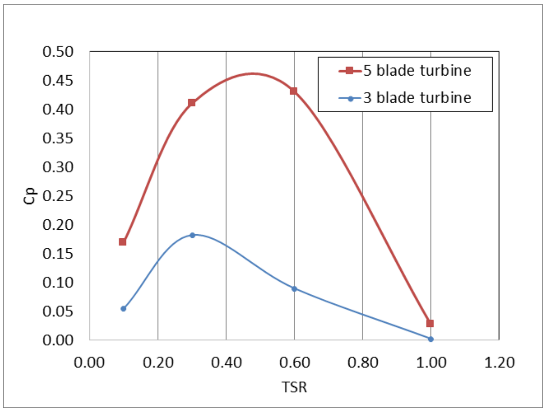


Fig. 8. Comparison of the two turbine models

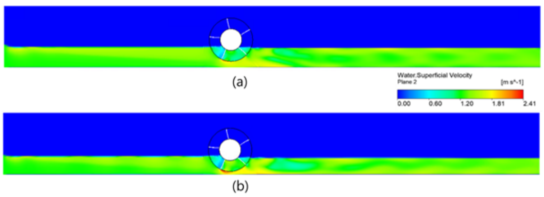


Fig. 9. Effect of bottom clearance on the 5-blade turbine performance: (a) 60cm clearance (b) 30cm clearance

The 5-bladed screw turbine produced a power of 1.6 kW at a 0.3m head condition. In order to increase the power output of the turbine, the clearance between the bottom channel and the turbine is decrease by from 60 cm to 30 cm. The power production increased by 74.4% with a decrease in clearance by 50%. The final power output of the turbine is obtained as 2.84 kW at 0.63 Cp. The water superficial velocity contour for the two bottom clearance cases is shown in Fig. 9. The velocity at the blade tip increases as the clearance decreases. Large volume of fluid passing through the turbine converts the kinetic energy of the water into useful torque at low clearances. Decreasing the turbine clearance less than 30cm increases the turbine blockage ratio which converts the undershot screw turbine into a hydrostatic pressure machine which is a developing technology and needless in this setting[17].

The screw turbine of 4.21 m diameter generates a power of 2.84 kW at 0.3 m head and 0.3 m clearance. Operating the turbine for 24 hours and 275 days would generate 18,744 kWh/year of electricity. With a System Marginal Price of 0.08 $ and an average price of the Renewable Energy Certificate of 0.03 $, an annual revenue of 2,200 $ can be generated from the turbine. The turbine feasibility study showed a practical and effective solution to generate electricity from ultra-low head sources.

6. Conclusion

A horizontal axis screw turbine concept is developed for extracting power from low-head streams and shallow industrial waterways. An initial screw turbine design is selected based on a real site dimension. The initial design is then tested for various shape parameters using CFD. The shroud diameter, the hub-to-shroud ratio and, the submergence depths of the turbine were studied for a range of dimensions in a steady state condition. A turbine of 2 m diameter and a hub-to-shroud ratio of 0.5 submerged at 30% depth in water showed higher performance. The rotational speed of the turbine is selected by conducting the TSR test in an unsteady multiphase free-surface CFD model. The turbine showed maximum coefficient of power at a TSR of 0.3. However, the power output at no-head condition is too low for economic recovery. The model turbine is then scaled up by 210% by keeping the same cord length to test the turbine at a real site operating condition with a 30 cm head. For the prototype turbine, a maximum Cp of 0.18 is obtained at TSR of 0.3. Huge flow losses were observed in the prototype turbine. Since the cord length were same as the model turbine, the blade pitch of the prototype increased by 110%. This large pitch gave way to wider gap between the blades allowing a large volume of water to pass through the turbine without energy transfer.

The prototype turbine was modified by reducing the cord length by 54% and pitch by 9.8% while increasing the blade number from three to five. The new prototype design was tested for different TSR ratios using the unsteady multiphase model. The power coefficient of the new prototype turbine increased by 236% compared to the latter. The maximum Cp value was obtained at a higher TSR value than the prototype design. To further improve the turbine performance, the turbine clearance height between the turbine and the bottom wall of the channel was decreased by 50%. The 50% decrease in the clearance height increased the power production by 74.4%. The final design produced a power output of 2.84 kW at 0.3 m head. The design feasibility study of the horizontal-axis screw turbine proved to be a viable solution for rural electricity needs with low-initial investment. The turbine is in the initial development stage and this study is conducted to determine the design feasibility of the selected turbine prior to manufacturing. This is an ongoing research. The final turbine model would be manufactured and tested in a laboratory scale in the near future.

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