



PRELIMINARY DESIGN AND CONSTRUCTION OF SMALL HYDRO TURBINE FOR OFF-GRID POWER GENERATION

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Abstract

The growing global demand for environmentally friendly energy solutions has increased the need for sustainable approaches to addressing the persistent problem of electricity supply. This study investigated the preliminary design and construction of a small-scale hydro turbine for power supply that could be deployed in riverine, remote, and off-grid communities. The prototype hydro turbine that offers an attractive solution for power generation at the scale envisaged for small rural communities is the propeller-type axial-flow turbine. This turbine offers several key advantages, including ease of construction, reliability, ease of operation and maintenance, efficiency, and adaptability to various environmental conditions. This paper details a comprehensive design methodology that began with the geometric optimisation of the turbine components of a 1kW axial-flow turbine operating with 1.5 m head and a flow rate of 0.07 m³/s. CFTurbo, a hydro-turbine design software package, was used with SOLIDWORKS to design and perform numerical simulations of the flow in the vicinity of the turbine components using Computational Fluid Dynamics (CFD). This process involved detailed analysis and iterative steps that led to modifications that ensured the turbine performed optimally across a spectrum of operating conditions. Both an analytical approach and physical tests were used to assess the turbine's operational characteristics. For the physical test, running water from a dam spillway was channeled through a funnel-shaped penstock into the turbine spiral casing to operate the turbine under off-design conditions. The maximum turbine output power during an off-design point test was 54W at 189 rpm, a flow rate of 0.0205 m³/s, and a head of 0.55 m, giving an efficiency of 61%. The study concluded that there is sufficient local capacity to produce an axial-flow propeller-type turbine that can be deployed as a viable option for small-scale hydropower supply to riverine and off-grid locations in Nigeria using simple technology.

Keywords: axial-flow turbine, CFD, design, flow rate, head, Small-scale hydropower

Introduction

The rising worldwide need for eco-friendly energy has intensified investigation into efficient methods for harnessing renewable resources for power generation. Among these resources, water remains one of the most dependable and widely used because of its abundance, renewability, and comparatively minimal environmental impact (Ritchie *et al.*, 2020). In this context, small hydropower (SHP) systems, typically ranging from a few kilowatts to several megawatts (up to 30 MW), have gained increasing focus as a viable answer for decentralized electricity generation, particularly in riverine rural and off-grid regions (Ibegbulam *et al.*, 2023). SHP systems offer

distinct advantages, including low environmental disruption, suitability for local manufacturing, ease of maintenance, and adaptability to community-scale applications, while making the challenges associated with large hydropower projects less severe (Nautiyal and Goel, 2020).

Hydraulic turbines are central to hydropower systems, converting the potential head and kinetic head of moving water into total head or mechanical energy that can be converted into electrical power (Sarkar *et al.*, 2023). The flow within hydraulic turbines is exceptionally convoluted—three-dimensional, non-steady, and chaotic—necessitating detailed analysis to optimise

performance and ensure operational reliability (Tiwari *et al.*, 2020). Broadly, hydraulic turbines are classified as active or reactive types. In active turbines, the total head is fully converted into kinetic energy, delivered as high-velocity jets from a nozzle. The transfer of this kinetic energy to the runner is through jet interaction with buckets or vanes, where conversion of energy occurs predominantly through impulsive forces (Chitrakar *et al.*, 2020). Pelton and Turgo turbines are classic examples of this category (Kumar, 2021). Conversely, in reaction turbines, water passes through stationary guide vanes that partially convert head into kinetic energy before entering the runner with combined pressure and kinetic energy, as seen in propeller, Francis, and Kaplan turbines (Vagnoni, 2022). Hydraulic turbines can also be categorized with respect to flow direction relative to the shaft: tangential-flow turbines (e.g., Pelton wheel), radial-flow turbines (e.g., Thomson and Fournayron), axial-flow turbines (e.g., propeller and adjustable blade or Kaplan), and mixed-flow turbines, where water enters radially and exits along the axis, as in modern Francis turbines (EEEGUIDE, 2024) are practical examples.

Despite significant advances in hydraulic turbine development, the design and localised manufacturing of small-scale axial-flow turbines remain limited in regions such as Nigeria, where substantial hydropower resources remain underutilized. This write-up presents the preliminary design and construction of a 1 kW axial-flow (propeller-type) hydraulic turbine, examines the functional relationships among key design parameters, and characterises the performance of the fabricated unit under controlled conditions.

The significant contribution of this work lies in providing a practical framework for the design and construction of a low-capacity axial-flow hydraulic turbine of the propeller type. The study bridges the gap between theoretical design principles and hands-on construction and offers insights into experimental performance relevant to small hydropower development in Nigeria.

Components of an Axial Flow Turbine

With reference to Figure 1, a typical axial-flow hydraulic turbine is made up of the following: (a) Spiral Casing, (b) Guide Vanes, (c) Runner, (d) Shaft, (e) Draft Tube.

In a hydraulic turbine system, water typically comes in through the penstock and is gently guided toward the spiral casing, which provides an even flow distribution to achieve the required reaction and improve the performance of the runner (Shrestha and Choi, 2020). As the water flows, it meets the guide vanes, which control the flow and direct it at the appropriate angle to the runner inlet (Stojkovski

et al., 2021). Upon leaving the guide vanes, the water moves into the runner, which is the central rotating element composed of carefully shaped blades that change the flow's kinetic and pressure energy into rotational energy of the shaft. The water goes through the runner blades along the axis at both the inlet and the exit (Abeykoon, 2022). After transferring its energy to the runner, the water goes into the draft tube, a diverging tube that connects both the runner and the tailrace. The increasing shape of the tube allows the water velocity to decrease while its pressure increases, ensuring maximum pressure recovery and smooth discharge into the tailwater (Barkett Botan *et al.*, 2021). The design and optimisation of these constituent parts are crucial, as they work in unison to ensure the effective transfer of hydraulic energy into shaft power, which can ultimately be harnessed to generate electricity. The Kaplan turbine (Figure 1), a specialized axial-flow turbine, shares these parts but features a runner with several adjustable blades whose angle of incidence is regulated by a complex control system within the hub and shaft, allowing the blade pitch to adapt to flow conditions and maintain high efficiency across varying operating points (Ghenaiet, 2020). The hydro turbine considered in this work is a fixed-blade (propeller-type) axial-flow turbine, and the design process presented in the following sections is based on this configuration (Figure 2).

Theoretical Analysis

The design theory and process of the turbine, from one component to the other, were analyzed and computed in the CFTurbo software, following the design flow chart in Figure 3 for both modelling and parameter calculations. The ultimate challenge is to design and proportion the turbine's constituent components, based on flow characteristics, so that the maximum possible power (P_{hyd}) is extracted from the water flow and delivered to the rotating shaft. This defines how efficient the machine can be. The maximum possible power, P_{hyd} is given by

$$P_{hy} = \rho g H Q \quad (1)$$

where ρ is the density of water (kg/m^3), g is the acceleration due to gravity (m/s^2), H is the gross head (m), and Q is the volumetric flow rate (m^3/s). According to Euler's turbine equation, the energy per second extracted from the water by the blades of the runner is the product of the torque, T (N.m), and the angular velocity, ω (rad.s^{-1}). This can be related to the gross hydraulic power to determine the overall hydraulic efficiency.

$$\eta_{hy} = \frac{2\pi NT}{\rho g H Q} \quad (2)$$

η_{hyd} is the overall hydraulic efficiency, and N is the shaft rotational speed (rpm).

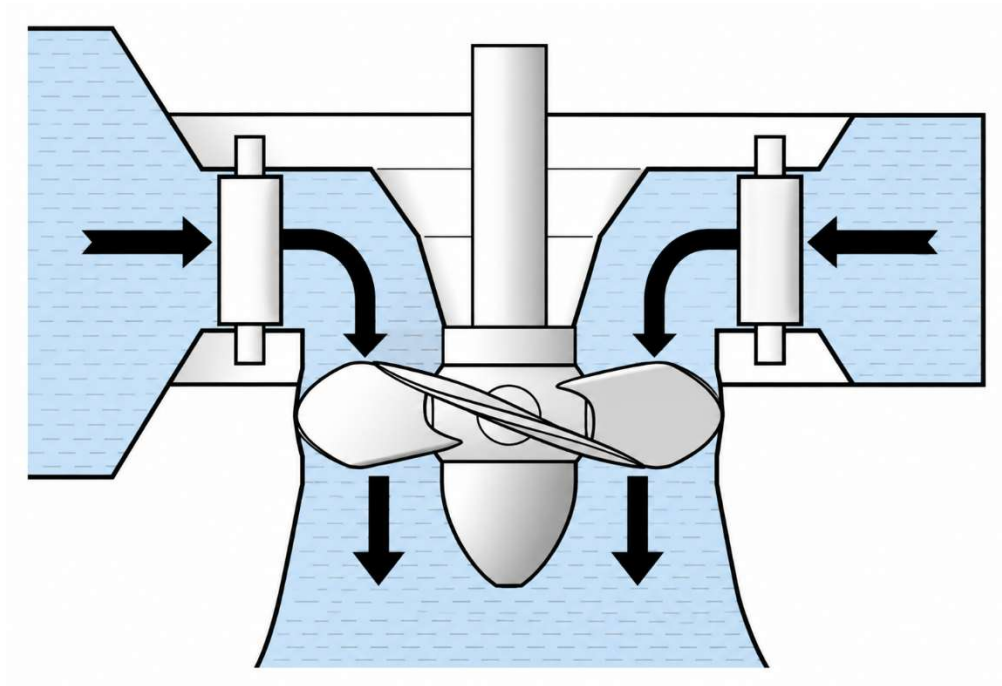


Figure 1. Kaplan turbine (adapted from Jahobr, 2016)

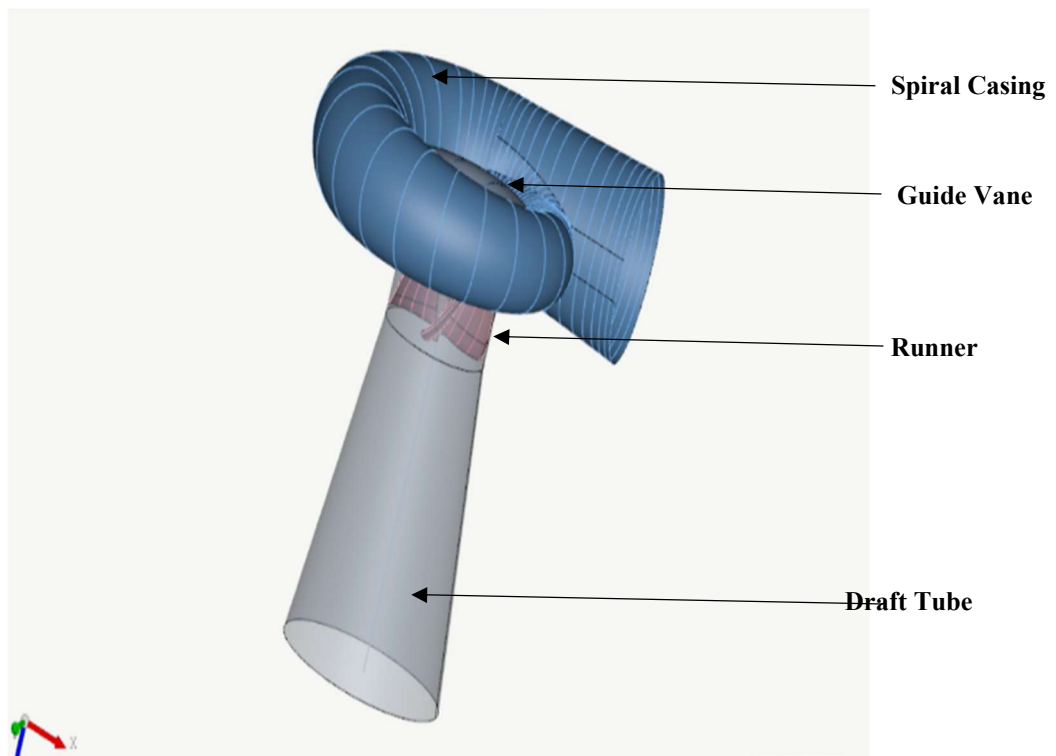


Figure 2. Axial-flow propeller turbine 3D model from CFTurbo.

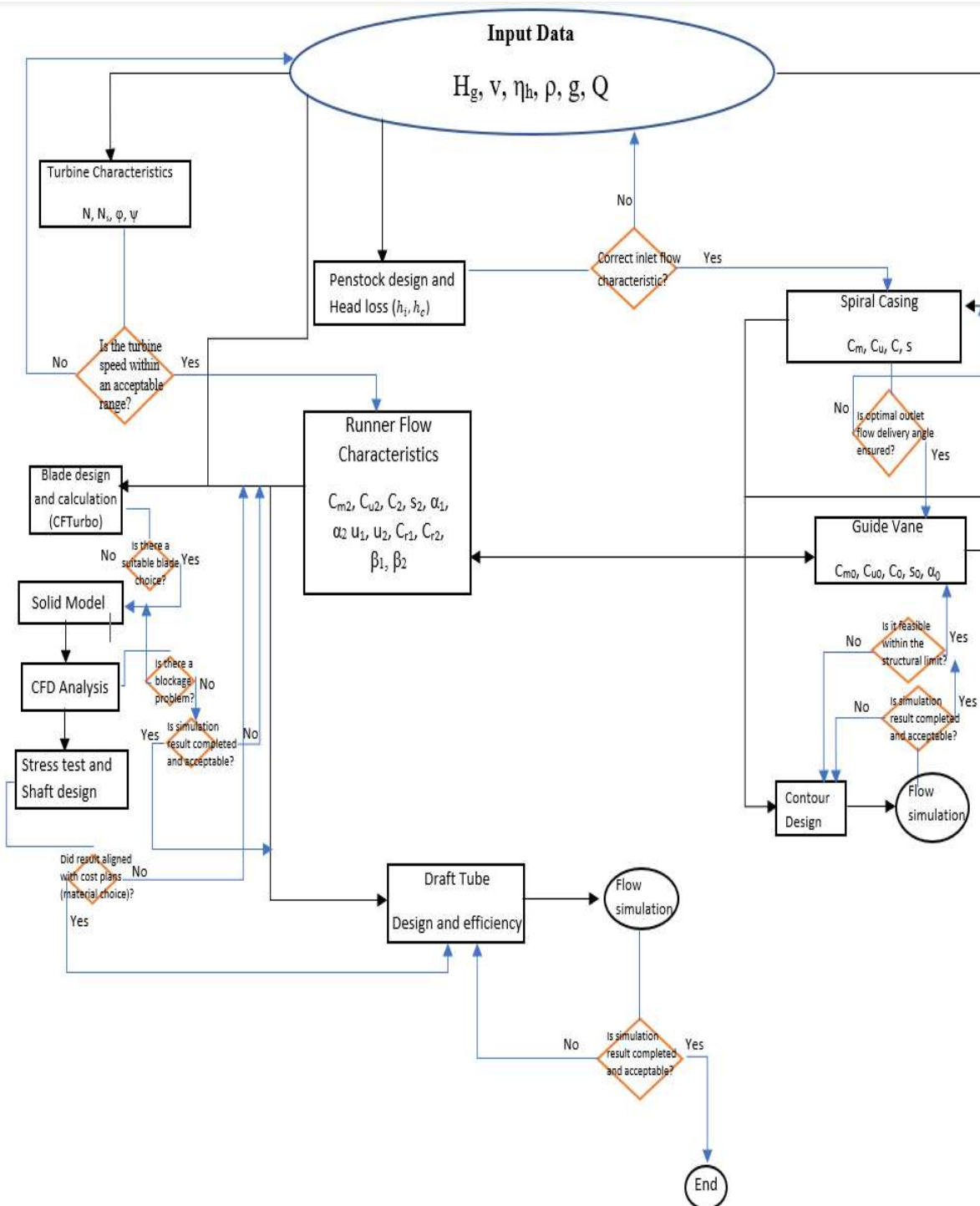


Figure 3. Flow chart for modelling and calculation of design parameters

Flow Parameters

Considering a propeller turbine with a gross head of 1.5 m, a flow rate of 0.07 m³/s, flowing water at a temperature of 20°C as the working fluid, and pressure equal to the atmospheric pressure (101.3 kPa). The total hydraulic power and shaft rotational speed of the turbine is given by

$$P_{hyd} = \rho g H Q = 1.028 \text{ kW}$$

The turbine shaft rotational speed and the specific speed N_s are expressed as (Abeykoon, 2022).

$$N = \frac{\sigma(2gH)^{0.75}}{2\sqrt{\pi Q}} \quad (3)$$

$$= 13.4 \text{ rev s}^{-1} \text{ or } 800 \text{ rpm}$$

$$N_s = \frac{N\sqrt{Q}}{H^{0.75}} \quad (4)$$

$$= 156$$

σ is the coefficient of specific speed for a given turbine type, and for a propeller type, it is 1 (See Figure 4). In addition, taking u as the peripheral or transportation speed of the runner, we can calculate the discharge coefficient and head coefficient using the following relations (Barsi et al., 2021):

$$\begin{aligned} \text{Discharge coefficient, } \varphi \\ = \frac{\text{Actual flow rate}}{\text{Theoretical flow rate}} \end{aligned}$$

$$\varphi = \frac{Q}{\frac{\pi D_1^2}{4} u} \quad (5)$$

$$= 0.627$$

$$\text{Head coefficient, } \psi = \frac{\text{Actual head}}{\text{Theoretical head}}$$

$$\psi = \frac{2g}{u^2} \quad (6)$$

$$= 0.743$$

where D_1 is the runner diameter.

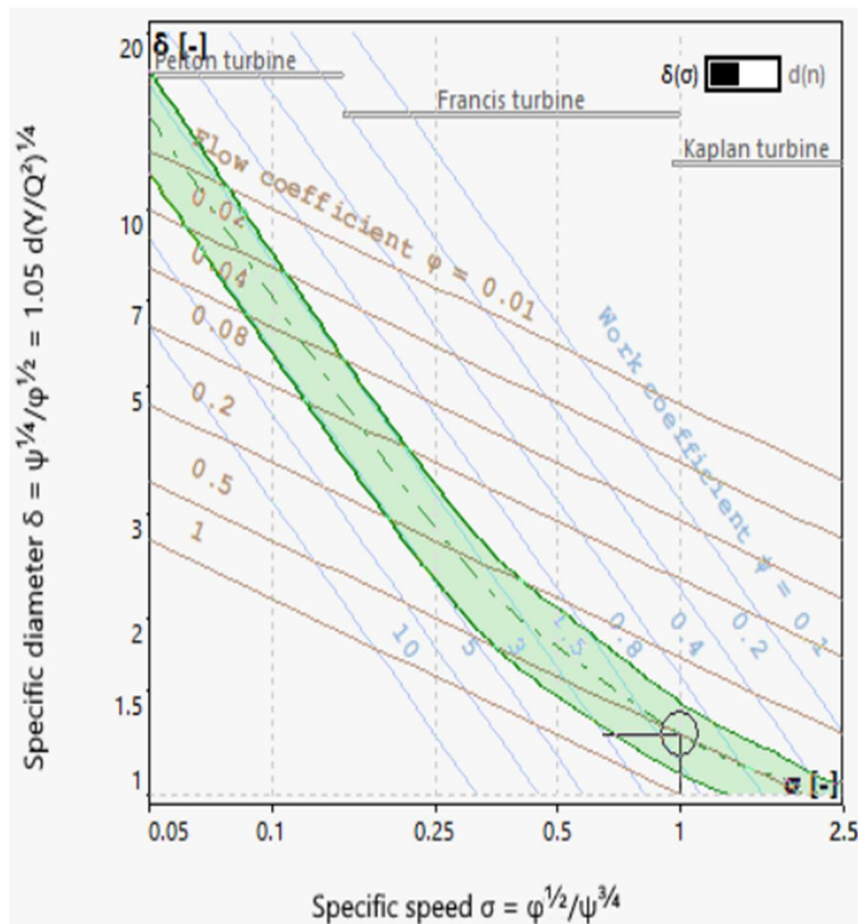


Figure 4. Coefficient of specific speed generated with CFTurbo

Runner Design and Flow Distributions

The parameter that sufficiently fixes the structural form of the turbine runner is the upstream geometrical feature called the guide vanes or wicket gate of the turbine (see Figure 1). It is related to $\sigma_d =$

NQ/gH , which is the *design parameter*. This parameter determines not only the structural or geometrical form but also the size of the axial flow turbine. The runner diameter D_1 can be estimated, commonly, with $b/D_1 = 0.3-0.4$. Velocity triangles

(Figure 5) are graphical representations that show the relative velocities and directions of the water as it flows through the turbine. These triangles help us comprehend how the water interfaces with the turbine blades and the subsequent transfer of energy during the process. C , C_m , C_u , and C_r are the absolute,

meridional, circumferential, and relative velocities, respectively. Angles α and β are the absolute and relative flow angles, in that order. The subscripts 1 and 2 denote the runner inlet and outlet, respectively.

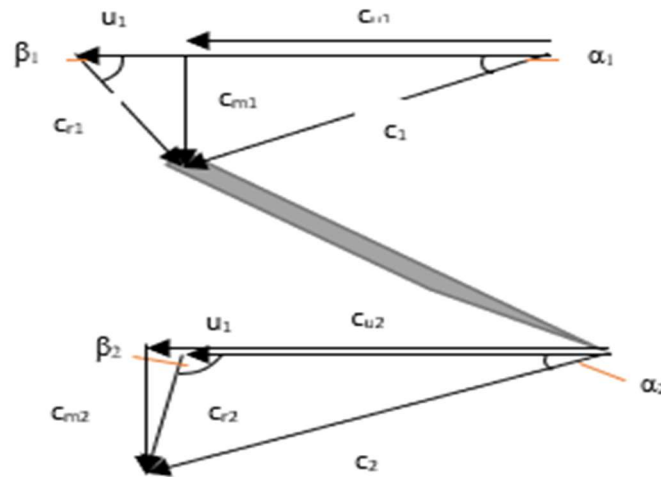


Figure 5. Velocity triangle of the runner

1. At Runner Inlet

A consideration of the velocity triangle in Figure 5 establishes the following relation

$$c_{r1}^2 = c_{m1}^2 + (u - c_{u1})^2 \quad (7)$$

The tangential velocity relates to the installation head and peripheral speed, as shown below (Abeykoon, 2022);

$$c_{r1} = \frac{2g(H-H_s)}{u_1} \quad (8)$$

$$c_{m1} = \frac{Q}{\pi(r_1^2 - r_{hu}^2)} = 2.8 \text{ m/s} \quad (9)$$

$$u_1 = \frac{\pi DN}{60} = 6.3 \text{ m/s} \quad (10)$$

Substituting the values from (8), (9), and (10) into (7),

$$c_{u1} = 5.3 \text{ m/s.}$$

The upstream flow down the guide vane and into the runner inlet can be assumed to be a free vortex flow since it rotates without any external forces acting on it. Therefore, the angular momentum is conserved from the outlet of the guide vane to the inlet of the runner blade. Considering a constant flow rate

through the guide vane outlet at a flow angle α_o , the angular momentum per unit time, $\dot{M}_0$, of the flow can be shown to be,

$$\dot{M}_0 = c_{u0} \frac{D_0}{2} \dot{m} = \frac{c_{m0}}{\tan \alpha_0} \frac{D_0}{2} \rho Q = \frac{1}{\tan \alpha_0} \frac{1}{2\pi b} \rho Q^2 \quad (11)$$

It follows that this value remains constant as the fluid reaches the runner inlet with an even distribution of the meridional velocity, which must fulfil the condition for free vortex flows (no energy loss). Therefore, the circumferential velocity with respect to the radial coordinate r can be expressed as;

$$c_{u1} = c_{u1,hub} \frac{r_{hu}}{r_0} \quad (12)$$

The angular momentum at the inlet, $\dot{M}_1$, is then given as;

$$\dot{M}_1 = \rho \int_0^Q c_{u1} r_0 dQ \quad (13)$$

$$= \rho \int_{r_{hub}}^{r_1} 2\pi r c_{m1} c_{u1,hub} r_{hub} dr = \rho Q c_{u1,hub} r_{hub}$$

$$\text{where } Q = \pi(r_1^2 - r_{hu}^2) c_{m1} \text{ and } dQ = 2\pi r c_{m1} dr.$$

$$\text{since } \dot{M}_0 = \dot{M}_1,$$

$$\frac{1}{\tan \alpha_0} \frac{1}{2\pi b} \rho Q^2 = \rho Q c_{u1,hu} r_{hu} = \rho Q c_{u1} r_0 \quad (14)$$

Then,

$$c_{u1} = \frac{1}{\tan \alpha_0} \frac{1}{2\pi b r_0} Q \quad (15)$$

Substituting the value of c_{u1} from (11) into (15), $\alpha_0 = \alpha_1 = 29^\circ$

The angle of relative flow, β_1 , can be obtained by examining the velocity triangle. Thus,

$$\tan \beta_1 = \frac{c_{m1}}{u_1 - c_{u1}} \quad (16)$$

$$\beta_1 = 70^\circ$$

2. At Runner Outlet

The flow angle of the outflow from the runner just before it enters the draft tube is fixed by the trailing-edge profile of the blades of the runner (Zhang, 2021). Therefore,

$$\tan \beta_2 = \frac{\frac{\varphi}{1 - \frac{r_{hub}^2}{r_1^2}} r_1}{r_1} \quad (17)$$

$$\beta_2 = 105^\circ$$

From the velocity triangle in Figure 5,

$$c_{r2} = \sqrt{c_{m2}^2 + (c_{u2} - u_2)^2} \quad (18)$$

where,

$$c_{m2} = \frac{Q}{A_2} \quad (19)$$

$$= 2.2 \text{ m/s}$$

$$u_2 = \frac{\pi D N \varphi}{60} \quad (20)$$

$$= 4.2 \text{ m/s}$$

$$c_{u2} = u_2 - \frac{c_m}{\tan \beta_2} \quad (21)$$

$$= 4.88 \text{ m/s}$$

Substituting values obtained in (17), (19), and (20) into (18) gives the blade outlet relative velocity, $c_{r2} = 2.3 \text{ m/s}$.

$$c_2 = \sqrt{c_m^2 + c_{u2}^2} \quad (22)$$

$$= 5.35 \text{ m/s}$$

Using the sine rule,

$$\frac{\sin \alpha_2}{c_{r2}} = \frac{\sin 105}{c_2}$$

$$\therefore \alpha_2 = 24.5^\circ$$

The distribution of velocity in the runner is depicted in Figure 6. Higher velocities appear at the inlet on the left in red, orange, and yellow, while the velocity gradually decreases through the transition section and becomes much lower toward the outlet on the right, shown in blue.

Blade Design

The flow inlet and outlet angles, including the blade angles for the runner, have been determined, but they are insufficient to define the blade profile completely. To comprehensively define the blade profile, CFTurbo software was used to compute data from the analysis of the calculated angles and angular momentum, which were then used to specify an airfoil with lift and drag coefficients that satisfy the equation below (Yang et al., 2023).

$$\eta_h = u c_{r2} \frac{l}{t} \left(k_z - \frac{k_x}{\tan \beta_2} \right) \quad (23)$$

where l is the blade length, t is the spacing between blades, while k_z is the coefficient of lift and k_x is the coefficient of drag of the airfoil.

$$\frac{\eta_h}{U_2 C_{r2}} \tan \beta_2 = (k_z \tan \beta_2 - k_x) \frac{t}{l}$$

To calculate the left-hand side (LHS) of the equation, we need to determine U_2 and C_{r2} which are the coefficients of peripheral and relative velocity.

$$U = \frac{u_2}{\sqrt{2gH}} = 0.77 \quad (24)$$

$$C_{r2} = \frac{c_{r2}}{\sqrt{2g}} = 0.99 \quad (25)$$

$$\frac{\eta_h}{U_2 C_{r2}} \tan \beta_2 = \frac{0.8}{0.77 \times 0.99} \tan 105 = 3.92$$

$$\text{Then, } 3.92 = (k_z \tan \beta_2 - k_x) \frac{t}{l}.$$

To satisfy the LHS with the RHS of the equation, CFTurbo was used to perform interpolations, from which the selected profile was NACA 6309, which satisfied the given direction for selecting the cross-section using equations.

$$k_z = \frac{2F_l}{\rho u^2 A} \quad (26)$$

$$k_x = \frac{2F_d}{\rho u^2 A} \quad (27)$$

where the lift and drag forces are F_l and F_d respectively, and with blade area represented by A .

— **Max Thickness:** The maximum thickness of the NACA 6309 airfoil occurs at 35% of the chord length. It is 9% of the chord length.

- **Max Camber:** The maximum camber (curvature) of this airfoil is 1.1%, and it is located at 50% of the chord length.

The airfoil profile (NACA 6309) is then used to design the blade in SolidWorks (see Figure 7). The runner is then printed using a 3D printer (Figure 8).

Table 1 shows the blade airfoil dimensions from hub to shroud, while Table 2 gives the blade local profile spanwise. Table 2 shows that the blade profiles and streamline velocities are described by the parameters in the first column, and these values change as the span moves from the hub to the shroud.

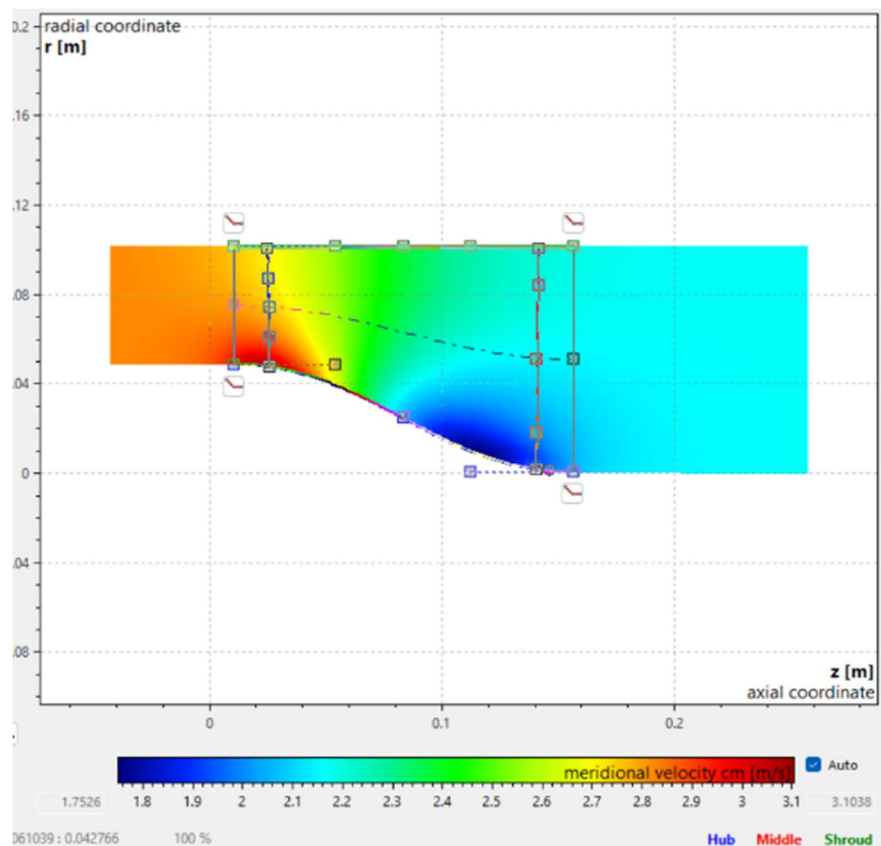


Figure 6. Velocity distribution in the runner (simulation result from CFTurbo)

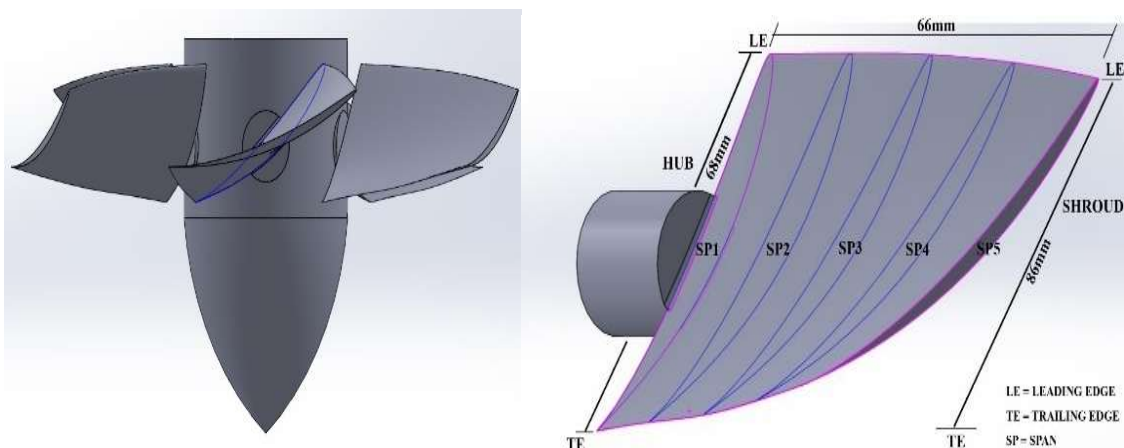


Figure 7. Left: SolidWorks model of the runner; Right: blade airfoil showing the spans.

Table 1. Blade Airfoil Dimensions

	LENGTH (mm)	LE ANGLE (°)	TE ANGLE (°)	THICKNESS (mm)	WIDTH (mm)
1 (HUB)	68	75	35	3.2	0
2	72	66.2	30.2	3.2	16.5
3	76	52.5	25.5	2.8	33
4	82	38.7	20.8	2.2	49.5
5 (SHROUD)	86	25	16	1.1	66

Table 2. Blade Local Profile

Parameter	SPAN 1 (HUB)		SPAN 2		SPAN 3		SPAN 4		SPAN 5 (SHROUD)	
	LE	TE	LE	TE	LE	TE	LE	TE	LE	TE
D [m]	0.070	0.003	0.103	0.052	0.136	0.1022	0.169	0.1516	0.202	0.202
α [°]	29	24	29	24	29	24	29	24	29	24
β_1 [°]	70		66.3		52.5		38.7		25	
β_2 [°]	35		30.2		25.5		20.8		16	
c_m [m/s]	3	13.1	3	2.3	2.9	2.2	2.9	2.2	2.9	2.2
c_u [m/s]	11.1	-18.5	9.1	-1.7	7.1	-0.4	5.2	0.5	3.2	0.7
c_t [m/s]	-0.5	-2.1	-0.4	-0.3	-0.3	-0.2	-0.1	-0.1	0	0
u [m/s]	4	0.1	5.1	2.2	6.2	4.3	7.3	6.3	0.84	0.84
w [m/s]	7.7	22.8	5	4.6	3.1	5.2	3.6	6.3	6	81
W_u [m/s]	7.1	-18.6	4	-4	0.9	-4.7	-2.2	-5.8	-5.2	-7.7
τ	1.093	6.03	1.063	1.064	1.046	1.038	1.036	1.03	1.029	1.03
i/δ	-77.5	0	-77.5	0	-55.2	0	-14.4	-0.1	-3.6	-0.1
w_2/w_1	2.945		0.918		1.709		1.739		1.347	
c_2/c_1	1.97		0.302		0.297		0.387		0.546	
$\Delta\beta_B$ [°]	-45		-36		-27		-17.9		-9	
$\Delta\alpha_F$ [°]	129.5		109.2		78.6		48.1		31	
$\Delta\beta_F$ [°]	-121.9		-113.4		-82.2		-32.2		-12.5	
$\Delta(c_u \times r)$ [m ² /s]	-0.558		-0.6		-0.55		-0.411		-0.25	
γ	1		1		1		1.0004		1.004	

LE = Leading Edge, TE = Trailing Edge



Figure 8. Printed runner.

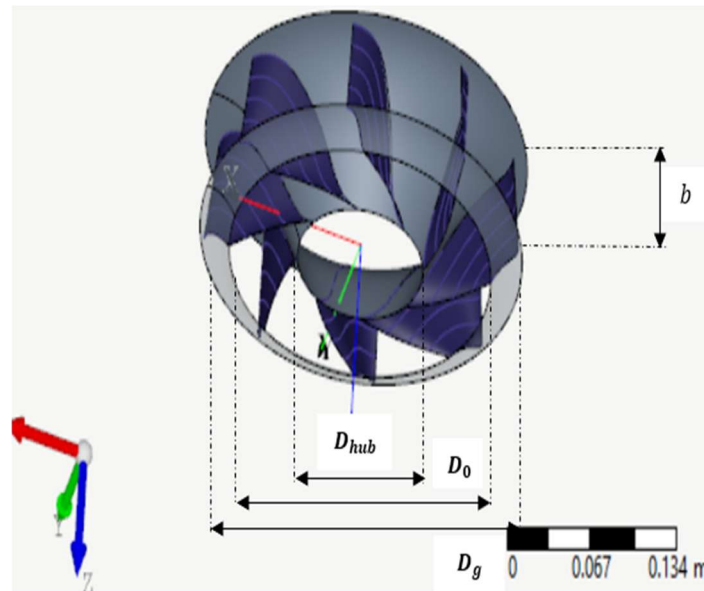


Figure 9. CFTurbo model of the Guide Vane

Overall Hydraulic Efficiency

The overall efficiency combines hydraulic and mechanical efficiencies. It varies with the turbine type and operating conditions. It typically ranges from 80% to 95%.

The specific energy expended by a unit mass of water on the runner and its blades is given by

$$\begin{aligned} gE = Y &= u_1 c_{u1} - u_2 c_{u2} \\ &= 12.3 \text{ m}^2/\text{s}^2 \end{aligned} \quad (28)$$

Application of Newton's second law of motion to the runner's rotational motion gives,

Torque = change of angular momentum per unit time

The fluid flow entering the blade has an angular momentum given by $\dot{m}c_{u1}r_1$ while that of the exiting fluid flow is $\dot{m}c_{u2}r_2$.

Since the fluid works on the blade, and the work done by the fluid per second is $T\omega$.

Work done/second:

$$E = T\omega = \dot{m}(c_{u1}r_1 - c_{u2}r_2)\omega \quad (29)$$

$$\text{But } \omega = 2\pi N \text{ and } u = \omega r$$

$$2\pi NT = \dot{m}(c_{u1}u_1 - c_{u2}u_2) \quad (30)$$

$$\therefore T = 10.29 \text{ Nm}$$

Power transmitted to the runner is

$$\begin{aligned} T\omega &= 2\pi NT \\ &= 862 \text{ W} \end{aligned}$$

From (2),

$$\begin{aligned} \eta_{hy} &= \frac{2\pi NT}{\rho g HQ} \\ &= \frac{862 \text{ W}}{1028 \text{ W}} = 0.84 \text{ or } 84\% \end{aligned}$$

This is the overall hydraulic efficiency of the turbine.

Guide Vane Design

The purpose of the guide vane is to supply water to the runner in a specific direction and at a specific velocity, ensuring these values remain unchanged throughout the entire circumference; that is, to impart a uniform whirl to the fluid across its entire periphery. The important parameters of this apparatus are the discharge rate and the outlet blade angle.

Flow through the stationary vane adheres to the fundamental physical laws of thermodynamics and can be characterized using velocity triangles. In a steady incompressible flow of fluid with density ρ , the mass flow rate remains constant across the area sections shown in Figure 9 (the inlet and outlet area of the stator).

$$\dot{m} = \rho A c_m = \rho A_0 c_{m0} \quad (31)$$

$$\therefore Q = A c_m = A_0 c_{m0} \quad (32)$$

where c_m and c_{m0} are the meridional (axial) velocities at the inlet and outlet of the guide vane, as shown in Figure 10.

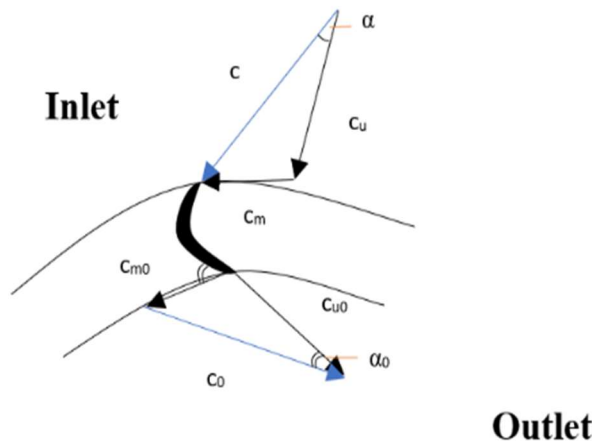


Figure 10. Velocity triangle of the guide vane.

Inlet meridional velocity:

$$\begin{aligned} c_m &= \frac{Q}{\pi D b} \\ &= 1.8 \text{ m/s} \end{aligned} \quad (33)$$

Outlet meridional velocity:

$$\begin{aligned} c_{m0} &= \frac{4Q}{\pi D^2} = \frac{Q}{\pi(r_1^2 - r_{hub}^2)} \\ &= 2.8 \text{ m/s} \end{aligned} \quad (34)$$

Inlet flow angle:

$$\tan \alpha = \frac{c_m}{c_u} \quad (35)$$

Absolute component:

$$\sin \alpha = \frac{c_m}{c} \quad (36)$$

Using the cosine rule,

$$c_m^2 = c^2 + c_u^2 - 2cc_u \cos \alpha \quad (37)$$

Substituting the value of c from (36) into (37) and expressing $\sin \alpha$ in terms of $\tan \alpha$,

$$\sin \alpha = \frac{2 \tan \frac{1}{2} \alpha}{1 + \tan^2 \frac{1}{2} \alpha} = \frac{\tan \alpha}{\sqrt{1 + \tan^2 \alpha}} \quad (38)$$

Then, in (35), the inlet flow angle $\alpha = 15^\circ$

From (36), absolute velocity $c = 6.95$ m/s

Therefore, when (37) is simplified, the inlet circumferential velocity is $c_u = 6.7$ m/s.

The main rule is to maintain a constant contraction of the guide duct area. This constant contraction from the inlet of the guide duct area to its outlet results in a steady increase in water velocity. The optimum number of guide blades can be selected using the relation (Aung et al., 2014):

$$Z = \frac{1}{4\sqrt{D_1}} + 4 \text{ to } 6 \quad (39)$$

Spiral Casing Design

The spiral casing (Figure 11), also called a volute, is an important component of a turbine that evenly distributes the water's kinetic and potential energy, thereby generating the reaction force required for improved runner performance. The spiral casing has a spiral design and creates an enclosed water path to maintain water pressure. The number one requirement of a spiral casing is to uniformly channel the water flow along the guide vane circumference. The implication is that the discharge through a meridional section of the spiral casing is ascertained by the angle ϕ given as:

$$Q_\phi = Q \frac{\phi}{360^\circ} \quad (40)$$

where Q [m³/s] is the total delivery through the spiral casing and ϕ [°] is the angle measured from the end of the spiral tooth called the wrapping angle.

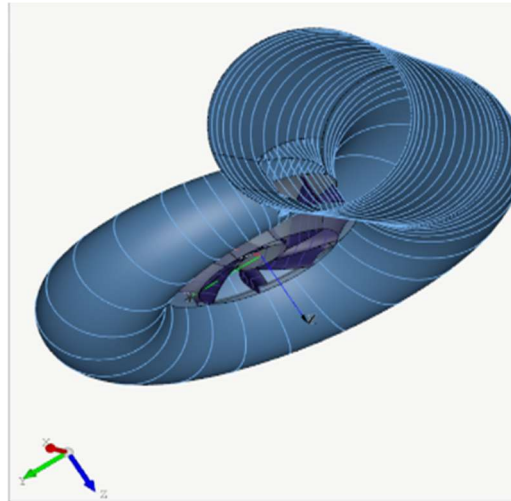


Figure 11. CFTurbo model of the Spiral Casing with guide vane.

The procedure for design begins with the spiral casing inlet section defined by the coverage angle ϕ_{cov} , inlet water flow rate Q_{in} and inlet peripheral velocity V_{in} . The coverage angle is the angle between the inlet section of the spiral case and the end of the spiral tooth. If the spiral tooth angle is taken as $\phi_{s,t} = 15^\circ$, then we define the coverage angle as:

$$\begin{aligned} \phi_{cov} &= 360^\circ - \phi_{s,t} \\ &= 345^\circ \end{aligned} \quad (41)$$

Now, there are two approaches to determining the peripheral velocity at the inlet. The first approach is based on the net head H_n of the turbine, and this is

given by the expression

$$V_{in} = 1.93 H_n^{0.33} \quad (42)$$

The spiral casing is made up of circular and elliptical segments. Either segment is cut out vertically with height represented by t_s , and this varies along the periphery of the spiral outlet. Then, the inlet section has an area given by

$$A_{in} = \frac{Q}{V_{in}} \quad (43)$$

from which the diameter of the full circle at the inlet could be determined. That is,

$$D_{in} = 2 \sqrt{\frac{A_{in}}{\pi}} \quad (44)$$

The cut circle has an area determined using

$$A_{ef,0} = r_{in}^2 = \left[\pi \left(1 - \frac{\alpha}{360} \right) + \sin \frac{\alpha_{in}}{2} \cdot \cos \frac{\alpha_{in}}{2} \right] \quad (45)$$

Centroidal radius of the shell from the spiral casing, $R_c = 0.343$ m

Diameter of the spiral casing inlet, $D_{in} = 0.250$ m

Radius up to the end of the transient piece from the spiral casing, $R_h = 0.125$ m

Diameter of extension to penstock, $D_e = 0.253$ m

Number of casing sections, $n = 19$

Draft Tube Design

The shape of the draft tube is that of a diffuser, and the description of a draft tube from a fluid flow perspective is one of the most challenging things to do. This is due to the interplay of many complex flow features such as unsteadiness, turbulence, flow separation, curved streamlines, secondary flow, swirl, and vortex breakdown.

A low-pressure region, which the turbine can utilise, is generated by the draft tube beneath the runner. The static pressure drop and dynamic pressure drop h_s and $\frac{\alpha_1 V_2^2}{2g} - \frac{\alpha_2 V_3^2}{2g} - h_f$ respectively constitute the lower pressure. The static pressure drop is not influenced by the discharge, whereas an increase in flow rate generally leads to an increase in dynamic pressure drop. The drop in dynamic pressure can be achieved by increasing the wall angle of the diffuser and/or by increasing the length of the draft tube. In the two cases, however, there will be an increase in

hydraulic losses. Therefore, there is an optimal angle of the diffuser and length for which reduction of pressure below the runner will be maximal for an efficient draft tube.

A practical method used in determining the draft tube efficiency (conical type only) is to calculate the recovery head H_r which is divided by the actual head loss to determine the efficiency of the draft tube (Bhattacharjee *et al.*, 2021). That is,

$$H_r = \frac{(V_2^2 - V_3^2) - 0.25V_3^2}{2g} \quad (46)$$

Efficiency;

$$\eta_{cp} = \frac{H_r}{\frac{V_2^2 - V_3^2}{2g}} \quad (47)$$

Factors that affect the outlet velocity, pressure, and consequently the efficiency of the turbines are the Draft tube length along the axis, the diameter D_3 at the outlet, and the divergence angle θ (See Figure 12) are the variables that influence the outlet velocity, the developed pressure, and consequently the efficiency of a hydro turbine. Several L/D_3 factors were tested, and it was notable that the draft tube achieved the best range of efficiencies at $L/D_3=2$, $\theta = 6^\circ$ where L is not greater than $\frac{1}{2}H$.

At $L = 0.6$ m, $\theta = 6^\circ$,

$$D_3 = 2L \tan \theta + D_2 = 0.33\text{m}$$

$$L/D_3 = 1.83 \sim 2 \text{ (satisfactory)}$$

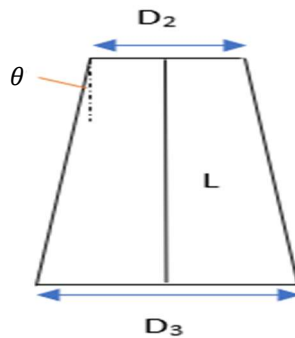


Figure 12. Important Draft Tube Parameters

Using these calculated parameters, the CFTurbo software provides the inlet and outlet velocity along the meridional contour, as shown in Figure 13. The red region on the left represents the draft tube inlet, where the flow exhibits relatively high velocity and pressure. The flow velocity decreases substantially

toward the outlet; this is shown in blue. This permits effective pressure recovery and thus improved overall efficiency of the turbine. From Figure 13,

$$V_2 = 2.95 \text{ m/s}; V_3 = 0.64 \text{ m/s}$$

Substituting these into (46) and (47),

Recovery Head $H_r = 0.41m$; Efficiency $\eta_{cp} = 0.98$

Experimental Setup and Testing

The experiments to characterize the performance of the fabricated turbine were conducted under various

operating conditions, and the data collected included the power output of the turbine, its volumetric flow rate, and the rotational speed of the shaft. The data were analyzed to identify patterns and correlations among the operating parameters, and the performance of the turbine was evaluated in terms of efficiency and power output.

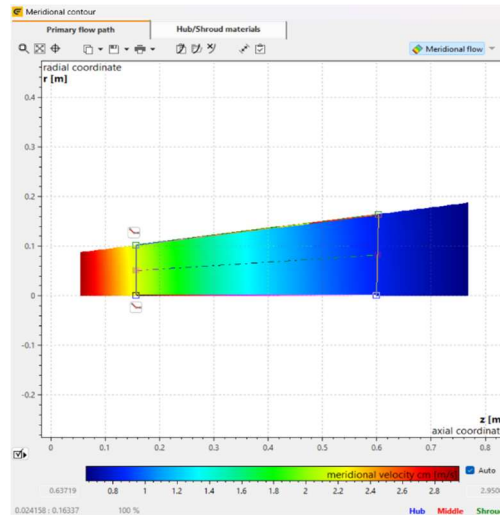


Figure 13. Distribution of Velocity in the Draft Tube (Result of Simulation from CFTurbo)



Figure 14. Axial-flow propeller-type turbine with frame

The turbine (Figure 14) was temporarily installed at the spillway of the Obafemi Awolowo University dam. It was properly aligned and rigidly secured between the penstock and the spiral casing inlet, and the turbine shaft was connected to the alternator, as shown in Figure 15. The stability of the turbine and

the correct installation head were ensured. The following measurements were taken during the tests:

1. Mechanical measurements such as discharge Q at the penstock and shaft rotational speed N at the corresponding flow rate. The shaft speed was measured with a tachometer under a

constant operating head H throughout the experiment.

2. The alternator voltage reading as well as the current drawn by a load connected across it were taken using a digital multimeter placed as

part of the circuit. This happened at the specified flow rate Q_1 and corresponding rotational speed N_1 .



Figure 15. Performance Testing of Axial-flow Turbine

Results and Discussion

Test Results and Calculations

1. Head Losses

There are head losses in the installation setup, such as the intake loss at the inlet of the funnel h_i contraction loss at the funnel-pipe connection h_c through the penstock and at the inlet of the spiral casing (neglected). The gross head H_g measured is 0.55 m, and the net head H_n can be determined from

$$H_n = H_g - h_i - h_c \quad (48)$$

$$h_i, h_c = k \frac{v^2}{2g} \quad (49)$$

($k = 0.25$ for intakes, $k = 0.09$ for contractions)

$$H_n = 0.55 - 0.025 - 0.009 = 0.516 \text{ m}$$

($v = 1.4 \text{ m/s}$)

2. Flow Rate Measurements, Shaft Speed, and Turbine Performance

The flow rates were obtained using the flow velocity and penstock area, while the corresponding shaft speeds were measured using a tachometer. Electrical voltage and current were measured using a digital multimeter. At the first operating condition, the measured electrical output was 15.15 W.

Efficiency losses can be significant when conventional generators are operated at rotational speeds considerably below their rated operating speed, particularly in low-speed micro-hydropower applications (Wei *et al.*, 2020; Safdar *et al.*, 2020). Efficiency is expected to drop from 60-70% at 1500 RPM to 30-40% at 150 RPM. Assuming an alternator efficiency of 35% at 174 rpm, the mechanical power input to the alternator is

$$P_1 = P_m = \frac{P_e}{\eta_m} \quad (50)$$

According to the Similarity law, the power coefficient

$$K_p = \frac{P}{\rho N^3 D^5} \quad (51)$$

where K_p is the power coefficient. Hence, for constant fluid density and runner diameter,

$$\therefore \frac{P_1}{N_1^3} = \frac{P_i}{N_i^3} \quad (52)$$

where P_1 and N_1 represent the turbine power and runner shaft speed at the first operating condition, while P_i and N_i represent the corresponding values at the subsequent operating conditions. The turbine efficiency at each operating condition was then determined from

$$\eta_i = \frac{P_i}{\rho g_i H_n} \quad (53)$$

where P_i , Q_i , and n_i are the turbine power, flow rate, and efficiency, respectively, at each operating condition. The value of torque was then calculated as follows.

$$P = \frac{2\pi NT}{60}$$

The measured and calculated performance parameters are summarised in Table 3. As operating conditions approach the design point, turbine efficiency improves because the flow enters the runner at a more optimal angle and the increased rotational speed enhances alternator performance.

The efficiencies obtained under these test conditions can be compared with the results obtained by Chaulagain *et al.* (2024) for an ultra-low head turbine. They developed a turbine with a maximum efficiency of 53.8% for a head of 0.5m and discharge of 0.088 m³/s. The turbine in this study has reached a maximum efficiency of 70.2% at a net head of 0.516 m and at a flow rate of 0.035 m³/s. Deviation from design efficiency is due to off-design conditions such as reduced flow rate, fabrication deficiencies, non-optimal blade angle, and real fluid effects. Figure 16 shows the turbine being tested with a 60 W light bulb.



Figure 16. Powering a 60 W light bulb using the turbine-driven alternator

Table 3. Performance results for the axial-flow turbine

Parameter	1 st Reading	2 nd Reading	3 rd Reading
Net Head, H (m)	0.516	0.516	0.516
Flow rate, Q (m ³ /s)	0.0140	0.0175	0.0350
Flow rate, Q (L/s)	14.0	17.5	35.0
Shaft Speed, N (rpm)	174	189	248
Turbine Power, P (W)	43	55	125
Efficiency, η	60.7	62.3	70.2
Torque, T (N. m)	2.36	2.78	4.80

Characterisation of the Turbine

The universal characteristics chart for the turbine is a graphical representation of the relationships between speed and discharge characteristics, plotted at constant values of runner diameter $D_1 = 0.21$ m and net head $H_n = 0.52$ m. The variables N' , Q' are plotted on the X and Y axes, respectively. Other quantities, such as the opening B of the wicket gate and efficiency η , were also plotted; the flow coefficient and cavitation coefficient were not included. Using the recalculation equation, one can

predict the required parameters for plotting the turbine characteristics under various operating conditions and for assessing the turbine's performance.

$$Q' = \frac{Q}{D^2 \sqrt{H_n}} \quad (54)$$

where $1/D^2 \sqrt{H_n}$ is constant

$$\therefore Q_{11} = Q_1 K_y \quad (55)$$

Also,

$$N' = ND/\sqrt{H_n} \quad (56)$$

where $D/\sqrt{H_n}$ is constant

$$\therefore N_{11} = N_1 K_x \quad (57)$$

Substituting the measured values into (55) and (57), $K_y = 1.25$ and $K_x = 1.09$

The chart shows the relationships among the turbine parameters, enabling analysis of the performance of the turbine. The curves represent the turbine's efficiency, power output, volumetric flow, and speed characteristics.

In the universal characteristics chart for the axial-flow hydro turbine that is presented in Figure 17, the efficiency at a volumetric flow rate of 70 l/s is 79%, which is less than the 84% from the design calculation. This deviation is attributed to the reduced head at which the turbine operates, which reduces pressure in the system. Observing the hill

chart, one can observe that for rotational speeds below 250 rpm, the efficiency is higher at low discharges than at higher rotational speeds (>250 rpm). This condition is evident for flow rates $Q' = 40\text{--}70$ l/s, where efficiency ranges from 75% to 90%. At higher flow rates $Q' = 120\text{--}150$ l/s, efficiency decreases (less than 75%). The efficiency of the turbine decreases at higher flow rates due to incidence losses, viscous effects, turbulence, and vortex formation within the impeller. At 400 rpm, flow rates of 40–70 l/s yield efficiencies below 60%, while 120–150 l/s achieve above 60%. Overall, the chart shows efficiency rising with flow rate up to about 90–110 l/s, after which it begins to decline. Subekti *et al.* (2025) reported a similar trend in their experimental study of a very low head propeller turbine. They found that the turbine efficiency increased with increasing flow rate up to an optimum condition, and beyond that, the turbine efficiency decreased with further increase in flow rate.

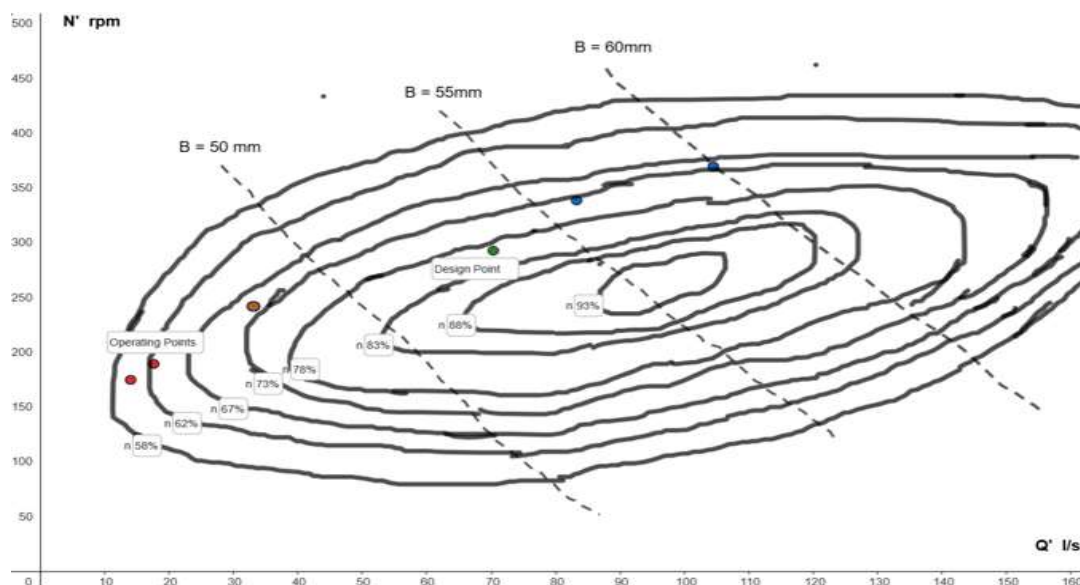


Figure 17: Universal characteristics of the fabricated axial-flow (propeller-type) turbine ($D_1 = 210$ mm, $H = 0.52$ m).

Conclusion

It has been demonstrated sufficiently through this work that there exists the local capacity to design and construct an axial-flow hydraulic turbine (propeller-type) which employs simple hydraulic technology in its design, construction, operation and maintenance. The axial-flow hydraulic turbine is suitable for meeting the electricity demand of riverine communities that are not connected to the

national grid system. The turbine is suitable for low-head water and “run-of-river” flow. In this work, the turbine was designed for 1kW, with a head of 1.5m and a flow rate of 0.07m³/s. Design calculations were based on an overall efficiency of 84% and a shaft rotational speed of 800 rpm. However, the maximum turbine output power obtained during actual testing at the spillway of Opa Dam at Obafemi Awolowo University was 54W at 189 rpm and a

flow rate of 0.0205 m³/s, giving an efficiency of 61%. The power reduction was due mainly to the fact that the tests were conducted at an off-design duty point necessitated by insufficient water flow rate and the available low head of 0.55 m at the spillway during the period of the season when the tests were carried out. Future work should focus on the design and fabrication of a suitable mechanical or electronic governor to regulate the flow rate to allow the turbine to respond adequately to varying load demand and not peak load only.

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