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Innovative design of crossflow hydro turbine system for Hhaynu micro-hydropower plant – Mbulu, Tanzania

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Abstract

Micro-hydropower plants are very applicable in rural and off-grid areas where water resources become available. This is because they can be installed with fraction of the cost as compared to large hydropower plants or even grid extension. Also, in rural and off-grid areas, the population density is small and very sparsely distributed which makes it un-economical for the development of large electricity supply projects. In this case the mini and micro energy projects are the suitable technology to be used to supply power to the consumer load demand in the rural and off-grid areas. One of the sustainable ways to do is to use the available water resources like small rivers to develop micro-hydropower plants. The effective use of water from local rivers to develop micro-hydropower plants have proven to be sustainable way of electricity generation. But despite all these positive outcomes, studies have shown that many of the available rural areas micro-hydropower potential sites are facing reduced water volumetric flow due to irrigation activities and also lack high site heads due to the nature of the landscape. In this case, the development of a micro-hydropower for electricity generation is limited to specific type of hydro turbine technology called crossflow turbine. This is because this type turbine technology can accommodate wider range of flow discharge and head values in the micro and mini scale of hydropower technology range. The crossflow turbines can also be developed locally and adapted to the local rural environments and also have been proven to be very robust with less operational and maintenance costs. Thus, why there is a need to customize this technology in the local rural area in terms of turbine design in order to standardize the local manufacturing and this is the main motivation that this design study have been addressing.

Keywords: *Crossflow, Hhaynu, Micro-hydropower, Hydro turbine, Generator, Electricity, Mbulu, Tanzania*

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1. Introduction

The type of turbine to be used in Hhaynu micro-hydropower project has been selected based on the data obtained from the site area. In this case, two site measured parameters are required for the determination of turbine type selection and these parameters are the available volumetric flow of the river (Q) and site head (H). On the other hand, in literature, there are two general types of a water turbines, Reaction turbine and Impulse turbines on which each type fits on specific site characteristic for it to be selected for a particular application. In terms of their characteristics reaction turbines requires a substantial amount of water flow discharge while impulse turbines are much more dependable on medium and high head site locations [1].

From the site feasibility study report, the determined volumetric flow for Hhaynu river is 0.9 m^3/s on which the results from the hydrological study shows that only $0.45m^3/s$ (50% flow) is the feasible discharge flow for the micro-hydro turbine system on which it will make provision for environmental flow during the dry season. In addition to that, the maximum gross head obtained after taking the site measurements is 25m and this is the distance from the forebay area to the power house location[2]. Using these two values which have been obtained from the site measurements, then the type of turbine technology to be used for this research project is obtained by using the following two diagrams in Figure 1.1 and Figure 1.2.

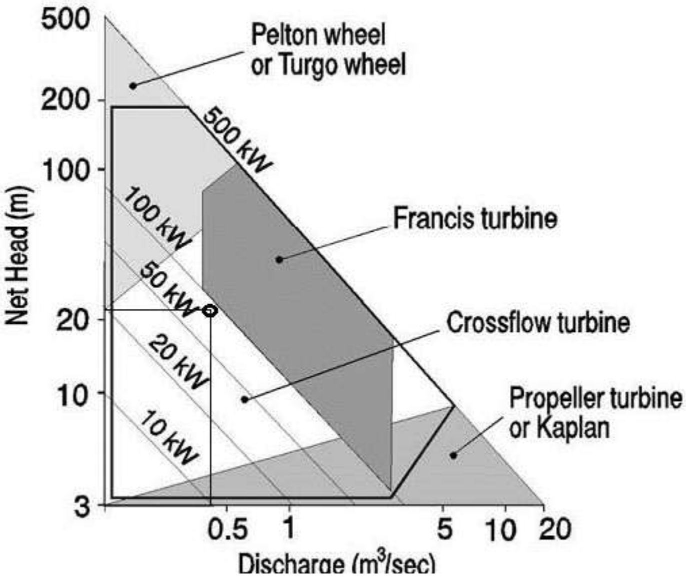

Figure 1.1: Turbine selection chart based on head and flow discharge [3]

As shown in Figure 1.1 the cross-reference lines for the site measured values of the net head of 22.71 m (25 m gross head) and turbine discharge of 0.45 m^3/s with the 79.5 kW turbine mechanical power which falls under crossflow turbine category and therefore this type of turbine technology will be used in this research project. Alternative turbine selection chart has been used based in Figure 1.2 below which also shows similar results for the selected crossflow turbine technology. Both charts show that crossflow turbines have a wider range of head and flow discharge values from low to medium values, thus why they are mostly applicable to micro-hydropower systems in developing countries like that of Hhaynu micro-hydropower plant in Mbulu, Tanzania as shown in Figure 1.2.

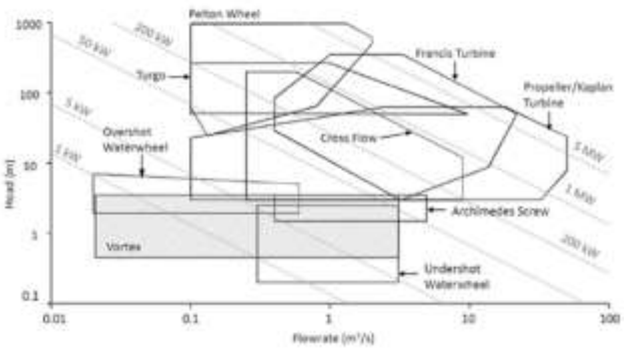

Figure 1.2: Turbine application range chart with the crossflow turbine [4]

2. Material and Methods

2.1 Turbine transmission

The micro-hydro turbine transmission involved the turbine rotational speed in RPM and also transmission speed ratio between the turbine and generator pulleys.

2.1.1 Turbine speed

Cross flow turbines use pulley and belt as the transmission drive from the turbine to the generator unit which means that the turbine speed is usually lower than the generator speed. In calculating the generator speed the following formula is used[5].

$$N = 513.25 \text{ x } \frac{(\mathrm{Hg})^{0.745}}{\sqrt{\mathrm{Pm}}} \qquad (1)$$

Where; N = Turbine speed in RPM, Hg = Gross head (m) and Pm = Mechanical/Turbine Power (kW)

So, substituting the values we get;

$N = 513.25 \text{ x } \frac{(25)^{0.745}}{\sqrt{79.5}} = 513.25 \text{ x } 1.234$

N = 633.30 RPM

Choose the standard turbine speed to be **635 RPM**

2.1.2 Turbine – Generator Speed Ratio

Speed ratio = $\frac{\text{Generator speed}}{\text{Turbine speed}} = \frac{1500\,\text{RPM}}{635\ \text{RPM}}$ = 2.3622

Thus, the Turbine – Generator speed ratio is **1:2.3622**

Table 1: Summary of main turbine design values

S/N	Parameter	Value	Unit	Remarks
1	Turbine speed	635	RPM	Standard value
2	Speed ratio	1:2.36	-	-
3	Turbine torque	1.5	kNm	-
4	Turbine shaft diameter	60	mm	Standard value

2.2 Turbine parameters

2.2.1 Turbine shaft diameter

The turbine shaft diameter is calculated using the Power – Torque equation as follows;

$$P = T.\omega \qquad (2)$$

Where; P = Mechanical Power, T = Torque, ω = Rotational speed in radians per second and N = speed in revolution per minute (RPM)

Making T the subject gives the following;

T = $\frac{P}{\omega}$ and ω is given as $\frac{2 \times \pi \times N}{60}$

So, T = $\frac{P \times 60}{2 \times \pi \times N} = \frac{79.5 \times 10^3 \times 60}{2 \times \pi \times 635}$ = 1,196.154 NM

Then, T_{max} = 1.25 x T = 1.25 x 1,196.154 = **1,495.1928 Nm**

Also, the torque can be calculated using the following shear stress equation as follows;

$$T = \frac{\pi d^3 \tau_s}{16} \qquad (3)$$

Where: τ_s = shear stress of steel = 42MPa

Re-arranging the equation gives the following;

$d^3 = \frac{16 \times T}{\pi \times \tau_s} = \frac{16 \times 1{,}495.1928}{3.14 \times 42 \times 10^6} = 1.814 \times 10^{-4}$

d = 0.0566m = 56.61mm = ≈**60mm**

So, the diameter of the turbine shaft is **60mm**

Figure 1.3: Turbine shaft diameter

2.2.2 Turbine runner outer diameter

The outer diameter of the turbine runner is calculated using the following equation;

$D_o = \frac{40 \times \sqrt{H}}{N} = \frac{40 \times \sqrt{25}}{635}$ = 0.31496m

D_o = **315mm**

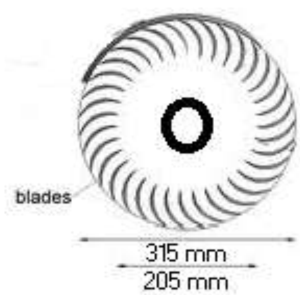

Figure 1.4: Turbine runner outer diameter

2.2.3 Turbine runner length and runner blade spacing

The turbine runner length is given using the following equation;

$$L = \frac{QxN}{50xH} \qquad (4)$$

$L = \frac{0.45 \text{ x } 635}{50 \text{ x } 25}$ = 0.2286 m
Turbine length, L = 228.6 mm $\approx$ **230 mm**

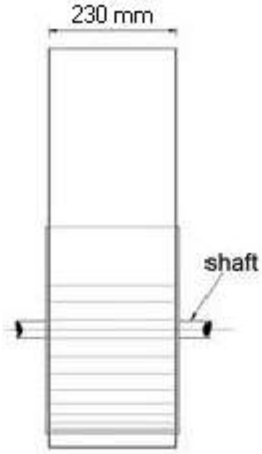

Figure 1.5: Turbine runner length

Turbine runner blade spacing:
Blade jet entrance (te) = K. D_o
Where; K = constant value = 0.087
te = 0.087 x 315mm = 27.405mm $\approx$ 28 mm
Take standard blade spacing value of
te = $\approx$ **30 mm**

2.2.4 Turbine runner tangential spacing and runner blade number
Tangential spacing (tb) = 0.174 x D_o
tb = 0.174 x 315mm = 54.81mm ≈55mm
Take the standard tangential spacing value, tb = **55mm** [This is also called Radial Rim Width (a)]

Turbine runner blade number:
Minimum number of runner blade (Z) = $\frac{\pi \text{ x Do}}{\text{tb}} = \frac{3.14 \text{ x } 315}{55}$ = 17.98 $\approx$ 18
But, the recommended number of runner blade should be at least 30% more than the calculated value in order to make it effective, so in this case the design number of runner blades,
Z = ($\frac{1}{3}$ x 18) + 18 = 24
Thus, the turbine runner blade number is **24**

2.2.5 Water jet thickness and nozzle area to the runner
Water flow jet thickness (tj) = 0.29 x D_o
tj = 0.29 x 315mm = 91.35mm $\approx$ **92 mm**
So, the water jet thickness to the turbine entry is **92mm**

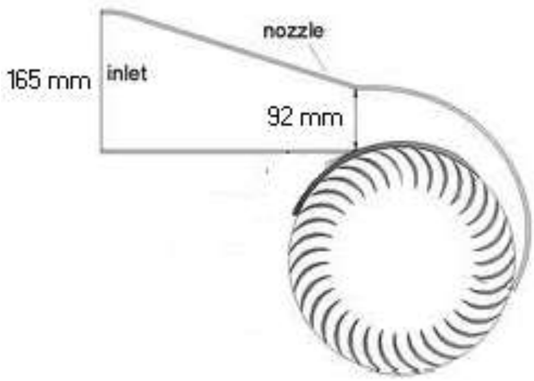

Figure 1.6: Turbine nozzle area

2.2.6 Nozzle area
Area = L x W
Area = 0.23 x 0.092 = 0.02116 m^2
Area = 2.116 x $10^{-2}$$m^2$
Size of penstock pipe diameter to connect the nozzle
The area is given using the following formula,

$$\text{Area (A)} = \frac{\pi \times d^2}{4} \quad (5)$$

Re-arranging the equation gives, $d^2 = \frac{4 \times A}{\pi} = \frac{4 \times 2.116 \times 10^{-2}}{3.14} = 2.6955 \times 10^{-2}$

D = 0.16418m = **164.18mm**

So, the size of the penstock pipe diameter to connect to the turbine nozzle is **164.18mm (take 165 mm)** which is equivalent to **6.5”** pipe size.

2.2.7 Inner diameter of the turbine runner and radius of blade curvature

Inner diameter of the runner (Di) = D_o – 2 x radial rim width

Di = 315 – 2 x 55 = **205mm**

Radius of blade curvature:

Radius of the blade curvature (Rc) = 0.163 x D_o

Rc = 0.163 x D_o = 0.163 x 315mm = 51.345mm

Rc = ≈ **52 mm**

So, the diameter of the pipe for blades = 2 x Rc = 2 x 52mm = **104mm**

Thus, the selected value of the blade curvature should be from class C steel pipe of size 4”.

Alternatively, $D_o = D_i + 2Rc$ which gives $Rc = \frac{Do - Di}{2}$

2.2.8 Water jet velocity and turbine arc length

The water jet velocity is calculated on the impulse turbines (Crossflow and Pelton) using the following equation;

$$V_j = \sqrt{2gH} \quad (6)$$

$Vj = \sqrt{2 \times 9.81 \times 25} = \sqrt{490.5} = 22.147$ m/s

Vj = **22.147m/s**

Thus, the water jet velocity, Vj is **22.15m/s**

Turbine arc length:

The arc length (S_1) = rQ = $\frac{\pi \times Rc \times Q^0}{180}$

$S_1 = \frac{3.14 \times 52 \times 72^0}{180}$ = **65.312 mm**

S_1 = ≈ **66 mm**

Pipe cut for turbine blades:

Size diameter 104 mm (4”)

Then, C = π x D = 3.14 x 104 = 326.56 mm

So; $\frac{326.56}{S1} = \frac{326.56}{66}$= 4.95 pieces

Note: Cut 4 pieces of 4” diameter steel pipe to make the turbine blades.

2.2.9 Water Pressure and Force to the turbine nozzle

In water power systems, the power inside the turbine is given by;

$$P_t = pQ \quad (7)$$

Where: p = pressure (kN/m^2), Q = Flowrate (m^3/s), P_t = Turbine (kW)

Then, $p = \frac{Pt}{Q} = \frac{75.5 \text{ kW}}{0.45 \, m^3/s}$ = **166.67kN/m^2**

But pressure (p) = $\frac{F}{A}$ when re-arranging this equation gives the following

F = p x A

But, Area = L x W

A = 0.23 x 0.092 = **0.02116 m^2**

So; Force (F) = p x A = 166.67 x 0.02116 = 3.5267 kN

Therefore, the Force exerted to the turbine runner through the nozzle, **F = 3.53 kN**

Alternatively, Power (Pt) = Water Force (Fw) x Velocity (Vj)

Force (F_w) = $\frac{Pt}{Vj} = \frac{75.5 \text{ kW}}{22.147 \text{ m/s}}$ = **3.386 kN**

Then the distributed force at the turbine entry = F_1 = F/L = 3,530N/m x 0.23

Therefore, the distributed water force on the turbine = **811.9N**

2.3 Penstock pipe calculations

The penstock is the steel pipe that conveys water with pressure from the forebay to the turbine unit in the power house. The water in the forebay is stored in a form of potential energy and when delivered to the turbine through a penstock pipeand produce kinetic energy which rotates the hydro turbine. On the other hand, the penstock steel pipe wall thickness depends on the pipe materials, diameter and operating pressure and is calculated as follows:

2.3.1 Penstock diameter and thickness[6]

$$D_p = 2.65 \text{ x } (n^2 \text{ x } Q^2 \text{ x } L_p/H_g)^{0.204} \qquad (8)$$

Where: D_p = penstock diameter,
n = manning coefficient for the mild steel penstock pipe = 0.012
Q = design flow discharge = 0.45 m^3/s
L_p = length of the penstock = 162 m
H_g = Gross head = 25 m
So, in this case;
$D_p = 2.65 \text{ x } ((0.012)^2 \text{ x } (0.45)^2 \text{ x } 162/25)^{0.204}$
$= 2.65 \text{ x } (0.000144 \text{ x } 0.2025 \text{ x } 6.48)^{0.204}$
$= 2.65 \text{ x } (1.88956 \text{ x } 10^{-4})^{0.204} = 0.460918$ m
D_p = **460 mm**
Therefore, the size of the penstock pipe selected to supply the design discharge to the turbine is **460mm** dimeter (**18”**)
From literature, the minimum wall thickness is given by the following equation) [7]

$$t_p = \frac{D_p + 508}{400} + 1.2 \text{ mm} \qquad (9)$$

t_p = minimum penstock thickness (mm)
D_p = penstock diameter = 460 mm
So,
$t_p = \frac{460\text{mm} + 508}{400} + 1.2$ mm
t_p = 2.42 mm+ 1.2 mm
t_p = 3.62 mm = ~ 4mm
Therefore, the thickness of the steel penstock pipe selected is **4 mm**

2.3.2 Water flow velocity in the penstock

The water flow discharge to the turbine passes through the penstock pipe and is given by the following equation:

$$Q = AV \qquad (10)$$

Where;
Q = water flow discharge, A = penstock pipe area and V = water flow velocity
Making V the subject from the above equation gives;
$V = \frac{Q}{A} = \frac{4 \text{ x } Q}{\pi d^2}$
$V = \frac{4 \text{ x } 0.45}{\pi \text{ x } (0.46)^2} = \frac{1.8}{0.664424} = 2.7091$
Water flow velocity V in the penstock is **2.7 m^3/s**
(Recommended water flow velocity in the penstock pipe range between 1 m/s – 2.8 m/s) [8]

2.3.3 Area of the Penstock pipe

Area of the penstock pipe is given by the following equation;
$A = \frac{\pi \text{ x } (d)^2}{4} = \frac{\pi \text{ x } (0.46)^2}{4} = 0.166$
Therefore, the computed area of the penstock is **0.166 m^2**

2.3.4 Penstock head loss

Head loss in the penstock can be calculated using the following equation:

$$\text{Head loss} = \left[\frac{10 \text{ x } n^2 \text{ x } Q^2}{D_p^5}\right] \text{x Lp} \qquad (11)$$

Where; n = Manning value, Q = Design flow discharge, Dp = Diameter of the penstock and Lp = Length of the penstock
Head loss = $\left[\frac{10 \text{ x } (0.012)^2 \text{ x } (0.45)^2}{(0.46)^5}\right]$ x 162 = 0.014158 x 162 m = 2.294m
Head loss = **2.29m**
Therefore, the net head is given;

Net head = Gross head – head loss = 25m – 2.29m = 22.71 m
Net head is **22.71 m**

2.3.4.1 Percentage of head loss

% Head loss = $\frac{\text{Head loss}}{\text{Gross head}} = \frac{2.29\text{m}}{25\text{m}}$ = 0.0916
Head loss = **9.16 %**
(The recommended economical head loss for most small hydropower schemes should be below 10%) [9].

Table 2: Summary of penstock pipe design values

S/N	Parameter	Value	Unit	Remarks
1	Penstock diameter	460	mm	18" Standard value
2	Water velocity in the penstock	2.7	m/s	1m/s – 2.8m/s so, OK
3	Area of the penstock pipe	0.166	m^2	-
4	Penstock head loss	2.29	m	-
5	Percentage of head loss	9.16	mm	< 10% so, OK

2.4 Pulley and belt design

In order to calculate the pulley and belt parameters, the following standard values have been selected based on the turbine and generator design calculations:

Turbine design parameters:

(a) Turbine speed = 635 RPM
(b) Turbine Power = 79.5 kW
(c) Turbine operation time 24 hours

Generator design parameters:

(a) Generator speed = 1500 RPM
(b) Generator Power = 75.5 kW
(c) Synchronous, 3 phase, 90 KVA with power factor 0.8
(d) Generator pulley diameter = 150mm (OK because > 140)
(e) Generator operation time 24 hours

2.4.1 Turbine pulley design

2.4.1.1 Turbine pulley diameter using speed ratio

In calculating the turbine-generator speed ratio the turbine and generator speed and diameter ratios need to be considered.

Speed ratio = $\frac{N_1}{N_2} = \frac{d2}{d1}$

Speed ratio = $\frac{N_1}{N_2} = \frac{d2}{d1} = \frac{635}{1500} = \frac{150}{d1}$

$d_1 = \frac{150 \ x \ 1500}{635}$ = 354.33mm

Therefore, the pulley diameter d_1 of the turbine side is **354.33 mm** as shown in**Table 3**.

Table 3: Generator and Turbine pulley dimensions.

Generator Pulley diameter (mm)	Turbine pulley diameter (mm)	Closest standard Turbine pulley diameter (mm)	Turbine speed (RPM)
150	354.33	355	635

2.4.1.2 Sizing of the belt length

The belt length is determined using the following relation

$$\text{Belt length (L)} = 2C + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4C} \quad (12)$$

Where: C = centre distance and d = diameter of pulley

$L = 2 \text{ x } 505 + \frac{\pi(505+150)}{2} + \frac{(355-150)^2}{4 \text{ x } 505}$

L = 1802.85mm + 20.8045 mm
L = 1823.654 mm

The closest standard v-belt length is SPB wedge belt SPB1825 with 17mm top width and 13mm height.

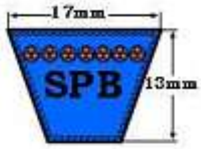

Figure 1.7: Designed V-belt section

2.4.1.3 Turbine pulley belt speed
The turbine velocity is given with the following equation:

$$\text{Velocity} = \frac{d}{2} \text{ x } \omega \qquad (13)$$

Where; ω is the rotational turbine speed given as; $\omega = \frac{2\pi N}{60}$
So;
Velocity = $\frac{0.355}{2}$ x π x $\frac{633.80}{30}$ = 11.77 m/s (OK because < 40)
Therefore, the belt velocity of the turbine pulley is **11.17 m/s**
2.4.1.4 Belt centre distance
Centre distance C is calculated as follow;
C = d + D = 150mm + 355mm = 505mm
But the exactly centre distance can be calculated using the following formula;

$$C = A + \sqrt{A^2 - B} \qquad (14)$$

But $A = \frac{L}{4} - \frac{\pi}{8}(d + D)$ and $B = \frac{(D-d)^2}{8}$
Taking L = 1825mm and substituting the values to the above A and B equations gives;
$A = \frac{1825}{4} - \frac{\pi}{8}(150 + 355) = 258.0375$mm
$B = \frac{(355-155)^2}{8} = 5{,}253.125$mm 247.
Then, $C = 258.037 + \sqrt{(258.037)^2 - 5253.125} = 505.6869$mm
Therefore, the actual centre distance for the belt is **505.69mm**
2.4.1.5 Power per belt
Interpolating from the given Table on page 63 [10]
- Rated power per belt = (20.75 + 23.56)/2 = 22.155 kW
- Additional power = (1.50 + 1.75)/2 = 1.625 kW

Therefore, the basic power per belt = 22.155 kW + 1.625 kW = **23.78 kW**
2.4.1.6 Correction power per belt
From Table on page 58 the combined correction factor is 0.85[10]
Therefore, correction power per belt = 23.78 kW x 0.85
Correction power per belt = **20.213 kW**
2.4.1.7 Number of belts
Number belt for the belt design is given as follow;
Number of belts = $\frac{\text{Total power}}{\text{Correction power per belt}} = \frac{75.5 \text{ kW}}{20.213 \text{ kW}}$ = **3.735**
Therefore choose **4 belts** as standard
2.4.1.8 Net driving force
The net driving force from the turbine pulley is given from the following equation;
$F_d = F_f - F_b$
The above equation can also be derived from the torque formula as;

$$\text{Torque} = F_d \text{ x } r \qquad (15)$$

where r = radius of the turbine pulley (177.5mm) and T = Torque (1,495nm)
Then, $F_d = \frac{T}{r} = \frac{1{,}495.185\text{nm}}{0.1775\text{m}} = 8{,}423.58$N
Therefore, the driving force for the turbine pulley system is **8.423 kN**

Table 4: Summary of pulley and belt drive design values

S/N	Parameter	Value	Unit	Remarks
1	Turbine pulley diameter	355	mm	Standard value
2	Generator pulley diameter	150	mm	
3	Length of the belt	1825	mm	**SPB1825**
4	Belt velocity	11.7	m/s	< 40 so, OK
5	Belt centre distance	505.69	mm	-
6	Number of belts	4	pcs	-
7	Net driving force	8.42	kN	-

2.5 Electrical system components

The electrical components consist of the generator system and transmission system. These components are related to the electrical power produced and transmission to the consumers.

2.5.1 Generator system

The generator system will be operated with the following rated design values:
Frequency, f = 50 Hz
Rated power, Pe = 75.5 kW
Efficiency, eff = 95%
Rated speed, N = 1500 RPM
Number of poles, p = 4
Power factor, p.f. = 0.8
The generator power is calculated based on the available turbine mechanical power using the following equation;
Pe = turbine mechanical power x generator efficiency
(Refer to the turbine mechanical power, Pm = $0.45m^3/s$ x $9.81m/s^2$ x 25m x 0.72 = 79.46kW) Therefore, Electrical Power,
Pe = 79.46 x 0.95 = **75.49 kW**
Pe = **75.5 kW**

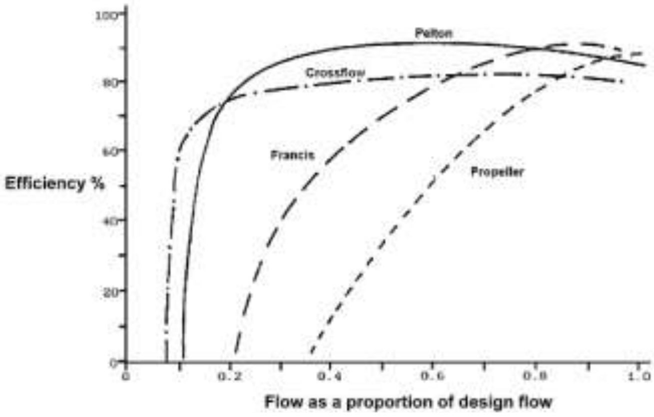

Figure 1.8: Efficiency range with design flow discharge for different turbines[5]

The generator power output will be supplied to the consumer load through the local grid connections in the area. The generator design output value is very much influenced by the turbine output power.
The synchronous speed, ω_s of the generator is related to the speed N and is given as:

$$\omega_s = \frac{2\pi N}{60} \tag{16}$$

At the rated condition N = 1500 RPM
So, synchronous speed $\omega_s = \frac{2\pi \times 1500}{60}$ = 157 rad/s
This can also be calculated using the following equation;
$\omega_s = \frac{4\pi f}{p} = \frac{4\pi \times 50}{4}$ = 157 rad/s
In the actual speed of the generator changes due to load changes and this affects the actual speed of the generator. This speed value is the difference between the actual speed and synchronous speed and is the one that determines the instantaneous generator speed under different loading condition as shown inTable 5.

Table 5: Generator power and speed at different rated capacity

Generator Power (kW)	Generator speed (RPM)	Generator speed (rad/s)	Remarks
75	1500	157	Rated capacity
40	1520	159.09	Minimum load
101.8	1494	156.37	Maximum load

Therefore, the difference in speed (RPM) from the above three scenarios can be calculated as follows;

$$\Delta N = \frac{60}{2\pi}\Delta\omega \quad (17)$$

At low power demand, $\Delta N = \frac{60}{2\pi} x\ (159.09 - 157)$ = **19.97 RPM** (this indicate a generator speeds up which is over speed during low demand)
At high power demand, $\Delta N = \frac{60}{2\pi} x\ (156.37 - 157)$ = **- 6.01 RPM** (this indicate a generator slows down which is under speed during high demand)

3. Conclusion

In designing hydro turbines, many parameters need to be considered on which most of them are obtained from a particular site. Two of the most deterministic parameters to be considered are the design flow discharge (Q) and site head (H) which are obtained from the particular site during feasibility study and data measurements. In additional to that, the two measured parameters can also be used to determine the type of turbine technology to be used in a particular hydropower project by using standard turbine selection charts. On the other hand, when turbine design capacity needs to be determined, the value of flow discharge (Q) and site head (H) are also used as inputs to determine the turbine power using the standard formulas. Thus, using the results from the current study for the turbine design, the flow discharge for Hhaynu micro-hydropower plant was 0.45 m^3/s with the site head of 25m and based on the turbine selection charts, the selected turbine type falls under the crossflow turbine technology. The crossflow turbine technology has wider values of flow discharge and head in the medium range which make them to be widely used for micro-hydropower projects in off-grid rural areas. The crossflow turbines can also be adapted locally with a simple design which results to low in maintenance cost and thus makesmost of the micro-hydropower projects sustainable.

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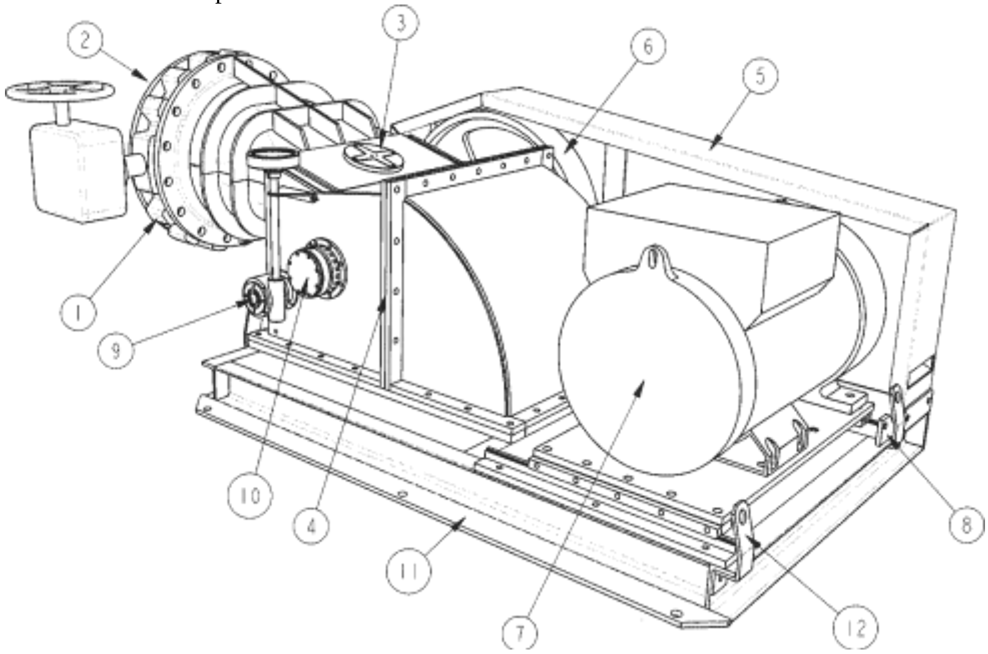
Reference

[1]. Energy.gov. *Water power technologies office; hydropower turbines*. Available from: https://www.energy.gov/eere/water/types, accessed on 20 May 2022.
[2]. Ngoma, D.H. and Y. Wang, *Hhaynu micro hydropower scheme: Mbulu–Tanzania comparative river flow velocity and discharge measurement methods.* Flow Measurement and Instrumentation, 2018. **62**: p. 135-142.
[3]. Paish, O., *Small hydro power: technology and current status.* Renewable and sustainable energy reviews, 2002. **6**: p. 537-556.
[4]. Timilsina, A.B., S. Mulligan, and T.R. Bajracharya, *Water vortex hydropower technology: a state-of-the-art review of developmental trends.* Clean Technologies and Environmental Policy, 2018. **20**(8): p. 1737-1760.
[5]. Penche, C., *Layman's guidebook on how to develop a small hydro site, published by European Small Hydropower Association (ESHA), Second edition, Belgium.* 1998.
[6]. Chitrakar, P., *Micro-hydropower design aids manual.* Small Hydropower Promotion Project (GTZ) and Mini-Grid Support Program, Alternate Energy Promotion Center, Government of Nepal, 2004.
[7]. Nasir, B.A., *Design considerations of micro-hydro-electric power plant.* Energy Procedia, 2014. **50**: p. 19-29.
[8]. Mohibullah, M., A.M. Radzi, and M.I.A. Hakim. *Basic design aspects of micro hydro power plant and its potential development in Malaysia*. in *PECon 2004. Proceedings. National Power and Energy Conference, 2004.* 2004. IEEE.
[9]. Singh, D., *Micro Hydro Power, Resource Assesment Handbook, An Initiative of the Asian Pacific Center for Transfer of Technology*. 2009.
[10]. Wales, N.S., *Technical and Education Commission. Manufacturing and Engineering Educational Services. Bankstown, N.S.W.: Manufacturing and Engineering Educational Services, N.S.W. TAFE Commission, 3rd Edition.* 2000.

Appendix: Crossflow turbine parts



1.Manual / semi-automatic floodgate
2.Connection to the penstock
3.Inspection door
4.Turbine case
5.Guard for pulley and belt
6.Pulleys with a cogged driving belt
7.Generator
8.Generator fixing slide with belt tightening pulley
9.Flow regulation (manual or automatic)
10. Bearing
11. Frame
12. Lifting points