

# Crossflow Turbine Design Specifications for Hhaynu Micro-Hydropower Plant-Mbulu, Tanzania

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## Abstract

Micro-hydropower plants are mostly 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 relatively 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 those rural and off-grid areas. One of the sustainable way to do this 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. However, 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 of 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 manufactured 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; Mbulu; Tanzania

## Introduction

The type of hydro turbine unit that have been installed at Hhaynu micro-hydropower plant was selected based on the data obtained from the site feasibility study. In this case, two site measured parameters have been used 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  $0.9 \text{ m}^3/\text{s}$  on which the results from the hydrological study show that only  $0.45 \text{ m}^3/\text{s}$  (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 25 m and this is the distance from the forebay area to the powerhouse location. 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 and Figure 2 below [2,3].

As shown in Figure 1 below, 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  $0.45 \text{ m}^3/\text{s}$  with the 79.5 kW turbine mechanical power that 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 2 below which also shows similar results for the selected crossflow turbine technology. Both charts shows that crossflow turbines have a

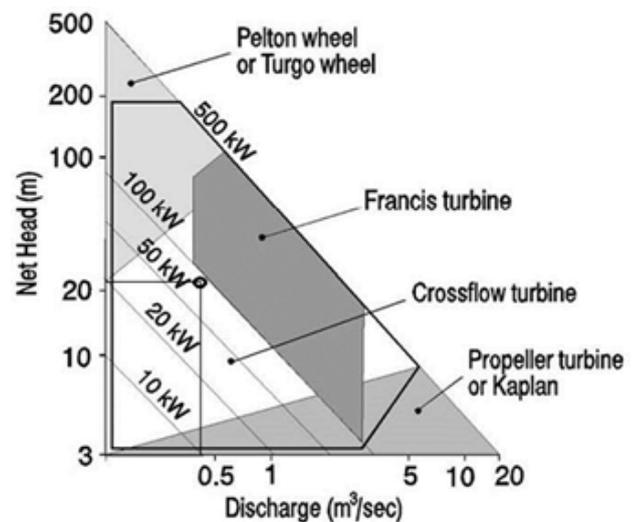


Figure 1: Turbine selection chart based on head and flow discharge

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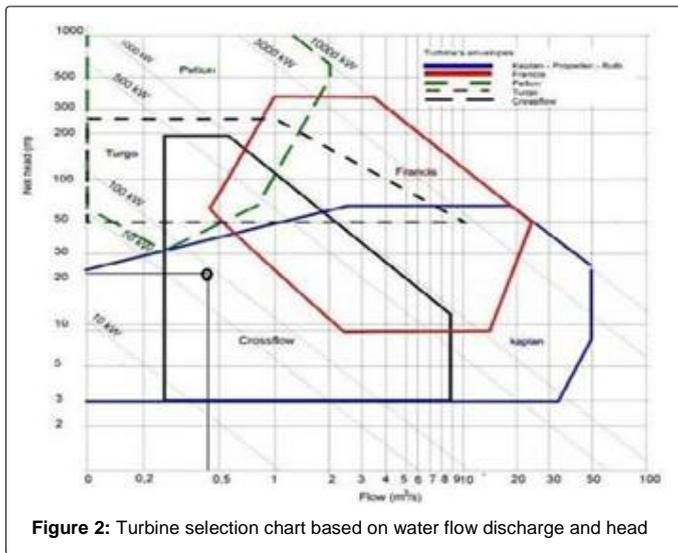


Figure 2: Turbine selection chart based on water flow discharge and head

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 Hhaynumicro-hydropower plant in Mbulu, Tanzania as shown in Figure 2 above.

**Materials and Methods**

**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.

**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 (1) is used [4].

$$N = 513.25 \times \frac{(Hg)^{0.745}}{\sqrt{Pm}} \tag{1}$$

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

So, substituting the values we get;

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

$$N = 633.30 \text{ RPM}$$

Choose the value of the turbine speed to be 635 RPM (satisfying the condition that turbine and generator speed ratio should be between 2-3) [5].

**Turbine- generator speed ratio:**

$$\text{Speed ratio} = \frac{\text{Generator speed}}{\text{Turbine speed}}$$

$$= \frac{1500 \text{ RPM}}{635 \text{ RPM}} = 2.3622$$

Thus, the Turbine- Generator speed ratio is 1:2.3622 (Table 1).

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

Table 1: Summary of main turbine design values.

**Turbine parameters**

**Turbine shaft diameter:** The turbine shaft diameter is calculated using the Power- Torque equation (2) as follows;

$$P = T \cdot W \tag{2}$$

Where; P=Mechanical Power, T=Torque, W=Rotational speed in radians and N=speed in RPM

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

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

$$\text{Then, } T_{\text{max}} = 1.25 \times T = 1.25 \times 1196.154 = 1495.1928 \text{ Nm}$$

In addition, the Torque can be calculated using the following shear stress equation (3) as follows;

$$T = \frac{\pi d^3 \tau_s}{16} \tag{3}$$

Where:  $\tau_s$ =shear stress of steel=42 MPa

Re-arranging the equation gives the following;

$$d^3 = \frac{16 \times T}{\pi \times \tau_s} = \frac{16 \times 1495.19}{\pi \times 42} = 1.814 \times 10^{-4}$$

$$d = 0.0566 \text{ m} = 56.61 \text{ mm} \approx 60 \text{ mm}$$

So, the diameter of the turbine shaft is 60 mm (Figure 3).

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

$$D_o = 40\sqrt{H}$$

$$D_o = 315 \text{ mm (See Figure 4)}$$

**Turbine runner length and runner blade spacing:**

The turbine runner length is given using the

following equation (4);

$$L = \frac{Q \times N}{50 \times H} \tag{4}$$

$$L = \frac{0.45 \times 635}{50 \times 25} = 0.2286 \text{ m}$$

Turbine length, L=228.6 mm  $\approx$  230 mm (Figure 5)

**Turbine runner blade spacing:**

$$\text{Blade jet entrance (te)} = K \cdot D_o \text{ Where; } \tag{5}$$

K=constant value=0.087

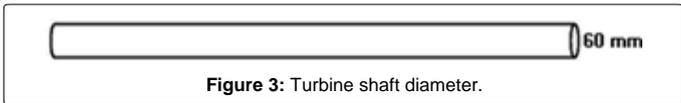


Figure 3: Turbine shaft diameter.

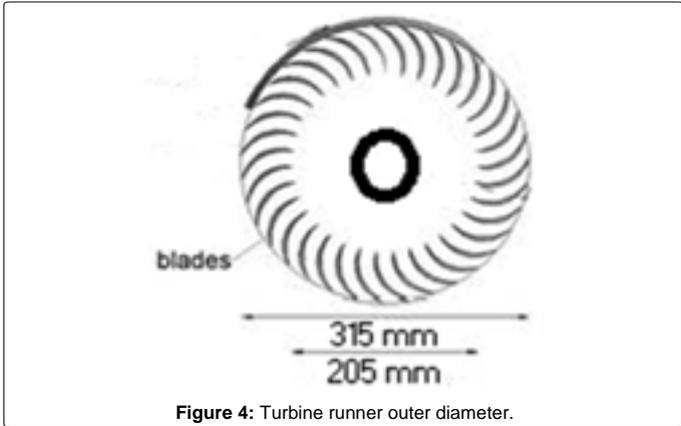


Figure 4: Turbine runner outer diameter.

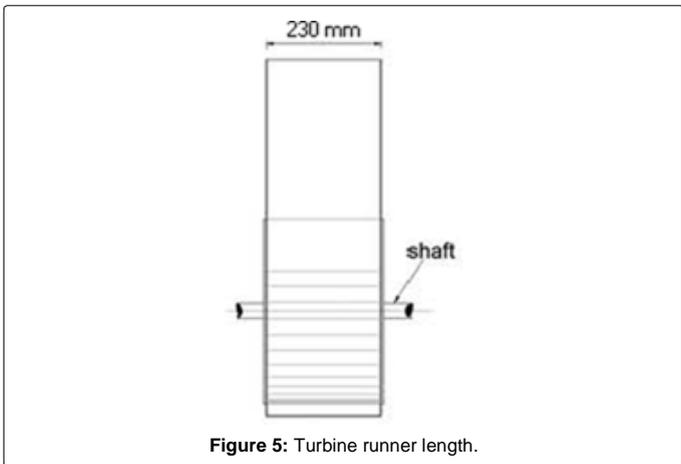


Figure 5: Turbine runner length.

$$t_e = 0.087 \times 315 \text{ mm} = 27.405 \text{ mm} \approx 28 \text{ mm}$$

Take standard blade spacing value of  $t_e \approx 28 \text{ mm}$

**Turbine runner tangential spacing and runner blade number:**

$$\text{Tangential spacing } (t_b) = 0.174 \times D_o \tag{6}$$

$$t_b = 0.174 \times 315 \text{ mm} = 54.81 \text{ mm} \approx 55 \text{ mm}$$

Take the standard tangential spacing value,  $t_b = 55 \text{ mm}$  [This is also called Radial Rim Width(a)]

**Turbine runner blade number:**

Minimum number of runner blade;

$$Z = \frac{\pi \times D_o}{t_b} \tag{7}$$

$$Z = \frac{3.14 \times 315}{55} = 17.98 \approx 18$$

However, 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,

$$Z = \left(\frac{1}{3} \times 18\right) + 18 = 24$$

Thus, the turbine runner blade number is 24

**Waterjet thickness and nozzle area to the runner:**

**Water flow jet thickness:**

$$t_j = 0.29 \times D_o \tag{8}$$

$$t_j = 0.29 \times 315 \text{ mm} = 91.35 \text{ mm} \approx 92 \text{ mm}$$

So, the water jet thickness to the turbine entry is 92 mm (Figure 6).

**Nozzle area:**

$$\text{Area} = L \times W \tag{9}$$

$$\text{Area} = 0.23 \times 0.092 = 0.02116 \text{ m}^2 \text{ Area} = 2.116 \times 10^{-2} \text{ m}^2$$

Size of penstock pipe diameter to connect the nozzle

The area is given using the following formula,

$$\text{Area}(A) = \frac{\pi \times d^2}{4} \tag{10}$$

Re-arranging the equation gives;

$$d^2 = \frac{4 \times A}{\pi} = \frac{4 \times 2.116 \times 10^{-2}}{3.14} = 2.6955 \times 10^{-2}$$

$$D = 0.16418 \text{ m} = 164.18 \text{ mm}$$

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

**Inner diameter of the turbine runner and radius of blade curvature:**

Inner diameter of the runner ( $D_i$ ) =  $D_o - 2 \times$  radial rim width

$$D_i = 315 - 2 \times 55 = 205 \text{ mm}$$

**Radius of blade curvature:**

Radius of the blade curvature ( $R_c$ ) =  $0.163 \times D_o$

$$R_c = 0.163 \times D_o$$

$$= 0.163 \times 315 \text{ mm} = 51.345 \text{ mm}$$

$$R_c \approx 52 \text{ mm}$$

So, the diameter of the pipe for blades =  $2 \times R_c = 2 \times 52 \text{ mm} = 104 \text{ mm}$

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

Alternatively,  $D_o = D_i + 2R_c$  which gives;

$$\frac{D_o - D_i}{2}$$

**Water jet velocity and turbine arc length:** The water jet velocity is calculated on the impulse turbines (Crossflow and Pelton) using the following equation (11);

$$V_j = \sqrt{2gH} \tag{11}$$

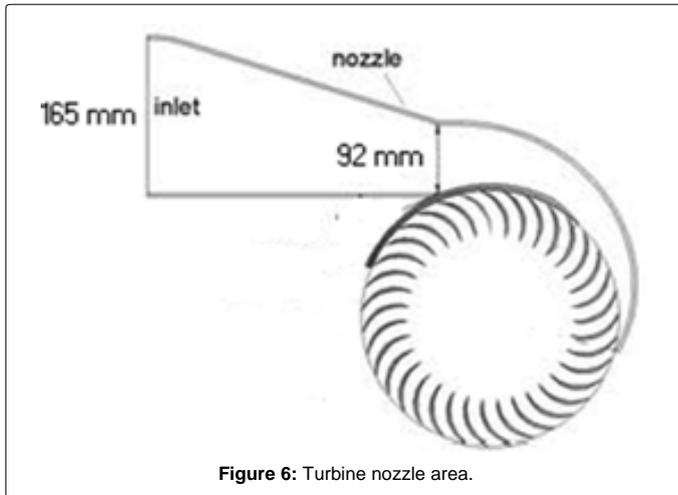


Figure 6: Turbine nozzle area.

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

$$V_j = 22.147 \text{ m/s}$$

Thus, the water jet velocity,  $V_j$  is 22.15 m/s

**Turbine arc length:**

$$S_1 = \frac{\pi \times R_c \times Q^\circ}{180} \quad (12)$$

$$S_1 = \frac{3.14 \times 52 \times 72^\circ}{180} = 65.132 \text{ mm}$$

$$S_1 \approx 66 \text{ mm}$$

**Pipe cut for turbine blades:**

Size of pipe diameter 104 mm (4")

Then the pipe circumference is given by;

$$C = \pi \times D$$

$$C = 3.14 \times 104 = 326.56 \text{ mm}$$

$$\text{So, } \frac{326.56}{66} = 4.95 \text{ pieces}$$

Where:  $S_1$  = the arc length (equation 12)

**Note:** Cut 4 full pieces of 4" diameter steel pipe to make the turbine blades.

**Water pressure and force to the turbine nozzle:** In water power systems, the power inside the turbine is given by;

$$P_t = p \times Q \quad (13)$$

Where:  $p$  = pressure (kN/m<sup>2</sup>),  $Q$  = Flow rate (m<sup>3</sup>/s),  $P_t$  = Turbine (kW)

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

But pressure,  $p = \frac{F}{A}$  when re-arranging this equation gives the following (14);

$$F = p \times A \quad (14)$$

But, Area =  $L \times W$

$$A = 0.23 \times 0.092 = 0.02116 \text{ m}^2$$

$$\text{So; Force (F)} = p \times A = 166.67 \times 0.02116 = 3.5267$$

kN

Therefore, the Force exerted to the turbine runner through the nozzle,  $F = 3.53 \text{ kN}$

Alternatively;

Power ( $P_t$ ) = Water Force ( $F_w$ ) x Velocity ( $V_j$ )

$$\text{Force (F}_w) = \frac{75.5 \text{ kW}}{22.147 \text{ m/s}} = 3.386 \text{ kN} \quad (15)$$

Then the distributed force at the turbine entry =  $F_1 = F/L = 3,530 \text{ N/m} \times 0.23$

Therefore the distributed water force on the turbine = 811.9 N

### Penstock pipe calculations:

The penstock is the steel pipe that conveys water with pressure from the forebay to the turbine unit in the powerhouse. The water in the forebay is stored in a form of potential energy and when delivered to the turbine through a penstock pipe and 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:

**Penstock diameter and thickness:**

$$D_p = 2.65 \times (n^2 \times Q^2 \times L_p / H_g) 0.204 \quad (16)$$

Where:  $D_p$  = penstock diameter,

$n$  = Manning coefficient for the mild steel penstock pipe = 0.012

$Q$  = design flow discharge = 0.45 m<sup>3</sup>/s

$L_p$  = length of the penstock = 162 m,  $H_g$  = Gross head = 25 m So in this case;

$$\begin{aligned} D_p &= 2.65 \times ((0.012)^2 \times (0.45)^2 \times 162 / 25) 0.204 \\ &= 2.65 \times (0.000144 \times 0.2025 \times 6.48) 0.204 \\ &= 2.65 \times (1.88956 \times 10^{-4}) 0.204 = 0.460918 \text{ m} \end{aligned}$$

$$D_p = 460 \text{ mm}$$

Therefore, the size of the penstock pipe selected to supply the design discharge to the turbine is 460 mm diameter (18") [6].

From literature, the minimum wall thickness is given by the following equation (17) [7]

$$t_p = \frac{460 \text{ mm} + 508}{400} \pm 1.2 \text{ mm} \quad (17)$$

$t_p$  = minimum penstock thickness (mm)

$D_p$  = penstock diameter = 460 mm

$$\text{So, } t_p = \frac{460 \text{ mm} + 508}{400} \pm 1.2 \text{ mm} \quad t_p = 2.42 \text{ mm} + 1.2 \text{ mm}$$

$$t_p = 3.62 \text{ mm} \approx 4 \text{ mm}$$

Therefore the thickness of the steel penstock pipe selected is 4 mm

**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 (18):

$$Q=AV \tag{18}$$

Where;

Q=water flow discharge, A=penstock pipe area and V=water flow velocity

Making V the subject from the above equation gives (19);

$$V = \frac{Q}{A} = \frac{4 \times Q}{\pi d^2} \tag{19}$$

$$V = \frac{4 \times 0.45}{\pi \times (0.46)^2} = \frac{1.8}{0.664424} = 2.7091$$

Water flow velocity V in the penstock is 2.7 m<sup>3</sup>/s [Recommended water flow velocity in the penstock pipe range between 1 m/s – 2.8 m/s][8].

**Area of the penstock pipe:** Area of the penstock pipe is given by the following equation (20);

$$A = \frac{\pi \times (d)^2}{4} \tag{20}$$

$$A = \frac{\pi \times (0.46)^2}{4} = 0.166$$

Therefore, the computed area of the penstock is 0.166 m<sup>2</sup>

**Penstock head loss:** Head loss in the penstock can be calculated using the following equation (21):

$$\text{Headloss} = \frac{10 \times n^2 \times Q^2}{D^5} \times L_p \tag{21}$$

Where; n=Manning value, Q=Design flow discharge, D<sub>p</sub>=Diameter of the penstock and L<sub>p</sub>=Length of the penstock

$$\text{Headloss} = \frac{10 \times (0.012)^2 \times (0.45)^2}{(0.46)^5} \times 162$$

$$= 0.014158 \times 162 \text{ m} = 2.294 \text{ m}$$

$$\text{Head loss} = 2.29 \text{ m}$$

Therefore, the net head is given;

$$\text{Net head} = \text{Gross head} - \text{head loss} = 25 \text{ m} - 2.29 \text{ m} = 22.71 \text{ m}$$

Net head is 22.71 m.

**Percentage of head loss:**

$$\% \text{ Head loss} = \frac{\text{Head loss}}{\text{Gross head}} \tag{22}$$

$$\% \text{ Head loss} = \frac{2.29 \text{ mm}}{25 \text{ m}} = 0.0916$$

$$\text{Head loss} = 9.16 \%$$

The recommended economical head loss for most small hydropower schemes should be below 10% (Table 2)[9].

**Pulley and belt design**

In order to calculate the pulley and belt parameters, the following

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

**Table 2:** Summary of penstock pipe design values. standard values have been selected based on the turbine and generator design calculations:

**Turbine design parameters:**

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

**Generator design parameters:**

- (α) Generator speed=1500RPM
- (β) Generator Power=75.5kW
- (χ) Synchronous, 3phase, 90KVA with power factor 0.8
- (δ) Generator pulley diameter=150mm (OK because >140)
- (ε) Generator operation time 24hours

**Turbine pulley design:**

**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 as follows (23);

$$\text{Speed ratio} = \frac{N_1 \times d_2}{N_2 \times d_1} \tag{23}$$

$$\text{Speed ratio} = \frac{N_1}{N_2} = \frac{d_2}{d_1} = \frac{635}{1500} = \frac{150}{d_1}$$

$$d_1 = \frac{150 \times 1500}{635} \quad d_1 = 354.33 \text{ mm}$$

Therefore, the pulley diameter d<sub>1</sub> of the turbine side is 354.33mm as shown in Table 3 below.

**Sizing of the belt length**

The belt length is determined using the following relation (24);

$$\text{Belt length (L)} = 2C + \frac{\pi(D+d)(D-d)^2}{4C} \tag{24}$$

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

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

$$L = 1802.85 \text{ mm} + 20.8045 \text{ mm} \quad L = 1823.654 \text{ mm}$$

The closest standard v-belt length is SPB wedge belt SPB1825 with 17 mm top width and 13 mm height (Figure 7).

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

Table 3: Generator and Turbine pulley dimensions.

**Turbine pulley belt speed**

The turbine velocity is given with the following equation (25):

$$\text{Velocity} = \frac{d}{2} \times W \tag{25}$$

Where; w is the rotational turbine speed given as (26);

$$W = \frac{2\pi N}{60} \tag{26}$$

$$\text{Velocity} = \frac{0.355}{2} \times \pi \times \frac{633.80}{30} = 11.77 \text{ m/s}$$

(OK because < 40)

Therefore the belt velocity of the turbine pulley is 11.17 m/s.

**Belt centre distance**

Centre distance C is calculated as follow;

$$C = d + D = 150 \text{ mm} + 355 \text{ mm} = 505 \text{ mm}$$

However, the exact centre distance can be calculated using the following formula (27);

$$C = A + \sqrt{A^2 - B} \tag{27}$$

$$\text{But } A = \frac{L}{4} - \frac{\pi}{8}(d+D)$$

$$\text{and } B = \frac{(D-d)^2}{8}$$

Taking L=1825 mm and substituting the values to the above A and B equations gives;

$$A = \frac{1825}{4} - \frac{\pi}{8}(150+355) = 258.0375 \text{ mm}$$

$$B = \frac{(355-150)^2}{8} = 5253.125 \text{ mm} = 505.6869 \text{ mm}$$

$$\text{Then, } C = 258.037 + \sqrt{(258.037)^2 - 5253.125}$$

Therefore the actual centre distance for the belt is 505.69 mm.

**Power per belt**

Interpolating from the given Table [10]

$$\text{-Rated power per belt} = (20.75 + 23.56)/2 = 22.155 \text{ kW}$$

$$\text{-Additional power} = (1.50 + 1.75)/2 = 1.625 \text{ kW}$$

Therefore the basic power per belt = 22.155 kW + 1.625 kW = 23.78 kW

**Correction power per belt**

From Table the combined correction factor is 0.85 [10]

$$\text{Therefore correction power per belt} = 23.78 \text{ kW} \times 0.85$$

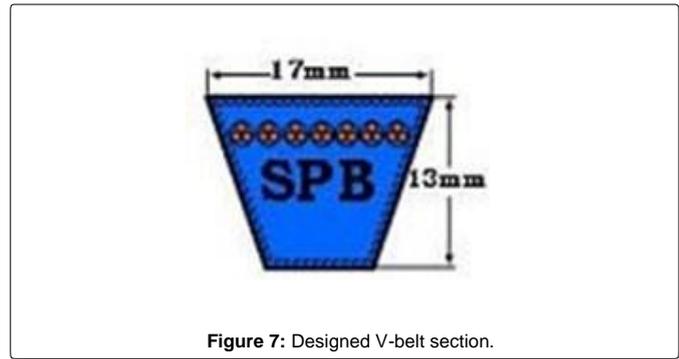


Figure 7: Designed V-belt section.

Correction power per belt = 20.213 kW

**Number of belt**

Number belt for the belt design is given as follow;

$$\text{Number of belts} = \frac{\text{Total power}}{\text{Correction power per belt}} = \frac{75.5 \text{ kW}}{20.213 \text{ kW}} = 3.735$$

Therefore choose 4 belts as standard

**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 (28) as;

$$\text{Torque} = F_d \times r \tag{28}$$

where r = radius of the turbine pulley (177.5 mm) and T = Torque (1,495 Nm)

$$\text{Then, } F_d = \frac{T}{r} = 8423.58 \text{ N}$$

Therefore, the driving force for the turbine pulley system is 8.423 kN (Table 4).

**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.

**Generator system:** The generator system will be operated with the following rated design values:

Frequency, f = 50 Hz

Rated power, P<sub>e</sub> = 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;

$$P_e = \text{turbine mechanical power} \times \text{generator efficiency}$$

(Refer to the turbine mechanical power, P<sub>m</sub> = 0.45 m<sup>3</sup>/s × 9.81 m/s<sup>2</sup> × 25 m × 0.72 = 79.46 kW)

Therefore, Electrical Power, P<sub>e</sub> = 79.46 × 0.95 = 75.49 kW

S/N	Parameter	Value	Unit	Remarks
1	Turbine pulley diameter	355	mm	Standard design 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 centredistance	505.69	mm	
6	Number of belts	4	pcs	
7	Net driving force	8.42	kN	

**Table 4:** Summary of pulley and belt drive design values.

Pe=75.5 kW (Figure 8) [11].

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, Wof the generator is related to the speed N and is given as (29):

$$W = \frac{2\pi N}{60} \quad (29)$$

At the rated condition N=1500 RPM

So, synchronous speed

$$W = \frac{2\pi \times 1500}{60} = 157 \text{ rad/s}$$

This can also be calculated using the following equation;

$$W = \frac{4\pi f}{pp} = \frac{4\pi \times 50}{50} = 157 \text{ rad/s}$$

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 in Table 5 below.

Therefore, the difference in speed (RPM) from the above three scenarios can be calculated as follows (30);

$$\Delta N = \frac{60}{2\pi} \Delta W \quad (30)$$

At low power demand,

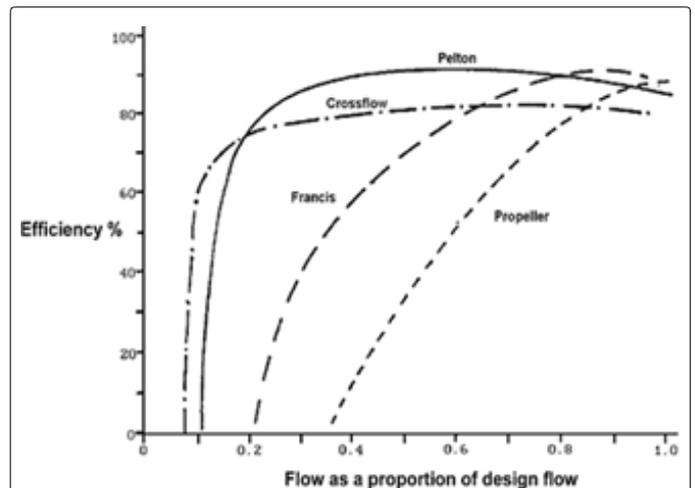
$$\Delta N = \frac{60}{2\pi} \times (156.37 - 157) = -6.01 \text{ RPM}$$

(This indicate a generator speeds up which is overspeed during low demand)

At high power demand, (this indicate a generator slows down which is under speed during high demand)

### Discussion and 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



**Figure 8:** Efficiency range with design flow discharge for different turbines.

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

**Table 5:** Generator power and speed at different rated capacity.

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 need 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 currents study for the turbine design, the flow discharge for Hhaynu micro-hydropower plant is 0.45 m³/s with the site head of 25 m 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 hence make the micro-hydropower project sustainable.

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