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DESIGN AND FABRICATION OF A COST EFFECTIVE CROSS FLOW HYDRO TURBINE FOR LOW HEAD MICRO HYDRO POWER SYSTEM

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Abstract*-Hydraulic turbine is the main building block of any micro hydro power plant (MHPP). In this paper the selection criterion of a suitable type of turbine for low head micro hydro schemes has been discussed briefly. Again a general design procedure and fabrication process of Cross-Flow Hydro Turbines (CFHT) has been described in this paper. A soft program has been developed to calculate the dimensional parameter of CFHT, where some optimum design parameters were used that amplifies the efficiency of CFHT. A detailed analysis of fatigue strength has been performed to calculate safe length of runner blades and maximum head of water for which the CFHT is able to withstand the occurring forces. The designed CFHT is flexible and portable so it can effectively be used in remote areas where the infrastructure still not developed for electrification. Preliminary tests were carried out in BUET fluid mechanic laboratory are aimed at observing performance characteristics of the designed CFHT.*

Keywords: Cross flow hydro turbine, Micro hydro power, Low head, Fabrication, Induction generator.

1. INTRODUCTION

Micro Hydro Power (MHP) schemes with outputs of less than 100 kW can provide cost effective power to remote rural communities with water resources. The prospect of MHP is large in Bangladesh since the dimension of the project is small. There are many Micro Hydro (MH) potential sites has been selected by BPDB/BWDB joint study in 1981, Flood Action Plan in 1992 and LGED especially in the northeast and southeast parts of the country [1-3]. Most of these potential sites have low head and it varied from 2 to 10 meter on the other hand there are huge flow variations during dry to rainy season. So it is very important to select suitable type of turbine for the reliable operation of MHP Plant. Among these potential sites MHP Plant is only operated at Banskhali, Bamer-chara irrigation project by LGED and Monjaipara micro-hydro power unit at Bandarban in our country.

The objective of this paper is to design and fabrication of a cost effective CFHT for low head MH schemes by using indigenous technology and raw materials available in local market. The CFHT was invented by Australian engineer A.G.M. Mitchell Banki about a century ago right from the day. Since the initiation of CFHT much advancement has been made in its design through experimental studies and research [4-9]. The CFHT consist of two parts, a nozzle and turbine runner. The runner is build up of two parallel circular disks joined together at the rim with a series of curved blades. Water is guided into the rotor by rectangular shaped nozzle and discharges the jet full width of the wheel and enters the wheel at an angle α. The shape of the jet is rectangular, wide and not very deep tangent to the periphery of the wheel. The water then strikes the blades on the rim of the wheel, flows through the rotor blade, through the interior, through the outer rim, through the exit the machine at ambient pressure. The wheel is therefore an inward jet wheel and because the flow is essentially radial, the diameter of the wheel is practically independent of the amount of water impact, and the desired wheel breadth can be given independent of the quantity of water. Due to the change in angular momentum of the water across the turbine rotor, a torque is applied to the output power shaft. The output power shaft can be used to drive generator.

The organization of this paper is as follows: an In Section 2 Micro hydro resource of Bangladesh is presented. Selection criteria of turbine for low head micro hydro scheme is described in Section 3. The design process of CFHT, its strength analysis and fabrication process is described in section 4, 5, 6 respectively. In section 7 and 8 financial aspects of CFHT and its experimental test result is presented. Finally conclusions are drawn in Section 9.

2. MICRO HYDRO RESOURCES IN BANGLADESH

The potential of MHP especially in Chittagong, Sylhet, Dinajpur, Jamalpur and Rangpur regions of our country have been reported in various studies[1-2] is presented in table 1. In most cases micro-hydro power MHP schemes are 'run of river' types so quite small dam or barrage is required to regulate the level of water at the intake for the micro-hydro power plant so it has no adverse effect on the local environment as large hydro. The power developed by a water turbine is proportional to the volumetric rate of water flow through it and the pressure head of water. The general equation (1) for any hydro system's power output is

$$P_0 = \eta_o Q H_o g \ \text{kW} \tag{1}$$

Where P is the mechanical power produced at the turbine shaft in kW, η_o is the overall efficiency of the MHPP which is in the range of 50-70% [10], g is the acceleration due to gravity (m/s^2), Q is the flow rate in m^3/sec, and H is the effective pressure head of water across the turbine (m).

Table: 1 Micro hydro prospect in different region of Bangladesh

Region	Potential electric energy in kW
Chittagong	352
Sylhet	735
Dinajpur	171
Jamalpur	98
Rangpur	80

3. SELECTION CRITERIA OF LOW HEAD TURBINE

All hydro turbines convert the energy from falling water into rotating shaft power, so it is necessary to select a suitable type of turbine that can be adopted for a particular project. The type, geometry and dimensions of the turbine for a particular project depend on the net head available, range of discharge through the turbine, rotational speed and techno-economic considerations of the generating equipment. The head is the first criterion for the selection of the type of turbine for a particular site and other important parameters is the specific speed n_s to be considered. In table 2 specifies the range of head for different types of turbine with their specific speed ranges [11-12]. It is given by the following relation:

$$n_S = \frac{N\sqrt{P \times 1.358}}{H^{5/4}} \tag{2}$$

Where, n_s is specific speed of turbine in r.p.m., N is rated speed of turbine in r.p.m, P is turbine output in kW, and H is Rated head in m.

Table: 2

Head and specific speed ranges for various types of turbine runners

Turbine type	Head range in (m)	Specific speed range
Kaplan	2-40	200-1000
Bulb turbine	0-10	260-360
Cross-flow	1.5-150	20-200
Francis	10-350	30-400
Pelton	400-2000	10-30
Turgo	50-250	20-70

The seasonal variation of flow rate and net head available is also necessary to be considered while selecting suitable turbine for a low head micro hydro schemes. In Fig. 1 the relative efficiencies for a part flow condition of different turbines is shown. The full Kaplan turbine and cross flow turbine can operate with a better efficiencies in part flow condition than the propeller turbine. In this research work the Cross flow turbine is selected as a suitable turbine for low head micro hydro schemes because of its simple construction, cost is much cheaper than Kaplan turbine of the same size and easy to manufacture by using our indigenous technology.

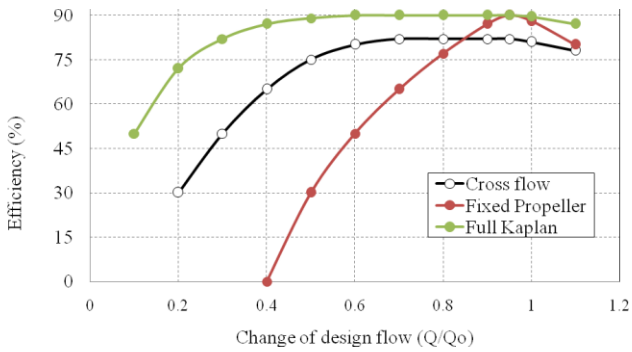

Fig.1: Relative efficiencies of the different turbine with the change of design flow

4. DESIGN PARAMETERS OF CFHT

By means of all the extensive literature research that has been conducted throughout the course of the project, some optimum design parameters have identified that help to amplify the efficiency of CFHTs [4-9]. An interactive soft-program has been developed that incorporates these optimal design values and helps to calculate and identify the best design parameters for the site under consideration. This soft-program has been developed by C programming language. The design procedure follows a series of logical steps. The calculations for each step have been put into the soft-program. The key design parameters for a turbine are head H, volume flow or discharge Q and rotational speed N. Starting from the site data the turbine speed should be chosen to give a specific speed which fits with the CFHT range. The choice of the speed, N, depends on the speed of the generator and the type of drive used.

In this work a double stage belt drive is used which allows the possibility of changing the turbine operating speed. This gives more flexibility in the turbine design and in matching to site conditions. After that, choosing water jet entrance angle α to the runner and then calculates the dimensions of the runner length L, runner main diameter D_1, and finally the curvature of blade to give the correct swirl velocity at runner inlet and outlet. The dimensions of the runner may need to be adjusted through several iterations to avoid large twist in the blades. In addition the Distributed load on Blade W, Bending moment M, Flexural stress σ on the blade also calculated by this soft-program. In the design of CFHT average discharge Q is 120 l/sec and net head H is 2m has been considered.

4.1 The Efficiency of CFHT

The velocity of the water before entering turbine rim through nozzle is given by;

$$V_1 = C_v \sqrt{2gH} \tag{3}$$

Where, C_v is the coefficient of velocity dependent upon the nozzle, V_1 absolute velocity of water and H is net head. Hence the Force acting on the vane in the direction of motion is denoted by F_x defined as the product of "mass of fluid striking the vane per sec and change of velocity of whirl" is found from the velocity triangle shown in Fig. 2.

$$F_X = \frac{w}{g}\left(V_1 \cos\alpha_1 + V_2 \cos\alpha_2\right) \tag{4}$$

Now net work done per second on vane is found by:

$$P = \left(\frac{w}{g}u_1\right)\left\{\left(V_1\cos\alpha_1 - u_1\right)\left(1+\psi\frac{\cos\beta_2}{\cos\beta_1}\right)\right\} \tag{5}$$

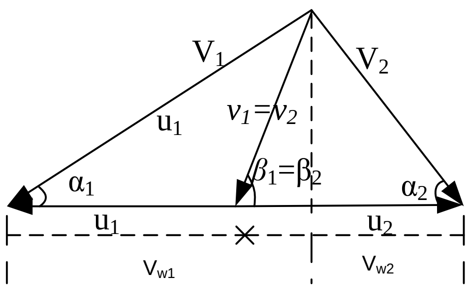

Fig. 2: Inlet-outlet velocity triangle of cross-flow turbine

The hydro power supplied to cross-flow turbine rotor (P_{in}) due to head H is define as product of mass flow rate of water and available head.

$$P_{in} = \frac{wV_1^2}{2gC_v^2} \tag{6}$$

Finally the efficiency η_h is defined as ratio of the mechanical energy delivered by the rotor of the cross-flow turbine to the available hydro energy supplied to cross-flow turbine.

$$\eta_h = \frac{P_{out}}{P_{in}} = 2C_v^2 \frac{u_1}{V_1}\left(\cos\alpha_1 - \frac{u_1}{V_1}\right)\left(1+\psi\frac{\cos\beta_2}{\cos\beta_1}\right) \tag{7}$$

When, $\beta_1=\beta_2$, then efficiency.

$$\eta_h = \frac{P_{out}}{P_{in}} = 2C_v^2 \frac{u_1}{V_1}\left(\cos\alpha_1 - \frac{u_1}{V_1}\right)(1+\psi) \tag{8}$$

Now consider all variables as constant except efficiency and u_1/V_1. The optimum efficiency in respect of the velocities ratio is;

$$\frac{\partial\eta_h}{\partial\frac{u_1}{V_1}} = 0, \text{ then } u_1 = \frac{1}{2}V_1\cos\alpha_1 \tag{9}$$

$$\frac{\partial^2\eta_h}{\partial\left(\frac{u_1}{V_1}\right)^2} = -4C_v^2(1+\psi) \tag{10}$$

This optimum efficiency is a maximum. Then the maximum efficiency is;

$$\eta_{h\max} = \frac{1}{2}C_v^2(1+\psi)\cos^2\alpha_1 = k\cos^2\alpha_1, \tag{11}$$

where $\mathrm{k} = \frac{1}{2}\mathrm{C}_\mathrm{v}^2(1+\psi)$

It is result the maximum efficiency $\eta_{h\max} = 1$ only when the flow is in-viscous (assuming no loss of head due to friction in nozzle or on the blades) then $k=1$ and for tangential entrance angle of water jet $\alpha_1 = 0^0$. The theoretical approaches are impossible because the real flow in the nozzle is with hydraulic losses and the tangential entrance implies zero rate of flow through the runner. The real entrance angles α_1 limited in the range of 10^0 to 20^0, depending on specific speed and it is associated with central angles extension of the nozzle exit reflected at the CFHT axis. In literature the values mostly met are $\alpha_1=16^0$ for getting the highest mechanical efficiency.

4.2 Runner Angular Velocity

The calculation of runner angular velocity is based on the following assumptions:
Induction Generator (4pole, 50Hz) rpm: 1565
Velocity ratio: 13:1
Therefore runner rpm = 1565/13 = 120 rpm

4.3 Design of Blade Angle

Choose angle of attack (α_1) for water jet entering at inlet to optimize the efficiency $\alpha_1 = 16^0$

Required blade angle (β_1) for CFHT is:

$$\beta_1 = \tan^{-1}(2\tan\alpha_1) = 29.83^0$$

4.3 Design of Radial Rim Width and Diameter

Runner outer diameter and radial rim width is calculated after fixing angular velocity of runner by the following equations;

$$D_1 = 360 \times \frac{C_v\sqrt{2gH}\cos\alpha_1}{\pi \times N} = 47\text{cm}$$

Radius of inner circle: $r_2 = 0.66r_1$ =15.44 cm

Width of the Turbine Runner (L):

$$L = \frac{144Q}{C_v k D_1\sqrt{2gH}} = 47\text{cm}$$

Velocity of jet at inlet: $V_1 = C_v\sqrt{2gH}$ = 6.14 m/sec

Area of the Jet: $A = \frac{Q}{V_1}$ =195.47 cm^2

Thickness of the Jet: $S_o = \frac{A}{L}$ = 4.09 cm

Blade spacing between two neighbor blades (t):

$$t = \frac{S_o}{\sin \beta_1} = 8.21 \text{ cm}$$

Number of blade in the wheel is: $n = INT(\frac{\pi D}{t})$ =18

Radius of Blade Curvature (ρ):

$$\rho = \frac{r_1}{\cos \beta_1}\left\{1-(\frac{r_2}{r_1})^2\right\} = 7.68 \text{ cm}$$

4.4 Distance of Jet from Center of Shaft and Inner Periphery

Thickness in the inner part of the wheel is:

$$Y = \frac{2S_0 \cos \alpha_2'}{\frac{r_2}{r_1} \times \cos \alpha_1} = 7.78 \text{ cm}$$

Distance of jet from centre of shaft:

$$Y_1 = r_2 \sin(90-\alpha_2') - y - \frac{d_0}{2} = 5.40 \text{ cm}$$

Distance of Jet from the inner periphery of wheel:

$$Y_2 = (0.134 - 0.94k)D_1 = 2.84 \text{ cm}$$

5. CALCULATION OF BENDING STRESS ON BLADE SECTION

Bending Stress (M) and tensional stress (σ) on a blade section is calculated after fixing the diameter of runner and wall thickness of runner blade. This enables the turbine to specify the maximum head of water for which it is suited. Conversely it also gives the safe length of blades between supports if the head of water under which the turbine is going to be used is specified [6]. In this analysis the blade is treated as a beam of constant cross section, rigidly fixed at both ends as shown in Fig. 3.

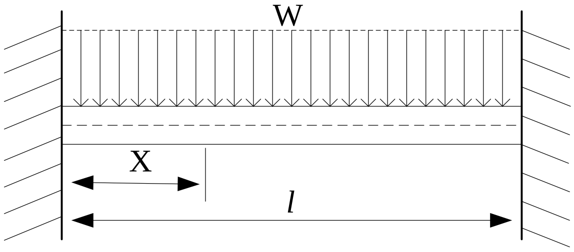

Fig. 3: Blade section as a rigidly fixed beam.

Hydraulic force on blade: $F = \frac{T_c}{r}$ = 420.77 N

Distributed load on Blade (W):

$$W = 11217.4(\frac{9.619\rho}{n} - t)H = 1037.65 \text{ N/m}$$

Extreme fiber distance (e):

$$e = \frac{10^{-4}(2631\rho^2 + 10644\rho t + 6219t^2)}{(2\rho + t)} = 1.16 \text{ cm}$$

Bending moment: $M = -\frac{Wl^2}{12}$ =19.80 N-m

Flexural stress or tensional stress (σ):

$$\sigma = 934.78(\frac{9.619}{n} - t)\frac{Hl^2 e}{I_X} = \frac{Wl^2 e}{12 I_X} = 33.26 \text{ N/mm}^2$$

The tensional stress σ occurs in each runner blade once per revolution. In order to make a safe runner construction the value for σ has to be chosen according to fatigue strength criteria. As the alternating load takes place in the presence of water a low fatigue strength results. A value of flexural stress for mild steel is $\sigma \leq 22$ N/mm^2 for mild steel seems in the correct order of magnitude.

6. FABRICATION PROCESS OF CROSS FLOW TURBINE

The fabrication process of different parts of the CFHT is briefly described in this section and it has been fabricated in BUET mechanical workshop. The general arrangement drawing, scheme of the nozzle and rotor combination is shown in Fig.4.

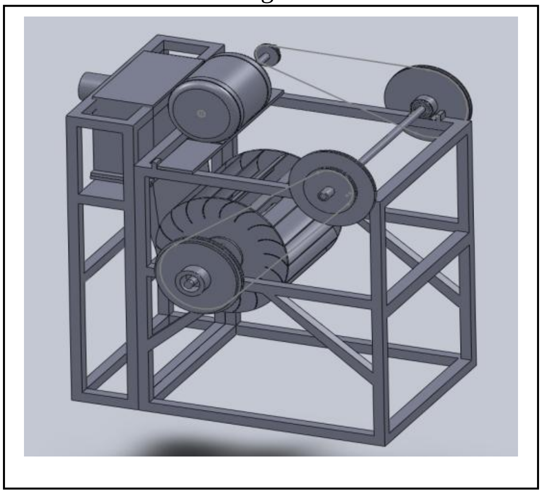

Fig. 4: Side view of the designed CFHT

6.1 Fabrication of CFHT Rotor

The turbine has a squirrel cage shaped rotor, whose rotor diameter and breadth are 47cm respectively. The optimum speed under the design head of 2m is 120 rpm. It has 18 blades, each blade simply a circular segment. Here the turbine runner disks were prepared by mild steel plates, whose dimension is 7mm x 51cm x 51cm. Firstly these plates has been faced and then rounded to desire thickness and diameter as shown in Fig. 5 by lathe machine. After that trace inner circle and pitch circle, has drawn on the rotor plate. The pitch circle is divided in to 18 sections and points each section. To form blade profile a curvature has drawn from each points on the pitch circle whose radius is 7.68 cm. Finally the traced blade profile slot on rotor disk has been cut by Jig boring machine in Bangladesh Machine Tools Factory Ltd as shown in Fig. 6.

Fig. 5: Turbine runner disk is faced by lath machine

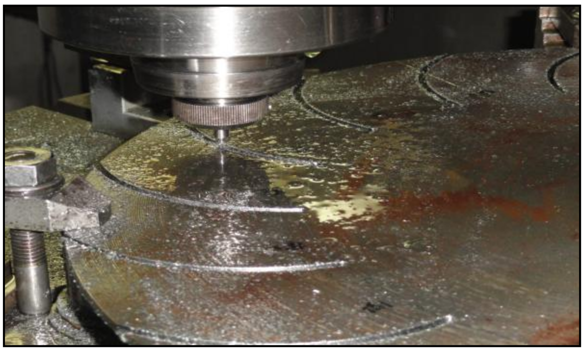

Fig. 6: Cutting of blade profile on runner disk by Jig boring machine

The rotor blades have been rolled out of 3mm mild steel sheet to desire curvature by rolling machine. These blades are placed between the disks in grooves spaced and arc welded to the disk as shown in Fig. 7 and in Fig. 8 complete assembling of turbine runner is shown. The runner end wheel was mounted on a 30mm steel shaft with the help of flange and keyed.

Fig. 7: Runner end pieces are attached on shaft

6.2 Fabrication of CFHT Nozzle

The CFHT Nozzle size has been determined by using the following formula [13].

$$\text{Nozzle length} = \frac{210 \times \text{Flow(cubic feet/ second)}}{\text{runner outside diameter (in)} \times \sqrt{\text{head(ft)}}}$$

The nozzle has been built up of 0.3mm mild steel plate. Cut side sections and flat front and back sections of the nozzle. Width of the front and back pieces equal to width of the turbine wheel width but in practical case the nozzle length should be 1.5cm to 3cm less than the inside of the turbine. The left and right side plate has been cut to the desire design size and weld all sections together as shown in Fig. 9.

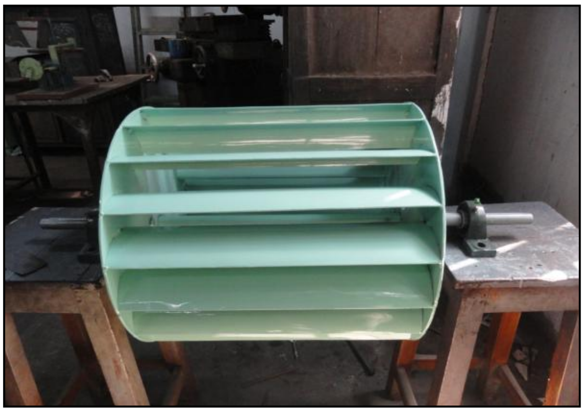

Fig. 8: Picture of fabricated CFHT

Fig. 9: end pieces of nozzle are jointed together and spot welded.

6.3 Fabrication of CFHT Frame

The frame is totally made of angles, and bolted together to the turbine, nozzle and generator. The generator mounted on the top of the frame work and its based has made such manner that different types of generator can be attached to the frame work. The system is simple and it can be decouple easily and allows for easy access to the turbine and nozzle for repair and maintenance

6.4 Power Transmission of CFHT

There are two possible solutions for power transmission from the turbine shaft to the generator shaft, these are: Gears, Belts and Pulleys. Gears are usually avoided in micro-hydro schemes due to their high cost and high maintenance. Belts and pulley mechanism, if properly designed, can well serve the purpose with efficiencies of about 98% [14]. In this work the shaft power of the CFHT is transmitted to induction generator in two stages. The speed ration in first stage and second stage are 4.66 and 2.88 respectively.

6.5 Design of Penstock Size

The size of penstock is designed considering the average flow 120 l/sec under a head of 2m. Equation (12) is used to determine the cross-section area of the penstock [15]:

$$d = \left(\frac{flQ^2}{3h_f} \right)^{1/5} \quad (12)$$

Where d is the penstock diameter in m, f is the Darcy's coefficient or friction coefficient, l length of the penstock in m, h_f head difference on both ends of the penstock in m, Q flow through the penstock in m^3/s

7. FINANCIAL ASPECTS OF THE DESIGNED CFHT

The Material used to develop the CFHT is available in the local market so the cost of the fabricated CFHT is much cheaper than the existing low head turbine around the world. A comparative per kW capital cost of the fabricated low head CFHT with other low head turbines system is presented in table 3.

Table 3: Comparative per kW Capital cost of fabricated LHCFHT with existing turbine

Name of Plant	Investment Cost BDT per kW
Fabricated Low head CFHT	43,000.00
Existing Low head CFHT [16]	1,56,000.00
Low Head Kaplan turbine [17]	3,10,000.00
Low Head Propeller turbine [18]	2,39,600.00

8. EXPERIMENTAL TEST RESULTS

The laboratory test of cross flow turbine is performed at fluid mechanics lab in BUET. Here the turbine directly connected to the centrifugal pump through a 6 inches pipe, the pump flow controlled by gate valve fixed just after the pump. The test system is shown in Fig. 10.

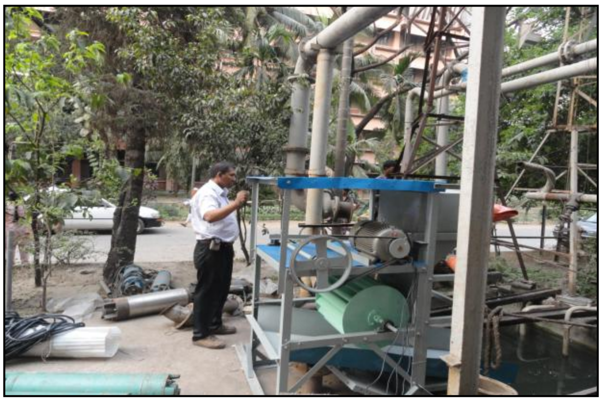

Fig. 10: The designed CFHT under test

In laboratory test the flow rate and pump head were maintained to 65 l/sec and 1.4 m respectively. Under this condition the no-load voltage of the IG is build up to 230 volt at 1250 rpm. When three 100 watt and 60 watt incandescent lamp were used as electric load on IG terminal then the voltage was fallen to 190 volt and rotor speed decreases to 1215 rpm. At this time 0.75A per-phase load current was delivered by IG. The experimental data is shown in table 4. From this test result overall efficiency of the system has been calculated 43.34 % and the efficiency of CFHT 59% respectively. In this laboratory test the speed of the IG could not be increased further due to the limitation of experimental test setup. Because of this limitation the induction generator was operated at near the unstable region of its stability curve for that any further increase in load causes a huge drop in terminal voltage. The load characteristic of the CFHT is shown in Fig. 11.

Table 4: Experimental test data of CFHT

Applied electric load (Watts)	Runner speed N(rpm)	Discharge Q (l/ sec)	Effective head H (m)
0	93	50	1.4
180	92	55	1.4
283	91	60	1.4
384	90	65	1.4

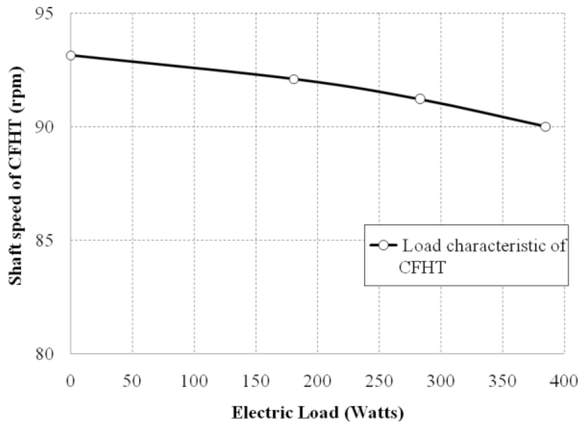

Fig. 11: Load characteristics of the CFHT

9. CONCLUSION

The aim of this research is to design and developed a cost effective CFHT especially for the developing country. Per kW installation cost of fabricated CFHT is much cheaper than other low head turbine exist around the world. The maximum electric power developed by the CFHT under this test condition is 384 watt at 190 volt at runner speed of 90 rpm. The overall system efficiency of the system and the efficiency of CFHT is found 43% and 59 % respectively. The efficiency of the fabricated CFHT is lower than the anticipated maximum efficiency of the system due to losses in bearings, belt drive system and finally the system was driven at reduce effective head and discharge. Technologically the fabricated CFHT is feasible, and economically viable since it can be easily developed locally even by small workshops. The designed CFHT is flexible and portable so it can effectively be used in remote rural areas where MHPP is suitable for electrification.

10. ACKOWLEDGEMENT

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11. NOMENCLATURE

Symbol	Meaning	Unit
η_h	Efficiency of CFHT	Dimension less
Q	Flow rate	(m^3/sec)
H_o	Net head available	(m)
P_o	Power developed by a water turbine	(kW)
ω	Angular velocity of the rotor	(rad/s)
α	Angle of water jet entry to runner	(degree)
β	Blade angle Beta	(degree)
t	Blade spacing between two adjacent blade	(cm) (cm)
σ	Flexural stress	(N/mm^2)
ψ	empirical coefficient	Dimension less
T_C	Torque transmitted by a blade channel	N-m
e	Extreme fiber distance	cm