EXPERIMENTAL INVESTIGATION OF CROSS-FLOW TURBINE

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Abstract- The purpose of this study is both to test the performance of a low cost cross-flow turbine and to study on the efficiency of a low cost cross-flow turbine. In this research the maximum effective head 4.79 m and maximum flow rate 5.65×10^{-3} m^3/s were occurred at gate valve fully open. The speed of the turbine is between 89 and 163 rpm depending on the flow rate. The maximum shaft power is 236.66 W and the maximum generator output power is 143 W at the gate valve fully open condition. The other objective of this work is to characterize the two stage power transfer in the runner. Theoretically, the first stage contribute 76.2% to the total torque (or power) production while the second stage produce the remaining, 26.8%. The maximum overall efficiency is 57%.

Index Terms- Cross-Flow Turbine, Effective Head, Efficiency Performance, Torque.

I. INTRODUCTION

Hydropower was considered as one of the most desirable source of electrical energy due to its environment friendly nature and extensive potential available throughout the world [8]. The enormous dependency on the conventional energy resources has resulted serious ecological impact, which have heightened the global awareness of the need of renewable energy resources. Among several renewable resources, hydropower is an important and sustainable renewable energy in the world today that can be exploited to the maximum due to its attractive economics and the availability of basic technology [1]. The most critical component of a hydropower system is the turbine.

A hydraulic turbine converts hydraulic power into mechanical power in form of rotating shaft. The power transfer process takes place on the surfaces of the blades. The blades are assembled on a disc that is supported by a shaft; the whole assembly is technically known as runner [3].

Nowadays, the cross-flow hydraulic turbine is getting popularity in low head and small water flow rate establishments, due to its simple structure and ease of manufacturing in the site of the power plant. A typical cross-flow turbine (CFT) technology consists of two main components namely the nozzle and the runner. The flow of water crosses the runner blades twice, from the periphery towards the centre, and after crossing the open centre space, from the inside to the outward periphery. This turbine is therefore a turbo-machine with two velocity stages, and water filling only part of the runner at any time of operation. There is some portion of water is entrained within the runner blade and flung out of the runner tangentially. This is the entrained flow. A nozzle, which converts potential energy to kinetic energy of the water in form of a high velocity jet, is an important component for impulse turbines. The CFT turbine nozzle is basically a rectangle in cross-section. The two surfaces are planes and the other two surfaces are typically curved. To guide and control water flow to the runner blades, the nozzle may incorporate a guide vane or the one of the nozzle curved surfaces can be made movable to control the outlet Cross-sectional area. The cross-section of the blades was circular with specific radius of curvature. The runner blade s can be cut from standard steel pipes [3]. For this reason, cross-flow turbine technology has been promoted in developing countries.

II. PREVIOUS WORK

Several studies, both experimental and numerical methods had been done to determine the optimal configuration of the cross-flow turbine. Most of these studies included details concerning the influence of nozzle shape, diameter ratio of the runner, admission arc of the nozzle and the number of blades on the turbine efficiency.

M. Sinagraa, V. Sammartanoa, C. Aricòa [7], they conducted a research on title “Cross-Flow Turbine Design for Variable Operating Condition” in 2013. This paper had outlined a simple but rigorous procedure for the design of a cross-flow turbine with discharge regulator. The researchers had observed reduction of turbine efficiency along with the inlet surface reduction and computational fluid dynamics (CFD) simulations were also observed.

Vincenzo Sammartano, Costanza Aricò, Armando Carravetta, Oreste Fecarotta and Tullio Tucciarelli [6] they conducted a theoretical framework for a sequential design of the turbine parameters, taking full advantage of recently expended computational capabilities. In this paper a novel two-step procedure was described. In the first step, the initial and final blade angles, the outer impeller diameter and the

shape of the nozzle are selected using a simple hydrodynamic analysis, based on a very strong simplification of reality. In the second step, the inner diameter, as well as the number of blades and their shape, are selected by testing single options using CFD simulations.

III. METHODOLOGY

A.Experimental Approach

Experiments have been carried out in the laboratory of the Department of Mechanical Engineering, Mandalay Technological University. The experimental facility is made of water pumping system and a test stand with the cross-flow turbine with a generator. A small cross-flow turbine, designed according to the previous study [2] and has been installed in a hydraulic circuit. A single phase AC synchronous generator is used for power generation system. The photographic view of the experimental facility of cross-flow turbine is as shown in Fig.1.

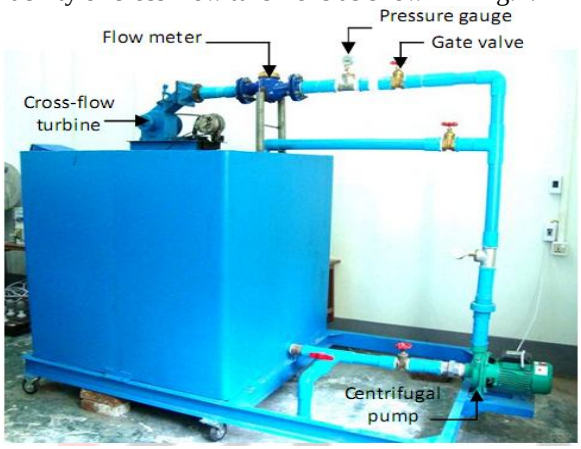


Fig. 1 Experimental set-up of cross-flow turbine

The test stand is made of the cross-flow turbine coupled with a generator by using pulley system. Pulley systems are commonly used in small scale hydropower systems. The speed ratio used in this system is 3:1. The turbine used in the experiments was built at the laboratory of Renewable Energy, Mechanical Engineering Department, Mandalay Technology University. The main components of the cross-flow turbine are nozzle, runner, casing and draft tube as shown in Fig. 2.

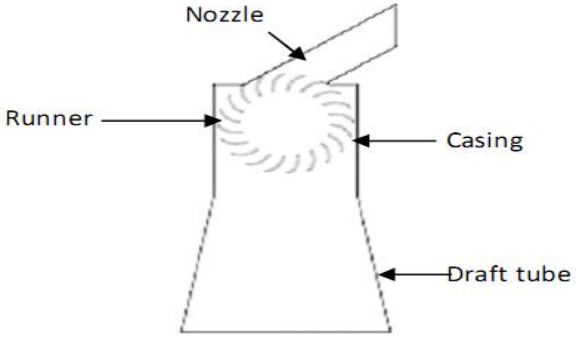


Fig. 2 Cross-flow turbine geometry

B.Efficiency of Turbine

The efficiency of the turbine is defined as the ratio of the power output from the turbine to the generator and the energy of the water flowing into the turbine.

$$\eta_T = \frac{T_s \omega}{\rho Q g H_e} \qquad (1)$$

The variables and constants for equation 1;
T_s – Shaft torque, N-m,
ω – Angular velocity, rad/s,
ρ – Water density, 1000 kg/m^3,
Q – Volumetric flow rate, m^3/s,
H_e – Effective head, m.
There are some quantities that have to be measured. Net head is calculated with use of equation 2. Volumetric flow rate is measured with a water meter from the turbine inlet. The water velocity at turbine inlet can be calculated from the measured flow rate Q (m^3/s) using the continuity equation, equation (3).

$$H_e = \frac{v^2}{2g} + \frac{p}{\gamma} + z \qquad (2)$$

The governing variables for calculating net head;
v - Water velocity at turbine inlet, m/s
p - The gauge in pressure at turbine inlet, N/m^2,
z - The height difference between the pressure gauge and the centre of the cross-flow turbine, m.

$$v = \frac{Q}{A} = \frac{4Q}{\pi (D)^2} \qquad (3)$$

Where, D is the diameter of the inlet pipe, 0.0508 m. To calculate the theoretical efficiency of the turbine, the shaft torque T_s can be estimated based on a flow analysis. The angular velocity of the rotor, ω, is set to be equal to the tangential flow velocity at inlet, U.

C.Flow Through a Cross-flow turbine and Velocity Diagram

There are two major flow patterns through a cross-flow turbine, cross flow and entrained flow shown in figure 3. When water follows the cross flow pattern from point 2 to 3, it hits the turbine blades at two stages before it exits the turbine. Water travels diagonally through the runner after exiting the first stage of the torque transfer at point 2. The angle and curvature of the blades are designed so the water will transfer a significant amount of torque at stage two at point 4 before exiting the turbine.

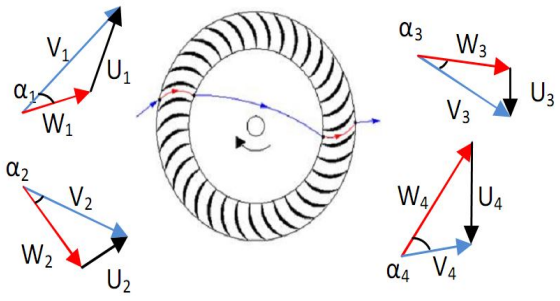


Fig. 3 Cross-flow turbine geometry and velocity triangles

A part of the flow is trapped by the blades and swung around to the bottom of the turbine. This is what is called entrained flow. This flow pattern will only contribute with one stage of torque transfer. Rotational speed of the rotor and nozzle opening affect the cross flow through the runner. Inappropriate configuration of the turbine can make the water hit the axle or most of the flow gets entrained.

Leakage flow is a loss due to water flowing through gaps between the rotor and turbine casing. There are two leakage flows, one between the end of the top cover .The other leakage flow is between the side casing and the rotor. Figure 3 shows the flow through the runner and velocity diagram.

$$\omega = \frac{20 \times \sqrt{H_e}}{R_1} \times \frac{2\pi}{60} \qquad (4)$$

R_1 is the radius from the centre of the rotor to the outer edge of the turbine blades.

D.Theoretical Torque Transfer in the Two Stage

The other objective of this work is to characterize the two stage power transfer in the runner. Power transfer is the result of the resultant torque exerted on the blades in both stages as a result of change of angular momentum of the jet. Due to complex action of the Cross-flow turbine, the theoretical torque transfer analysis can be possible if assumptions are made. These assumptions are; there are no losses incurred such as incidence, shocks and friction.

Consider the flow through the runner passing through first stage as schematically shown in Fig. 3. If Newton Second Law of Motion is applied to a mass of water flowing with an infinitesimal velocity (dV) in a unit time in the direction of the velocity vector, then an infinitesimal force ($dF = \rho Q d$V) is generated by the fluid. The infinitesimal reaction force (dF$_R$) on the blade is equal and opposite of this force, according to Newton Third Law of Motion. Therefore, the forces on the runner in the first stage (F$_R$ 1→2) and second stage (F$_R$ 3→4) are as follows:

$$(F_R)_{1\to 2} = \int_1^2 \rho Q d\vec{V} = \rho Q(\vec{V}_1 - \vec{V}_2) \qquad (5)$$

$$(F_R)_{3\to 4} = \int_3^4 \rho Q d\vec{V} = \rho Q(\vec{V}_3 - \vec{V}_4) \qquad (6)$$

To generate power, it is important to calculate torque due to these forces about the axis of rotation of the runner. When using the equations (5) and (6), the absolute velocity vectors can also be expressed in terms of their orthogonal components, tangential component (V$_t$) and radial components (V$_r$). The velocity vector responsible for torque production is the tangential component also known as velocity of whirl. The radial component does not produce torque because its line of action passes through the axis of rotation. Therefore, taking multiplication of the force due to change in whirl velocity and its corresponding radius:

$$T_{1\to 2} = \rho Q(\vec{V}_{t1} R_1 - \vec{V}_{t2} R_2) \qquad (7)$$

$$T_{3\to 4} = \rho Q(\vec{V}_{t3} R_2 - \vec{V}_{t4} R_1) \qquad (8)$$

The ratio of power transfer during both stages can be given as:

$$\frac{T_{1\to 2}\omega}{T_{3\to 4}\omega} = \frac{(\vec{V}_{t1} R_1 - \vec{V}_{t2} R_2)\omega}{(\vec{V}_{t3} R_2 - \vec{V}_{t4} R_1)\omega} \qquad (9)$$

From geometry of the velocity triangles and noting that the water jet must leave the first stage with radial relative velocity and enter the second stage in radial direction:

$$\vec{V}_{t1} = U_1 + W_1 \cos\beta_1 \qquad (10)$$

$$\vec{V}_{t2} = U_2 = \vec{V}_{t3} \text{ and } \vec{V}_{t4} = V_4 \cos\alpha_4 \qquad (11)$$

It can be shown from design optimization[2] that for maximum efficiency, that the flow must leave the second stage without whirl, therefore:

$$\vec{V}_{t4} = 0 \qquad (12)$$

Further, it can be shown from optimisation of the blade angle with respect to runner periphery speed that for maximum efficiency,

$$U_1 = W_1 \qquad (13)$$

Angular velocity of the runner ω is constant,

$$U_1 = \omega R_1 \text{ and } U_2 = \omega R_2 \qquad (14)$$

$$\vec{V}_{t2} = U_1(1 + \cos\beta_1) = \omega R_1(1 + \cos\beta_1) \qquad (15)$$

Substituting in equation (9)

$$\frac{T_{1\to 2}\omega}{T_{3\to 4}\omega} = \frac{\omega R_1(1 + \cos\beta_1) \times R_1 - \omega R_2 \times R_2}{\omega R_2 \times R_2}$$

$$\frac{T_{1\to 2}\omega}{T_{3\to 4}\omega} = \left(\frac{R_1}{R_2}\right)^2 (1 + \cos\beta_1) - 1 \qquad (16)$$

The equation (16) shows that, for an optimised design, the ratio of power transfer in the two stages theoretically depends on the size of the runner and the blade angle. It can be seen from the equation that it is the first stage always contribute more power than second stage. For the simple Cross-flow turbine runner under, using the design parameters laid down in Table 1.

Table I. Design Parameter of the Cross-Flow Turbine

No.	Description	Symbol	Value	Unit
1.	Outer diameter of runner	D_1	150	mm
2.	Inner diameter of runner	D_2	100	mm
3.	Width of runner	B_o	76	mm
4.	Angle of attack	α_1	16	degree
5.	Radius of blade curvature	R_{bc}	24.46	mm

6.	Radius of pitch circle	R_0	55.35	mm
7.	Radial rim width	a	25	mm
8.	Central angle of each blade	δ	73	degree
9.	Spacing of blade in wheel	t	22.5	mm
10.	Inlet blade angle	β_1	30	degree
11.	Number of blade	z	20	-
12.	Width of nozzle	b	57	mm

Therefore percentage contribution of each stage to total torque production

$$\%T_{1\to2} = \frac{T_{1\to2}}{T_{3\to4} + T_{1\to2}} = 76.2\% \qquad (17)$$

$$\%T_{3\to4} = \frac{T_{3\to4}}{T_{3\to4} + T_{1\to2}} = 23.8\% \qquad (18)$$

Therefore, theoretically, the first stage contribute 76.2% to the total torque (or power) production while the second stage produce the remaining, 23.8%.

$$T_s = T_{1\to2} + T_{3\to4} \qquad (1$$

Where, $T_{1\to2}, T_{3\to4}$ - Torque transmitted by a blade at stage 1 and 2, Nm.

IV. TEST PROCEDURE

Firstly, the water is pumped to the inlet pipe from the open tank. The flow rate to the turbine is controlled by gate valve which has eight cycles (fully open). The valve was marked at opening one cycle. In this test, the flow was controlled by opening gate valve starting from two cycles to eight cycles (fully open). In the event of an overflow, there is a by-pass pipe line by opening by-pass valve to the tank.

The flow rate in the discharge pipe is measured by a flow-meter located in a section shortly upstream the inlet of the turbine. The pressure is measured by the pressure gauge on the discharge pipe. The runner rotational speed is measured by tachometer. The voltage and current are measured by clamp meter. The measuring instrumentation are shown in Fig. 4.



Fig. 4 Measuring Instrumentation, (a) Tachometer and (b) Clamp meter

V. EXPERIMENTAL DATA AND RESULT

Table II. Experimental Data with Gate Valve Adjustment

Gate valve adjustme-nt ,(cycle)	Flow rate, ($m^3/s \times 10^{-3}$)	Turbine inlet pressure, (kPa)	Runner speed, (rpm)	Generator speed, (rpm)
2-cycle	3.65	17.23	89	182
3-cycle	4.29	24.13	93	285
4-cycle	4.71	31.03	123	375
5-cycle	5	33.09	140	428
6-cycle	5.26	34.47	152	467
7-cycle	5.48	37.92	161	493
8-cycle	5.65	41.37	163	502

Table III. Result Data based on Experimental Data

Gate Valve Adjustment	Effective Head, (m)	Hydrauli c Power, (W)	Angular Velocity, (rad/s)	Shaft Power, (W)
2-cycle	2.1	74.68	40.47	66.53
3-cycle	2.87	120.77	47.3	107.58
4-cycle	3.62	167.1	53.12	148.86
5-cycle	3.86	189.52	54.89	168.83
6-cycle	4.04	208.25	56.11	185.52
7-cycle	4.42	237.26	58.69	211.37
8-cycle	4.79	265.66	61.14	236.66

Table IV. Result Data based on Experimental Data

Gate Valve Adjustment	Voltage, (V)	Current, (A)	Output Power, (W)
2-cycle	33	1.2	39.6
3-cycle	56	1.2	67.2
4-cycle	79	1.17	92.43
5-cycle	90	1.2	108
6-cycle	97	1.17	113.49
7-cycle	103	1.24	127.72
8-cycle	106	1.35	143.1

The experimental data were shown in Table. I and result data based on experimental data were shown in Table. II and Table. III.

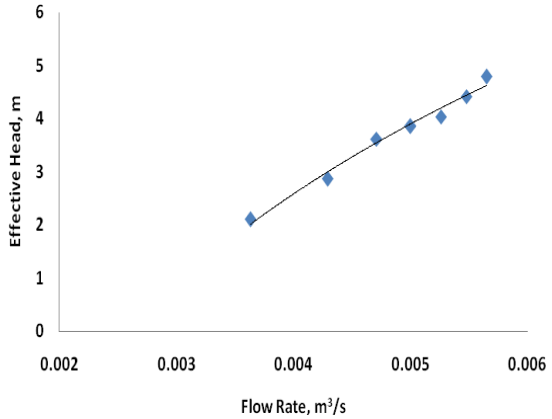


Fig. 5 Effective Head Vs Flow Rate

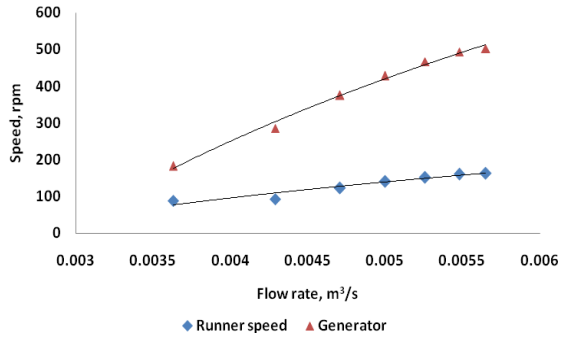


Fig. 6 Speed Vs Flow Rate

From the Fig. 5, it can be seen that the effective head increased with the flow rate increased. From the Fig. 6, it can be seen that the optimum speed of runner is occurred at the maximum flow rate.

Table V. Velocity Components based on Experimental Data

Gate Valve Adjustment	V_1, (m/s)	U_1, (m/s)	W_1, (m/s)	$U_2=U_3$, (m/s)	$W_2=W_3$, (m/s)	$V_2=V_3$, (m/s)
2-cycle	6.2907	3.0239	3.4913	2.0157	2.6749	3.3493
3-cycle	7.3519	3.5335	4.0802	2.3557	3.1261	3.9143
4-cycle	8.2565	3.9683	4.5823	2.6455	3.5107	4.3953
5-cycle	8.5326	4.1010	4.7355	2.7340	3.6281	4.5429
6-cycle	8.7219	4.1920	4.8405	2.7947	3.7086	4.6437
7-cycle	9.1239	4.3850	5.0639	2.9233	3.8794	4.8575
8-cycle	9.5034	4.5676	5.2743	3.0451	4.0409	5.0598

Where,

W - Relative velocity, (m/s)

V - Absolute velocity, (m/s)

U - Peripheral velocity, runner velocity, (m/s)

According to the Fig. 7 and Fig. 8 the maximum output power is at the gate valve fully open condition, however the maximum efficiency is occurred at the gate valve five cycle open condition.

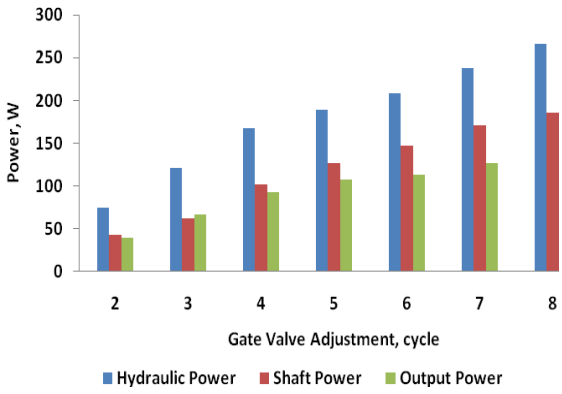


Fig. 7 Power for various gate valve adjustments

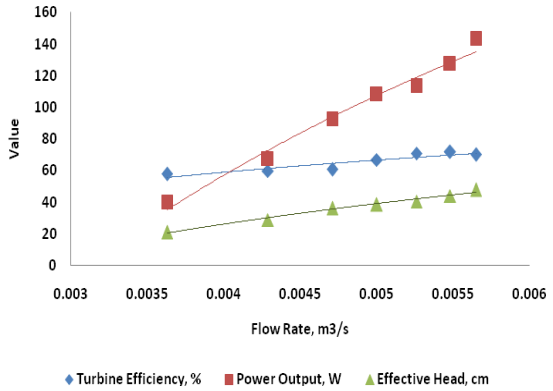


Fig. 8 Performance curve of the Cross-flow turbine in form of gate valve adjustment

CONCLUSION

In this study the experiment of cross-flow turbine has been carried out. In this experiment the maximum effective head 4.79 m and maximum flow rate 5.65×10^{-3} m^3/s were occurred at gate valve fully open. The maximum absolute velocity is 9.5 m/s. The speed of the turbine is between 89 and 163 rpm depending on the flow rate. The turbine maximum efficiency for conditions under study is 71.9% at a head of 4.79m, guide vane opening of 80%. For a small simplified Cross-flow turbine this measured efficiency of 71.9% is relatively high. The theoretical value of torque transfer at stage one and stage two are express.

In this experiment, some measuring instruments such as torque meter, data locker are not available, so the shaft torque value cannot collected by experiment.

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