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Performance Characteristics Of Cross – Flow Impulse Turbine

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Abstract

An experimental investigation was conducted of cross-flow hydraulic turbine, known as Michell-Banki turbine. The turbine runner is 250 mm outside diameter, 168 mm inside diameter and 150 mm length. The test system is provided with guide vanes, centrifugal pump, pitot-static meter, head gauge, tachometer, disc brake dynamometer, and load meter.
Many relevant variables have been studied, e.g. runner speed, water flowrate, head, load, and guide-vane angle. Characteristic curves have been performed. These curves relate unit power and unit flow with unit speed. Results indicate that optimum runner speed is 250 rpm corresponding to maximum power generated and maximum efficiency of 180 W and 45% respectively. The turbine specific speed is found to be 0.4. Empirical correlation relating power coefficient to flow coefficient, Reynolds number, head coefficient, and guide-vane-angle was carried out. Experimental data has been correlated as followed. The model may be used for preleminery prediction of the power of cross-flow turbines having different scales.

$$\frac{P_M}{N^3 D^5 \rho} = 1.076 \times 10^{-2} \left(\frac{Q}{ND^3}\right)^{9.8} \left(\frac{\rho N^2 D}{\mu}\right)^{1.19} \left(\frac{gH}{N^2 D^2}\right)^{-1.21} \theta^{-1.562}$$

Keywords: turbine, cross flow, power, performance, correlation.

Nomenclature

F : Load, N
g : Acceleration gravity, m/s^2
H : Net turbine head, m
N : Shaft speed, rpm
N_s: Dimensionless specific speed, rad.
N_u: Unit speed, rpm
P_M : Mechanical power, W
P_{exp}: Experimental power, W
P_{corr}: Correlated power, W
P_u: Unit power, W
Q : Water flowrate, m^3/s
Q_u: Unit flowrate, m^3/s
r : Runner radius, m

T : Torque, J

Greek symbol

η : Efficiency

ρ: Density, kg/m^3

θ : Guide vane angle, rad.

μ : Viscosity, kg/m.s

1- Introduction

In most of the micro-hydro power systems operating today either have the cross-flow turbine or single to multi-jet Pelton wheel depending on the heads and flow normally encountered. The reason for using these turbines is the advantages offered by these in comparison to the rest of the impulse turbines. That, they are the simplest impulse turbines in construction, operation, and maintenance. Multi-nozzles Pelton-wheel and partitioned cross-flow turbines have generally well part load efficiencies and hence can cope with considerable flow variations normally occurring in micro-hydro systems. At high head Pelton wheel is the most efficient device among the impulse turbines and for a given head and power output cross-flow turbine covers the widest range of application (2-200m) among the several types of turbines that are available and yet offers reasonably good efficiency within this range [1]. Besides , Their floor space requirements are comparatively small.

The cross-flow turbine has attracted the attention of several investigators working in the area of micro hydroelectric power generation. This type of turbine, although primarily an impulse type, is suitable for operation at low and medium heads. It meets the specific requirements of economical utilization of small water resources and is very well suited to fluctuating water-courses. These powerful units are increasingly used as a basis for town and rural power supply generation in the Asia-Pacific region to replace costly diesel imports and reduce noxious greenhouse missions [2]. They are designed to aid people living in remote areas are often denied the benefits of electricity and provides a real low-cost option to using diesel, kerosene, batteries and firewood which are expensive and damaging to the environment. The principle of operation of this turbine is based on a broad jet of water passing twice through the rectangular openings between the adjacent blades of the runner. Water flows as a sheet crossing through the empty center of the turbine and exiting just below the center on the opposite side. In this process the water strikes blades on both sides of the runner providing up to 75% of the power extracted from the entry sheet of water and 25% from the exit flow of the same water.

The aim of the present work is to identify the characteristic curves, specific speed, optimum runner speed and optimum input water flowrate for cross-flow turbine. Also, empirical model will be developed since no attempt had been carried out for this purpose.

2. Theory

2.1 Mechanical power of the turbine is calculated by,

$$P_M = \frac{2\pi.N.T}{60} \qquad (1)$$

Where,

$$T = F.r \qquad (2)$$

2.2 Water Power is given, as

$$P_W = \rho.g.H.Q \qquad (3)$$

2.3 Turbine efficiency is determined as followed,

$$\eta\% = \frac{mechanicalpower}{waterpower} \times 100 = \frac{P_M}{P_W} \times 100 \qquad (4)$$

3. Dimensional Analysis

Fig. 1. shows a control volum which represents a turbine through which water of density ρ and viscosity μ flows at a volume rate of Q, controlled by guid vane angle θ . The head difference across the control volume is H. Turbine diameter of D develops a shaft power P_M at a speed of rotation N , then the power output is a function of all of above variables, or

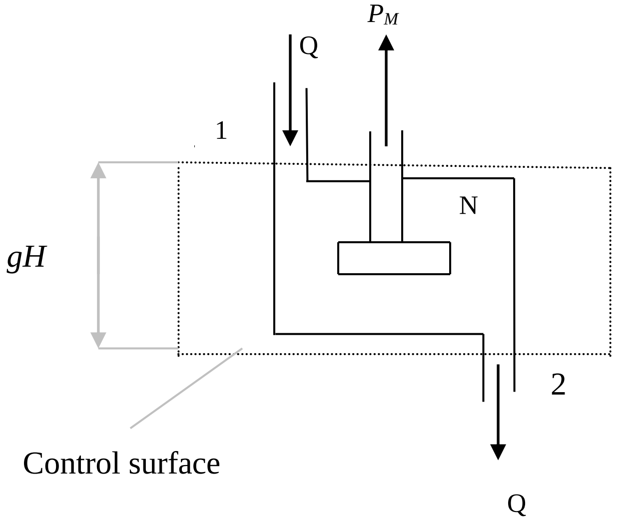

Fig.1. Cross flow turbine control volume

$$P_M = f(\rho, N, \mu, D, Q, (gH), \theta) \qquad (5)$$

Using Buckingham's Π theorem [3], the following empirical correlation is obtained,

$$\frac{P}{N^3 D^5 \rho} = K\left(\frac{Q}{ND^3}\right)^a \left(\frac{\rho N^2 D}{\mu}\right)^b \left(\frac{gH}{N^2 D^2}\right)^c \theta^d \qquad (6)$$

where: K, a, b, c, d are constants.

4.Experimental Study:

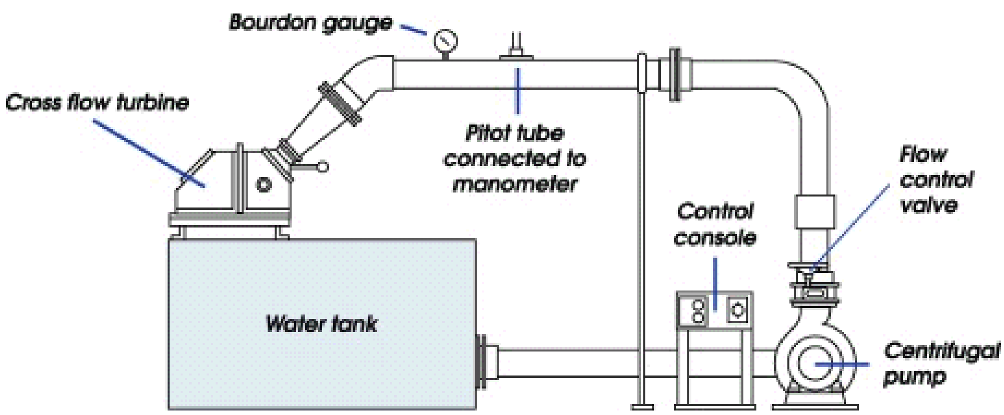

Fig. 2. Schematic of circuit (brake and tachometer omitted for clarity) [4] .

The apparatus that shown diagrammatically in Fig. 2. comprises a steel tank on which is mounted a cross flow turbine. The turbine is a low radial impulse type horizontal shaft machine and is flexibly coupled to a disc brake dynamometer by which the output torque is measured. The shaft speed of the turbine is directly indicated on a belt-driven tachometer. The flow of water through the machine, and hence the power output, is controlled by a moving guide vane. This vane is connected by levers to an operating linkage which in turn is actuated by a single hand lever incorporating a vane opening position lock. The turbine pressure is indicated on a bourd0n type gauge mounted on the inlet transition. The complete turbine assembly is mounted on a substantial steel bedplate which in turn is bolted to the steel tank. An electrically driven centrifugal pump floor mounted alongside the tank draws water from the tank through a suction pipe and delivers it to the turbine via a valve and pipework system. An averaging Pitot tube fitted on the delivery pipe indicated flow measurement directly on an inclined manometer.

4. Results and Discussion

Figure 3. relates turbine brake power with input flowrate at different guide vane angles, (5^0 – 20^0). At fixed angle, the power initially increases to a maximum value with water flowrate , then it decreases due to the design of the runner vanes which cause power losses at high water flow. The optimum water flow is approximately 26.5 *l*/s. Figure 4. shows brake power versus runner speeds, (110 – 500) rpm. It is obvious that the relationship is of the second degree as in Fig. 3. There are no power at zero and maximum runner speed. For the former the torque is maximum due to high load but the speed is zero while it is the reverse for the second case and the runner rotates freely without load. Therefore, to have a power one needs both the force and revolution speed. At certain angle, the power increases with runner speed to a maximum value which corresponds to the optimum speed, then it decreases due to reduction in the torque. At fixed runner speed, the power increases with guide vane angle, since the flow is increased. Figure 5. demonstrates the angular momentum (torque) against turbine shaft speed at different guide vanes angles. The torque reduces with the runner speed as result of lowering the load on turbine. At a definite speed, the torque increases with the angle since water flow increases which requires more load to keep same speed.The effect of shaft speed on turbine efficiency has been investigated as shown in Fig. 6. Comparing Fig. 6. with that of Fig. 4., prevailing mechanical power, it is found that the relations are of similar trend with concave second degree curves which agree with theoretical approaches [5]. Moreover, the optimum runner speed for power and efficiency curves is approximately overlapped at 250 rpm which occurs at nearly maximum mechanical power and maximum efficiency of 180 rpm and 45 %, respectively.

Performance characteristics of turbines are often identified in terms of unit speed, unit power, and unit flowrate as shown in Fig. 7. and Fig. 8. which arose from the need to be able to compare hydraulic turbines tested under a set of standard conditions. These parameters are:

$$N_u = \frac{N}{H^{1/2}} \quad , \quad P_u = \frac{P}{H^{3/2}} \quad , \quad Q_u = \frac{Q}{H^{1/2}}$$

Dimensionless specific speed is a design parameter at maximum efficiency, used frequently to indicate the type of turbine that should be used for a given P, N, and H. Fig. 9. illustrates efficiency versus N_s. The dimensionless specific speed for the turbine under study is found to be 0.4 which is within that of pelton wheel turbine, 0.05-0.4 [6]. The high value of N_s returns to the low head required for cross-flow turbine.

Empirical model has been characterized relating power coefficient, $\frac{P_M}{N^3 D^5 \rho}$, with flow coefficient, $\left(\frac{Q}{ND^3}\right)$, Reynolds number, $\left(\frac{\rho N^2 D}{\mu}\right)$, head coefficient, $\left(\frac{gH}{N^2 D^2}\right)$, and guide vane angle, θ in radian as followed.

$$\frac{P_M}{N^3 D^5 \rho} = 1.076 \times 10^{-2} \left(\frac{Q}{ND^3}\right)^{9.8} \left(\frac{\rho N^2 D}{\mu}\right)^{1.19} \left(\frac{gH}{N^2 D^2}\right)^{-1.21} \theta^{-1.562}$$

It is obvious that flow coefficient has pronounced effect on turbine power among other parameters. Least square method has been used to correlate the experimental data with the aid of basic language program. The mechanical power from above correlation is plotted versus experimental value, as shown in Fig. 10. with an upper and lower limit of $\pm 25\%$ within which 94% of the experimental data lies. Thus, it may be served as preliminary prediction of cross-flow turbine power.

5. Conclusion

- Maximum output power and efficiency achieved from cross flow turbine under study are 180 W and 45%, respectively.
- The optimum input water flow and optimum shaft speed of the turbine corresponding to the maximum efficiency and power are 26.5 *L*/s and 250 rpm, respectively.
- Performance characteristics curves for cross- flow turbine have been performed.
- Dimensionless specific speed of the turbine is determined to be 0.4 .
- For initial power estimate of cross-flow impulse turbines having various scales, it is recommended to use empirical model developed in the present study.

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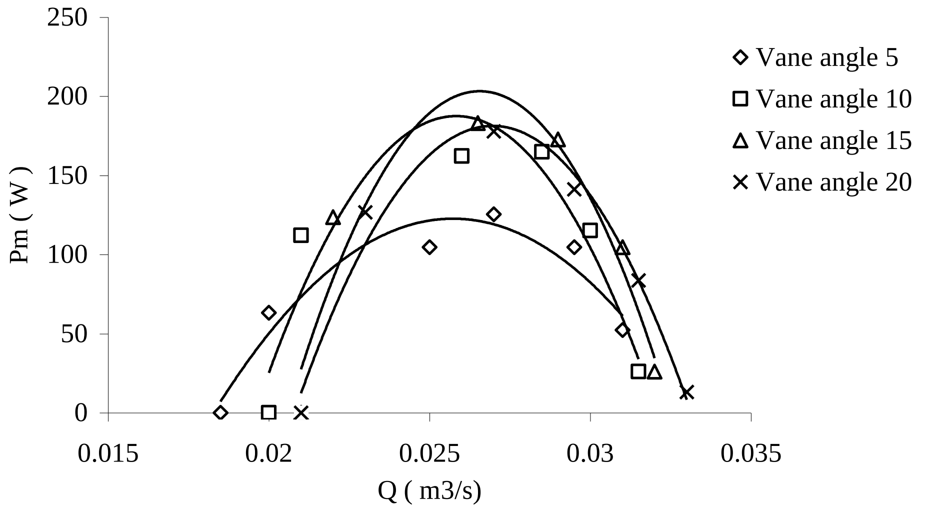

Fig. 3. Turbine mechanical power versus input water flowrate

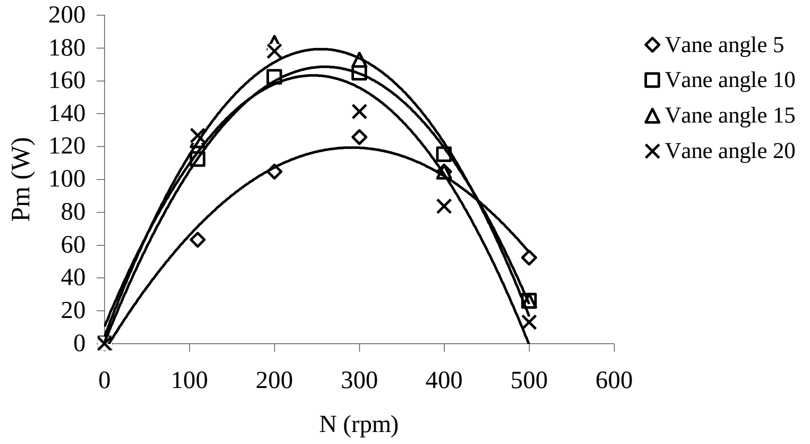

Fig. 4. Turbine mechanical power versus shaft speed

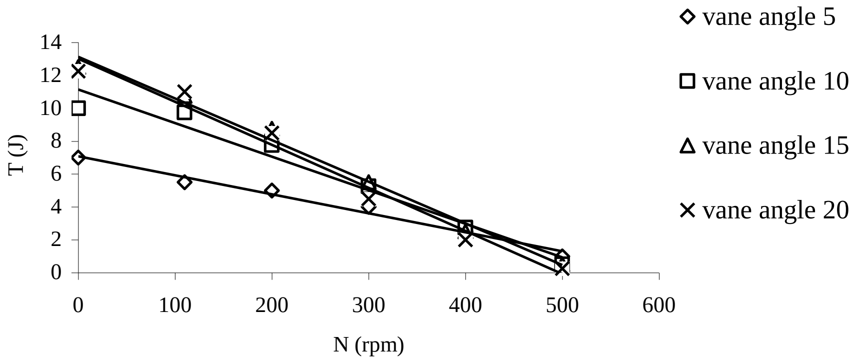

Fig. 5. Torque versus shaft speed

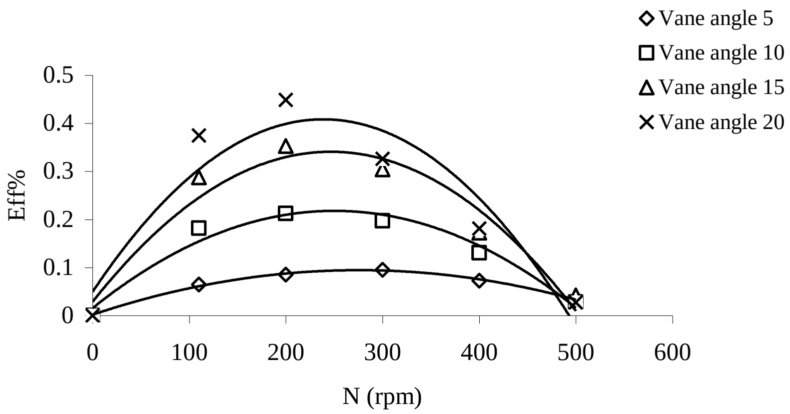

Fig. 6. Turbine efficiency versus shaft speed

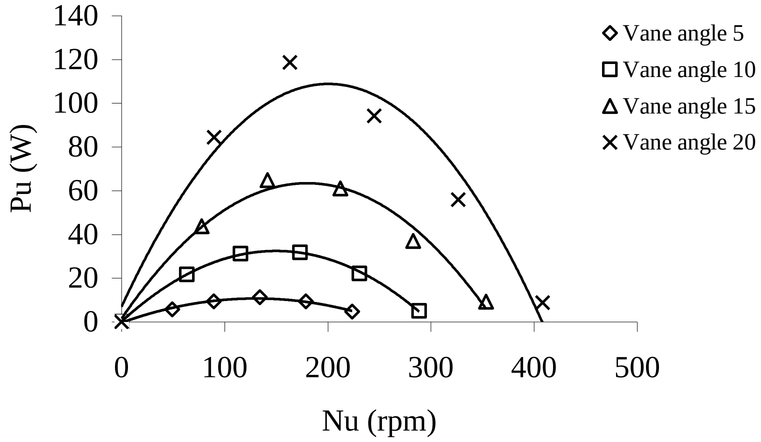

Fig. 7. Unit power versus unit speed

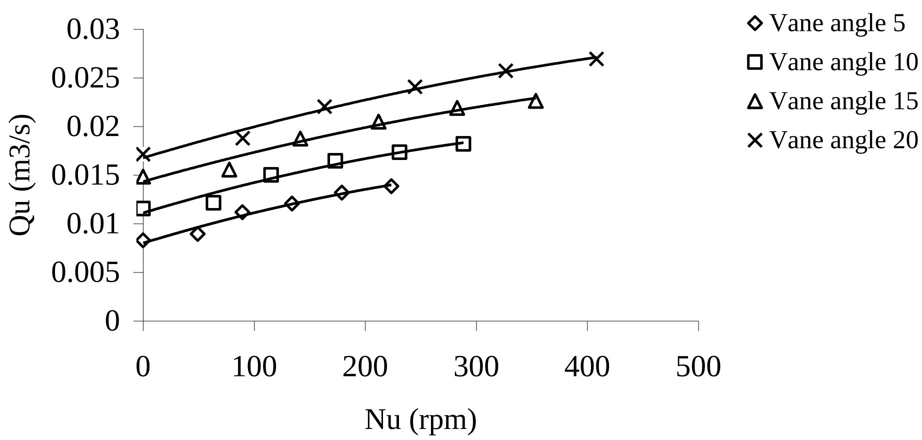

Fig. 8. unit flow versus unit speed

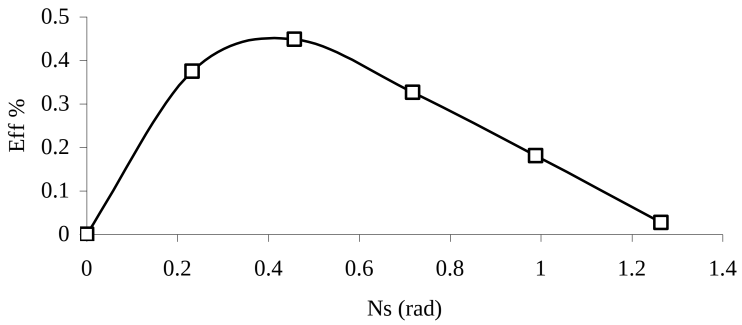

Fig. 9. Turbine effeciency versus dimensionless spesific speed

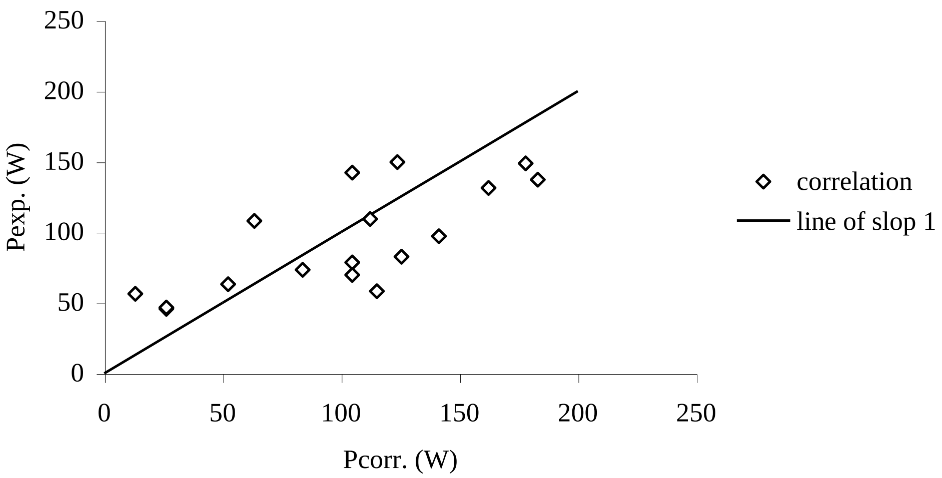

Fig. 10. Experimental versus correlated power