Modifications in a cross flow turbine for performance improvement

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Several modifications in the standard design of a cross flow turbine have been incorporated to improve the performance of the turbine. Experimentally it is observed that optimum shaft diameter and adding a well designed draft tube improves the performance of the turbine. The other two modifications (provision of a nozzle insert and an additional nozzle) did not improve the performance of the turbine. It was also found that the optimum size of the shaft diameter for the present cross flow turbine lies in the range proposed by Banki.

Micro-hydropower systems have been universally recognized as an attractive alternative to conventional sources for meeting the ever increasing energy needs of the society. Cross flow turbines because of their flatter characteristics with respect to head are frequently used in such systems. These turbines are essentially scaled down versions of larger turbines to match the lower head and power requirements. Cross flow turbine was first invented and patented by A G M Michell and further developed by Banki. This turbine did not gain much importance because it had low efficiency in comparison to Pelton wheel and was not seen to be economically viable. Importance of Micro-hydro systems world wide, did rekindle interest in this turbine. Mockmore and Merrifield carried out detailed experimental investigations and showed that cross flow turbines could operate efficiently under a wide range of flow variations than most other turbines. Some experimental studies have been made on the effect of runner shaft diameter, runner width, blade number and draft tube on the performance of cross flow turbine. It has been noted that the optimum shaft diameter for maximum efficiency agreed with the range given by Banki, however, the draft tube had adverse effect on the performance of the cross flow turbine. Other important investigations have also been made. Investigations carried out by Khosrowpanah and Joshi et al. have been more wide ranging where they have been able to clearly identify the effect of blade number, nozzle entry arc and head on the performance of cross flow turbine. The use of draft tube with these turbines has also been investigated by Vanleer and Heimert and reinvestigated by Toyokura et al.

From their studies, they concluded that for using the draft tube effectively, the water level in the runner chamber should be kept at 90° of the height of the runner diameter measured from lower tip at the runner. However, the results available in literature are some what contradictory. The aim of the present study is to relook into the effect of draft tube and shaft diameter on the performance of this turbine when all its other parameters have been optimised. The other modification attempted was that in the nozzle so as to use the leakage water from the nozzle which does not go through the runner.

**Experimental Procedure**

The schematic diagram of the experimental set-up of a prototype cross flow turbine used in the study is shown in Fig. 1. The main components of the set-up are supply pipe line, nozzle, runner system, prony brake system and measuring instruments. The detailed operational concept of this set-up has already been described. The important dimensions of the turbine set-up are as below.

Nozzle
- Maximum nozzle entry arc: 36°
- Entry angle to the runner: 16°
- Width of the nozzle: 25 mm

Runner
- Number of blades: 20 (original)
- Length of the runner: 0.325 m
- Diameter of the runner: 0.3 m
- Shaft diameter: 63 mm (original)
- Housing dimensions: (80 x 54 x 41) cm³

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Draft Tube Details

Inlet dimension: \(820 \text{ mm} \times 420 \text{ mm}\)
Exit dimension: \(203 \text{ mm diameter}\)
Length: \(750 \text{ mm}\)
Straight transition piece length: \(300 \text{ mm}\)
Conical diffuser length: \(780 \text{ mm}\)
Conical exit diameter: \(284 \text{ mm}\)
Overall length: \(1830 \text{ mm}\)

The water was supplied to the turbine from a large sized overhead tank situated at an elevation of approximately 15 m above the ground level. The level of water in this tank was maintained constant by operating suitable pumps. The flow rate into the turbine \(Q\) was measured by a pre-calibrated orifice meter provided in the supply line. The inlet head \(H\) to the turbine could be varied by operating a gate valve in the supply pipe. The turbine could be loaded by a prony brake system and from the measured values of speed of the runner at various torques the output of the turbine is calculated. At any given setting of the turbine, the input power is given by

\[
\text{Power input} = \rho Qgh
\]

The output of the turbine is

\[
\text{Power output} = Tw
\]

where \(T\) = torque on the prony brake which is \(= (W - S)D/2\)
where \(W\) = weight hung from the brake drum, \(S\) = spring balance reading, \(D\) = diameter of the brake drum and \(w\) = angular velocity of the turbine which is \(= 2\pi N/60 \text{ (N= rpm)}\)

The model had the facility of systematically varying various parameters like inlet head, nozzle entry arc and blade numbers. The results from such a study are reported elsewhere. In addition, a draft tube with an inlet size of 250 mm and having a geometry as shown in the figure has been designed and fabricated. This was incorporated in the turbine and its effect on the performance of the turbine has been investigated at various inlet heads. During these tests the casing of the turbine was ensured to be air tight.

An attempt has also been made to utilize the leakage water from the nozzle. An insert and a second nozzle were incorporated in the turbine and the effect of these on turbine performance has been evaluated individually. The geometrical details of these modifications are given in Fig. 2. The housing of the runner was made of perspex sheet so that the flow could be visualised. The torque for evaluating the performance was measured using prony brake system. The effect of draft tube has also been established for a runner with 24 blades and a shaft diameter of 63 mm. The nozzle was modified as per the suggestion of Sonnek and this has been achieved by fixing a specially designed insert at the end of internal nozzle end surface (Fig. 2a). Another attempt was made by installing a second nozzle for reutilizing the issuing flow through the nozzle/runner gap (Fig. 2b) having identical shape at the exit as the first nozzle but it covered four blades as compared to 2 blades by the first nozzle.

In the present study, the effect of shaft diameter on the performance of the cross flow turbine is reported. In particular, four shaft diameters namely 53, 63, 68 and 77 mm have been tried and these are chosen to be in the range suggested by Banki. For each shaft, the head \(H\) is varied in the range 2-9 m and the performance characteristic curves (efficiency vs speed) have been obtained at each head. The performance of the tur-
bine is quantified by the (a) unit discharge, $Q_u = Q/\sqrt{H}$, (b) $\phi_0 =$ speed ratio at the b.e.p. = $N_0/\sqrt{2gH}$, (c) $\phi_R =$ speed ratio at the run away conditions = $N_R/\sqrt{2gH}$ and (d) $\eta_{\text{max}} =$ maximum efficiency of the turbine.

Results and Discussion

The effect of various modifications attempted$^{12}$ have been quantitatively presented in terms of variation in different performance parameters like $Q_u$, $\phi_0$, $\phi_R$ and $\eta_{\text{max}}$ in the present study.

Effect of shaft-diameter—The performance of cross flow turbine has been evaluated as a function of shaft-diameter ($d_s$) in terms of its effect on various parameters for a turbine having 20 blades with various nozzle entry arcs in the range of 23-36°. Fig. 3 shows the variation of unit discharge ($Q_u$) as a function of shaft diameter at various heads ($H$). It is observed, that at all heads within the experimental uncertainties the unit discharge remains fairly constant for all shaft diameters. This implies that the shaft diameter does not have any significant effect on the flow rate and hence on the input power. It was further observed that both $\phi_0$ and $\phi_R$ did not show any systematic variation with change in shaft diameter. This was found to be true at all heads tested. The values of $\phi_0$ were in the range 0.42-0.44 whereas the corresponding range for $\phi_R$ was 0.90-0.95. It was also noticed that at the largest shaft diameter tested ($d=77$ mm) the value of $\phi_R$ at higher heads

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Fig. 2—Details of the modifications attempted, (a) Additional Nozzle, (b) Nozzle insert and (c) Draft tube (all dimensions in mm).

Fig. 3—Effect of shaft-diameter on the unit discharge for different supply heads ($\lambda=32^\circ$, $n=20$).
tended to show a decreasing trend thereby indicating higher losses.

Fig. 4 shows the variation of maximum efficiency as a function of shaft-diameter for \( H = 9 \) and \( 10 \) m and for \( \lambda = 23, 26 \) and \( 32^\circ \). It is observed that for both the heads and all nozzle-entry-arcs the peak efficiency is obtained at a particular shaft-diameter ratio of 0.19 corresponding to the shaft-diameter of \( 63 \) mm which is within the range given by Banki. For all other shaft-diameters tested during the study the peak efficiencies are lower. Similar trends were observed at all other heads. It is further observed that the peak efficiency is obtained at the largest nozzle-entry-arc of \( 32^\circ \) and the variation of maximum efficiency with shaft-diameter for this is not that pronounced. This is shown by the flattening of the curve at that nozzle-entry-arc implying that the effect of shaft-diameter at large nozzle-entry-arc is somewhat inhibited: The reason for the peak efficiency to have highest value at a particular shaft-diameter is attributed to its contribution in streamlining the flow coming out of the first stage and guiding it properly to the second one. To understand this effect, flow was visualized from the side plates which were made of plexiglass sheet for the three shaft diameters with all other parameters being identical. The flow pattern through the runner at the optimum speed (\( \phi_0 \)) has been sketched in Fig. 5 for the three diameters. For the small diameter it is seen that influence of shaft-diameter is not felt on the jet due to significant gap between the top surface of the jet and the shaft surface. For shaft-diameter of \( 63 \) mm, the jet top surface slides just along the surface of the shaft diameter which imparts momentum to the jet which implies that jet spreads without losing its momentum and, therefore, for this diameter, maximum power output is achieved. For the larger diameter, the flow is directly obstructed by the shaft which leads to drop in achievable efficiency. From this visualization, one can easily conclude that there is always an optimum shaft-diameter for achieving highest efficiency. Flow visualization done by Gautschi et al.\(^3\) also showed similar results. The present experimental measurements also support the observations made on the basis of flow visualization.
Effect of draft tube—In the initial stages of development of cross-flow turbines, these were considered to be pure impulse devices and hence were operated without any draft tube. Later it was found that the flow at the runner exit still had some 8% of kinetic energy, from which, according to the prediction, some 3–6% could be regenerated, through the use of draft tube. Today most of the cross-flow turbines are operated with draft tubes wherever the heads are small. There are, however, cases where they are not used because the use of draft tube requires somewhat sophisticated technology to make it air tight and this always involves higher skills and higher investment. Another reason for not using the draft tube, specially in the developing countries, is the prevailing prejudice of the manufacturers, who believe that its use would reduce the performance of the turbine because of the wading of its runner. However, many authors have advocated the use of draft tube. It seems, hence, necessary to carry out the experiments to establish the actual fact. For this reason experimental studies have been carried out in the present study with a draft tube for a turbine having 24 blades, a nozzle entry arc of 36° and shaft diameter of 63 mm.

Fig. 6 presents the effect of draft tube on the basic parameters namely \( Q_u \), \( \phi_R \), \( \phi_0 \) and \( \eta_{\text{max}} \) with head as the primary variable and are compared with results without draft tube. Fig. 6a shows the variation of \( Q_u \) with head with and without draft tube. It is seen that at lower heads, the value of \( Q_u \) is significantly large with draft tube and the gap reduces as head is increased. This implies that at lower heads, draft tube is able to extract more input power for the same head. Fig. 6b shows the variation of \( \phi_R \), and it is seen that only for very low heads, this value is higher with the draft tube and with increase in head this value drops with draft tube whereas without draft tube it increases. This can be attributed to the limited capacity of the draft tube. With the present sizing of the draft tube, at higher heads it could not handle the increased flow. The variation of optimum speed ratio \( \phi_0 \) with head is shown in Fig. 6c. It is seen that \( \phi_0 \) for a runner with draft tube is not a function of head and remains constant for all heads tested and the value is close to 0.48 which has been established to be the optimum value. It is also seen that the addition of draft tube has no significant effect on the value of \( \phi_0 \).

Fig. 6d shows the variation of maximum efficiency with head with and without draft tube. It is seen that the maximum efficiency (\( \eta_{\text{max}} \)) with draft tube is significantly higher at lower heads and this difference decreases with increase in head and for larger heads it becomes even lower in comparison to the value without draft tube. The remarkable increase in efficiency at low head is not quite understood. At this head the suction pressure in the housing was only nominal and visual observation showed that no water was raised up to the runner level. Therefore, the runner was not wading in water. At higher heads the suction pressure in the housing as well as at runner exit was substantial. At 4 m head the water inside the housing was very turbulent at higher runner speeds which could be the reason for drop in efficiency. At larger heads and higher speeds the runner was partially immersed inside the water. Only at lower speed the level of water decreased gradually and only part of the runner was wading. It was found that the raising of water inside the housing at higher heads (i.e., higher flow rates) was caused merely by the incomplete drainage of exit flow and the reason behind it was obviously the undersizing of the draft tube.

Thus it can be concluded that as long as the draft tube is sized properly, it is possible to achieve improvement in the performance of crossflow turbine even at higher heads. However, additional data with different sizes of draft tubes is needed to support this conclusion. On the basis of the present study, it may be concluded that it is definitely advantageous to use draft tube at low heads.

Effect of modifications in the inlet nozzle—Two modifications have been attempted in the nozzle.
which are described in the following sections. These modifications were tried for \( n = 24, d_s = 63 \) mm and flow entering through the nozzle with \( \lambda = 36^\circ \). The first modification was based on the suggestion made by Sonnek in 1923. He observed that the exit flow through the nozzle/runner gap did not proceed parallel to the nozzle-end surface but diverted away from it by a certain angle. So he proposed to design the end of this nozzle surface in such a way that this diversion is avoided and the uniform angle of attack \( \alpha_1 \) is maintained. This suggestion has been incorporated in the present modification by fixing a specially designed insert at the end of internal nozzle-end-surface as shown in Fig. 2. It was observed that there was hardly any significant visible change either in the direction of flow or in the volumetric loss through the nozzle/runner gap. Further there was no significant change in the measured values of maximum efficiency at various heads. Thus it can be concluded that the addition of an insert is not beneficial.

The second modification in the nozzle tried was to enhance the performance of the turbine by reutilizing the flow issuing through the nozzle/runner gap which otherwise would have been wasted without doing any work. To achieve this an addition nozzle (Fig. 2) was fabricated which is identical to the lower end of the main nozzle but covered four blades instead of two only. It was fixed very close to the runner at a distance of one bladespacing below the original nozzle. This arrangement enabled the water issuing through the nozzle/runner gap to restrike the runner blades at the same angle of attack \( \alpha_1 \).

It was observed that at higher runner speeds a significant part of water issuing through the gap could not pass through the additional nozzle and was overflowing from its top. But as the speed decreased, i.e., when the turbine was gradually loaded, the amount of water flowing through the gap as well as that over flowing through the top of the additional nozzle reduced. This phenomena was observed at all the heads. It was also observed that a considerable part of flow issuing through the additional nozzle was turned over by the blades in the first stage itself and could not cross the runner.

Measurements showed that even with this kind of modification the performance of the turbine did not show any improvement. Instead there was a slight deterioration both in terms of maximum efficiency and runaway speed at lower heads. The deterioration in the performance at low heads could be attributed to two factors. Firstly, the jet of flow passing through the turbine from the main nozzle might have been distributed and secondly the flow through the additional nozzle could hardly impart any energy on the turbine as it is already at a very low velocity. From the above observations, it can be concluded that the use of additional nozzle does not contribute to enhancement in the performance of the turbine.

Conclusions
The present investigation has shown that the size of the shaft in a cross flow turbine has a significant effect on its performance and the optimum value for the diameter ratio \( (d_s = d_s/d) \) is 0.19. Further, the draft tube always improves the efficiency of the turbine as long as it is properly sized to handle the flow. The various suggestions reported in the literature to reutilise the leakage water are found to be not effective.

Nomenclature

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\begin{align*}
D &= \text{diameter of the brake drum, m} \\
d &= \text{impeller diameter, m} \\
d_s &= \text{diameter ratio, } d_s/d \\
d_e &= \text{shaft diameter, m} \\
g &= \text{acceleration due to gravity, } m^2/s \\
H &= \text{inlet head, m} \\
N &= \text{speed, rpm} \\
n &= \text{numbers of blades} \\
N_o &= \text{optimum speed at best efficiency point, rpm} \\
N_r &= \text{run away speed, rpm} \\
Q &= \text{flow rate, } m^3/s \\
Q_u &= \text{unit discharge} \\
S &= \text{spring balance reading, kg} \\
W &= \text{weight hung from brake drum, kg} \\
w &= \text{angular velocity of the turbine} \\
\phi_0 &= \text{speed ratio at best efficiency point} \\
\phi_r &= \text{speed ratio at run away condition} \\
\rho &= \text{density of fluid, } kg/m^3 \\
\eta_{\text{max}} &= \text{maximum efficiency of the turbine} \\
\lambda &= \text{nozzle entry arc, degree} \\
\alpha_1 &= \text{angle of attack, degree} \\
\end{align*}
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References
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