CHAPTER 2
LITERATURE REVIEW

2.1 HISTORY

The inventor of cross flow turbine (CFT) was the Australian Engineer **A.G.M. Mitchell** who obtained a patent for the machine in 1903. The turbine was based on the theory of **Poncelet**, a French engineer (1788-1867) who improved the classical horizontal axis water wheel. No further information is available about Mitchell's patent. For a long period the investigations on CFT was not reported probably because the thrust was for Mega Hydel Power Plants during this period. Due to the present shift in importance for Mini and Micro Hydel systems and the environmental concerns, a considerable amount of work has been reported in the recent past. A brief and in-depth investigation of the available literature is presented in this chapter, which has led to the problem for the present investigation.

The Hungarian professor **Donat Banki** in West Germany did extensive work on his CFT between 1912 and 1919 and popularized it through his series of publications. So CFT is sometimes known as the Banki Turbine. Through his work, Banki concluded that to get maximum efficiency, the angle with which the water jet hits the blade should be as small as possible. Based on this assumption, he calculated the blade angle at the outer and inner periphery of the runner, the radial rim width, the path of flow through the runner and the blade curvature. He considered all possible losses that occurred in the nozzle and the runner and expressed the maximum possible efficiency as:

$$\eta_{max} = 0.771 - (0.384\, D_m/H) \qquad \text{-- (2.1a)}$$

$$\text{Also, } \eta_{max} = \cos^2\alpha_d \qquad \text{-- (2.1b)}$$

Where D_m is the runner diameter and H is the total Head.

Fritz Ossberger, a German mechanical engineer has refined Mitchell's original design and obtained a patent for the turbine developed. He manufactured a range of units handling flow rates of 200 - 700 litters /sec, heads of 1-m- 200 m, ratings varying from 1 kW to 1 MW and output speeds vary between 50 and 100 rpm.

Sonnek (1923) modified the Banki theory by assuming a constant blade angle of 30° instead of assuming a constant angle of attack. He obtained an expression for the maximum efficiency considering all the possible losses through the nozzle and runner, which is

$$\eta_{max} = 0.863 - (0.264\, D_m / H) \qquad \text{--(2.2)}$$

Sonnek recommended that the nozzle width be narrower than the runner width to reduce the separation on the backside of blades.

Van Leer (1929) compared the effect of head and speed on cross flow turbine performance with undershot and overshot wheel arrangement. He pointed-out that the Banki turbine filled the specific speed gap between the Pelton wheel and Francis turbine i.e. it had a specific speed range of 6 to 20 units. Van Leer suggested the cross flow machines are more suitable in small isolated farms or remote plants.

2.2 REVIEW OF EXPERIMENTAL ANALYSIS

Mockmore and Merryfield (1949) tested a model of the CFT with a 33-cm diameter runner, inner to outer diameter ratio of 0.66 and 20 blades. They investigated the performance of the machine under five different heads between 2.74 and 5.49-m for different nozzle width (gate opening). The maximum efficiency obtained was 68 percent with 4.88-m Head.

The major observations were

1. The break power output is proportional to $H^{1.5}$.
2. The CFT could be operated efficiently on a wider range of openings than most turbine
3. Maximum efficiency occurred at the same speed for all gate opening for a given head.
4. The cross-flow machine's characteristic speed occupied a position between impulse and that of reaction turbines.

Shepherd (1956) calculated the rate of energy transferred from the water to the blades in the first and second stages of the turbine. He pointed out that about 72 percent of the fluid energy were transferred to the runner at the first stage and the remaining 28 percent of the energy at the second stage.

Varga (1959) conducted experiments to measure the pressure distribution along the nozzle walls. A total of 24 gauging points were provided covering the upper and lower sections as well as the sidewalls of the nozzle. All the gauging points were connected to a multi manometer system. Pressure distributions along the walls were measured with and without the runner, and a trajectory network was constructed, similar to a flow net. The maximum efficiency reported in these experiments was 77 percent. Varga inferred that the CFT works as an impulse turbine only in the range of well-defined speeds with its upper limit being the point of maximum efficiency. It was also confirmed that at the maximum efficiency the ratio of the peripheral and absolute velocities are equal to half the cosine of the angle attack at the first stage inlet. He also concluded that the trajectory network and the pressure distributions along the nozzle walls could determine the momentum required for regulation of the CFT.

Haimerl (1960) introduced the possibility of adding a draft tube to the turbine, to increase the effective head on the runner. Also he mentioned that the cross flow turbine was not a pure impulse turbine as classified previously, because at the exit of the nozzle the pressure is slightly greater than atmospheric. He compared a cross-flow turbine and a Francis turbine for the amount of power generation during a period of three months. He also pointed out that to avoid fatigue failure, runners of large width should have one or two intermediate discs to reduce the unsupported blade span and runners should be balanced before being assembled into the turbine.

Arter (1979) extensively reviewed the theory of the cross-flow turbine. He derived the value of 0.68 for the inner to outer diameter ratio of the runner by considering the blade thickness and the losses that occur in the turbine. He also calculated the stresses and moments on the blades and the shaft to improve the design of the turbine components. He presented a simplified chart for the design of the cross-flow machine.

Johnson *et al* (1982) designed and tested a CFT with a wooden casing, PVC runner plates and vanes, and polymer coated wooden nozzles and guide vanes. The tests indicated that a single non-segmented design could achieve efficiencies of 60 to 80 percent over a wide range of flow rates and heads, at various runner speeds. For a runner with 18 blades, the maximum efficiency was obtained at a head of 0.91-m to 1.07-m. The root mean square uncertainty in the measured turbine efficiency was ±6 percent. The best results were obtained when the water in the draft tube was just at the bottom of the runner. They also concluded that with proper nozzle design, efficiency would remain high up to a nozzle entry arc of 120^0.

Nakase *et al* (1982) studied experimentally the effect of nozzle shape on the performance of the CFT. They used nozzles of different rear wall shapes, and concluded that circular and logarithmic spiral shape were both equally more efficient than an intermediate shape. Similarly, a nozzle with an entry arc of 90^0 was found to more efficient than nozzles with entry arcs of 30^0, 60^0, and 120^0. The most suitable value of the nozzle throat width ratio ($2S_0/D_1\delta$) was found to be 0.26, changing slightly with the nozzle entry arc. The absolute maximum efficiency obtained was 82%.

Toyokura *et al* (1986) derived the design data for an air – water mixed flow and investigated the optimum matching condition of turbine elements. They concluded that the runner, whose inside jet flow passes through underside of the shaft, gives the best efficiency for low heads. In this case a high water level in the runner chamber is desirable to utilize a draft head as possible. Furthermore, the effects of the head and the runner diameter on the performance were analyzed. They concluded that the turbine performance is scarcely affected by the head, and the efficiency does not lower even at an extremely low head.

Khosorowpanah *et al* (1988) were probably were the first to attempt a parametric study of the CFT nozzle as well as the runner. The experimental work involved the study of the effect of the number of blades, runner diameter, and nozzle entry arcs under flow/head variations. Water was admitted vertically through nozzles with entry arcs of 58^0, 78^0, and 90^0. They concluded that the maximum efficiency of the CFT increases with an increase in the nozzle entry arc from 58^0 to 90^0 and decreases slightly with the decrease in runner diameter at constant runner width. For the four runners tested, it was observed that maximum efficiency 80% was when the nozzle throat width ratio ($2S_0/D_1\delta$) was 0.41 and the speed ratio was 0.54, and that the number of blades has only a moderate effect on efficiency and power output. The runner with 15 blades was found to be more efficient than runners with either 10 or 20 blades.

Durgin and **Fay** (1984) built an Acrylic model of the CFT, and tested nozzles with entry arcs ranging from 50° to 80° and obtained a maximum efficiency of 66%. This maximum efficiency compared very well with their theoretical prediction. They also extracted the cross-flow making it to flow through a pipe with a slot to determine the power produced by the first stage which is about 83% while the rest of the power came from the second stage.

Chiatti and **Ruscitti** (1988) introduced an internal rotating deflection element inside the blade crown to guide the stream and to avoid focusing effects. Their prototype runner was 250-mm in outer diameter, having 24 blades of 145-mm length. The jet width, design power and the design speed were 4-mm, 140 kW and 3000 rpm respectively. The test bed was equipped with a digital speed meter and a brake mounted on a oscillating support with a dynamometer having a least count of 0.05 N. The maximum over all efficiency obtained was above 75% with an uncertainty of ± 3%. The optimum inlet angle and deviator-to-blade phase angle were close to the theoretical values.

Fiuzat and **Akerkar** (1989, 1991) conducted experiments to prove the effects of factors such as the angle of attack, nozzle entry arc and nozzle configuration on efficiency. Five nozzles were constructed, each with a throat width ratio of 0.41. All the nozzles had circular back walls with angles of attack of 16°, 20°, 24° and nozzle entry arcs ranging from 90° to 120°. Three runners were constructed with angles of attack 16°, 20° and 24°, outer diameter of 305-mm, inner to outer diameter ratio of 0.68 and 20 blades. All the runners had interior guide tube inside the runner. The results indicate that a vertical nozzle orientation is more efficient than horizontal and slant orientations. They also observed that the nozzle entry arc of 90° was an optimum and efficiency increases with increase in angle of attack from 16° to 24° with maximum efficiency of 89% with 24° and vertical orientation.

Ott and Chappell (1991) described an actual case in which a CFT was installed at the Arbuckle mountain site in Northern California. The project is reported to have a capacity of 336 kW, a turbine manufacturing cost of only US $ 304 per kW and a peak efficiency of 79.3% which occurred at about half opening and full opening. Efficiency was observed to increase with the wading depth (i.e., submerging depth). When the gate was fully open, two thirds of the runner was submerged and the efficiency was at its peak.

Desai and Aziz (1994a, 1994b) carried out an extensive experimental investigation with CFT. A total of 39 runners and 11 nozzles to cover 75 different combinations were fabricated. The angle of attacks of 24^0, 28^0, 32^0 and D_2/D_1 of 0.6, 0.68, 0.75 were tested. Numbers of blades used were 15, 20, 25, and 30. All nozzles had nozzle entry arc of 90^0, nozzle-throat width ratio were varied as per the equation $2S_0/D_1\delta = \sin\alpha_1$.

Aspect ratio (B/D_1) of 0.33 and 0.5 and flow speed ratio (B/W) of 1.0, 1.5, 2.0, and 3.0 were employed. The results were analyzed using multiple regression analysis.

i) When α_1 is decreased from 32^0 to 24^0, the maximum efficiency of CFT tends to improve and the highest efficiency with $\alpha_1 = 24^0$.

ii) With best $\alpha_1 = 24^0$ reported, the runner with $D_2/D_1 = 0.68$ outperforms when compared to $D_2/D_1 = 0.16$ and 0.75. However, when $\alpha_1 = 28^0$, $D_2/D_1 = 0.6$ does slightly better than $D_2/D_1 = 0.68$

iii) When the numbers of blades were increased from 15 to 30, the performance is found to improve. However, the relative increase is more when n_b is increased from 15 to 20 and is small when increased from 25 to 30

iv) The flow speed ratio (B/W) of 1.5 was found to perform better than 1.0, 2.0 or 3.0

v) The runner aspect ratio (B/D_1) of 0.33 performs better than B/D_1 of 0.5 by over 5 to 7% for higher speed ratios. However, the differences were narrow at lower speed ratios.

It is suggested that the maximum efficiency of 88% ± 2% can be achieved by careful selection of CFT parameters. The empirical relations have been developed for maximum efficiency in terms of geometric and flow-parameters, which is

$$\eta_{max} = 26.70 - 41.11(D_2/D_1) + 0.86N_b + (2{,}063.53/\alpha_1) - 4.16(B/W) + 0.31\beta_2 \qquad \text{------}(2.3)$$

Joshi *et al* (1995) carried-out extensive experimental studies to investigate the effect of blade number, nozzle entry arc, and head on the performance characteristics of a Cross Flow Turbine. The range of parameters investigated by them were blade numbers 8, 10, 16, 20, 24, and 30, nozzle entry arc 12^0, 26^0, 32^0, and 36^0, head 2, 4, 6, 8, and 9-m respectively. They had shown that in a CFT, the full conversion of the pressure head into velocity head does not takes place in the nozzle and the residual static head at the nozzle exit increases with an increase in supply head. The performance of the turbine improves with the increase in the nozzle entry arc and supply head provided that the flow passages as well as the housings are designed properly to handle the increased flow. As in the case of an impulse turbine, the unit discharge does not change significantly with variations in other parameters. They concluded that the turbine performance improves with the increase in number of blades up to a limit, beyond which it deteriorates due to an increase in blockage.

2.3 REVIEW OF THEORETICAL ANALYSIS

The recent theoretical studies commence with **Balje** (1981), who showed that the flow at the center of the runner approaches a free vortex flow. The vortex center is located between the inlet and exit at the runner inner radius of the runner, implying that the absolute flow angle at the entrance to the second stage is nearly the same as the absolute flow angle at the trailing edge of the first stage. Considering the incident losses at the second stage in the theoretical analysis, it was reported that an optimum blade configuration was determined and the maximum efficiency occurred at a speed ratio between 0.4 and 0.5. The maximum efficiency was computed as 73% and 82% for two different cases.

Durgin and **Fay** (1984) derived an expression for the maximum efficiency of the turbine as

$$\eta_{max} = \frac{1}{2C^2\left[(1-x)(1-\psi)+x\right]\cos\alpha_1} \qquad \text{-----------------(2.4)}$$

Where C is nozzle entrance loss coefficient,

x is ratio of cross flow to total flow and

ψ is the ratio of relative velocities for the crossed flow at the second stage exit to that at the first stage inlet.

Fukutomi *et al* (1985) numerically analyzed the flow from the cross flow turbine nozzle with arbitrary asymmetric curved surface, using Schwarz-Christopher transformation. The flow from a CFT nozzle gives some circumferential velocity and an optimum angle to flow at the nozzle exit, where it has free boundaries. The results of the numerical analysis were compared with the experimental results and both were found to agree well.

Further in 1991 **Fukutomi** *et al* made a two dimensional analysis of the turbine. The flow was divided into six regions, radially inward in the first three regions and radially outward in the last three regions. The study also included numerical calculations of the unsteady flow along streamlines in the relative system of the runner, and an investigation of the flow along the runner periphery. There was close agreement between the results of the numerical analysis and the experiments. The research succeeded in predicting the flow pattern for the top and bottom streamlines for various values of shaft rotational speed. The experiment indicated that the flow inside the runner was exceedingly non-uniform along the periphery and this was the reason for the decrease in CFT efficiency. In addition, flows at turn over regions (the uncrossed flow region) and at the last outward flow region had velocity components in the rotational direction only. At these regions the loss in the flow rate was high. If the runner diameter ratio were small, the regions would extend, resulting in a higher loss of flow rate.

Hothersall (1985) was first to attempt a two dimensional study of CFT. The analysis was used to illustrate the effect of the number of blades on the first stage when operating partially as reaction turbine. The results of study indicated that CFT is quite tolerant of blade number. It was also suspected that at low blade numbers the separation occurs at the suction faces of the blades and CFT operates as impulse turbine. The analysis demonstrated the importance of the Coriolis forces, which contributed about 40% of the circumferential pressure force on the blades.

Kong *et al* (1992) carried out theoretical studies of the absolute path of the water jet in the first stage of the CFT runner and determined that at a constant inlet angle, the absolute flow path is independent of the flow rate of water. The research concluded that the path line is dependent only on the runner geometry and that the butterfly valve located in the nozzle inlet acts more as a guide vane than as a flow control valve. It was recommended that more attention be focused on the butterfly valve, as it significantly affects the overall turbine performance.

Table 2.1 shows the consolidated position of details available from the literature so far.

TABLE 2.1 CONSOLIDATED LITERATURE OF EARLIER INVESTIGATIONS

Sl. No	REFERENCE		Angle of Attack, 'α_d' deg.	Runner diameter Ratio, 'D_2/D_1'	No. of Blades, 'N_b'	Nozzle entry arc, 'δ' deg	Efficiency 'η_{max}' % (reported)
1	Mockmore and Maryfield (1949)		16	0.66	20	-	68
2	Varga	(1959)	16	0.66	30	-	77
3	Johnson et al.	(1982)	16	0.68	18	-	80
4	Nakase et al.	(1982)	15	0.68	26	30,60,90, 120	80
5	Durgin and Fay	(1984)	16	0.68	20	50-80	66
6	Khosorowpanah	(1984)	16	0.68	10,15,20	58,78,90	80
7	Hothersal	(1985)	16	0.66	21	-	-
8	Fiuzat and Akerkar	(1989)	16,20,24	0.68	20	90-120	89
9	Ott and Chappell	(1989)	16	0.68	20	-	79.3
10	Desai et al.	(1994)	22,24,26, 28,32	0.6,0.68, 0.75	15,20,25,30	90	88
11	Joshi et al.	(1995)	16	0.66	8,10,16,20, 24,30	12,26,32, 36	62

2.3. PROBLEM STATEMENT:

After a thorough review of the literature, it was observed that the previous investigators have analysed the CFT with only one nozzle that loads the runner unevenly as only the bottom segment of the runner is loaded with water. Therefore, introduction of a second nozzle on to the runner periphery would enable to obtain an increase in power augmentation along with an increase in efficiency of the turbine had been visualised to be a fruitful improvement in the design of CFT.

Also, it was observed that the nozzle that is an all-important part of CFT had not been subjected to complete analytical scrutiny with a proper scientific basis for its different asymmetric shapes with multiple profiles. Hence, an attempt to ascertain the ideal shape of the nozzle of a CFT for the given conditions employing Ideal Fluid Theory has been aimed.

From the above considerations the work carried out are:

1. Design of nozzle wall shapes using Ideal Fluid Theory and analysis of flow through nozzle with designed profiles for front and rear walls to compare with already reported profiles using Finite Element Analysis.
2. Finite Element Analysis of flow through blade passages using the results available from nozzle analysis.
3. Experimental investigations – the effect of supply head, flow rate, number of blades, runner aspect ratio, inlet flow angle and nozzle entry arc on CFT performance with single nozzle.
4. A new approach – a CFT with two nozzles which might lead to effective utilization of total runner periphery and hence improvement in performance/power output – has been planned for experimental investigations.

The following parameters are identified as the pertinent variables on which the attention needs to be focussed for the present study.

Angle of attack in the 1st stage inlet, α_1	: 16^0, and 24^0
Diameter ratio, D_2/D_1	: 0.66
Number of blades, N_b	: 20, 24, and 30
Aspect ratio, B/D_1	: 0.33, and 0.5
Nozzle entry arc, δ	: 36^0, 60^o, and 90^0