Effect of Variation of Design Parameters on Cross Flow Turbine Efficiency Using ANSYS

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Abstract: Most of the major micro hydro power plants in Nepal uses Crossflow turbine for power generation which are manufactured locally. However, efficiency of these turbines has not been tested and verified. In this research, Cross flow turbine designs were obtained from local manufacturers. Efficiencies of these turbines were determined using simulation under steady state condition. Efficiencies were verified using the data from the installation site where these designs were used. Different Cross flow turbine models were prepared by varying the curvature radius of the blade and the ratio of inner to outer radius of the runner. The efficiencies of such models were determined using simulation.

Keywords: Cross flow turbine, efficiency, design parameters, blade angle

1. Introduction

The overall efficiency of a micro hydro power plant is considered in the range of 50 to 60 percentages. Turbine contributes to the major portion of the reduction of the overall efficiency in hydropower plant. The higher turbine efficiency would mean higher power output and consequently higher income for the same amount of investment. During 1996 to 2011, out of 308 numbers of Micro hydropower systems, 174 schemes used Cross flow turbines installed through Rural Energy Development Program (REDP) in Nepal [14].

Studies were performed to improve the efficiency of cross flow turbine. Effects of design parameters on the efficiency have been matter of extreme interest for researchers. There have been numbers of experimental researches testing the effects of variation of design parameters on the efficiency [2, 10, 12, 13, 16, 17]. Most of the researches have followed the standard design where the outlet angle of the turbine blade is always 90°. The studies have tried to study the effect of variation of outlet angle of the blade on the efficiency by varying diameter ratio [12, 13]. Research has also been done using commercial CFD code investigating the power output of cross flow turbine while varying inlet and outlet angles [17]. In the present study, a standard design model being using by manufacturers in Nepal is tested using CFD analysis. For other models, the diameter ratio and the radius of the blade curvature is varied to produce nine different models for the study.

2. Literature Review
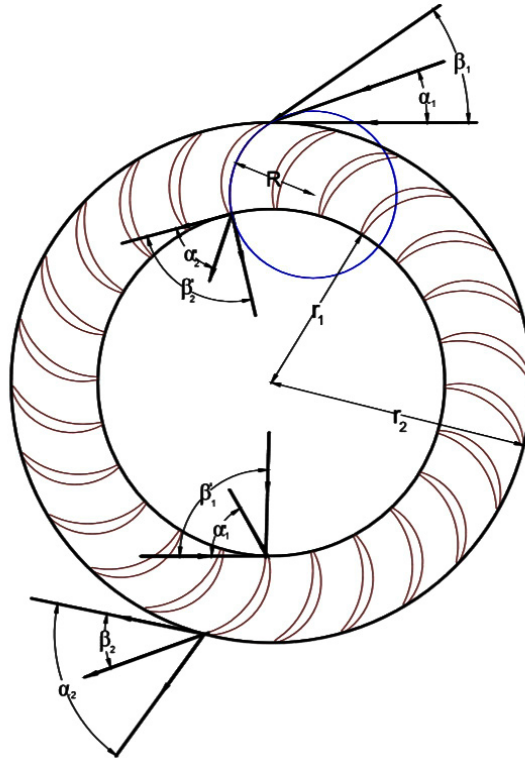
2.1 Nomenclature and Design

Fig.1: Cross flow Turbine Runner

- R is the radius of blade curvature
- β is the angle between relative velocity and peripheral velocity (also can be termed as angle between the tangent of the blade and runner periphery or *blade angle*)
- α represents the angle between absolute and peripheral velocity
- the sub-scripts '1' and '2' in case of α and β represents inlet first stage and inlet second stage respectively
- the super-script 'I' in case of α and β represents at the inside diameter of the wheel
- r_1 is the inner radius of the turbine runner
- r_2 is the outer radius of the turbine runner
- r_1/r_2 is the diameter ratio
- H is the design head
- Q is the design flow

2.2 Cross flow turbine design

The cross flow turbine design prevailing in most manufacturers of Nepal was collected from Kathmandu Metal Industry, a leading designer of cross flow turbine in Nepal. The design is based for maximum head of 88m. The turbine was designed for 1500 rpm and outer diameter of the turbine is taken 0.25 m. The breadth of the turbine is 150mm. Design value of flow at maximum head was calculated to be 308 lps. The blade angle at inlet (β_1) is 30º. The *angle of attack* (α_1) as

given by the surface of the nozzle is 16°. The blade angle inside the runner (β_1^{I} and β_2^{I}) is 90°. The number of blades of the runner is taken 24.

2.3 Analyzing the design

The major design parameters of the obtained design are based on design suggested in [7]. However, the improvement is done in the surface and structure of the blade which differs from the normal blade design used in cross flow turbines.

2.4 Identifying design parameters to vary

The blade angles inside the runner (β_1^{I} and β_2^{I}) are equal to each other as they are the angles given by the runner blade which is consistent throughout the runner. Also, α_2^{I} equals α_1^{I} for the same reasons [7]. Considering, the absolute velocity of water at outlet of first stage is equal to the absolute velocity of water at the inlet of second stage of runner, the relative velocity of water at the second stage of the runner equals to the resultant velocity of water at the outlet of first stage. Due to this, blade angle corresponds to the angle made by the relative velocity at the inlet of second stage ($\beta_1^{\text{I}} = \beta_2^{\text{I}} = 90°$), so that there is no shock loss. However, the maximum efficiency is not a function of the angles β_1^{I} and β_2^{I}. So, considering this fact, the blade angle was considered as the parameter to vary to study its effect on efficiency. The blade angle of the cross flow turbine could be varied by varying either the radius of curvature of the blade or the diameter ratio of the runner (r_1/r_2) or both. The standard value of radius of curvature of the blade is 0.05 m and the diameter ratio is 0.67. These parameters were varied with the purpose of testing its effect on the efficiency of the runner.

3. Preparation of Different Models

CAD model of the standard design obtained from Kathmandu Metal Industry was varied to obtain the nine different CAD models of the turbines. The varied parameters were radius of curvature of the blade and the diameter ratio. The radiuses of curvatures used were 0.045, 0.05 and 0.055 m. The diameter ratios used were 0.6, 0.67 and 0.7. Using these parameters, nine different models were made. The runner of the standard model was used for all varied runner models.

Table 1: Formulated Crossflow Turbine Models with Varying Design Parameters

Model	Diameter Ratio (r_1/r_2)	Blade Curvature Radius (R) (m)	Remarks
1	0.6	0.045	
2	0.67	0.045	
3	0.7	0.045	
4	0.6	0.05	
5	0.67	0.05	Standard Model
6	0.7	0.05	
7	0.6	0.055	
8	0.67	0.055	
9	0.7	0.055	

For all the models the angle of attack and the number of blades were kept constant as that of the

standard model. The outer radius of turbine is 250mm, the angle of attack is 16° and the number of blades is 24.

4. Methodology

4.1 Rotor mesh preparation

Among nine different models prepared, the rotor mesh was obtained using ANSYS Turbogrid. For the mesh preparation on turbogrid, the CAD model from AutoCAD was transferred to Catia V5 to obtain the co-ordinates of the blade, hub and shroud of each model of the cross flow turbine. Thus obtained co-ordinates were transferred in to a notepad file and consequently to Turbogrid to obtain the mesh files. The mesh for the casing of the runner was prepared using ANSYS Mesh.

4.2 ANSYS CFX Setup

The simulation of the rotor was done using ANSYS CFX on ANSYS Workbench as platform. The casing CAD file from Catia V5 was imported to ANSYS Design Modular. The mesh of the casing was produced using ANSYS ICEM CFD. Thus produced mesh file was used in ANSYS CFX along with the mesh file of the rotor produced from Turbogrid. The following major parameters were used for CFX Setup for the simulation of the nine formulated runner models.

- Analysis Type: Steady State
- Fluid: Water, Air
- Homogenous Model
- Turbulence: k-Epsilon
- Boundary Type: Mass Flow Inlet Pressure Static Outlet
- Mass Flow Inlet: 308 kg/s
- Pressure Static Outlet: 1 atm
- Interface: Frozen Rotor
- Interface Type: Fluid
- Water Volume Fraction at Inlet: 1
- Air Volume Fraction at Inlet: 0
- Convergence: 0.0001

For the standard model, Transient analysis type was also used. Also for the standard model, different boundary conditions (Total Pressure Inlet Mass Flow Outlet and Total Pressure Static Inlet Pressure Outlet) were used to evaluate the efficiency under different boundary conditions. For the total pressure inlet, the value of pressure was taken as 862,207 kPa which is equivalent of 88 m maximum design head of the standard model of the turbine.

4.3 Field Verification Data

The power output testing data of three micro hydro sites were obtained where crossflow turbine was installed. The micro hydro powers were Nurkhuwa Micro Hydro Power System (Khotang District), RoshiKhola MHS (Kavre District) and SobuwaKhola MHS (Taplejung District) in Nepal. The data were used to estimate the efficiency of the crossflow turbines on the site.

5. Results and Discussion

5.1 Efficiency from onsite Power Output Testing

The POT data were obtained for three sites as mentioned above. However, the data from RoshiKhola and NurkhuwaKhola were not reliable as the power testing did not yield stable results during different phase of testing. For the estimation of turbine efficiency, the generator efficiency and drive efficiency in assumed to be 80% and 95% respectively. The total power was measured by allowing the ballast load to consume all the produced power, and hence the turbine efficiency was calculated to be 71%. This efficiency is low compared to the efficiency as mention in various literatures.

Mesh Statistics of Rotor

The mesh of a single blade obtained from turbogrid was transformed in to the mesh of rotor with 24 blades on ANSYS CFX Setup. The following table shows the mesh statistics of the rotors of all models as shown in CFX Setup.
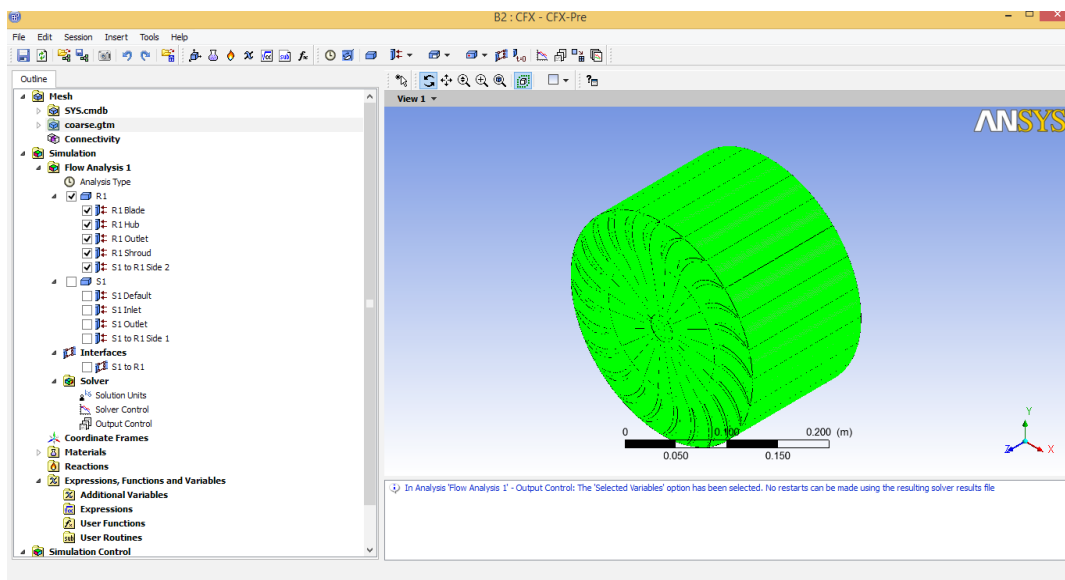
Fig. 2: Mesh Statistics of the Rotors

Table 2: Mesh Statistics for Different Rotor Models

Model No.	Total Number of Nodes	Total Number of Elements	Total Number of Hexahedrons
1	3819024	3538080	3538080
2	4342272	4040064	4040064
3	4653792	4338432	4338432
4	1322112	1164240	1164240
5	4399824	4096512	4096512
6	4283664	3984120	3984120

7	3879744	3594528	3594528
8	4260960	3960432	3960432
9	1368576	1209600	1209600

5.3 Mesh Statistics of Casing

The casing used for all rotors were same. The number of nodes in the mesh of the casing was 44,478. Total number of elements was 36960 out of which total numbers of prism were 640 and total numbers of hexahedrons were 36320.
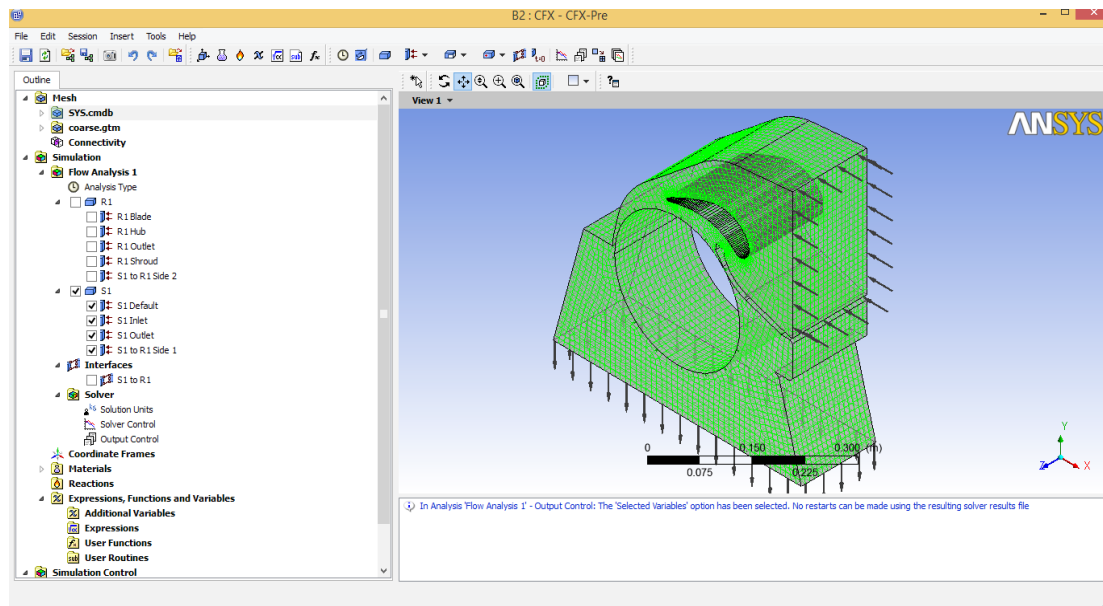

Fig. 3: Mesh Statistics of the Casing

5.4 Efficiency of Rotors

The power developed at the shaft of the runners and the hydraulic power was calculated using the following expression in ANSYS CFD Post.

$$Pshaft = torque_{z()}@R1Blade * 2 * Pi * \frac{omega}{360} \ [degree] \tag{1}$$

$$Phydraulic = (massFlowAve(Pressure)@S1 \ Inlet - massFlowAve(Pressure)@S1 \ Outlet) *$$
$$massFlow()S1\frac{Inlet}{1000} \ [\frac{kg}{m^3}] \tag{2}$$

The efficiency was calculated using following expressions.

$$Efficiency = \frac{Pshaft}{Phydraulic} * 100 \tag{3}$$

Table 3: Efficiency of Different Models

Model Number	P_{shaft}(Watt)	$P_{hydraulic}$ (Watt)	Efficiency (%)	(r_1/r_2)	Radius of curvature of Blade
1	291178	442005	65.87	0.6	45
2	290456	434105	66.9	0.67	45
3	289469	430097	67.3	0.7	45
4	291282	438678	66.4	0.6	50
5	294107	437966	67.153	0.67	50
6	288130	434807	66.26	0.7	50
7	291415	448552	64.96	0.6	55
8	287809	443486	64.897	0.67	55
9	290155	438964	66.1	0.7	55

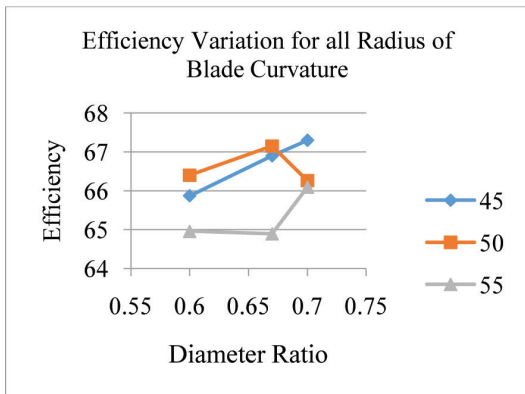## 5.5 Variation of Efficiency

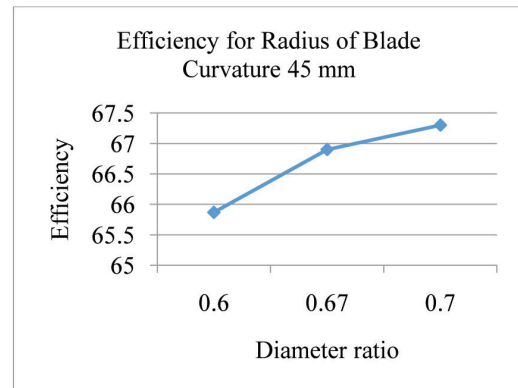**Fig. 4: Efficiency variation of different model**

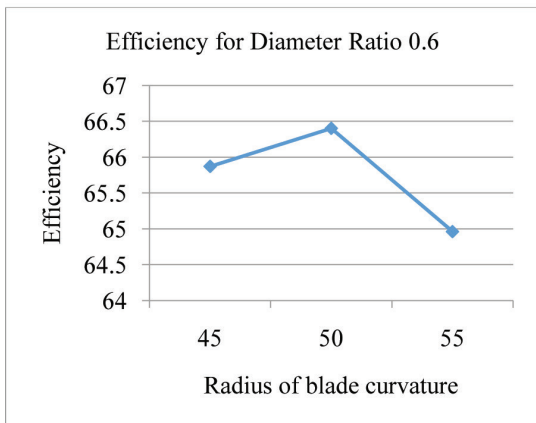**Fig.7: Efficiency for radius of blade curvature 45 mm**

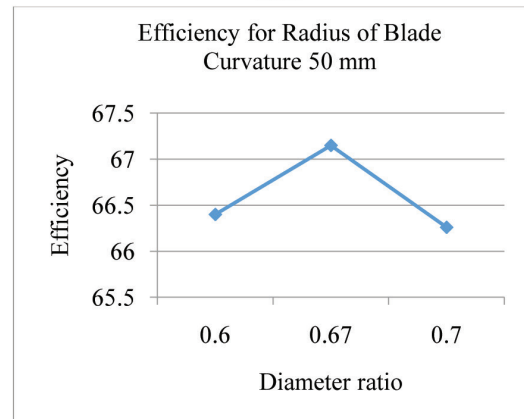**Fig. 5: Efficiency for Diameter Ratio 0.6**

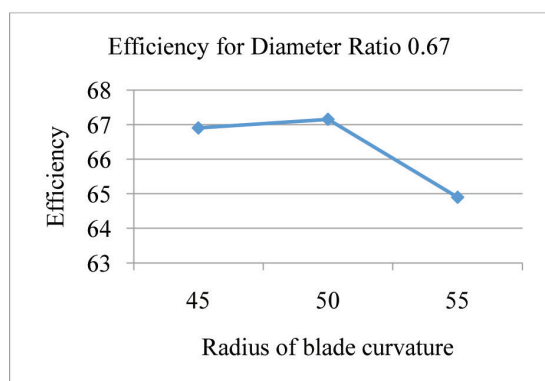**Fig. 8: Efficiency for Radius of Blade Curvature 50 mm**

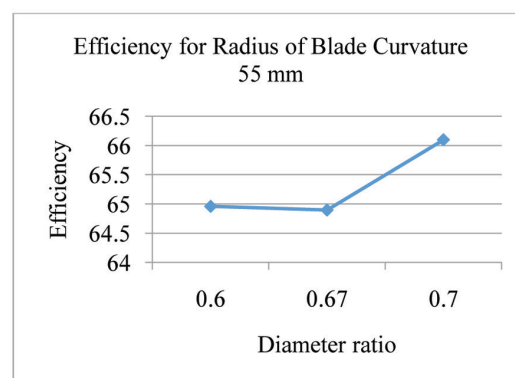**Fig. 6: Efficiency for Diameter Ratio 0.67**

Fig. 9: Efficiency for Radius of Blade Curvature 55mm

The efficiency vs. diameter ratio graph for different blade curvature radius does not show a common pattern for efficiency variation for the different radius. However, by varying the radius of blade curvature and diameter ratio the efficiency is increased in case of blade curvature radius 0.055m and diameter ratio 0.7.

6. Conclusion

Among the various Cross flow turbine models which were prepared by varying the design parameters, most efficient model was found to be Model 3 which has radius of blade curvature 0.045 m and diameter ratio 0.7. The efficiency of the Model 3 is 67.3%. The design value is different from that of the standard design which has 0.05 m blade curvature radius and diameter ratio of 0.7. The efficiency of the standard model (Model 5) was found to be 67.15% in steady state simulation. The onsite data yielded efficiency as 71% which was close to the result from the simulation. Conclusion that can be derived from this research is that the efficiency has increased at outlet angle which is different from traditional angle of 90°. This conclusion is supported by the conclusion derived by Young Do Choi [17] where the research shows that the power produced in the second stage of the cross flow turbine is highest in case of outlet angle of 80°. It implies that the power produced in the second stage of the turbine is greater when the outlet angle deviates further away from 90°. In this research, the total power in both first and second stage is highest with lowest inlet angle and outlet angle of 87°.

The variation of parameters has led to increase in efficiency even though the increased value is insignificant. As the value of the varied design parameters were taken to the close proximity of the standard design values, the results shows room for improvement of the efficiency by taking wide range of variation of the radius of blade curvature and the diameter ratio. The maximum efficiency is also dependent upon the angles β_1' and β_2' so, more relevant theory and experimental verification is required to identify the optimum values of β_1' and β_2' for maximum efficiency.

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