Volume 1, Issue 7, pp. 37-41, 2017.
International Journal of Scientific Engineering and Science
ISSN (Online): 2456-7361

# Design of Propeller Turbine for Micro Hydro Power Station Using CFD

Ravi Shankar Shukla[1], Dr. Charu Parashar[2]
[1]Scholar (Hydro Power Engineering), Department of Civil Engineering, Maulana Azad National Institute of Technology (MANIT), Bhopal, India
[2]Professor, Department of Civil Engineering, Maulana Azad National Institute of Technology (MANIT), Bhopal, India

***Abstract**—Low head power plants are expected to be implemented increasingly in the future for economical, geographical and environmental purposes. Propeller turbines are well suited for these types of applications. They operate at higher flow rate, smaller head and faster rotational speed, thus being more compact than other types of machines; its design and performance improvement is of great concern. In the present work Runner and distributor of a Propeller turbine are designed using Bladegen software and a conical draft tube is designed in ICEM CFD. Blades geometry generated in Bladegen is exported to TURBOGRID to do its volumetric meshing. The complete passage of flow including distributor, runner and draft tube is numerically simulated using SST model in Ansys CFX. Simulations have been carried out at different speed of rotation varying from 316 rpm to 516 rpm keeping the discharge constant. The results obtained from flow simulations are found to be in accordance with turbine characteristics curves. Results and their variations with speed are represented graphically.*

***Keywords**— Propeller Turbine, Runner, Distributor, CFD, Efficiency, Head, Mesh.*

## I. INTRODUCTION

Turbines convert hydraulic energy of water into mechanical energy which is further converted into electrical energy. This energy obtained is known as hydro-electric power which is one of the cheapest forms of energy generation. The amount of production of electricity from a hydropower installation depends on the quantity of water passing through a turbine and head of water available. The greater the flow and head the more electricity is produced. The role of hydro plants becomes more and more important in today's global renewable energy. Principal components consist of a spiral casing, stay vanes, wicket gates/guide vanes, runner and a draft tube.

*Classification of Hydro Power Plants*

Hydroelectric plants based on their installation capacity are classified as follows:

- Large : > 100 MW
- Medium : 25 MW - 100 MW
- Small :1 MW- 25 MW
- Mini : 100 KW- 1 MW
- Micro : 5 KW – 100 KW
- Pico : < 5 KW

*[A.] According to Action of Water Flowing Through Runner*

*Impulse Turbine*: Impulse turbines work based on momentum principle.

*Reaction Turbine*: The flow is fully pressurized and it works according to conservation of angular momentum.

*[B.] According to Flow Direction of Water in Runner*

*Tangential Flow Turbine*: In this type of the turbines, water strikes in the runner in a direction tangential to wheel. Example: Pelton turbine

*Radial Flow Turbine*: In these turbines, water enters from the radial direction. Example: Francis turbine.

*Axial Flow Turbines*: The water flows in a direction parallel to the axis of the runner shaft. Example: Kaplan turbine, propeller turbine and bulb turbine.

Axial flow turbines are available with both fixed blades (Propeller) and variable pitch blades (Kaplan) which may be mounted either horizontally or vertically. It allows the fluid to enter the runner axially and discharge the fluid axially. The propeller shaped runner evolved from the Francis mixed flow runner that fulfills the need for a faster unit using a large quantity of flow with low head. The turbine having a propeller shaped runner is known as propeller turbine, which is an axially flow reaction turbine. Propeller turbine has fixed runner blades whereas the Kaplan turbine is just a Propeller turbine in which the runner blades are made adjustable. The propeller turbine can be employed economically when it has to work constantly under full load; otherwise Kaplan turbine will be preferred.

Improvement in performance of Propeller turbine is of great concern and also a wide scope in research area. The performance of propeller turbine can be measured experimentally as well as with the help of advanced numerical tool called, computational fluid dynamics (CFD). The experimental methods for numerical simulation are costly and time consuming, Computational fluid dynamics (CFD) become a cost effective tool to provide detailed flow information inside the complete turbine space as a whole so that interaction between different components could also be considered. CFD has been widely used by designer and researchers to optimize its design. Ferro[1] et al. [2010] designed a guide vane system of a mini hydraulic turbine using a quasi-three-dimensional method by prescribing constant angular-momentum distribution along the vanes. Prasad[2] V. [2011] simulated viscous 3D turbulent flow at different guide vane opening and at different rotational speeds of an experimentally tested axial turbine model using SST Ķ-ω model in Ansys CFX software and the numerically simulated results for variation of discharge factor, efficiency and specific energy were agreeing with experimental results of any axial turbine. Rivetti[3] et al. [2012] carried out numerical simulations for incompressible transient flow through a prototype Kaplan turbine by applying RANS equations under fixed volume scheme using Ansys CFX codes. Good agreement was found in pressure fluctuations both in shape and amplitude at lower frequencies. Yu War Myint 4 et al.(2014) described design calculation of runner blade for that they utilized Solid Works flow simulation for predicting the flow analysis of runner. Pankaj[5] et al. (2016) reviewed design work performed on Micro Hydro Kaplan Turbine. This study mainly deals with an effective study of different profiles of turbine blade for maximum efficiency.

Performance characteristics of propeller turbine are derived by plotting graphs for various local and global flow parameters computed in non-dimensional form.

## II. Propeller Turbine Specifications

The Propeller turbine is designed for 1499 kg/s discharge, speed 416 rpm, head 4m and power output is 58.48 kW. Runner and distributor of a Propeller turbine are designed using Bladegen software and a conical draft tube is designed in ICEM CFD. TURBOGRID is used to do its volumetric meshing.

## III. Design Methodology

In this work, the 1-D flow method is chosen for designing the runner of propeller turbine and calculations are done as below:

The power developed by a turbine is calculated by following formula

$$P = \rho g Q H \eta_0$$

Specific speed of the turbine $N_s = \dfrac{885.5}{H^{0.25}}$

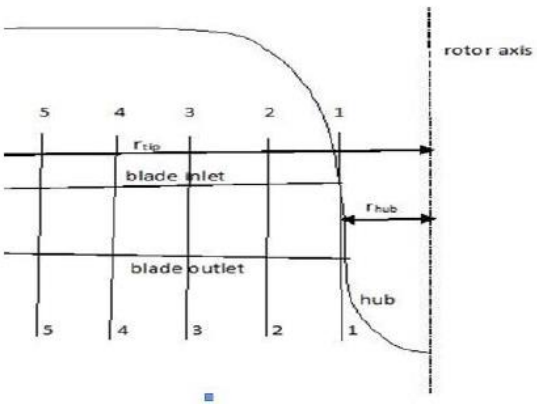
Speed of the turbine runner $N = \dfrac{N_s H_d^{1.25}}{\sqrt{P}}$

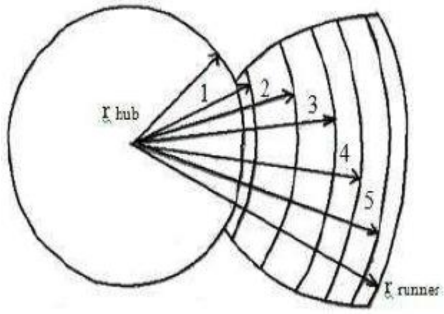
Based on specific speed, values of flow coefficient, speed coefficient and hub to shroud diameter ratio (d/D) are selected. Then, meridional velocity component is calculated by using equation

$$C_m = k_f \sqrt{2gH}$$

Computed value of Cm is taken as constant from inlet to outlet at any radius and also from hub to shroud.

$$Q = \frac{\pi (D^2 - d^2) C_m}{4}$$




Tangential velocity (u) at any section: $u = \dfrac{\pi DN}{60}$

Blade angles and flow angles for each cylindrical section at inlet and outlet are calculated as,

$$\beta_1 = \tan^{-1} \frac{C_{m1}}{u - C_{u1}}$$

$$\beta_2 = \tan^{-1} \frac{C_{m2}}{u - C_{u2}}$$

$$\alpha_1 = \tan^{-1} \frac{C_{m1}}{C_{u1}}$$

$$\alpha_2 = \tan^{-1} \frac{C_{m2}}{C_{u2}}$$

*Formulae Used for Calculated in Non Dimensional Parameters,*

Different flow parameters are computed in non-dimensional form to show runner performance at different operating conditions. Formulae used for these flow parameters are given below:

| | | |
|---|---|---|
| Specific energy coefficient | $\psi = \dfrac{gHD^4}{Q^2}$ | (1) |
| Speed factor | $SF = \dfrac{ND}{\sqrt{gH}}$ | (2) |
| Discharge factor | $DF = \dfrac{Q}{D^2\sqrt{gH}}$ | (3) |
| Runner head | $H_R = \dfrac{(p_{T1} - p_{T2})_{Stat\ frame} - (p_{T1} - p_{T2})_{Rot\ frame}}{\rho g}$ | (4) |
| Hydraulic efficiency (%) | $\eta_h = \dfrac{H_R}{H} * 100$ | (5) |
| Hydraulic losses (%) | $H_L = \dfrac{H_{loss}}{H} * 100$ | (6) |
| Specific meridional velocity | $c_m^* = \dfrac{C_m}{\sqrt{2gH}}$ | (7) |
| Specific whirl velocity | $c_u^* = \dfrac{C_u}{\sqrt{2gH}}$ | (8) |
| Specific absolute velocity | $c^* = \dfrac{C}{\sqrt{2gH}}$ | (9) |

International Journal of Scientific Engineering and Science
Volume 1, Issue 7, pp. 37-41, 2017. ISSN (Online): 2456-7361

## IV. Geometric Modelling and Simulation

The runner and the distributor are modelled in Ansys BladeGen and it consists of inlet, outlet, hub, shroud and blades. The complete geometry of propeller turbine model is shown in the Fig. 1. The specifications of Propeller turbine model is shown in table 1.

TABLE 1. Specifications of designed propeller turbine.

| | |
|---|---|
| Distributor, runner diameter | 0.68 m |
| Hub diameter (distributor, runner) | 0.26 m |
| Number of runner blades | 4 |
| Number distributor blades | 10 |
| Inlet diameter of draft tube | 0.68 m |
| Outlet diameter of draft tube | 1.3 m |
| Length of draft tube | 3.41 m |

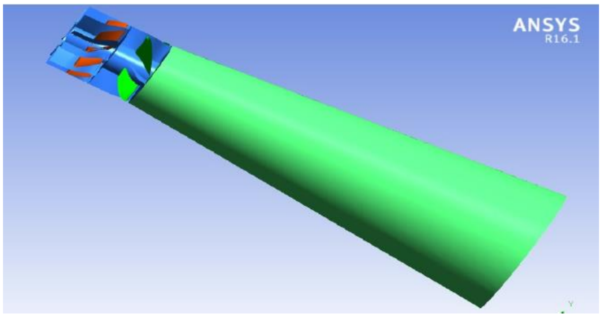



Fig. 1. Designed propeller turbine.

## V. Results and Discussions

The flow has been simulated for a value for rotational speed varying from 316 rpm to 516 rpm at regular intervals of 50 rpm .The variations of hydraulic efficiency with speed is shown in fig. 2. Hydraulic efficiency shows a parabolic variation with speed which is a characteristic of an axial flow turbine. The turbine is found to have a maximum hydraulic efficiency of 76.38 % at a discharge of 1499 kg/s and rotational speed of 416 rpm.

The variation of degree of reaction of the turbine with speed is presented in Fig. 3. It is seen from the graphs that degree of reaction decreases with increase in speed.

The variation of discharge factor at constant discharge with different speed is shown in Fig.4. As speed increases net head decreases and in turn speed factor also increases. Also with increase in speed the turbine sucks more volume of water and discharge factor also increases.

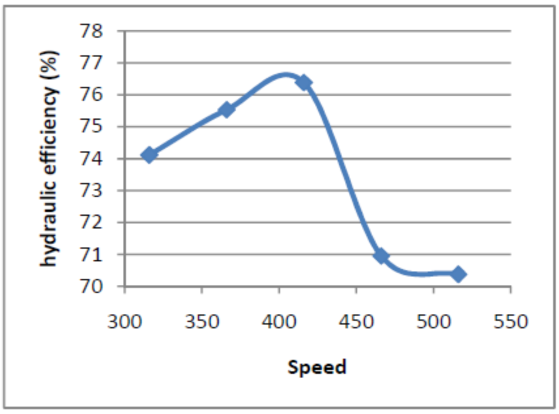



Fig. 2. Variation of hydraulic efficiency.

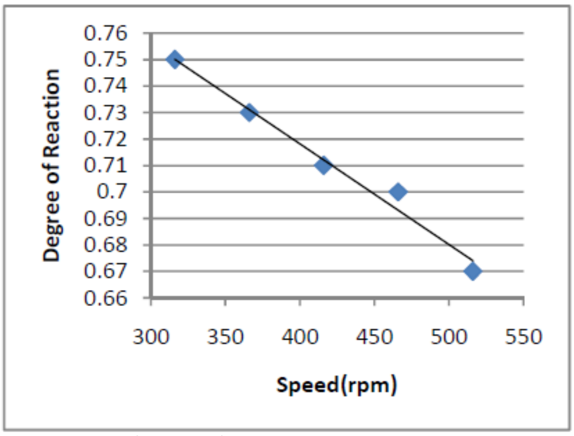



Fig. 3. Variation of degree of reaction at blade Q = 1499 kg/s with speed.

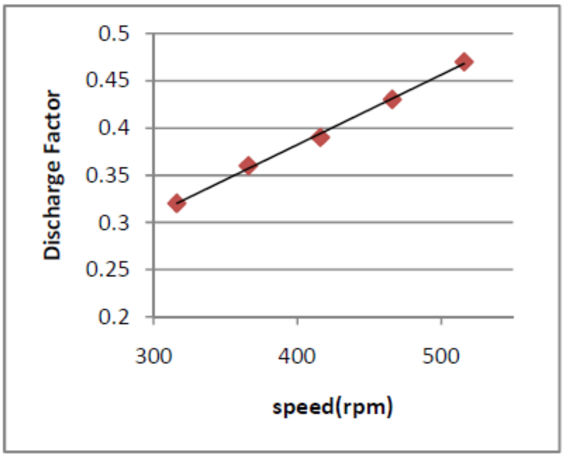



Fig. 4. Variation of discharge factor at Q = 1499 Kg/s with speed.

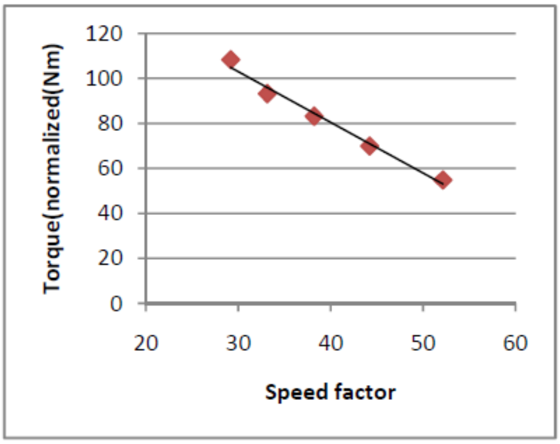



Fig. 5. Variation of torque developed on runner blade Q = 1499 kg/s with speed.

Fig. 5 shows variations of torque developed on runner blade with speed factor at Q=1499 kg/s with varying speed. Torque on blades and runner head have similar relationship with speed factor and so torque also decreases with increase in speed factor.

Pressure contours at constant discharge of 1499 kg/s and rotational speed 416 (design condition) rpm is shown in Fig. 6. Maximum pressure inside the runner decreases with increase in runner speed. Apart from design condition the pressure

International Journal of Scientific Engineering and Science
Volume 1, Issue 7, pp. 37-41, 2017. ISSN (Online): 2456-7361

difference is not enough to create the required torque on blades and to get optimum efficiency. In the below figure of pressure contours it is seen that pressure inside runner decreases from its inlet to outlet which is just because the pressure energy is being converted into mechanical energy.

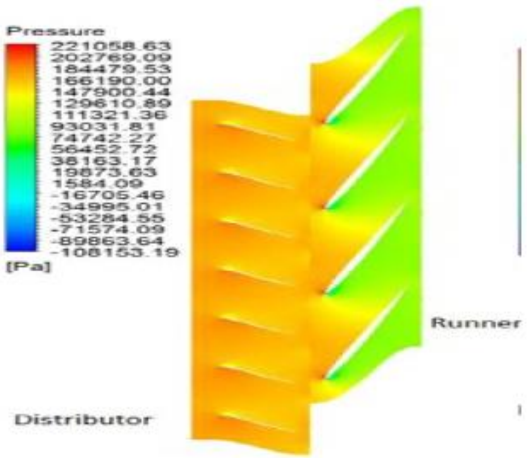



Fig. 6. Pressure contours on blade to blade view at mid span at Q = 1499 kg/s and at speed of 416 rpm.

The pressure distribution on runner blade surfaces from its leading edge to trailing edge at its mid span at a discharge of 1499 kg/s and speed of 416 rpm is shown in Fig. 7. The pressure difference between pressure and suction surface firstly increases from leading edge as water strikes on the blade and after that decreases smoothly to meet towards the trailing edge

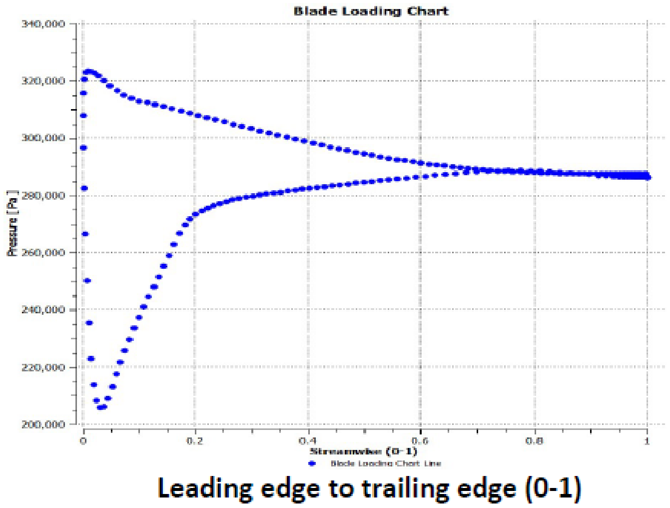



Fig. 7. Blade loading at mid span at rotational speed 416 rpm and discharge of 1499 kg/s.

Variations of velocity streamline patterns in distributor and runner domains for various speed of runner is shown in Fig.8. It is observed that that there is not much change in velocity inside distributor but inside runner it increases from its inlet to outlet.

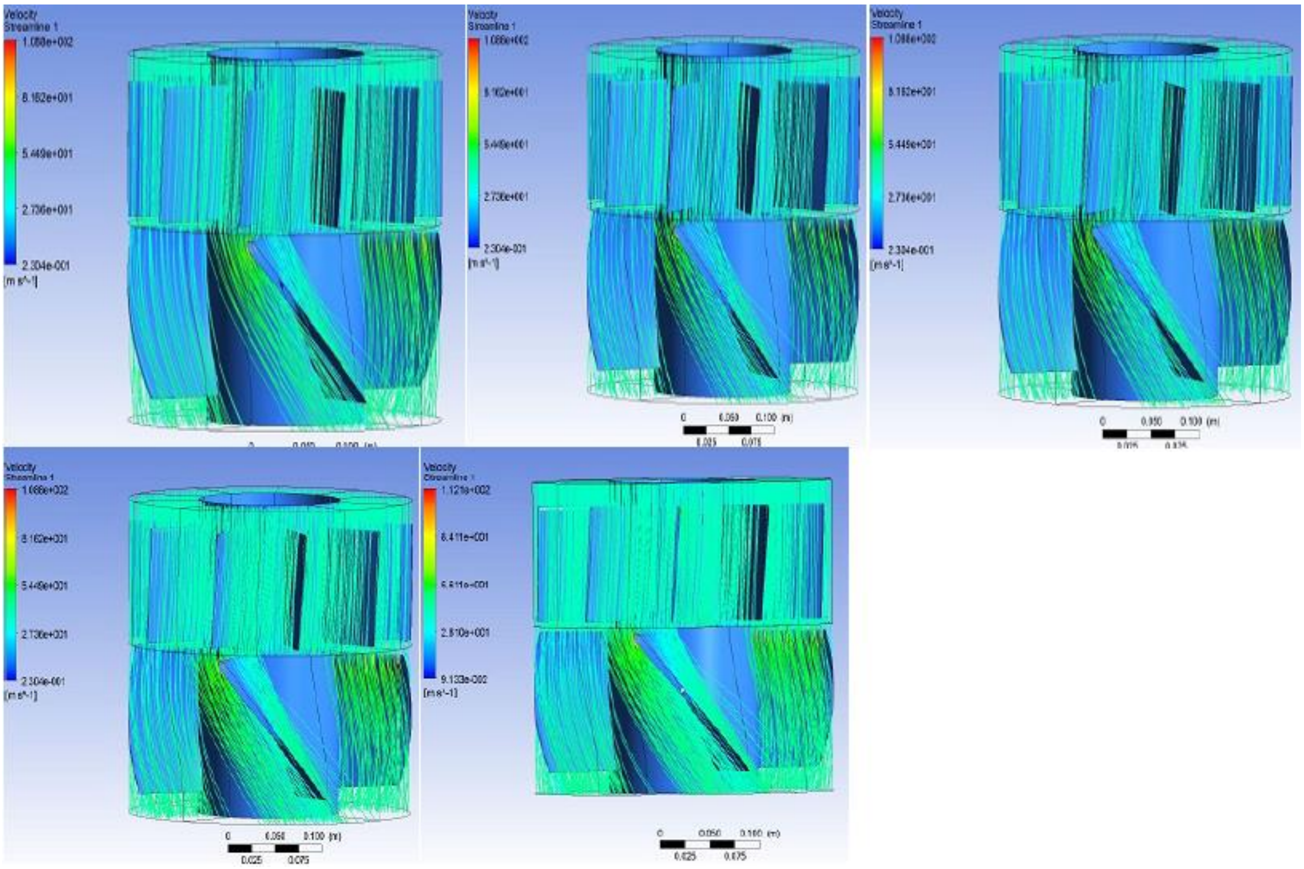


Fig. 8. Streamlines in distributor and runner at Q = 1499 kg/s and at rotational speed of (a) 316 rpm (b) 366 rpm (c) 416 rpm (d) 466 rpm and (e) 516 rpm.

## VI. CONCLUSIONS

It has been observed from the numerical simulations of Propeller turbine at different runner speed, that When discharge is kept constant as 1499 kg/s and runner speed is increased from 316 rpm to 516 rpm at an interval of 50 rpm, then hydraulic efficiency of turbine first increases up to 416 rpm and then it starts decreasing, along a parabolic curve, which shows that Q = 1499 kg/s and N = 416 rpm are design operating parameters. Maximum efficiency is found to be 76.38%. The net head decreases with increase in runner speed. The torque on runner blades decreases with increase in runner speed. Maximum pressure inside the runner decreases with increase in runner speed.

## REFERENCES

[1] L. M. C. Ferro, L. M. C. Gato, and A. F. O. Falcão, "Design and experimental validation of the inlet guide vane system of a mini hydraulic bulb-turbine," *Renewable Energy*, vol. 35, issue 9, pp. 1920-1928, 2010.

[2] V. Prasad, "Numerical simulation for flow characteristics of axial flow hydraulic turbine runner," *Energy Procedia*, vol. 14, pp. 2060-2065, 2012.

[3] A. Rivetti, C. Lucino, S. Liscia, D. Muguerza, and F. Avellan, "Pressure pulsation in Kaplan turbines: Prototype-CFD comparison," *In IOP Conference Series: Earth and Environmental Science*, vol. 15, no. 6, pp. 062035, IOP Publishing, 2012.

[4] Y. W. Myint and H. Htay Win, "Design and flow simulation of runner blade for propeller turbine," *International Journal of Scientific Engineering and Technology Research (IJSETR)*, vol. 03, issue 11, pp. 2259-2262, 2014.

[5] P. P. Amonkar, A. U. Naik, S. Devashetty, and V. Kolar, "Structural analysis on micro hydro Kaplan turbine blade," *International Journal for Innovative Research in Science & Technology (IJIRST)*, vol. 2, issue 11, pp. 2349-6010, 2016.

[6] A. L. A. Mesquita, C. M. V. Serra, D. O. A. Cruz, and N. M. Filho, "A simplified method for axial-flow turbomachinery design", *Journal of the Brazilian Society of Mechanical Sciences*, vol. 21, no. 1, pp. 61-70, 1999.

[7] S. Liu, S. Li, and Y. Wu, "Pressure fluctuation prediction of a model Kaplan turbine by unsteady turbulent flow simulation," *Journal of Fluids Engineering*, vol. 131, issue 10, 101102, 2009.

[8] M. Banaszek and K. Tesch, "Rotor blade geometry optimisation in Kaplan Turbine," *TASK Quarterly*, vol. 14, issue 3, pp. 39-55, 2010.

[9] L. M. C. Ferro, L. M. C. Gato, and A. F. O. Falcão, "Design of the rotor blades of a mini hydraulic bulb-turbine," *Renewable Energy*, vol. 36, issue 9, pp. 2395-2403, 2011.